International Journal of Energy and Environmental Engineering
ISSN: 2008-9163
Vol.1 / No.1(pp.27- 35) / Fall2010

Numerical simulation of a horizontal ground-coupled Heat pump

*K. Javaherdeh[1], H. Karimi[2]

[1]Faculty of mechanical engineering, Islamic Azad University, Takestan branch, Iran
[2] Member of young researchers club, Islamic Azad University of Takestan, Iran

Received Date: 21Jul; 2010 Accepted: 8 Nov; 2010

ABSTRACT

This study represents effect of four variables in performance of a heat-pump in two cities of Tehran and Rasht. A computer program has been developed, simulating a certain heat pump acting in different environments with different refrigerants in heating season. The analysis is based on a homogeneous model. Effect of different refrigerants, depth of the buried heat exchangers and moisture content of soil has been studied. The first and second law of thermodynamic has been used for evaluating energy and exergy performance of system and its components. The effect of moisture content of soil between two cities of Tehran and Rasht was compared. Results show that there is an optimum depth for horizontal heat exchanger. Also the analysis show that increasing refrigerant mass flow rate decreases energy efficiency where increases exergy efficiency.

KEYWORDS

Ground –coupled heat pump, Horizontal heat exchanger, Energy, Exergy

1. INTRODUCTION

Ground-coupled heat pumps are utilizing more and more in the developed countries. Researches have shown that these systems are very suitable for providing high level of comfort and an acceptable reduction of electrical energy consumption, where they also need very low maintenance and known as clean technologies [1, 4]. In Iran, there is a good opportunity, using and developing such these instruments. A few efforts have also been done for using this technology in Iran. Design and installation of a geothermal heat pump in Iran was done in Tabriz engineering research center. An 18000 BTU/hr air-to-air heat pump was modified and designed to a geothermal heat pump system for the first time in Iran [5].There have been no more reports of studies for using of this facility in Iran. The low prices of natural gas are main reason for paying no attention to energy saving technology in Iran. In this article a simulation program has been used for estimating energy and exergy performance of a typical heat pump in two cities of Iran in different heating season.

The first and second law of thermodynamic was used in this research, where the first one deals with the conservation of energy and the second one with quality of energy and material. The maximum coefficient of performance (COP) was obtained, using the first law of thermodynamic, and this means, maximum heat transfer for minimum power input. Using the second law of thermodynamic tends to obtaining maximum exegetic efficiency, and that means minimum entropy generation within the system. So heating can be maximized for the smallest destruction of available energy (exergy). Exergy loss is caused by irreversibility, and can be calculated by exergy method (second law analysis).Exergy is an important thermodynamic property and measures the useful work which can be produced

* Corresponding author: K.Javaherdeh is with the Faculty of mechanical engineering, Islamic Azad University, Takestan branch, Iran (email:Javaherdeh_k@yahoo.com)

by a substance [4].In energy conversion systems, the exergy analysis can appear as a powerful tool. This concept is extensively discussed by Koats [6], Nakanishi et al. [7] and Wei [8].

1. SYSTEM DESCRIPTION

A. *Heat pump and heat exchanger*

The properties of selected system equipments are shown in table 1. The main components in heat pump system are the compressor, the expansion valve and two heat exchangers, evaporator and condenser. The components are connected to form a closed circuit, as shown in figure 1. In the evaporator the temperature of liquid working fluid is kept lower than the temperature of the ground, causing heat to flow from the ground to the liquid refrigerant, where it evaporates the refrigerant. Vapour refrigerant from the evaporator is compressed to a higher pressure and temperature. The hot vapor then enters the condenser, where it condenses and gives of useful heat. Finally, the high pressure refrigerant is expanded to the evaporator pressure and temperature in the capillary tube. Then it is returned to its original state and once again enters the evaporator. The compressor is driven by an electric motor. The input power of compressor is inserted after subtracting of mechanical and electrical losses. The following assumptions have been made in this research:

(i) There is a safety margin of 10°C superheat to prevent liquid droplets entering the compressor.

(ii) The refrigerant in evaporator is subcooled 5°C.

(iii) Condenser and evaporator have a pressure drop of 5%.

Table 1.properties of system equipments[9]

Heat pump	
Average compressor adiabatic power input	0.9 kW
Compressor adiabaticefficiency	0.8
Refrigerant mass flow rate	0.016 kg/s
Refrigerant type	R134a
Condenser fan mass flow rate	0.136 kg/s
Horizontal ground heat exchanger	
Configuration type	Horizontal
Pipe diameter	0.016 m
Pipe friction coefficient	0.017
Water-antifreeze type	Propylene glycol solution 25%
Water-antifreeze mass flow rate	0.2
Soil type	Silt and clay
Moisture weight percentage	7.9
Soil dry density	1382
Soil dry specific heat	0.84

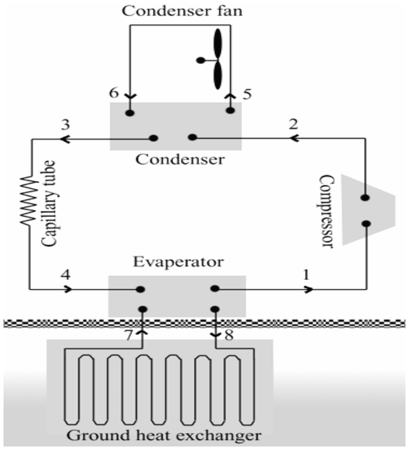

Fig.1: A schematic diagram of the system components.

