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Energy and exergy analysis of a ground-coupled heat pump system with two horizontal ground heat exchangers

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Abstract

In this paper we investigate of energetic and exergetic efficiencies of ground-coupled heat pump (GCHP) system as a function of depth trenches for heating season. The horizontal ground heat exchangers (HGHEs) were used and it were buried with in 1 m (HGHE1) and 2 m (HGHE2) depth trenches. The energy efficiency of GCHP systems are obtained to 2.5 and 2.8, respectively, while the exergetic efficiencies of the overall system are found to be 53.1% and 56.3%, respectively, for HGHE1 and HGHE2. The irreversibility of HGHE2 is less than of the HGHE1 as about 2.0%. The results show that the energetic and exergetic efficiencies of the system increase when increasing the heat source (ground) temperature for heating season. And the end of this study, we deal with the effects of varying reference environment temperature on the exergy efficiencies of HGHE1 and HGHE2. The results show that increasing reference environment temperature decreases the exergy efficiency in both HGHE1 and HGHE2.

Keywords: Ground coupled heat pump; Horizontal ground heat exchanger; Energy analysis; Exergy analysis

1. Introduction

The use of ground-coupled heat pumps (GCHPs) in commercial and residential facilities is a remarkable example. GCHP systems exchange heat with the ground, and maintain a high level of performance even in colder climates. This results in more efficient use of energy, for this reason many society utilities support the use of GCHP systems. The first patent on using ground as a heat source for GCHP systems, issued in Switzerland in 1992, is due to Heinrich Zoley. This date may be regarded as the official date of birth of GCHP systems, but the concept of using the ground as a heat source is much older [1].

In history, there has been a noticeable increase of interest in the applications second law analysis to the design of thermal systems [2]. A typical thermal design based on the first law thermodynamics allows us to address issues related to the energy balance of the system. The second law of thermodynamic analysis combined with a standard design procedure of a thermal system gives us invaluable insight into the operation of the system. Exergy (or availability) analysis is a powerful tool in the design, optimization and performance evaluation of energy systems. This analysis can be used to identify the main sources of irreversibility (exergy loss) and to minimize the generation of entropy in a given process where the transfer of energy and material take place [3,4]. According to Dincer and Rosen [5], exergy analysis is an effective thermodynamic scheme for using the conservation of mass and energy principles together with the second law of thermodynamic for the design and analysis of thermal systems, and is an efficient technique for revealing whether or not and by how much it is possible to design more efficient thermal systems by reducing the inefficiencies. The concepts and definitions of exergy analysis are well recognized [6–9].

Various theoretical [10–13] and experimental [14–18] studies based on the exergy concept with heat pump systems have been published. Nakanishi et al. [10] investigated exergetic performance of various heat pump units. Hiharat [11] has been carried out theoretically heat

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Nomenclature

$COP_{overall}$	heating coefficient of performance of ground coupled heat pump system (−)
$C_{p,air}$	specific heat of air (kJ/kg K)
$C_{p,wa}$	specific heat of water–antifreeze solution (kJ/kgK)
$\dot{E}$	energy rate (kW)
$\dot{E}x$	exergy rate (kW)
$\dot{F}$	exergy rate of the fuel (kW)
h	specific enthalpy (kJ/kg)
$\dot{I}$	rate of irreversibility (kW)
$\dot{m}$	mass flow rate (kg/s)
P	pressure (Bar)
$\dot{P}$	exergy rate of the product (kW)
$\dot{Q}$	heat transfer rate (kW)
s	entropy (kJ/kg K)
T	temperature (°C)
$\dot{W}$	work rate or power (kW)

Greek letters

ε	exergy (second law) efficiency (dimensionless)
ψ	specific exergy (kJ/kg)

Subscripts

0	dead state
act	actual
comp	compressor
cond	condenser
cp	circulating pump
cfan	condenser fan
ct	capillary tube
d	destroyed, destruction
e	entropy
eva	evaporator
gen	generation
ghe	ground heat exchanger
HE	heat exchanger
i	inlet
o	outlet
overall	system
ref	refrigerant
r	location
tot	total
wa	water–antifreeze

