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EXERGETIC ANALYSIS OF A VERTICAL GROUND-SOURCE HEAT PUMP SYSTEM WITH WALL HEATING/COOLING

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ABSTRACT
The present study deals with an exergetic analysis and assesment of a Vertical Ground Source Heat Pump System (VGSHP) combined with a Wall Heating System (WHS) in a building. This study is an experimental investigation of a real building's heating system. System is located at Yildiz Renewable Energy House (YREH) in Yildiz Technical University and fulfills the heating demand of YREH and a living room of the neighbor dormitory. In order to validate an exergetic model, system is divided into three subsystems such as: (i) the ground coupling circuit, (ii) the refrigerant circuit and (iii) the WHS circuit. The schematic diagram of the constructed experimental system is given in Fig 1. Exergetic model is obtained by applying mass, energy and exergy equations for each system component. YREH has 4 rooms each one's floor area is 8 m^2 and neighbor dormitory has a 50 m^2 living room. In this study 3 rooms of YREH and living room have been heated in heating season. Heating season was assumed to be between 1 January and 31 March. As average results on heating season 6.509 kW heat energy was extracted from ground and 5.799 kW was used in the WHS. In this process electrical energy consumption of system components are; compressor 1.711 kW, ground heat exchanger pump 0.092 kW, accumulator tank circulation pump 0.114 kW and WHS circulation pump 0.108 kW. For heating season, calculated overall system efficiency was 67.36% while GSHP unit's efficiency was 85%. In addition, overall system COP was 2.76 while GSHP unit's COP 4.13. Total exergy destruction was found 1.759 kW and largest exergy destruction has occurred in the compressor as 0.714 kW. The exergy efficiency values for the individual components of the system have been found ranging from 58.3% to 98.4% according to P/F concept. It is expected that the model would be beneficial for evaluating low exergy heating systems which use ground source as renewable energy.

INTRODUCTION
Ground-source heat pump (GSHP) systems make use of renewable energy stored in the ground for building heating and cooling. They are suitable for a wide variety of building types, provide high levels of comfort, and are particularly appropriate for energy saving and environmentally attractive (Sanner et al., 2003; Lund, 2003; Esen et al., 2006; Ozgener et al., 2007a; Kavanaugh et al., 1992; Agustos et al., 2008; Akbulut et al., 2012).
Many theoretical and experimental works on GSHP systems have been accomplished, since late 1940s. GSHP systems are inherently more effective than air source heat pumps because the ground maintains relatively stable source and sink temperatures. This situation provides a better COP for the GSHP system. For the ground heat exchangers, VGSHP systems are usually preferred over HGSHP systems. They are more efficient and less ground area is required (Grandall, 1946; Ingersol et al., 1948; Hepbasli et al., 2002 and 2007; Li et al., 2006; Esen et al., 2007; Lund et al., 2005; Urchueguia et al., 2008).

WHC systems have low operation temperature benefit. This creates substantial energy savings for heating and cooling compared with conventional systems. The energy savings from a WHC system may reach more than 30%, as demonstrated in some theoretical and experimental case studies. Main advantages of WHC systems are enabling better thermal comfort, providing better indoor air quality with low air velocities and homogenous heat distribution, being available to use waste heat and low-enthalpy renewable energy resources, and having low initial investment, maintenance, and operating costs (ASHRAE, 2004). Researchers have mostly determined the convection heat transfer coefficient of heated and cooled walls. They have solved the natural convection problems in enclosures. In these studies, heat distribution from floors, walls, and ceilings were investigated for heating and cooling situations. All studies considered in the literature about WHC systems emphasized that this system was comfortable, economical, and very suitable to use with renewable energy systems. Because of this reason, a real VGHP system, combined with a WHC system, has been analyzed experimentally in our previous research. Energetic aspects of the system were introduced. However, exergy analysis of the system hasn't been conducted yet (Yoru et al., 2010; Kincay et al., 2010; Akbulut et al., 2011; Başkal et al., 2012).

Exergy analysis is widely used for to scale a process's thermodynamic ideality. This helps designing efficient and cost effective systems that also meet environmental conditions. In addition to the energy analysis, exergy analysis must be used to identify the components where inefficiencies occur. Improvements should be done to these components to minimize the irreversibility and optimize the system (Moran, 1989; Kotas, 1995; Bejan et al., 1996; Rosen, 1999; Dincer et al., 2001). Recent years, exergy analyses of space heating/cooling in buildings and GSHP systems have been developed (Bejan et al., 2002; Yildiz et al., 2009; Kincay et al., 2004; Hepbasli et al., 2004; Utlu et al., 2007; Ozgener et al., 2007b; Bi et al., 2009; Madani et al., 2010). These studies focused on the building heating mode, and few on the building cooling mode. This paper presents an exergy analysis of a real VGSHP system, combined with a WHC system. Real-time data has been obtained and used to represent destroyed exergy and exergy efficiency for both heating and cooling modes.