B. *Environment*

Soil and climate Information was needed for this simulation. Tehran and Rasht have been chosen as samples for respectively dry and wet cities in Iran. Climate data such as average air temperature, air temperature annual amplitude; minimum and maximum relative humidity, day of the year with maximum surface temperature, and heating design day were taken from associated climatology centers [10].

The soil thermal diffusivity, which is the ratio of the thermal conductivity and heat capacity, considers as known input value. The conductivity and volumetric heat capacity increase with water content so the diffusivity is also dependent upon soil water content. For mineral soils, the thermal diffusivity increases with water content at low water contents and then gradually decreases with increasing water contents at high water contents [11-13]. These values can be estimated from the following equations.

The equation of thermal conductivity for silt and clay soils is given by Hepbasli [14]:

$$k_s = 0.14423(0.9\log\lambda - 0.2)10^{0.000624\,\rho_{sd}} \quad (1)$$

And for sand soils:

$$k_s = 0.14423(0.7\log\lambda + 0.4)10^{0.000642\,\rho_{sd}} \quad (2)$$

Thermal diffusivity of soil αs is given by:

$$\alpha_s = \frac{k_s}{C_{psc}\rho_{sc}} \quad (3)$$

Where:

$$C_{psc} = \lfloor \omega C_{PW} + (100-\lambda)C_{psd} \rfloor / 100 \quad (4)$$

$$\rho_{SC} = [\omega\rho_{PW} + (100-\lambda)\rho_{sd}]/100 \quad (5)$$

C_{psd} is specific heat of dry soil which is about 0.4-1.05kJ/kgK.

Diffusivity of soil is used to evaluate earth temperature profile. Soil temperature fluctuates annually and daily affected mainly by variations in air temperature and solar radiation. The annual variation of daily average soil temperature at different depths can be estimated using a sinusoidal function [15]. The daily soil temperatures were calculated as functions of time or depth using this equation [9]:

$$T_\infty(\tau, z) = T_e + T_{e,am} e^{-z\times\sqrt{\frac{\theta}{2a_s}}} \cos(\theta\tau - z\times\sqrt{\frac{\theta}{2a_s}}) \quad (6)$$

1. ANALYSIS

A. Mass, energy and exergy analysis

The exergy theory discusses about availability of energy. To create the model the following assumptions were made:

(a) Steady state and steady flow processes, with negligible potential and kinetic energy effect and without chemical or nuclear reactions.

(b) Positive direction for heat transfer to the system and work transfer from the system.

(c)a 5% pressure drop in condenser and evaporator, but no pressure drop in the tubing connecting the components, because of their short lengths.

(d) Adiabatic compressor process and capillary tube.

(e) Compressor adiabatic efficiency is 80%.

(f) air was considered as an ideal gas which it's specific heat is a function of temperature.

(g) The power input to the fan coil fans and the circulating pumps are negligible compared with that the power input to the compressor.

(h) The refrigerant in dead state has a temperature of T0=274.15 K and a pressure of P0 = 1.013 bar.

Air and steam specific heat were needed to calculate enthalpy and entropy of air. Polynomial equations are used for estimating these values. The saturation tables for evaluating properties of refrigerants were used. To generate the unknown variables the four principle balance equations have been applied. A possible form of mass balance equation is:

$$\sum \dot{m}_i = \sum \dot{m}_o \quad (7)$$

The first law of thermodynamic can be expressed as:

$$E_i^\circ = E_o^\circ \quad (8)$$

Where Ei° is rate of net energy transfer in and Eo° is rate of net energy out by heat, work and mass. The common exergy balance can be expressed in the rate form as:

$$Ex_{heat}^\circ - Ex_{work}^\circ + Ex_{mass,i}^\circ - Ex_{mass,o}^\circ = Ex_{dest}^\circ. \quad (9)$$

Using last (number the equation) equation, the rate of information of the exergy balance can be obtained as:

$$\sum(1 - \frac{T_0}{T_r})Q_r^\circ - W^\circ + \sum m_i^\circ\psi_i - \sum m_o^\circ\psi_o = Ex_{dest}^\circ \quad (10)$$

Where

$$\psi = (h - h_0) - T_0(s - s_0) \quad (11)$$

B. Energy and exergy efficiencies

The common definition of COP is:

$$COP = \frac{(Energy\ \ in\ products)}{(Total\ energy\ input)} \quad (12)$$

In this study, COP can be defined as:

$$COP = \frac{\overset{\circ}{Q}_{condenser,out}}{\overset{\circ}{W}_{condenser,in}} \tag{13}$$