Superscripts

CH	chemical
εx	specific exergy (kJ/kg)
KN	kinetic
PH	physical
PT	potential

pump systems and analyzed each transformation from the exergetic point of view. Bejan [12] studied theoretically refrigeration systems and investigated based on entropy generation minimization. The study can be used for the thermodynamic optimization of refrigeration plants. Yumrutas et al. [13] studied exergy analysis of vapor compression refrigeration systems. A computational model based on the exergy analysis was presented for the investigation of the effects of the evaporating and condensing temperatures on the pressure losses, the exergy losses, the second law of efficiency, and the coefficient of performance (COP) of a vapor compression refrigeration cycle. Kaygusuz and Ayhan [14] presented experimentally a solar assisted heat pump system, and analyzed the data using exergy idea. Torres-Reyes et al. [15,16] studied a solar assisted heat pump experimentally, and optimized the system using exergy analysis. Bridges et al. [17] presented a second law analysis of domestic refrigerator and air-conditioning systems quantifies the destruction of available energy in each component to contribute to overall system efficiency. Smith and Few [18] presented second law analysis of an experimental domestic scale cogeneration plant incorporating a heat pump. The use exergy analysis in this work significantly contributed to the development of the combined heat and power plant concept. Bilgen and Takahashi [19] had been carried out exergy analysis of heat pump–air conditioner systems. The irreversibilities due to heat transfer and friction had been considered. Based on the exergy analysis, a simulation program had been developed to simulate and evaluate experimental systems. Badescu [20] investigated first and second law analysis of a solar assisted heat pump based heating system.

The studies on exergy analysis of GCHPs are relatively few. Piechowski [1] studied a relatively new approach to optimisation of a ground heat exchanger (GHE), based on the second law of thermodynamics and was adopted to test for an optimum combination of circulating water flow rate and pipe diameter. This method allows for identification and quantification of irreversibility taking place during a GHE operation. Hepbasli and Akdemir [21] described energy and exergy analysis of a GCHP system. The exergy transports between the components and the consumptions in each of the components of the GCHP system were determined for the average measured parameters obtained from the experimental results in February 2001. Ozgener and Hepbasli [22] investigated to the performance characteristics of a solar assisted GCHP greenhouse heating system with a 50 m vertical 1 × 1/4 in. nominal diameter U-bend GHE using exergy analysis method. Exergetic efficiencies of the system components were determined in an attempt to assess their individual performances and the potential for improvements was also presented.

This work examines energy and exergy analysis of a GCHP system with two different horizontal GHE. The influences of the buried depth of the earth coupled heat exchanger on the energetic and exergetic efficiencies are examined. The performance of a GCHP system along with its each effect was evaluated by using exergy analysis, which aims at better identifying process efficiencies and losses. The data used were obtained from the measurements made in a GCHP, which was designed and installed in the Technical Education Faculty of Fırat University, Elazığ, Turkey. Energy-exergy analysis results are discussed in the end section.

2. Description of experimental test apparatus and uncertainty

The schematic of the horizontal GCHP system constructed for space heating is illustrated in Fig. 1, while the view of the HGHE1 and HGHE2 at 1 and 2 m depths is shown in Fig. 2.

2.1. Experimental set-up

Table 1 summarizes the main components specification and characteristics of the GCHP system. The experimental set-up consists of three main components:

1. horizontal GHEs,
2. heat pump unit equipment,
3. auxiliary equipment.

2.1.1. Horizontal GHEs

There have been two GHEs installed at the University of Fırat. Each consists of a high density polyethylene tube, 16 mm diameter. The HGHE1 and HGHE2 are made as a single pass straight tube, buried at the depth of 1 and 2 m. The heat exchangers were been buried in the native ground. To allow for measurement of the circulating water–antifreeze solution and ground temperature a number of T-type thermocouples were installed. The pipe–ground interface temperature is measured in a similar fashion to the water–antifreeze solution-temperature measurement; except that thermal insulation is not used here since the thermocouple should have good contact with both the pipe and the ground.

2.1.2. Auxiliary equipment

As can be seen in Fig. 1, the collector valve allows for varying the circulating water–antifreeze solution flow rate. The flow rate of the circulated water–antifreeze solution through the closed loop GHE was measured by using a rotameter and controlled by a hand-controlled tap mounted on the collector. Anemometer has been used to measure the circulating air flow velocity. The electric power consumed by the system, (compressor, water–antifreeze circulating pump and fan), was measured by means of wattmeter. The inlet and outlet temperatures of the R-22 in the condenser, compressor and evaporator were measured with T-type (copper-constantan) thermocouples. In addition, temperatures of the circulated water–antifreeze solution at inlet and outlet of the GHEs and evaporator (Fig. 1) were measured. The ambient and