SYSTEM DESCRIPTION

The investigated VGSHP system with the WHC is shown in Fig. 1. This system consists of a ground heat exchanger loop, heat pump unit, accumulation tank, and a WHC loop in the building. The vertical ground heat exchanger consists of two boreholes, each containing a U-tube pipe. The depth of the boreholes is 65 m, and the diameter is 0.15 m. For generating the ground-loop heat exchanger, 240 m DN40 polyethylene composite pipe was used in a vertical direction and 45 m in horizontal. There are three circulation pumps and an isolated 500 lt accumulation tank in the system. In the WHC system, 840 m DN16 and 40m DN20 polyethylene pipe was used. Systems were set up for holding the test room temperature at 20°C on the heating and 24°C on the cooling mode. For this reason, 31°C and 18°C V-GSHP operating temperatures were chosen especially to maintain comfort and prevent condensation on wall-mounted coils. Heating season was analyzed using data collected between Jan. 1 and March 31, 2010. In this period, the GSHP unit was online a total of 863 hours for a mean of 30%. Furthermore, cooling season was analyzed using data collected between July 1 and September 30. In this cooling period the GSHP unit was online a total 120 hours for a mean of 5%.

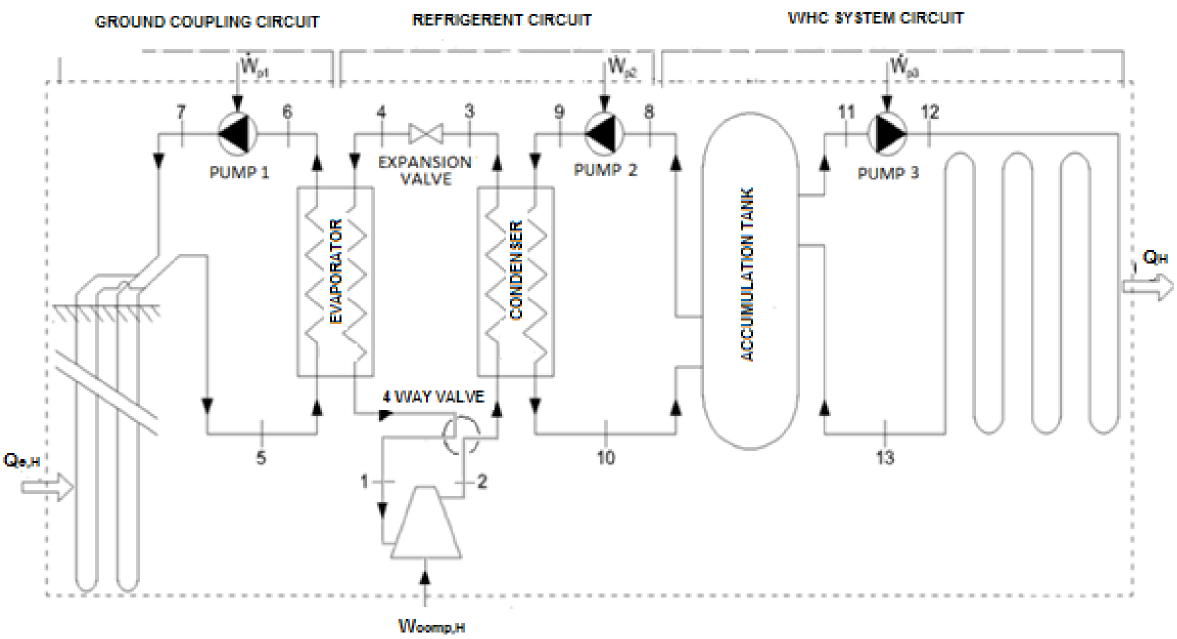

Fig.1. Schematic of investigated VGSHP and WHCS on heating mode.