In general, exergy efficiency is defined as the ratio of total exergy output to total exergy input:

$$\varepsilon = \frac{Ex^{\circ}_{output}}{Ex^{\circ}_{input}} \tag{14}$$

This can be expressed as:

$$\varepsilon = \frac{m^{\circ}_{air}(\psi_5 - \psi_6)}{W^{\circ}_{compressor}} \tag{15}$$

C. Exergy analysis in this system

Exergy destruction rate for different components of system is expressed as

Compressor:

$$Ex^{\circ}_{dest} = m^{\circ}_{ref}(\psi_1 - \psi_2) + W^{\circ}_{compressor} \tag{16}$$

Condenser:

$$Ex^{\circ}_{dest} = m^{\circ}_{ref}(\psi_2 - \psi_3) + m^{\circ}_{air}(\psi_6 - \psi_5) \tag{17}$$

Capillary tube:

$$Ex^{\circ}_{dest} = m^{\circ}_{ref}(\psi_3 - \psi_4) \tag{18}$$

Evaporator:

$$Ex^{\circ}_{dest} = m^{\circ}_{ref}(\psi_4 - \psi_1) + m^{\circ}_{water}(\psi_8 - \psi_7) \tag{19}$$

Condenser fan:

$$Ex^{\circ}_{dest} = m^{\circ}_{air}(\psi_5 - \psi_6) - Q^{\circ}_{con.fan}(1 - \frac{T_0}{T_{input\ air}}) \tag{20}$$

Ground-coupled heat exchangers:

$$Ex^{\circ}_{dest} = m^{\circ}_{water}(\psi_7 - \psi_8) + Q^{\circ}_{ghe}(1 - \frac{T_0}{T_{ground}}) \tag{21}$$

2. SIMULATION PROGRAM

According to analysis presented above, a simulation program was written on the Visual Basic platform, which can be extended for further industrial use. This program works with two main functions. The first one calculates thermodynamic cycle of refrigerant and then evaporator and condenser effective heat transfer surface for given energy input and output using Logarithmic Mean Temperature Difference (LMTD) method [16, 17]. As can be seen, it uses an initial cycle and manipulates it to find a cycle that fits desired conditions. Second function, uses first one to calculate energy and exergy results for a constant evaporator and condenser effective heat transfer surface. The flowchart of this process is shown in figure 3. Using these functions, the activity of a unique heat pump can be compared in different environment and with different refrigerants. Certainly these results are not suitable for a dynamic and continuous working condition, and so it can be used only by designer to evaluate local condition effects.

3. RESULTS

Simulation results with R-134a in Tehran and Rasht are shown in table2. Exergy rate of different points of cycle is calculated and used to obtain coefficients of performance which are given in table 2. Non-dimensional exergy destruction of system components is shown in fig. 2. As can be seen highest lost work occurs in condenser fan unit and latter in evaporator and compressor. There is not too much different between two cities Tehran and Rasht results, and that means that this lost work in intrinsic of refrigerant cycle and components. By 1st January 1996, all the CFCs phased out in developed countries, and by 1st January 2020, all HCFCs should be phased out [19]. These days, most industrial heat pumps installed in Iran work with R-22, and also using of R-134a is increasing, but R-22 has a High ODP (ozone depletion potential) and GWP (global warming potential). R-407C is one of refrigerant available for immediate use as an alternative for R-22 plants. Thermal properties and operating conditions of R407c are close to those of R-22. It has an ODP=0 and GWP = 1610, in comparison with R-22 with ODP =0.5 and GWB = 1300. R-134a is an automotive replacement for R-12, and is suitable for medium temperature refrigeration. Refrigerant R-404A is zero ozone depletion near azerotropic blend of HFC refrigerants (R-125, R-143a, and R-134a). R-404A is the ultimate and long terms HFC zero ozone depletion replacement for refrigerant R-502. Its properties closely match the properties of R-502. New system manufacturers have approved R-404A as their refrigerant in their new equipment. Result of simulation with these four refrigerants is shown in figures 3 and 4.As can be seen, R-407c has lower efficiencies than R-22, and engineers should notice this point while retrofitting. R404a has the lowest efficiency, so some changes should be taken in system before using this refrigerant. R-134a shows an acceptable behavior in energy and exergy analysis. Depth of buried ground heat exchangers determines temperature of soil which surrounds them. As can be seen in figures 5 and 6, by increasing depth, efficiency first rises and then slows down. This phenomenon is because of nature of temperature profile of earth in winters. Earth records all changes in surface temperature along the year. Changes in the Earth's surface temperature through time occur at several temporal scales. The largest of these changes are the daily and seasonal variations, both of which can have amplitudes of 10'C or more. The earth acts as a filter and attenuates these thermal waves with depth. It is very important to understand this behavior of earth. Soil diffusivity is an important factor in this behavior and is highly related to moisture content of soil. Moisture content is one of the major factors in determining the soil thermal conductivity. In these Analyses, a range of 3-30 has been used for soil moisture weight percent that is because of the equation which was used to calculate soil moisture weight percent gives valid value in this range. Results of increasing overall soil moisture content in heating season is decrease in energy and exergy efficiencies, As can be seen in figure 7 and 8. (This is because of increase in soil specific heat, and temperature drop in depth which ground heat exchangers are buried.) This should be noticed especially in cities with high rainfall and soil moisture like Rasht, that although adding moisture to the soil around GHEs would cause a better heat transfer, but overall moisture of the soil is destructive and should be noticed in design time.