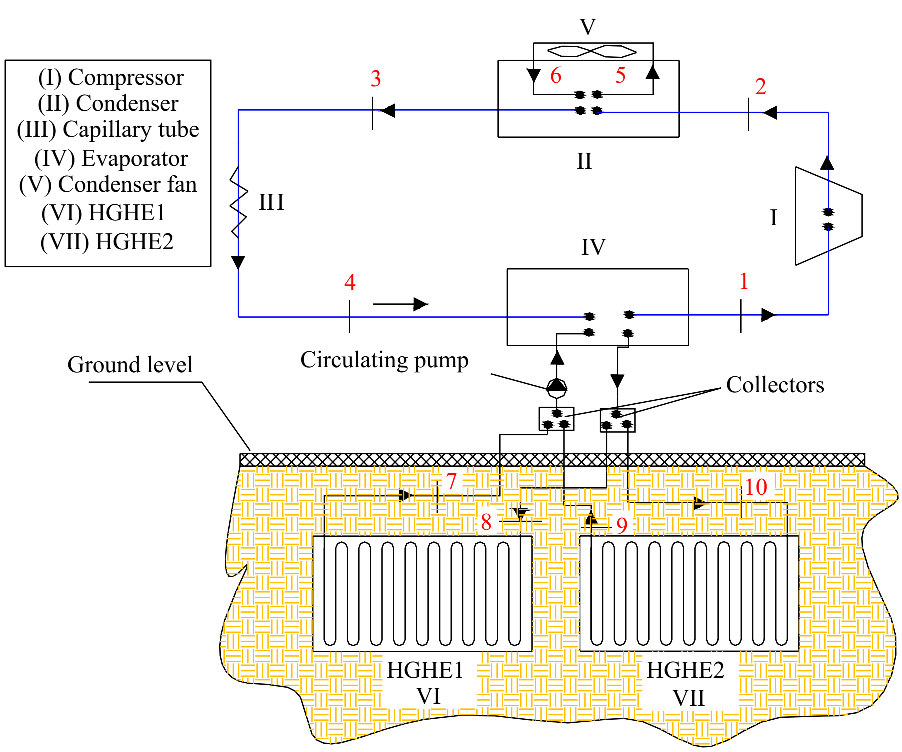

Fig. 1. Schematic diagram of the experimental apparatus.

Fig. 2. The view of the ground heat exchangers buried at 1 and 2 m depths.

indoor air temperatures were measured with thermometers. The inlet and outlet pressures of the compressor and evaporator were measured by using Bourdon tube type manometers.

2.1.3. Heat pump unit equipment

The heat transfer from Earth to the heat pump or from the heat pump to Earth is maintained with the fluid or water–antifreeze solution circulated through the GHEs. The fluid transfers its heat to refrigerant fluid in the evaporator (the water–antifreeze solution to refrigerant heat exchanger). The refrigerant, which flows through other closed loop in the heat pump, evaporates by absorbing heat from the water–antifreeze solution circulated through the evaporator and then enters the hermetic compressor. The refrigerant is compressed by the compressor and then enters the condenser, where it condenses. After the refrigerant leaves the condenser, the capillary tube provides almost 10 °C superheat that essentially gives a safety margin to reduce the risk of liquid droplets entering the compressor. A fan blows across the condenser to move the warmed air of the room. A non-toxic propylene glycol solution, 25% by weight, was circulated through the GHE. In the heating season, the heat exchange fluid (water–antifreeze solution) in the GHE loop collects heat from the earth and transfers that heat to the room. After the heat exchange fluid absorbs heat from the ground, the closed loop GHEs circulates the heat exchange fluid through pipes (see Fig. 1).

The GCHP system connected to a test room with 16.24 m^2 floor area in Fırat University, Elazığ (38.41° N, 39.14° E), Turkey, was designed and constructed. The heating and cooling loads of the test room were 2.5 and 3.1 kW at design conditions, respectively. The compressor and other part of the experimental system were selected the according to the heating and cooling load of test room from the manufacturer's catalog data.

Table 1
Technical features of the experimental set-up

Location: Elazığ, Turkey (lat. 38.41°N; long. 39.14°E)	
Weather information (*yearly average values*):	
Average outdoor temperature	286 (K)
Maximum outdoor temperature	291 (K)
Minimum outdoor temperature	280 (K)
Average relative humidity	56 (%)
Average solar radiation	14.9 ($MJ/m^2/d$)
Average wind velocity	2.5 (ms^{-1})
Average ground temperature at 1 m depth	289 (K)
Room information:	
Window area	2.24 m^2
Wall area	34.63 m^2
Floor area	16.24 m^2
Ceiling area	16.24 m^2
Comfort temperature	293 K
Dimensions of the building	55.21 m^3
Heat pump information:	
Capacity	4.279 kW
Compressor type	Hermetic
Evaporator type	TT3; copper and inner cooling aluminium
Condenser type	HS 10; friterm
Compressor power input	2 HP; 1.4 kW
Compressor volumetric flow rate	7.6 m^3/h
Compressor rotation speed	2900 tr/mn
Condenser fan	2350 m^3/h, 145 W
Evaporating temperature	0 °C
Condensing temperature	54.5 °C
Refrigerant type	R-22
Ground heat exchanger information:	
Configuration type	Horizontal
Pipe material	Polyethylene, PX-b cross link
Length of pipe	50 m × 2
Pipe diameter	0.016 m
Piping depth	1 and 2 m
Pipe distance	0.3 m
Circulating pump information:	
Type	Alarko, NPVO-26-P
Power	40, 62, 83 W

2.2. Uncertainty analysis

An important issue is the accuracy of the measured data as well as the results obtained by experimental studies. Uncertainty is a measure of the "goodness" of a result. Without such a measure, it is impossible to judge the fitness of the value as a basis for making decisions relating to health, safety, commerce or scientific excellence.