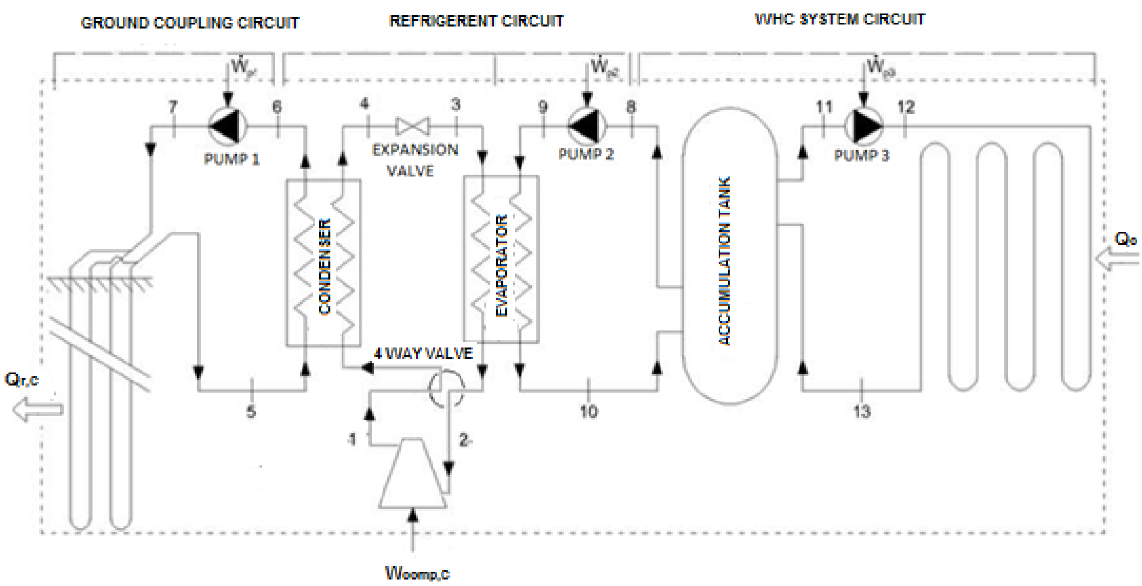

Fig.2. Schematic of investigated VGSHP and WHC system on cooling mode.

Schematic diagrams of the constructed experimental system on heating and cooling modes are illustrated in Fig. 1 and Fig. 2. This system mainly consists of three separate circuits as follows: (i) the ground coupling circuit, (ii) the refrigerant circuit, and (iii) the WHC system circuit. In the heating session, three rooms of YREH and the living room were heated; only two rooms of YREH were cooled in the cooling session. Conversion from the heating cycle to the cooling cycle is implemented by means of a four-way valve. The working fluid is R-410A. This system is installed at Yildiz Renewable Energy House (YREH) at Yildiz Technical University (latitude 41° N, longitude 29° E), Istanbul Turkey.

EXERGETIC MODELING

The general exergy balance can be expressed below Eq. (1), as the total exergy input is equal to total exergy output:

$$\sum(1-\frac{T_0}{T_K})\dot{Q}_k - \dot{W} + \sum \dot{m}_{in}\,\psi_{in} - \sum \dot{m}_{out}\,\psi_{out} = \dot{E}x_{dest} \tag{1}$$

where $\dot{Q}_k$ is the heat transfer rate crossing the boundary at temperature T_K at location k, $\dot{W}$ the work rate, ψ the flow exergy which can be calculated by Eq. (2).

$$\psi = h - h_0 - T_0(s - s_0) \tag{2}$$

Here h denotes the enthalpy, s the entropy, and the subscript zero indicates properties at the restricted dead state of P_0 and T_0.

The exergy rate is calculated by Eq. (3).

$$\dot{E}x = \dot{m}\,\psi \tag{3}$$

To obtain exergy destruction, the entropy generation $\dot{S}_{gen}$ is calculated first and used in the Eq. (4) which is called as Stadola Law.

$$\dot{E}x_{dest} = T_0\,\dot{S}_{gen} \tag{4}$$

The exergy destructions in the system components are calculated as follows, respectively:

- Compressor and pumps

$$\dot{E}x_{dest,c/p} = \dot{W}_{c/p} - (\dot{E}x_{out} - \dot{E}x_{in}) \tag{5}$$

- Heat exchangers (evaporator, condenser and accumulator tank)

$$\dot{E}x_{dest,HE} = \sum \dot{E}x_{in} - \sum \dot{E}x_{out} \tag{6}$$

- Expansion valve

$$\dot{E}x_{dest,expv} = \dot{m}_{ref}\,(\psi_{in} - \psi_{out}) \tag{7}$$

- Ground heat exchanger in heating session

$$\dot{E}x_{dest,grH} = \sum(1-\frac{T_0}{T_{gr}})\dot{Q}_{e,H} + \dot{m}_{in}\,(\psi_{in} - \psi_{out}) \tag{8}$$

