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Evaluation of low temperature geothermal energy through the use of heat pump

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Abstract

A water-to-water geothermal heat pump (GHP) running on R-22 was designed to evaluate disposed geothermal water from geothermal health resort centres in Erzurum. The GHP uses geothermal water at 35°C temperature and provides clean water at 45°C for a floor heating network. The GHP heating capacity was around 7.2 kW, and an electric driven hermetic R-22 compressor was used. The overall coefficient of performance was determined as 2.8. © 2000 Elsevier Science Ltd. All rights reserved.

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1. Introduction

Despite Turkey having very rich geothermal resources corresponding to 350 MW electrical power or 2000 MW thermal energy, evaluation of the resources is poor. There is only one electrical power plant (20.4 MW) using geothermal energy and a few direct use district heating applications across the country [1]. For direct use of geothermal energy in space heating applications, the well temperature should be higher than 80°C. The temperature gradient of geothermal water will vary by location, however, some reasonable predictions of well temperature versus well depth can be made using the formula given by Niess [2].

$$T = 12.8 + 27.4(Z),$$

where T and Z stand for temperature in °C and depth in km, respectively. From this correlation, it can be concluded that for obtaining geothermal water at 80°C or higher, deeper wells are required.

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Nomenclature

η	efficiency
ω	rotational speed (rad/s)
COP	coefficient of performance
c_p	specific heat at constant pressure (kJ/kg K)
h	specific enthalpy (kJ/kg)
I	current (A)
m	mass flow rate (kg/s)
P	power (kW)
Q	Heat rate (kW)
T	temperature (°C)
v	specific volume (m^3/kg)
V	volt
V_d	compressor displacement volume (m^3)
W	work (kW)

Subscripts

r	refrigerant
cd	condenser
ev	evaporator
w	water
i	inlet
o	outlet
cm	compressor
s	suction
v	volumetric

For instance, to provide hot water at 80°C, the indicated well depth would be over 2400 m. The cost of drilling wells to these depths could be prohibitive, considering the present cost of conventional energy sources. On the other hand, according to the Turkish Mineral Research and Exploration Foundation (MTA), there are more than 70 drilled wells, varying over 100–750 m depths, with 1–300 l/s flow rates and 30–50°C temperatures. There are also more than 170 hot springs at 30–50°C temperature range across the country [1]. Most of the wells, even at high temperature, are used as health resort centres, and the geothermal water after the baths is discharged to the environment. For instance, there are three wells below 45°C temperature around Erzurum city, where Ataturk University is located. In the *Pasinler* location, there are two wells at 42°C temperature, 200 m depth and 75–95 l/s flow rates. Another well is available in the *Ilıca* location at 39°C and 605 m depth. The wells are used only for health care, and the temperature of the discharged geothermal water is around 30–35°C. This is a very appropriate source temperature for a heat pump. There are many BTUs that can still be extracted from the disposed geothermal water via a heat pump. The basic principles underlying heat pump technology have been known for about a hundred years. A heat pump works by extracting heat from a low temperature source, such as water, air or the ground, and using it to provide useful energy for water heating.

Recent years, in the US, have brought new interest to geothermal heat pumps (GHPs) to evaluate low temperature geothermal resources for space heating rather than direct use of geothermal wells because of their higher drilling cost. For example, a GHP system was installed in the Daniel Boone High School in Washington County, resulting in \$37 000 per year saving. This project won the 1998 ASHRAE Technology Award in the category of alternative and/or renewable energy use [3]. Another GHP system was designed to absorb heat from a 30°C well and supply water at 52–65°C temperature to Grant County Courthouse's central heating system in Ephrata, Washington, resulting in an 80% decrease in energy consumption and an 85% decrease in the Courthouse fuel bill. This project also received national awards from the US Department of Energy and from ASHRAE [4].

Unlike the US, there is no GHP application in Turkey despite its good potential of low temperature geothermal sources just mentioned above. The main objective of this work is to evaluate a heat pump system using health resort discharge water as a source of heat to supply hot water for a hydronic heating network in Erzurum city where heating is a significant problem because of its harsh winter conditions.