The result R is a given function in terms of the independent variables. Let w_R be the uncertainty in the result and w_1, w_2, …, w_n be the uncertainties in the independent variables. The result R is a given function of the independent variables x_1, x_2, x_3,…,x_n. If the uncertainties in the independent variables are all given with same odds, then uncertainty in the result having these odds is calculated by [23]

$$w_R = \left[\left(\frac{\partial R}{\partial x_1} w_1 \right)^2 + \left(\frac{\partial R}{\partial x_2} w_2 \right)^2 + \cdots + \left(\frac{\partial R}{\partial x_n} w_n \right)^2 \right]^{1/2}. \quad (1)$$

The total uncertainties of the measurements are estimated to be $\pm 2.89\%$ for the water–antifreeze temperatures and refrigerant temperatures, $\pm 2.75\%$ for pressures, $\pm 4.35\%$ for power inputs to the compressor, condenser fan and circulating pump, and $\pm 3.00\%$ for electric currents. Uncertainty in reading values of the table is assumed to be $\pm 0.20\%$.

3. Thermodynamic analysis

This article focuses on the combination of the two laws of thermodynamic, which is described in the concept of exergy analysis.

The assumptions made in the analysis presented in this study are:

(i) steady state, steady flow operation,
(ii) negligible potential and kinetic energy effects and no chemical or nuclear reactions,
(iii) adiabatic compressor and capillary tube,
(iv) adiabatic compression efficiency is equal to 0.80,
(v) heat transfer and refrigerant pressure drops in the tubing connecting the components are ignored, since their lengths are short,
(vi) compressor mechanical and the compressor motor electrical efficiencies are 0.70 and 0.72, respectively. These values are estimated by using the actual data obtained from the power input to the compressor,
(vii) air is an ideal gas with a constant specific heat, and its humidity content is ignored,
(viii) the directions of heat transfer to the system and work transfer from the system are positive.

3.1. Energy and exergy balances

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

$$\sum \dot{m}_{\mathrm{i}} = \sum \dot{m}_{\mathrm{o}}, \quad (2)$$

where $\dot{m}$ is the mass flow rate, and the subscript "i" stands for inlet and "o" for outlet.

The first law of thermodynamics can be expressed as

$$\underset{\substack{\text{Rate of net energy}\\ \text{transfer in by heat,}\\ \text{work,and mass}}}{\dot{E}_{\mathrm{i}}} = \underset{\substack{\text{Rate of net energy}\\ \text{transfer out by heat,}\\ \text{work, and mass}}}{\dot{E}_{\mathrm{o}}}. \quad (3)$$

The common exergy balance can be expressed in the rate form as

$$\underset{\substack{\text{Rate of net exergy}\\ \text{transfer by}\\ \text{heat, work, and mass}}}{\dot{E}x_{\mathrm{i}} - \dot{E}x_{\mathrm{o}}} = \underset{\text{Rate of exergy destruction}}{\dot{E}x_{\mathrm{d}}}, \quad (4)$$

or

$$\dot{E}x_{\mathrm{heat}} - \dot{E}x_{\mathrm{work}} + \dot{E}x_{\mathrm{mass,i}} - \dot{E}x_{\mathrm{mass,o}} = \dot{E}x_{\mathrm{d}}. \quad (5)$$

Using Eq. (5), the rate of information of the general exergy balance can also be written as

$$\sum \left(1 - \frac{T_{\mathrm{o}}}{T_{\mathrm{r}}}\right) \dot{Q}_{\mathrm{r}} - \dot{W} + \sum \dot{m}_{\mathrm{i}} \psi_{\mathrm{i}} - \sum \dot{m}_{\mathrm{o}} \psi_{\mathrm{o}} = \dot{E}x_{\mathrm{d}}, \quad (6)$$

where $\psi = \varepsilon x^{\mathrm{PH}} = (h - h_{\mathrm{o}}) - T_{\mathrm{o}}(s - s_{\mathrm{o}})$ is flow exergy and the amount of thermal exergy transfer associated with heat transfer $\dot{Q}_r$ across a system boundary r at constant temperature T_{r} is

$$\sum \left(1 - \frac{T_{\mathrm{o}}}{T_{\mathrm{r}}}\right) \dot{Q}_{\mathrm{r}}. \quad (7)$$

The exergy associated with work is $\dot{W}$. The amount of exergy consumed due to irreversibilities during a process is

$$\dot{E}x_{\mathrm{d}} = \dot{I} = T_{\mathrm{o}} S_{\mathrm{gen}}. \quad (8)$$

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 T_{o} and pressure P_{o} of the environment are often taken as standard-state values, such as 1 atm and 25 °C. If the system uses atmospheric air, T_{o} might be specified as the average air temperature. If both air and water from the natural surroundings were used, T_{o} would be specified as the lower of the average temperatures for air and water [24].