- Ground heat exchanger in cooling session

$$\dot{E}x_{dest,grC} = \dot{m}_{in}\,(\psi_{in} - \psi_{out}) - \sum(1-\frac{T_0}{T_{gr}})\dot{Q}_{r,C} \tag{9}$$

- WHCS in heating session

$$\dot{E}x_{dest,whcs} = \dot{m}_{in}\,(\psi_{in} - \psi_{out}) - \sum(1-\frac{T_0}{T_{room}})\dot{Q}_{H} \tag{10}$$

- WHCS in cooling session

$$\dot{E}x_{dest,whcs} = \dot{m}_{in}\,(\psi_{in} - \psi_{out}) + \sum(1 - \frac{T_0}{T_{room}})\dot{Q}_C \quad (11)$$

System components' exergy efficiencies are calculated on product/fuel basis, by using Eq.12

$$\varepsilon_k = \frac{\dot{E}x_{P,k}}{\dot{E}x_{F,k}} \quad (12)$$

RESULTS AND DISCUSSION

During the calculations for the system energy losses from the ground heat exchanger, the expansion valve and isolated pipes were neglected. The dead state temperature, which is the dry-bulb temperature and the dead state pressure were taken 0.01°C and 101.325 kPa, respectively. After modeling the system and analysis of given data in Tables 1 and 2, exergetic results for the vertical ground-source heat- pump system with wall heating and cooling were obtained by using thermodynamic equations given in the exergetic modeling section. Results are shown by Tables 3 and 4. In the year 2010, heating season was assumed to be between Jan. 1 and March 31, and cooling season between July 1 and Sept 30. Data was recorded for each second and converted to a MySQL database. Collected data was arranged and transferred to other software for calculations and analysis. Refrigerant (R410A) properties were taken from Solkane 6.0, and water properties were taken from EES (Limited Academic version) software. Microsoft Excel pages were formed for calculations. After the calculations and analysis, for the heating and cooling seasons, average values of mass flow, temperature, pressure, enthalpy, and exergy rate are given in Tables 1 and 2, respectively. Also, the exergy rate of fuel, exergy rate of product, destroyed exergy, energy efficiency and exergy efficiency are shown in Tables 3 and 4.

Table 1. Measured data and calculated values of system in the heating session.

No	Name of element	$\dot{m}$[kg/s]	T[°C]	P[kPa]	h[kJ/kg]	s[kJ/kgK]	$\dot{E}x$ [kW]
0	Water (Dead state)	-	0.01	101.325	0.1032	4.279×10^{-6}	0
0'	Refrigerant (Dead state)	-	0.01	101.325	439.18	2.0982	0
1	Evaporator outlet/compressor inlet	0.035	7.97	795	429.62	1.84070	2.127
2	Compressor outlet/ condenser inlet	0.035	63.16	2021	465.84	1.86900	3.124
3	Condenser outlet	0.035	28.37	2021	245.49	1.15500	2.238
4	Evaporator inlet	0.035	-0.14	795	245.49	1.16670	2.127
5	Ground heat exchanger water outlet	0.44	7.48	257	31.53	0.11350	0.186
6	Ground heat exchanger water pump inlet	0.44	3.89	250	16.46	0.05945	0.052
7	Ground heat exchanger water inlet	0.44	3.94	300	16.67	0.06021	0.053
8	Water circulating pump inlet	0.57	29.65	250	124.3	0.43170	3.577
9	Water circulating pump outlet	0.57	29.70	300	124.5	0.43240	3.582
10	Heat pump outlet	0.57	32.66	293	136.9	0.47328	4.285
11	Accumulator tank outlet	0.27	30.60	250	128.3	0.44470	1.815
12	Heating water inlet	0.27	30.70	300	128.7	0.44610	1.820
13	Accumulator tank inlet	0.27	24.77	250	103.9	0.36378	1.196

Table 2. Measured data and calculated values of system in the cooling session.