1. Conclusion

In this paper the effect of depth, different refrigerants, and moisture content of soil have been studied. Performance of a heat-pump in the two cities of Tehran and Rasht was compared. Result shows that in similar conditions, studied heat pump has better performance in Rasht than Tehran, because of climate conditions of Rasht. Retrofitting of R-22 with other refrigerants is also discussed. The results indicate that, increasing ground temperature (By increasing depth which tubes are buried), can lead to a better energy and exergy efficiencies. The results also show that moisture content of soil can cause a drop in temperature of the earth and should be extensively studied at the design time, because of significant variations in energy and exergy efficiencies which it causes. At last, in design stage these parameters should be taken into consideration, in order to achieving an optimum design for various conditions.

2. Nomenclature

Specific heat of modified soil	C_{psc}
water specific heat	C_{PW}
specific heat of dry soil	C_{psd}
energy rate	$\overset{\circ}{E}$
exergy rate	Ex°
ground heat exchanger	*ghe*
specific enthalpy	h
thermal conductivity of soil	k_s
mass flow rate (kg/s)	m°
pressure (bar)	P
heat transfer rate	Q°
Entropy	S
mean earth temperature (°C)	T_e
Temperature	T
undistributed ground temperature (°C)	T_∞
earth surface temperature annual amplitude or swing (°C)	$T_{e,am}$
earth depth (m)	Z
Greek letters	
thermal diffusivity of soil	α_s
exergy efficiency	ε
angular frequency of annual earth temperature variation (θ = 0.000717 1/h)	θ
weight percentage of moisture content	λ
density of dry soil	ρ_{sd}
density of modified soil	ρ_{sc}
density of water	ρ_{PW}
time (h)	τ
specific exergy (kJ/kg)	ψ
Weight percentage of moisture content of soil	ω
Subscripts	
dead state	*0*
Condenser	*Con*
Destruction	dest
Inlet	i
Outlet	*O*
Location	*r*