3.3. Energy and exergy efficiencies

Exergy or availability, a gauge of how effectively the input is converted to the product is the ratio (product/input) [25], that is

$$\mathrm{COP} = (\text{Energy in products/Total energy input}), \quad (9)$$

$$\varepsilon = (\text{Exergy in products/Total exergy input}), \quad (10)$$

In this study, the exergy efficiency is calculated for the GCHP system on a product/fuel basis can be defined as

$$\varepsilon_{\mathrm{overall}} = \frac{\dot{P}_{\mathrm{tot,overall}}}{\dot{F}_{\mathrm{tot,overall}}}. \quad (11)$$

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:

$$\dot{m}_1 = \dot{m}_2 = \dot{m}_{ref}, \tag{12a}$$

$$\dot{W}_{comp} = \dot{m}_{ref}(h_{2act} - h_1), \tag{12b}$$

$$\dot{E}x_{d,comp,e} = T_0\dot{m}_{ref}(s_1 - s_{2act}), \tag{12c}$$

$$\dot{E}x_{d,comp} = \dot{m}_{ref}(\psi_1 - \psi_{2act}) + \dot{W}_{comp}, \tag{12d}$$

where the heat transfer versus the environment was neglected.

For condenser:

$$\dot{m}_1 = \dot{m}_2 = \dot{m}_{ref}; \quad \dot{m}_5 = \dot{m}_6 = \dot{m}_{air}, \tag{13a}$$

$$\dot{Q}_{cond} = \dot{m}_{ref}(h_{2act} - h_3);$$
$$\dot{Q}_{cond} = \dot{Q}_{cfan} = \dot{m}_{air}C_{p,air}(T_5 - T_6), \tag{13b}$$

$$\dot{E}x_{d,cond,e} = T_0[\dot{m}_{ref}(s_3 - s_2) + \dot{m}_{air}(s_5 - s_6)], \tag{13c}$$

$$\dot{E}x_{d,cond} = \dot{m}_{ref}(\psi_{2act} - \psi_3) + \dot{m}_{air}(\psi_6 - \psi_5) = \dot{I}_{cond}. \tag{13d}$$

For capillary tube:

$$\dot{m}_3 = \dot{m}_4 = \dot{m}_{ref}, \tag{14a}$$

$$h_3 = h_4, \tag{14b}$$

$$\dot{E}x_{d,ct,e} = T_0\dot{m}_{ref}(s_4 - s_3), \tag{14c}$$

$$\dot{E}x_{d,ct} = \dot{m}_{ref}(\psi_3 - \psi_4). \tag{14d}$$

For evaporator:

$$\dot{m}_4 = \dot{m}_1 = \dot{m}_{ref}, \tag{15a}$$

$$\dot{Q}_{eva} = \dot{m}_{ref}(h_1 - h_4); \quad \dot{Q}_{eva} = \dot{Q}_{ghe}, \tag{15b}$$

$$\dot{E}x_{d,eva,e} = T_0[\dot{m}_{ref}(s_1 - s_4) + \dot{m}_{wa}(s_7 - s_8)] \text{ with a HGHE1}, \tag{15c}$$

$$\dot{E}x_{d,eva,e} = T_0[\dot{m}_{ref}(s_1 - s_4) + \dot{m}_{wa}(s_9 - s_{10})] \text{ with a HGHE2}, \tag{15d}$$

$$\dot{E}x_{d,eva} = \dot{m}_{ref}(\psi_4 - \psi_1) + \dot{m}_{wa}(\psi_8 - \psi_7) \text{ with a HGHE1}, \tag{15e}$$

$$\dot{E}x_{d,eva} = \dot{m}_{ref}(\psi_4 - \psi_1) + \dot{m}_{wa}(\psi_{10} - \psi_9) \text{ with a HGHE2}. \tag{15f}$$

For condenser fan:

$$\dot{m}_{air,i} = \dot{m}_{air,o} = \dot{m}_{air}, \tag{16a}$$

$$\dot{Q}_{cfan} = \dot{m}_{air}C_{p,air}(T_{o,air} - T_{i,air});$$
$$\dot{Q}_{cfan} = \dot{Q}_{cond}; \dot{Q}_{sph} = \dot{Q}_{cond}, \tag{16b}$$

$$\dot{E}x_{d,cfan,e} = T_0\left[\dot{m}_{air}(s_6 - s_5) + \frac{\dot{Q}_{cfan}}{T_{i,air}}\right], \tag{16c}$$

$$\dot{E}x_{d,cfan} = \dot{m}_{air}(\psi_5 - \psi_6) - \dot{Q}_{cfan}\left(1 - \frac{T_0}{T_{i,air}}\right). \tag{16d}$$