No	Name of element	$\dot{m}$[kg/s]	T[°C]	P[kPa]	h[kJ/kg]	s[kJ/kgK]	$\dot{E}x$ [kW]
0	Water (Dead state)	-	0.01	101.325	0.1032	4.279×10^{-6}	0
0'	Refrigerant (Dead state)	-	0.01	101.325	439.18	2.0982	0
1	Compressor outlet/ condenser inlet	0.035	44.71	1668	450.21	1.81590	2.439
2	Evaporator outlet/compressor inlet	0.035	17.71	1113	431.75	1.83890	2.865
3	Evaporator inlet	0.035	10.87	1113	233.27	1.11490	2.194
4	Condenser outlet	0.035	21.08	1668	233.27	1.11680	2.176
5	Ground heat exchanger water outlet	0.44	20.80	257	87.27	0.30760	1.383
6	Ground heat exchanger water pump inlet	0.44	24.59	250	103.10	0.36110	1.918
7	Ground heat exchanger water inlet	0.44	24.63	300	103.30	0.36180	1.922
8	Water circulating pump inlet	0.57	17.66	250	74.14	0.26265	1.307
9	Water circulating pump outlet	0.57	17.70	300	74.31	0.26325	1.310
10	Heat pump outlet	0.57	14.53	293	61.05	0.21730	0.906
11	Accumulator tank outlet	0.27	17.91	250	75.19	0.26620	0.641
12	Heating water inlet	0.27	18.00	300	75.56	0.26750	0.645
13	Accumulator tank inlet	0.27	24.26	250	101.70	0.35650	1.138

Table 3. Exergetic analysis results for representative unit in the heating session.

No	Name of element	$\dot{E}x_F$ [kW]	$\dot{E}x_P$ [kW]	$\dot{E}x_{dest,k}$ [kW]	η [%]	ε[%]
I	Overall system	2.254	0.496	1.759	67.36	22.0
II	GSHP unit	1.840	0.698	1.142	85.00	37.9
III	Pump1	0.092	0.001	0.091	70.75	1.1
IV	Ground heat exchanger	0.230	0.133	0.097	100.00	57.9
V	Evaporator	0.134	0.001	0.133	97.63	0.5
VI	Compressor	3.838	3.124	0.714	74.10	58.3
VII	Expansion valve	2.238	2.127	0.112	100.00	95.0
VIII	Condenser	0.886	0.703	0.183	91.65	79.3
IX	Pump 2	0.114	0.005	0.109	87.69	4.4
X	Pump 3	0.108	0.005	0.103	83.08	4.4
XI	Accumulator tank	0.708	0.620	0.088	91.73	87.6
XII	WHCS	0.625	0.496	0.129	86.61	79.4

Table 4. Exergetic analysis results for representative unit in the cooling session.

No	Name of element	$\dot{E}x_F$ [kW]	$\dot{E}x_P$ [kW]	$\dot{E}x_{dest,k}$ [kW]	η [%]	ε[%]
I	Overall system	1.892	0.567	1.325	74.85	29.9
II	GSHP unit	1.470	0.535	0.935	80.76	36.4
III	Pump1	0.130	0.004	0.126	67.69	3.0
IV	Ground heat exchanger	0.539	0.436	0.103	100.00	80.9
V	Evaporator	0.404	0.263	0.141	91.91	65.1
VI	Compressor	1.066	0.426	0.640	60.60	40.0
VII	Expansion valve	2.194	2.176	0.018	100.00	99.2
VIII	Condenser	0.671	0.535	0.136	91.73	79.7
IX	Pump 2	0.130	0.003	0.127	74.54	2.7
X	Pump 3	0.130	0.004	0.126	76.85	3.1
XI	Accumulator tank	0.498	0.400	0.098	95.93	80.4
XII	WHCS	0.567	0.494	0.073	90.29	87.2

For the heating season, the exergy efficiency peak values for the expansion valve and accumulator tank were found 95% and 87.6% respectively. During the calculations energy loss from the expansion valve was neglected. Thereby, energy and exergy efficiency values for this component are suitable for these working conditions. Also, energy and exergy efficiency values for the accumulator tank seem to be normal and prove that the tank is well isolated. On the contrary, the exergy efficiency values are lowest for the evaporator and three circulation pumps. We should consider the energy efficiency values in order to decide whether these components are inefficient and should be replaced. These components energy efficiency values are 97.63%, 70.75%, 87.69% and 83.01% respectively. These values are suitable for a plate heat exchanger and small capacity circulation pumps. Thus, evaporator and pump1 have too small exergy efficiency values since their working temperatures which are too close to the dead state. Also, all three circulating pumps are not changing the temperature of water much. Whole exergy is destroyed on the pumping process. However, our overall system's energy and exergy efficiency values are 67.36% and 22.00% respectively and definitely we should develop some system components. Main indicator for this consideration is exergy destruction rate. That's why compressor, WHCS and the ground heat exchanger must be developed respectively. Improvements on the compressor will positively affect the performance of the condenser if the gas temperature of compressor outlet can be reduced.