Refrigerant *ref*

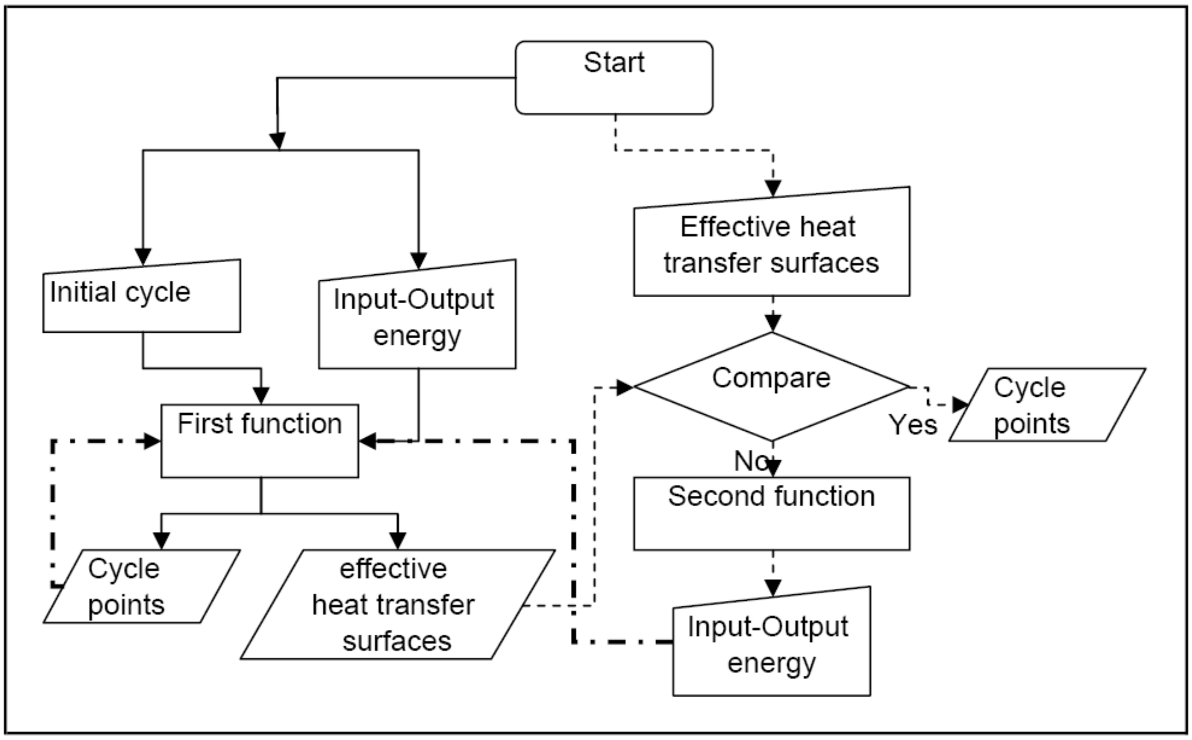

Fig.2: The flowchart of the simulation process

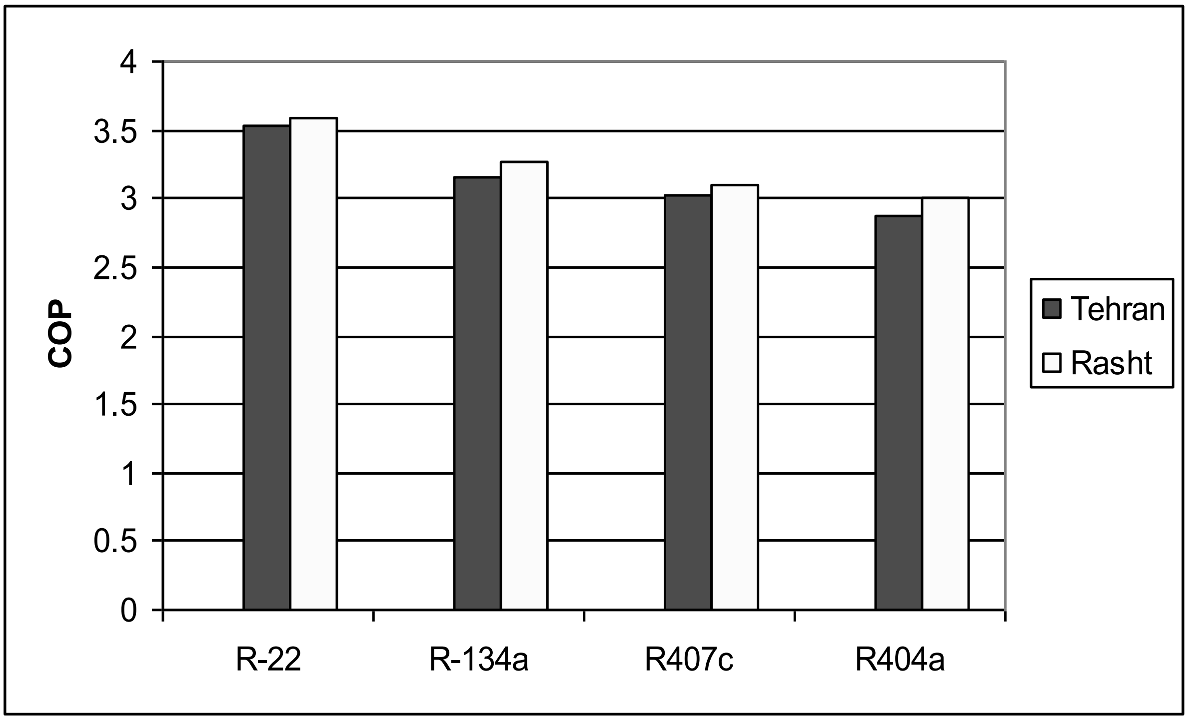

Fig.3: Energy efficiency with different refrigerants.