2. Experimental work

The GHP system consists of three water loops and a refrigerant circuit as shown in Fig. 1. Geothermal water flows through plate heat exchanger (22), where it's heat is transferred to the evaporator water loop. The plate heat exchanger isolates the geothermal water from the heat

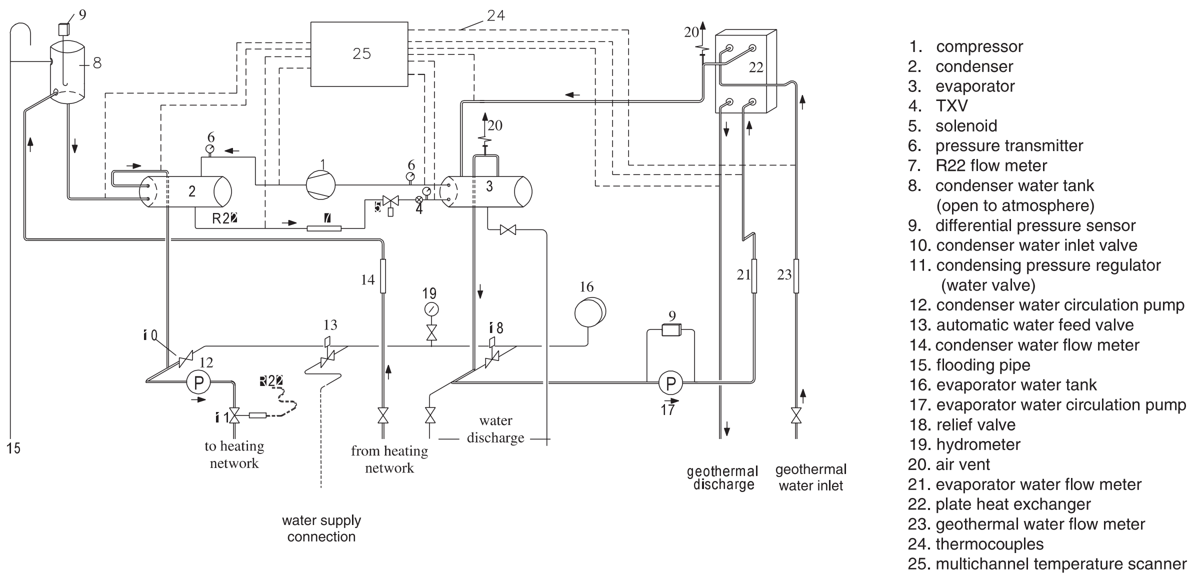

Fig. 1. Schematic diagram of experimental setup.

pump. The plate exchanger is Ti alloy, and it has good resistance against corrosion. In geothermal applications, corrosion is a very important problem and should be taken into account. Ti alloy plate heat exchangers work successfully against corrosion, especially at low temperature geothermal applications, and allow mechanical cleaning for scaling [5–7]. The heat extracted from geothermal water by the evaporator water loop is transferred to the refrigerant in the evaporator (3). The evaporator (3) is a shell and tube type heat exchanger in which the refrigerant flows inside the tubes, while water flows in the shell side.

Somehow, if there is no water in the evaporator loop, differential pressure sensor (9) produces an alarm signal and stops the compressor and all pumps. The refrigerant (R-22) evaporates by absorbing heat from the evaporator water and then enters the hermetic compressor (1). The refrigerant is compressed by the compressor and then enters the condenser (2), where it condenses. The condenser is a shell and tube type and has no subcooling, since the condensing refrigerant accumulates in it. The refrigerant condenses on the tube bundle, while the condenser water flows through in the tubes. After leaving the condenser, the refrigerant goes through thermostatic expansion valve (4), where it expands to the evaporation pressure. TXV provides almost 10°C superheat that essentially gives a safety margin to reduce the risk of liquid droplets entering the compressor.

The condenser water loop absorbs heat from the condensing refrigerant in the condenser and delivers it to the building via the floor heating distribution network. If there is no water in the loop, differential pressure sensor (9), placed on the condenser water tank (8), stops the compressor.

The water is supplied to both the evaporator and condenser loops by an automatic water feed valve (13). If there is no geothermal water circulation while the compressor is on, the temperature of the evaporator water loop will decrease below 0°C and freezing will occur. In order to avoid freezing, the evaporator water loop is equipped with a thermostat that stops the compressor if the temperature decreases below the set value (3°C).