For the cooling season, the exergy efficiency peak values for the expansion valve, ground heat exchanger and accumulator tank were found 99.2%, 80.9%, and 80.4% respectively. The expansion valve and accumulator tank have similar results in the heating and cooling seasons because of the reasons explained above. But the ground heat exchanger works very different. Much difference on the exergy efficiency values exist. Because the ground heat exchanger water inlet temperatures are too different, 3.94 °C and 24.63 °C respectively. There is a higher heat transfer coefficient due to water molecules activity at 24.63 °C temperature and also 3.94 °C is very close to dead state. However, in both cases the exergy destruction rates are similar, 0.097 kW and 0.103 kW respectively. The three circulating pumps have similar results working in both heating and cooling sessions. Therefore, the compressor, the WHCS and the ground heat exchanger must be developed to make the system more efficient in both sessions.

CONCLUSIONS

We have presented exergetic aspects of a vertical ground-source heat pump system combined with wall heating/cooling in general. We analyzed the overall system components such as compressor, condenser, evaporator, expansion valve, ground heat exchanger, accumulator tank and WHCS. We used energy and exergy analysis results of experimental data from our real system which works for heating and cooling purposes. Combining this study with our previous work which presents energetic aspects of the same system (Akbulut et al., 2012), we can point out some concluding remarks from this study as follows:

(a) In the literature, main indicator to choose system's inefficient components is exergy destruction rate. Nonetheless, exergy efficiency and exergy efficiency values must be considered too attentively if most of the components' exergy destruction rates are very similar like in our cases.
(b) Technical and economical availabilities and environmental aspects must be considered much before developing the system. That's why exergy economic and exergy environmental analysis of this kind of systems are needed.
(c) For the heating season, the exergy efficiency peak values for the expansion valve and accumulator tank are 95% and 87.6% respectively. For the cooling season, the exergy efficiency peak values for the expansion valve, ground heat exchanger and accumulator tank are 99.2% 80.9 and 80.4%.
(d) Calculated exergy efficiencies of the circulation pumps seem low at first glance. Because their operating temperatures are too near to dead state. Pumps are not changing the temperature of water much and all of the exergy is destroyed on pumping process. Their energy efficiency values are within normal limits.
(e) As a result of exergy analysis of this system, the order of development priority is the compressor, the WHCS and the ground heat exchanger, from much to less.
(f) Heat extraction rate of 50 W/m and heat rejection rate of 54 W/m of bore depth for the heating and cooling periods, remain within the range reported in the literature. But ground heat exchanger still has a development capacity.
(g) For heating season, calculated overall system efficiency was 67.36%, while the GSHP unit's efficiency was 85%. Adding to that, the overall system COP was 2.76, while the GSHP unit's COP was 4.13. As for the cooling season, calculated system efficiency was 74.90%, while the GSHP unit's efficiency was 80.76%. Furthermore, the overall system COP was 4.38 while the GSHP unit's COP was 4.78.
(i) The circulator wattage for the three closed loop of the system can be categorized as efficient and excellent. However frequency converter circulation pumps can be used for energy saving.
(h) The system can be improved by using a variable speed compressor. According to the literature, increase in the compressor speed lowers the heat pump COP. Optimum compressor speed is around 50 Hz (Madani et al., 2010).

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NOMENCLATURE

COP	coefficient of performance, dimensionless
$\dot{Ex}$	exergy rate, kW
h	specific enthalpy, kJ/kg
$\dot{m}$	mass flow rate, kg/s
P	pressure, kPa
$\dot{Q}$	heat transfer rate, kW
T	temperature, K
$\dot{W}$	rate of work or power, kW
s	specific entropy, kJ/kgK
$\dot{S}$	entropy rate, kJ/K

Greek Letters

ε	exergy (second law) efficiency, dimensionless
η	energy (first law) efficiency, dimensionless
ψ	specific exergy, kJ/kg

Subscripts

0	reference (dead) state
C	cooling
c/p	compressor or pump
dest	destroyed
expv	expansion valve

F fuel
gr ground
H heating
HE heat exchanger
in input
k k'th element
P product
ref refrigerant
out output
whsc wall heating cooling system

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