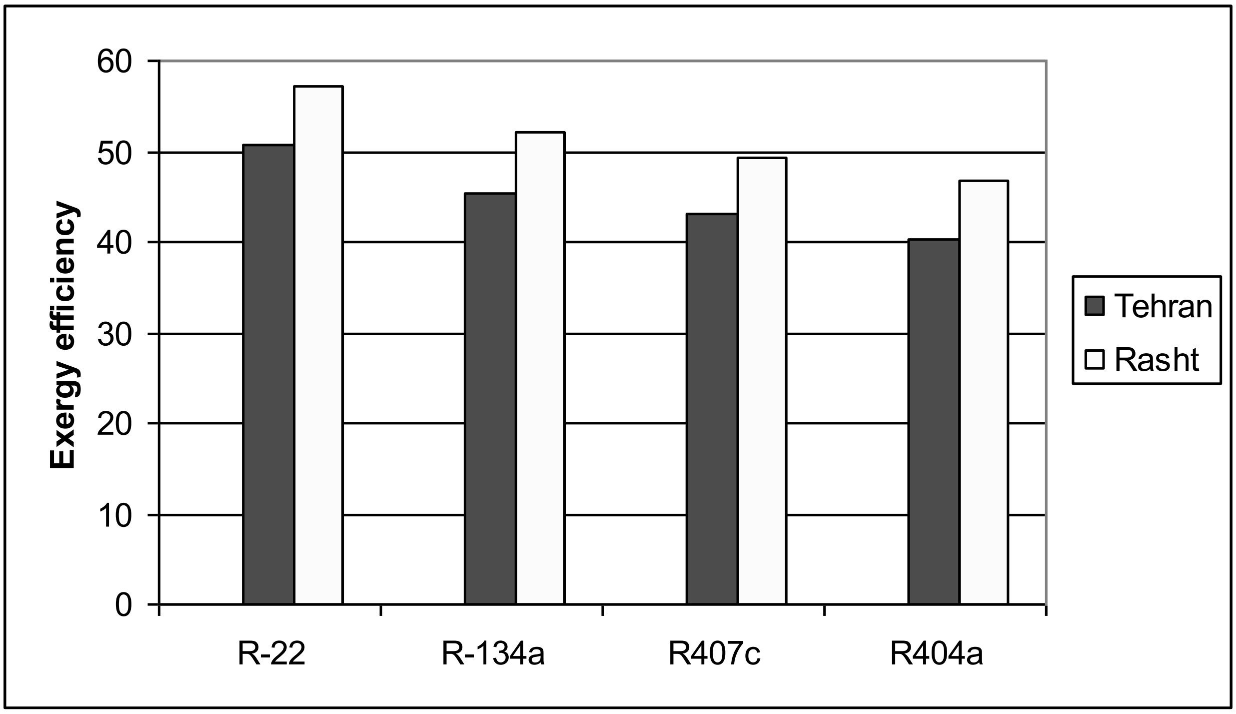

Fig.4: exergy efficiency with different refrigerants

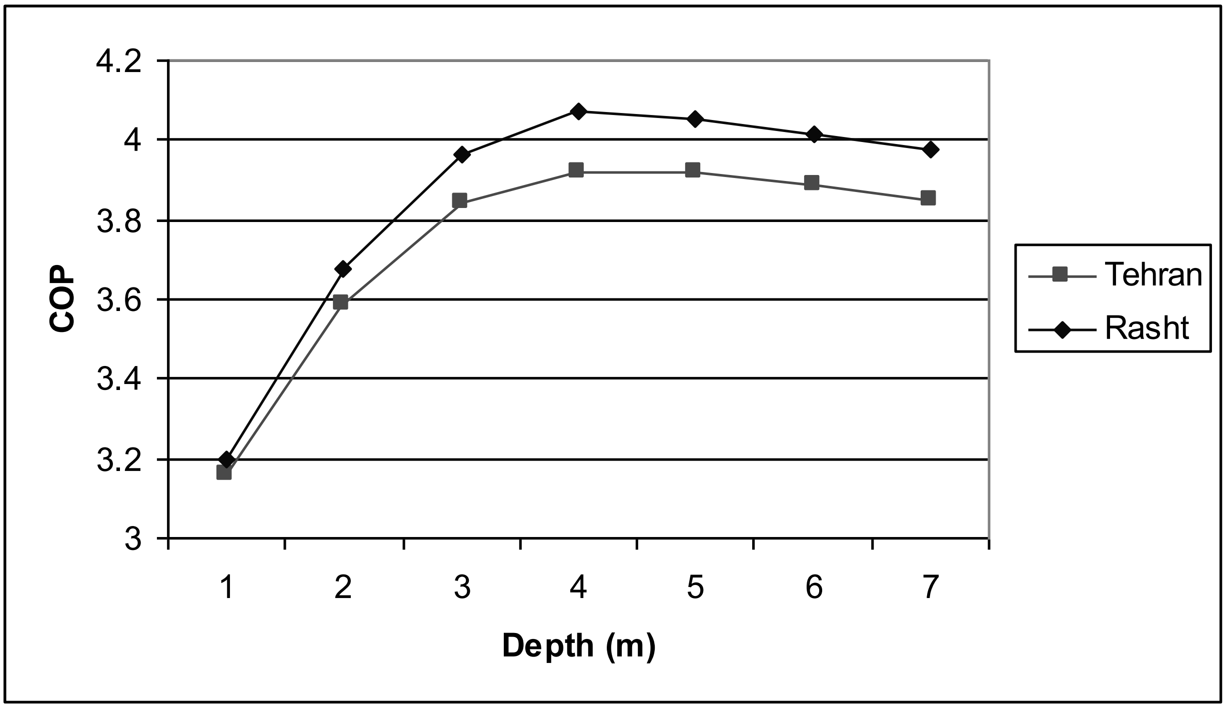

Fig.5: Energy efficiency in different depths (1^{st} January).