For ground heat exchangers (HGHE1 and HGHE2):

HGHE1:

$$\dot{m}_7 = \dot{m}_8 = \dot{m}_{wa}, \tag{17a}$$

$$\dot{Q}_{ghe} = m_{wa}C_{p,wa}(T_8 - T_7), \tag{17b}$$

$$\dot{E}x_{d,ghe,e} = T_0\left[\dot{m}_{wa}(s_8 - s_7) - \frac{\dot{Q}_{ghe}}{T_{ground}}\right], \tag{17c}$$

$$\dot{E}x_{d,ghe} = \dot{m}_{wa}(\psi_7 - \psi_8) + \dot{Q}_{ghe}\left(1 - \frac{T_0}{T_{ground}}\right). \tag{17d}$$

HGHE2:

$$\dot{m}_9 = \dot{m}_{10} = \dot{m}_{wa}, \tag{18a}$$

$$\dot{Q}_{ghe} = m_{wa}C_{p,wa}(T_{10} - T_9), \tag{18b}$$

$$\dot{E}x_{d,ghe,e} = T_0\left[\dot{m}_{wa}(s_{10} - s_9) - \frac{\dot{Q}_{ghe}}{T_{ground}}\right], \tag{18c}$$

$$\dot{E}x_{d,ghe} = \dot{m}_{wa}(\psi_9 - \psi_{10}) + \dot{Q}_{ghe}\left(1 - \frac{T_0}{T_{ground}}\right). \tag{18d}$$

4. Results and discussions

Annual ground temperatures were needed as boundary conditions for determining the year round performance of GCHP system. These temperatures, given in Fig. 3, were obtained from an experimental measurement.

The HGHE1 and HGHE2 with couples GCHP systems were tested under very similar heating loads of the test room, and then compared (25–26 January 2004). Temperature, pressure, and mass flow rate data for R-22, water–antifreeze solution, and air are given in Table 2

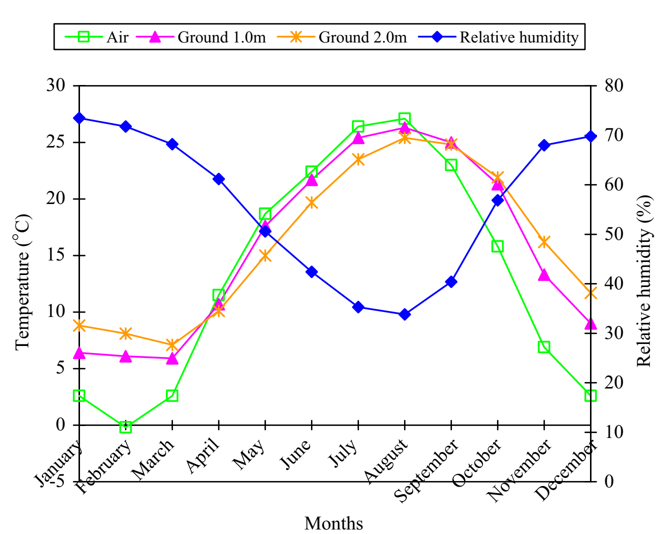

Fig. 3. Monthly ambient air and ground properties.