The refrigerant pressures at the condenser inlet and evaporator inlet/outlet were measured by using pressure transmitters (6). The accuracy of the pressure transmitters is 1% of full scale, and the effective measurement range is 0–25 bar. The standard output of the transmitters is 4–20 mA, and it is monitored by an LCD displayer. The accuracy and performance of the transmitters were satisfactory during the experiments. Because the tube lengths connecting the compressor, condenser and evaporator are small and insulated, the temperature and pressure losses in the connection tubes are neglected. The refrigerant pressure drop through the evaporator was measured, since the refrigerant flows inside the tubes, but the condenser refrigerant pressure drop was neglected, since the diameter of the condenser shell in which the refrigerant flows is large.

The refrigerant flow rate was measured by a metal tube magnetic flowmeter (7). Since the metal tube magnetic flowmeter is originally calibrated for water flow metering (i.e. its indicator scale is arranged for water), the measured data were corrected for R-22 by using the correction factor based on liquid density ratio given by the manufacturer in his product catalog. It is mounted at the condenser exit, where R-22 is in the liquid phase. The measurement range and accuracy of the flowmeter are 40–400 l/h and $\pm 2\%$ of full scale, respectively. Pressure loss across the metal tube is 20 mbar.

The refrigerant and water temperatures at the inlet and outlet of all heat exchangers (i.e. condenser, evaporator and plate) were measured by thermocouples (24). T-type (i.e. copper–

constant) thermocouples are used. They are calibrated at 0–90°C temperature range by using a standard water bath, and the accuracy is found as ±0.4. Thermocouple wires are isolated against environmental effects. All the temperatures measured are monitored and stored by a multi-channel temperature scanner.

The flow rates of the evaporator, condenser and geothermal water loops were measured by plastic tube flowmeters (14,21,23). The plastic tube flowmeters work based on the float principle that the float moves freely without friction in the measuring tube. The measurement range and accuracy of each flowmeter are 200–2000 l/h and 2% of full scale. The power consumptions of the compressor and pumps were also measured.

The tests were done under steady state conditions to determine the overall coefficient of performance of the system. Mean values of the measured data are given in Table 1.

Table 1
Measured parameters and experimental results

Measured parameters	
Evaporation pressure (MPa)	0.76
Compressor suction pressure (MPa)	0.68
Condensation pressure (MPa)	1.94
Refrigerant mass flow rate (kg/s)	0.0372
Refrigerant temperature at evaporator inlet (°C)	13.8
Refrigerant temperature at evaporator outlet (°C)	24
Refrigerant temperature at condenser inlet (°C)	98
Refrigerant temperature at condenser outlet (°C)	47
Compressor discharge temperature (°C)	98
Water temperature at condenser inlet (°C)	39.4
Water temperature at condenser outlet (°C)	45
Condenser water mass flow rate (l/h)	1100
Water temperature at evaporator inlet (°C)	27.5
Water temperature at evaporator exit (°C)	23.4
Evaporator water mass flow rate (l/h)	1100
Geothermal water inlet temperature (°C)	35
Geothermal water outlet temperature (°C)	30
Geothermal water mass flow rate (l/h)	1100
Compressor electrical current (A)	3.82
Total current of pumps (A)	2.7
Three-phase voltage (V)	375
Two-phase voltage (V)	215
Calculated results	
Condenser water side heat rate (kW)	7.02
Condenser refrigerant side heat rate (kW)	7.39
Evaporator refrigerant side heat rate (kW)	4.93
Evaporator water side heat rate (kW)	5.24
PHE geothermal water side heat rate (kW)	6.39
Volumetric efficiency (η_v)	0.82
Compression work (kW)	1.59
Actual compressor power consumption (kW)	1.98
Total pump power consumption (kW)	0.527
Actual COP	2.8

3. Results and discussion

The thermodynamic analysis of the system was based on the vapour compression cycle shown in Fig. 2. Heat rates in the condenser and the evaporator, both refrigerant side and water side, were calculated as follows.