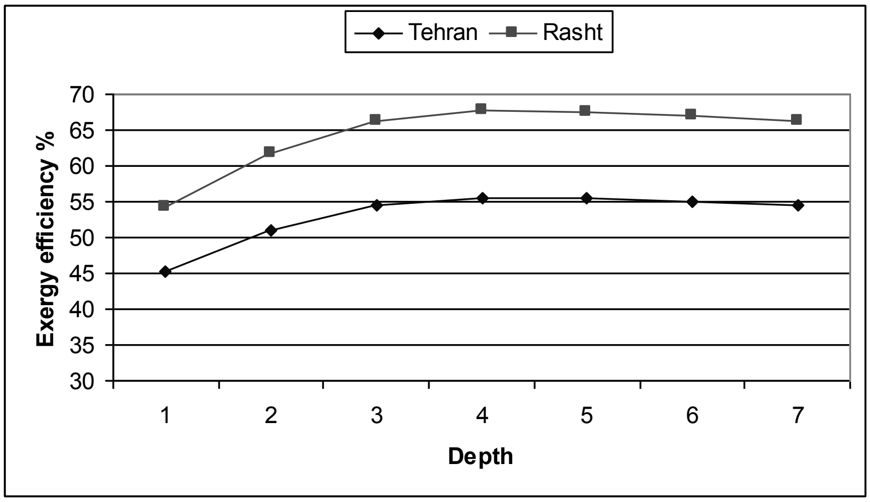

Fig.6: Exergy efficiency in different depths (1st January)

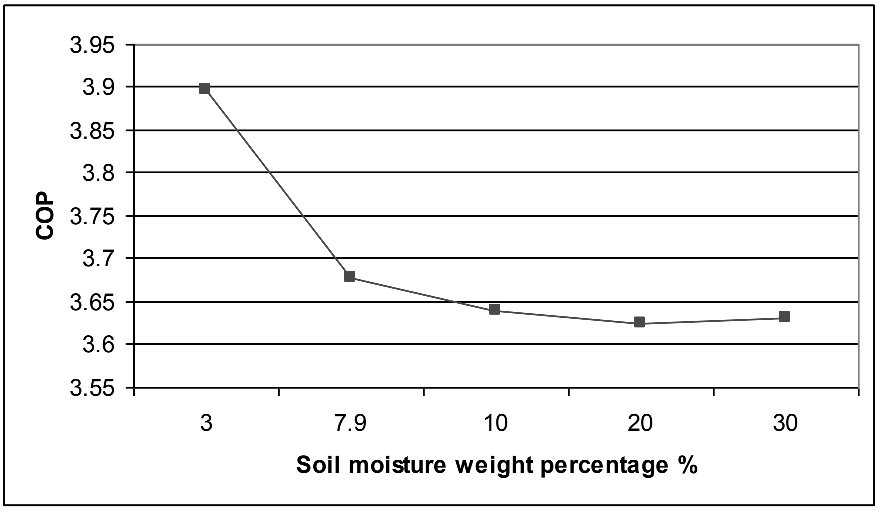

Fig.7: Effect of soil moisture in energy efficiency (1st January).

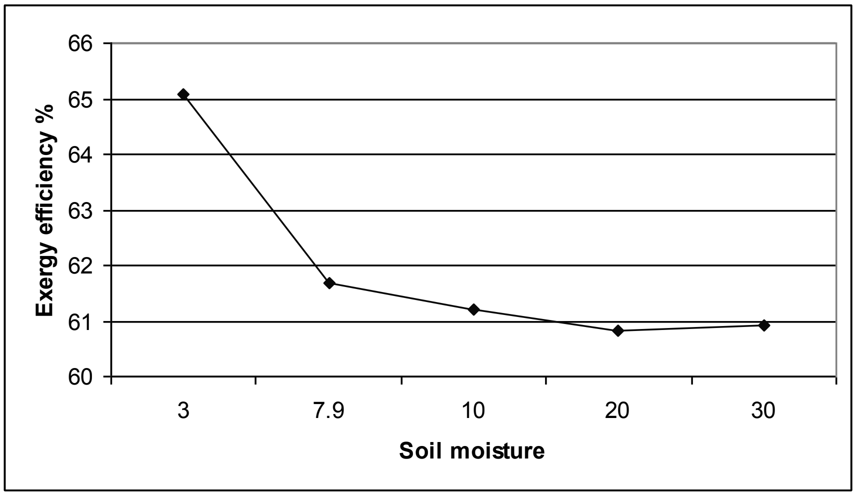

Fig.8: Effect of soil moisture in exergy efficiency (1st January).

Table 2. Results of energy and exergy analysis for selected system.