Table 2
Exergy analysis results of the GCHP system

Number HGHE1	Number HGHE2	Name of element	Fluid	Phase	Temperature, T (°C) HGHE1/HGHE2	Pressure, P (Bar) HGHE1/HGHE2	Specific enthalpy, h (kJ/kg) HGHE1/HGHE2	Specific entropy, s (kJ/kg K) HGHE1/HGHE2	Mass flow rate, $\dot{m}$ (kg/s) HGHE1/HGHE2	Specific exergy, ψ (kJ/kg) HGHE1/HGHE2	Exergy rate, $\dot{E}x = \dot{m}\psi$ (kW) HGHE1/HGHE2
0	0	—	R-22	Dead state	1.0	1.013	414.731	1.931	—	0	0
0	0	—	Air	Dead state	1.0	1.013	11.22	0.043	—	0	0
0	0	—	Water	Dead state	1.0	1.013	4.19	0.015	—	0	0
1	1	Evaporator outlet/compressor inlet	R-22	Super heated vapor	−8.79	3.7	401.62	1.7637	0.016	32.75	0.52
					−7.73	3.84	402.03	1.7620	0.016	33.35	0.53
2	2	Condenser inlet/compressor outlet	R-22	Super heated vapor	75.88	15.19	446.34	1.8097	0.016	64.86	1.03
					75.56	15.35	445.85	1.7877	0.016	70.13	1.12
3	3	Condenser outlet	R-22	Liquid	39.8	15.19	248.75	1.1637	0.016	44.36	0.70
					40.21	15.35	249.28	1.1653	0.016	44.19	0.70
4	4	Evaporator inlet	R-22	Mixture	−11.9	3..32	248.75	1.1883	0.016	37.63	0.60
					−10	3.55	249.28	1.1885	0.016	37.83	0.61
5	5	Condenser fan inlet	Air	Gas	29.85	1.013	99.12	0.3451	0.105	5.08	0.53
					30.07	1.013	103.006	0.3581	0.105	5.41	0.56
6	6	Condenser fan outlet	Air	Gas	23.54	1.013	69.66	0.2459	0.105	2.82	0.29
					23.10	1.013	68.08	0.24	0.105	2.86	0.30
7	9	Ground heat exchanger water–antifreeze pump outlet	Wat–antf	Liquid	7.33	3.5	31.22	0.1108	0.205	0.77	0.16
					10.19	3.5	43.49	0.154	0.205	1.19	0.24
8	10	Ground heat exchanger water–antifreeze pump inlet	Wat–antf	Liquid	4.9	2.5	20.8	0.074	0.205	0.44	0.09
					7.3	2.5	31.09	0.1103	0.205	0.78	0.16

according to their state numbers specified in Fig. 1. In this study, T_o and P_o were taken as 1 °C and 101.325 kPa, respectively, which were the values measured at the time when the GCHP system data were obtained. The thermodynamic properties of air, water and R-22 are obtained from Refs. [26,27].

4.1. Energetic evaluation

In present study, the results obtained from the experiments on 25–26 January 2004 were evaluated to determine the performance of the GCHP system. The $COP_{overall}$ were calculated from Eq. (9), and found to be on average 2.5 and 2.8, respectively, for the HGHE1 and HGHE2 with couples GCHP systems. The performance evaluation of the heating and cooling modes of operation of the system was given in other papers [28,29]. According to De Swardt and Meyer [30], the theoretical annual heating $COP_{overall}$ for a GCHP system is about 3.6 in three different depths (0.9, 1.5 and 3.0 m). In spite of the similar tendency of $COP_{overall}$, this value is higher than the results of our experiment. Because of the properties of the climate, ground etc. of the place where the evaluation is conducted, obtaining experimental values that are lower than the theoretical results is an expected result.

4.2. Exergetic evaluation

The exergy rate results are tabulated in Tables 2 and 3. These tables form is compatible with in the literature tables [21,22]. The values of HGHE1 and HGHE2 with couples GCHP systems are shown as HGHE1/HGHE2 in Tables 2 and 3. Table 3 indicates the values for exergy destruction rate, utilized power, exergy of the product of components (P), exergy of the fuel component (F) and exergy efficiency. These values are evaluated in terms of the GCHP unit (I–IV) and whole system (I–IV and I–VI + VII). Using Eq. (15), the exergy efficiency values for the HGHE1 and HGHE2 with couples GCHP system on a product/fuel basis are obtained to be 53.1 and 56.3%, respectively. These values are also shows in Table 2. It is obvious from the table that the irreversibility (exergy loss) of HGHE2 is less than of the HGHE1 as about 2.0%. This is due to the ground temperatures which 1 m is less than 2 m in heating season. The highest irreversibility on a system basis occurs in the condenser fan unit, followed by the evaporator, compressor, condenser and capillary tube for the GCHP unit and system, respectively. Experimental and exergetic results show that the mentioned system is very reliable, efficient, sensible and suitable for the purpose of the different studies [21,22].

Some energy and exergy values are dependent on the intensive properties of the dead state. Consequently the results of energy and exergy analyses generally are sensitive to variations in these properties. Before energy and exergy analyses can be applied with confidence to engineering systems, the significance of the sensitivities of energy and exergy analysis result to reasonable variations in dead-state properties must be considered [31].