$$\dot{Q}_{\rm rcd} = \dot{m}_{\rm r}(h_2 - h_3), \tag{1}$$

$$\dot{Q}_{\rm cdw} = \dot{m}_{\rm cdw} c_{p\rm w}(T_{\rm cdwo} - T_{\rm cdwi}), \tag{2}$$

$$\dot{Q}_{\rm evr} = \dot{m}_{\rm r}(h_1 - h_4), \tag{3}$$

$$\dot{Q}_{\rm evw} = \dot{m}_{\rm evw} c_{p\rm w}(T_{\rm evwi} - T_{\rm evwo}). \tag{4}$$

The work done on the refrigerant by the compressor was computed from

$$\dot{W}_{\rm r} = \dot{m}_{\rm r}(h_2 - h_1) \tag{5}$$

and the compressor power is

$$\dot{W}_{\rm cm} = \frac{\dot{W}_{\rm r}}{\eta_{\rm motor}\eta_{\rm mech}}. \tag{6}$$

The compressor actual power consumption was calculated from

$$W_{\rm cm,actual} = I_{\rm cm} V_{\rm cm} \sqrt{3} \cos\phi / 1000. \tag{7}$$

The actual volumetric efficiency of the compressor is

$$\eta_{\rm v,actual} = \frac{\dot{m}_{\rm r} v_{\rm s} 2\pi}{V_{\rm d}\omega_{\rm cm}}. \tag{8}$$

The pumps power consumption is

$$\sum P = V_{\rm pump}\left(\sum I_{\rm pump}\right)\Big/1000. \tag{9}$$

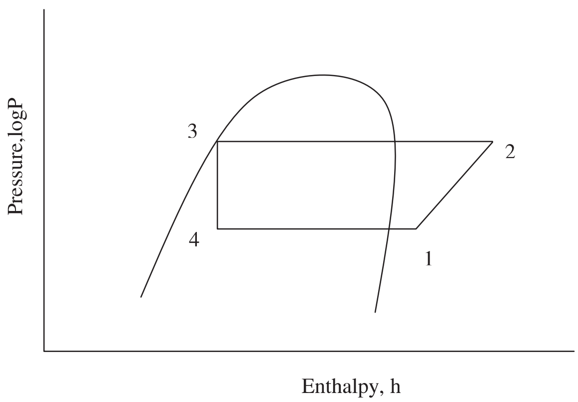

Fig. 2. Thermodynamic cycle of vapour compression heat pump cycle.

The overall coefficient of performance of the system is

$$\mathrm{COP} = \frac{\dot{Q}_{\mathrm{cdw}}}{\dot{W}_{\mathrm{cm,actual}} \sum P}. \tag{10}$$

The primer energy ratio is

$$\mathrm{PER} = \eta \times \mathrm{COP}, \tag{11}$$

where η is the efficiency of electric generation and distribution.

Thermodynamic properties of R-22 were calculated by a computer subroutine based on the equations given by Downing [8]. The results are shown in Table 1. According to Table 1, the condensation pressure is 1.94 MPa, which corresponds to 49.9°C saturation temperature, and this value is too close to the maximum discharge pressure limit of the compressor. According to the compressor manufacturer information, a safe discharge pressure limit of the compressor is around 22.4 bar. The compressor discharge valve will be damaged at discharge pressures higher than this value. As a result, if condensation temperatures higher than 50°C are desired, R-22 is not supposed to be used with a reciprocating compressor because of the over discharge pressure that is typically around 22 bar for a reciprocating compressor [9]. It can be concluded from this that a low temperature (i.e. 30–35°C well temperature) geothermal heat pump with R-22 reciprocating compressor is only suitable for floor heating. Since the condensation pressure is higher than that of a conventional refrigeration system with R-22, the equipment in the high pressure section of the GHP (i.e. condenser, sight glass, etc.) should be chosen accordingly.

The evaporation temperature is obtained as 13.8°C (see Table 1) because of the compressor maximum suction pressure limitation and would be expected to be around 18°C considering a 35°C well temperature. The thermostatic expansion valve ensures around 10°C superheat, which is a conservative value for reciprocating compressors. On the other hand, a disadvantage of superheat is that the compressor must have an increased size, and a more important problem is that the compressor discharge temperature is increased. Therefore, 7–8°C superheat would be better.

From Table 1, the refrigerant outlet temperature of the condenser is 47°C, and the condensation temperature is 49.9°C. There is 2.9°C subcooling, which is too small. Considering 39°C condenser water inlet temperature, 8°C subcooling would be expected.