Number	Cycle point name	Fluid	Phase	Temperature	Pressure	Specific enthalpy	Specific exergy	Exergy rate
0	_	R-134a	Dead state	1.0000	1.0130	403.7700	0.0000	0.0000
0	_	Air	Dead state	1.0000	1.0130	0.0000	0.0000	0.0000
0	_	Water	Dead state	1.0000	1.0130	0.0000	0.0000	0.0000
Results for Tehran,1st January								
1	Evaporator outlet	R-134a	Super heated vapor	-9.8730	1.6582	392.7400	10.9510	0.1752
2	Compressor outlet	R-134a	Super heated vapor	57.0360	11.4500	449.7500	58.2320	0.9317
3	Condenser outlet	R-134a	Liquid	34.5000	10.8770	248.0200	23.4290	0.3720
4	Capillary tube outlet	R-134a	Mixture	258.2700	1.7431	248.0200	17.6890	0.2830
5	Condenser fan inlet	Air	Gas	47.1100	1.0130	0.1564	5.0486	0.6866
6	Condenser fan outlet	Air	Gas	23.5400	1.0130	0.0795	1.6346	0.2223
7	Ground heat exchanger water-antifreeze pump outlet	Water-antifreeze	Liquid	12.0780	3.0000	47.6370	0.9374	0.1875
8	Ground heat exchanger water-antifreeze pump inlet	Water-antifreeze	Liquid	9.3855	2.0000	36.0570	0.5405	0.1081
Results for Rasht,1st January								
1	Evaporator outlet	R-134a	Super heated vapor	-8.4300	1.7598	393.6100	12.1640	0.1946
2	Compressor outlet	R-134a	Super heated vapor	56.9700	11.5093	449.1700	58.2400	0.9318
3	Condenser outlet	R-134a	Liquid	34.7000	10.9339	248.3100	23.4600	0.3754
4	Capillary tube outlet	R-134a	Mixture	-13.4300	1.8498	248.3100	18.0460	0.2887
5	Condenser fan inlet	Air	Gas	47.0100	1.0130	49.0330	6.2442	0.8492
6	Condenser fan outlet	Air	Gas	23.5400	1.0130	24.0200	2.2294	0.3032
7	Ground heat exchanger water-antifreeze pump outlet	Water-antifreeze	Liquid	13.5920	3.0000	54.1471	1.2067	0.2413
8	Ground heat exchanger water-antifreeze pump inlet	Water-antifreeze	Liquid	10.8880	2.0000	42.5220	0.7490	0.1498

3. REFRENCES

[1] Kavanaugh SP. Field test of vertical ground-coupled heat pump in Alabama. ASHRAE Trans 1992;98(2):607–16.

[2] Kavanaugh SP. Development of design tools for ground-source heat pump piping. ASHRAE Trans 1998;104(1B):932–7.

[3] Yumrutas R, Kunduz M, Kanoglu M. Exergy analysis of vapor compression refrigeration systems. Exergy, An International Journal 2002;2(4):266–72.

[4] Nakanishi S, Kawashima Y, Murai K. Thermodynamic analyses of performance of heat pumps and refrigerators. Part 1: exergy analysis of COP ratio as a performance index. Transactions of JAR 1993;10(1):1–9.

[5] Mortaza Yari, Nader Javani. Performance assessment of a horizontal-coil geothermal heat pump.International Journal of Energy Research. 2006(31), P. 288 – 299.

[6] Wei Bo Yang, Ming Heng Shi, Hua Dong. Numerical simulation of the performance of a solar-earth source heat pump system. Applied Thermal Engineering 2006; 26: 2367–76.

[7] Iranian meteorological organization (IMO), Data processing center.

[8] deVries, D. A., 1963. Thermal Properties of Soils. In W.R. van Wijk (ed.) Physics of Plant Environment. North-Holland Publishing Company, Amsterdam.

[9] deVries, D. A. 1975. Heat Transfer in Soils. In D.A. de Vries and N.H. Afgan (ed.) Heat and Mass Transfer in the Biosphere. Pp.5-28. Scripta Book Co., Washington, DC.

[10] Farouki, O.T.,Thermal Properties of Soils. Series on rock and soil mechanics. Vol. 11. Germany: Trans Tech Publ., Clausthal-Zellerfeld; 1986.

[11] Hepbasli and et al. Experimental study of a closed loop vertical ground source heat pump system.Energy Conversion and Management; 2003; 44: 527-48.

[12] Wu, J. and D. L. Nofziger. Incorporating temperature effects on pesticide degradation into a management model. J. Environ. Qual. 1999; 28; 92-100.

[13] Bowman, R. A.,A. E. Mueller, and W. M. Nagle. Mean Temperature Difference in Design, Trance. ASME, vol. 62, 1940.

[14] J. P. Holman, Heat transfer. 9th ed. New York: McGraw-Hill; 2001.

4. BIOGRAPHIES

Koroush Javaherdeh is the assistant Professor of Energy Conversion at the Islamic Azad University of Takestan. In 1987 He has graduated in Mechanical Engineering at the Ferdowsi University in Iran. Latter in 1996 he has done his MSc, PhD in mechanical and energy field at the Nancy University in France. His areas of interest include the design, development, analysis and optimisation of energy conversion systems and heat transfer in non-Newtonian fluid where he has published several papers. Since 1997, he has joint the Islamic Azad University of Takestan and involved in several projects in his fileds.