In this paper we deal with the effects of varying reference environment temperature on the exergy efficiencies of HGHE1 and HGHE2 with couples GCHP systems. This

Table 3
Exergetic analysis data of the GCHP unit and system

Item number		Component	Exergy destruction (kW)	Utilized power (kW)	Exergy of the product of a component P (kW)	Exergy of the fuel component F (kW)	Exergy efficiency (%)	Energy (first law) efficiency (%) or COP
HGHE1	HGHE2		HGHE1/HGHE2	HGHE1/HGHE2	HGHE1/HGHE2	HGHE1/HGHE2	HGHE1/HGHE2	HGHE1/HGHE2
I	I	Compressor	0.201	1.19	0.513	0.714	71.8	70
			0.112	1.174	0.588	0.7	84	70
II	II	Condenser	0.190	3.64	0.328	0.518	63.3	—
			0.185	3.69	0.415	0.6	69	—
III	III	Capillary tube	0.107	—	0.602	1.151	52.3	—
			0.101	—	0.605	0.706	85.6	—
IV	IV	Evaporator	0.467	2.44	0.067	0.534	12.5	—
			0.495	2.46	0.084	0.579	14.5	—
V	V	Condenser fan	0.74	3.64	0.236	0.976	24.2	65–80
			0.693	3.612	0.267	0.96	27.8	65–80
VI	VII	Ground heat exchangers	0.023	2.44	0.11	0.133	82.71	—
			0.027	2.46	0.121	0.148	81.76	—
I–IV	I–IV	GCHP unit	0.889	—	1.51	2.399	62.95	
			0.893	—	1.692	2.585	65.45	
I–VI	I–VI + VII	GCHP system	1.641	—	1.856	3.497	53.1	2.5
			1.613	—	2.08	3.693	56.3	2.8

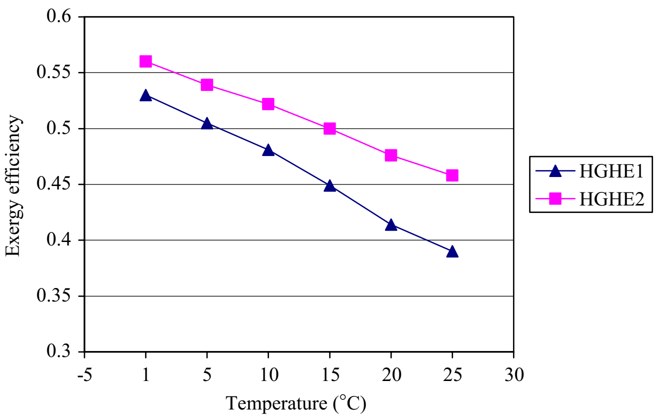

Fig. 4. Exergy efficiency versus reference environment temperature.

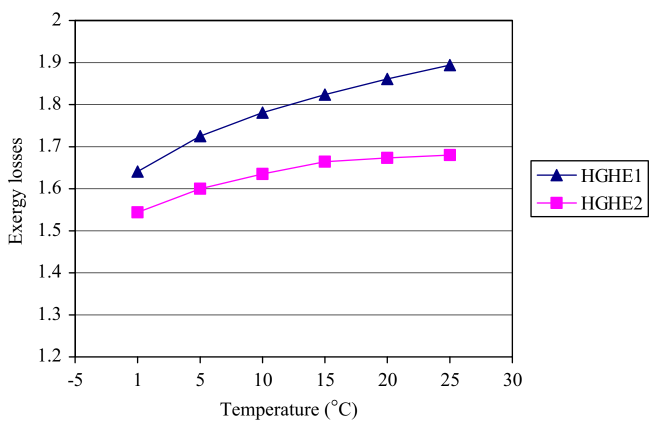

Fig. 5. Exergy losses versus reference environment temperature.

calculation in Fig. 4 shows exergy efficiency versus reference environment temperature for both HGHE1 and HGHE2. At 1 °C, exergy efficiency is 53.1% for HGHE1 and 56.3% for HGHE2. It is noticed that as the reference environment temperature increases, the exergy efficiencies decreases significantly for both the heat exchangers.

Fig. 5 illustrates the exergy losses (irreversibility, kW) between HGHE1 and HGHE2 with respect to the reference environment temperature changes. We notice from this figure that the losses are almost linear, and the reference more-or-less remains constant with respect to the reference environment temperatures. The variations of values in Figs. 4 and 5 are compatible with Ref. [32].

5. Conclusions

In this paper the exergy analysis and experimental study of a GCHP system for heating application is performed and exergy loss of each component is calculated. The first and second law efficiencies of the system working under varying operating conditions (HGHE1 and HGHE2) are investigated and compared. The results show that, as expected, the heating $COP_{overall}$ of the system increase slightly when increasing the heat source temperature (ground temperature). Also, the exergetic efficiency of the system increases when increasing the heat source temperature for heating season.

The results indicate that, for a same set of operating conditions (heat pump unit), changes in source temperature can lead to significant variations in exergy efficiency. This is because exergy recognizes that the energy content of a substance or flow of matter is usable only down to the environmental conditions.

Also, the results show that increasing reference environment temperature decreases the exergy efficiency in both HGHE1 and HGHE2 systems and also increases the difference between the exergy efficiencies of both systems. Thus, the exergy losses increase in the GCHP systems.

Finally, a system exergy-economics optimization study may be carried out to recognize each component's relative importance with respect to operational conditions.

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