The heat rates absorbed from geothermal water and transferred to evaporator water are determined as 6.39 and 5.24 kW (see Table 1), and the ratio of the heat rates is calculated as 0.82 which is the efficiency of the plate heat exchanger. In general, PHEs have higher efficiency. Because of fouling and scaling in the geothermal side of the PHE, the efficiency is lower. In order to avoid this problem, the PHE needs periodic cleaning.

The overall actual performance of the GHP system is calculated from Eq. (10) based on the ratio of heat obtained from the system to energy consumed by the system and is obtained to be 2.8 (see Table 1). An error analysis performed according to a method given Holman [10] indicated that the COP calculation could be in error up to 8.36%. The COP gives a measure of the usefulness of the heat pump unit in producing large amounts of heat from a small amount of work. It does not, however, express the fact that energy available as work is normally more valuable than energy available as heat. In order to compare a conventional heating system using a boiler fired by fossil fuels with an electric driven heat pump, the PER should be applied. Electrical power is a secondary energy and is generated inefficiently. The PER takes into account not only the heat pump COP but also the efficiency of conversion of the primary fuel into the work that drives the heat pump. According to Eq. (11), η stands for the efficiency of electric generation and distribution and can be assumed as around 30%. If the COP of a GHP is 3, then

$$\mathrm{PER} = 0.3 \times 3 = 0.9.$$

Assuming a boiler heating system efficiency, typically 70–80%, the ratio of PER to the efficiency of the boiler varies in the range of 1.12 and 1.28. This means that the heat pump gives 12–28% more heat than direct combustion of the fuel. Now, it can be concluded that the COP of a GHP should be 3 or higher to take an advantage over the conventional boiler heating system. As a result, the COP of the experimental system (see Table 1) is a little bit lower, but some basic modifications discussed in the recommended improvements section can enhance the overall performance of the experimental system.

4. Recommended improvements

In order to get better performance, the equipment of the heat pump should be sized carefully. For instance, according to experimental results, the maximum required water flow rates in the condenser, the evaporator and geothermal water loops for the current system are 1100 l/h, but the pumps can provide a flow rate higher than 2000 l/h. The centrifugal pump on the geothermal water loop and the other pumps on the condenser and the evaporator water loops are oversized. This is a very important point, since it decreases the COP. According to Eq. (10), the total power consumption of the pumps appears in the denominator, and if the pumps are oversized, the COP will decrease. As a consequence, great attention should be paid to pump sizing.

The condenser in the system did not provide enough subcooling, as mentioned in the previous section, but 8°C subcooling could be obtained by installing a subcooler or a larger condenser, and this would increase the COP. As is also mentioned in the results and discussion section, the evaporation temperature can be 18°C, resulting in a decrease in compression work. All these factors mentioned above will help the COP increase up to 3–3.5.

The most common heating system in Turkey is a hydronic system consisting of a boiler, fired by oil or solid fuels, a hot water circulating pump, distribution pipes and local heat exchangers, radiators or floor heating system, where heat is transferred to an individual room by means of free convection and radiation. The radiator system is more common than the floor heating system in Turkey. A GHP can be integrated readily into a hydronic system by replacing only the boiler with the GHP without changing the distribution network. When a GHP is integrated into the radiator system, the main problem is supply temperature. The design supply/return temperatures of the radiator system are 80°C and 60°C, respectively. With 80°C supply temperature, the radiator surface temperature is a very high value compared to room temperature and causes drying of the room air. This is not good considering room comfort, especially in some region such as Erzurum where the humidity is already low. Consequently, a 60°C supply temperature can be preferred to 80°C. For a GHP with 60°C supply temperature, 65°C condensation temperature is required and is hardly achievable with a R-22 heat pump, but it is possible with R-134a.Therefore, R-134a would be a much more proper refrigerant than R-22 for the GHP system.

5. Conclusion

As is mentioned in the introduction section, Turkey has a very good potential of low temperature geothermal resources, and they are only used for health care. Unfortunately, geothermal water is discharged to the environment when health resort facilities are heated by a boiler fired by fuel. The facilities, at least, can be more efficiently heated by a GHP than by a boiler. Since the GHP uses electric energy, it does not make any air pollution.

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