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Research Article

Experimental Performance Analysis of a Groundwater Heat Pump System in Erzincan, Turkey

Orhan KALKAN[1*], Mehmet KAYA[2]

[1] Erzincan University, Department of Mechanical Engineering, TR-24002, Erzincan Turkey
[2] Kocaeli University, Department of Airframe and Powerplant Maintenance, TR-41285, Kocaeli Turkey

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Abctract

In this study, a place with a floor area of 80 m² in Erzincan was heated by the Groundwater Heat Pump (GWHP). In this application experimentally investigated, energy and performance analysis of the heat pump and the entire system were performed. Evaporator water flow, condenser water flow and air flows of fan-coil devices was fixed at certain values. Then the measurements have been made under 9 different conditions consisting of these parameters. In the condition which the highest yield is obtained, performance coefficients of the heat pump (COP_{hp}) and the system (COP_{sys}) were determined 4.27 and 2.92, respectively. In this condition, the average amount of heat discharged from the condenser was calculated to be 14.3 kW. On the other hand, Groundwater quality and temperature that is an important criterion of the GWHP system was found to applicable to Erzincan.

Keywords: Energy, Heat pump, Groundwater, Refrigerant, COP

Erzincan İlinde Bir Yeraltı Suyu Kaynaklı Isı Pompası Sisteminin Deneysel Performans Analizi

Öz

Bu çalışmada, Erzincan İlinde bulunan 80 m^2 taban alanına sahip bir mekân Yeraltı Suyu Kaynaklı Isı Pompası (YASKIP) ile ısıtılmıştır. Deneysel olarak gerçekleştirilen bu çalışmada ısı pompasının ve tüm sistemin enerji ve performans analizi yapılmıştır. Buharlaştırıcı su debisi, yoğunlaştırıcı su debisi ve Fan-coil hava debisi belirli değerlerde sabitlenmiştir. Daha sonra bu parametrelerden oluşan 9 farklı koşulda ölçümler alınmıştır. En yüksek değerlerin alındığı koşulda, ısı pompasının ve tüm sistemin performans katsayıları sırasıyla 4.27 ve 2.92 bulunmuştur. Bu koşulda yoğuşturucudan atılan ortalama ısı yükü 14.32 kW olarak hesaplanmıştır. Diğer yandan, yeraltı suyu kalitesi ve sıcaklığı önemli bir kriter olan YASKIP sisteminin Erzincan için uygulanabilir olduğu belirlenmiştir.

Anahtar Kelimeler: Enerji, Isı Pompası, Yeraltı Suyu, Soğutucu Akışkan, ITK

1. Introduction

Since energy is directly related to human life, it is among the most important needs of man. When we look at the distribution of energy usage by sectors in Turkey; 42.4% of energy is used in climate, 36.2% in industry, 15.4% in transportation and 2.9% in agriculture (Turkish National Committee of the World Energy Council, 2016). It seems that air-conditioning has the largest share of energy use. For this reason, the importance of systems for the use of new and renewable energy sources in climate is great. Today, it is known that heat pump systems come first in the air conditioning systems (Szreder, 2014).

A heat pump is a system that briefly carries heat energy from one medium to another and is fed by electricity. Heat pumps, which have the advantage of being able to perform both heating and cooling processes with a single system, are increasingly used in air

*Corresponding Author: orhan.kalkan@erzincan.edu.tr

conditioning. Heat pumps are widely used in developed countries such as America, Canada, Germany, Switzerland. In particular, 25% of the buildings built after 1978 in America were planned and implemented to be heated by heat pump (Yamankaradeniz et al., 2013). In this regard, it is an important factor for heat pumps to achieve economical and efficient heating with low emissions (Chua et al., 2010). When we look at the literature, many studies related to heat pumps have been made. Some of these studies are summarized below:

Errera et al. (2014) investigated the use of underground-type heat exchangers in heat pump applications at different capacities and the effect of different combinations of these heat exchangers on system performance. Three types of design structures have been systematically compared. The first of these is an installation of equal-sized heat pumps. In the second, the heat pumps were distributed at equal distances to the cycles. The third is a design in which a number of heat pumps in different capacities are distributed in a single cycle. As a result, the third type of design seems to be more efficient than the others.

Haiwen et al. (2010) compared the seawater source heat pump with the conventional boiler house district heating system. They calculated the critical COP values of the heat pump system based on sea water depths, assuming a natural gas combustion rate of 60%. However, it has been stated that heat pump systems will save energy as long as there is no proper research and design. It also has been determined that energy saving will be achieved in the system when the criteria such as high temperature regions of sea water, reduction of seawater pumping power, keeping suction and injection points of water away from each other are taken into consideration.

Energy and exergy analyze of a GWHP system were conducted in an apartment building in Wuhan, China. In Wuhan, where 20 GWHP applications were made in 2000, ground water temperature is the range of 18-22 °C. The heating and cooling peak loads of the building tested are 2950 kW and 3687 kW respectively. Each of the 8 wells opened for groundwater use is 47 m deep and 500 mm in diameter. From June 2007 to February 2008, exergy was calculated as 0.50 and 0.19 in the heating and cooling season, respectively. The COP values of the heat pump are 4.1 (cooling) and 4.57 (heating). The total system COP values are 2.68 (cooling) and 3.1 (heating). As a result, GWHP system has been determined to be more efficient than air-source heat pumps (Fei and Pingfang, 2012).

Wang et al. (2015) conducted a measurement and evaluation of the GWHP system, in which 4 apartment buildings were heated and cooled in an area surrounded by 69 000 m². The energy consumption measured during the 92-day winter season was 2 346 000 kWh. The electricity cost for heating is 1 809 919 Yuan. The energy consumption measured during the 152-day summer season was 434 500 kWh. The electricity cost of cooling is 361 842 Yuan. The average COP value of the heat pump obtained during the heating season is 2.33. The average COP value of the system is 2.65. Compared to conventional air conditioners, the YASKIP system has lower operating costs and saves 21.39% in annual operating costs.

Kural (2012) compares the initial investment and annual operating costs of the GSHP (Ground Source Heat Pump) system with central air conditioning and natural gas systems. According to the cost analysis of these three separate systems made in a building with a total heating load of 810 kW in Malatya, Turkey. GSHP system was found to be the most economical system. The theoretical COP value of the system is 3.76. As a result of comparison, according to the central air conditioning system of the GSHP system, the depreciation time was found to be shorter than 5 months and 16 months compared to the natural gas system.

Bakirci (2010) studied the performance of the GSHP system in Erzurum, a cold climate region. In the experimental study, 2 vertical 53 m deep underground heat exchangers were applied. R134A refrigerant is used in the system with 8 kW heating power. According to the test results in the heating season, the annual average COP values of the heat pump and the system are found to be 3.1 and 2.7 respectively. As a result, the GSHP system has been found to be useful in residential heating in Erzurum, Turkey.

Aksu (2010) experimentally analyzed the performance of an open system ground-water source heat pump using a lake as a heat source in Balikesir, Turkey. The system works from water to air. Experiments were carried out at air velocities of 100 l/h, 300 l/h and 500 l/h at 1 m/s, 2 m/s and 3 m/s air velocities and the results were compared. The highest heating COP value was measured as 3.15 at an air speed of 2 m/s and a water flow rate of 500 l/h. As a result, the heating COP value of the system increases with increasing air velocity and water flow.

In this study, a single-storey building with a floor area of 80 m² was heated by GWHP system in Erzincan, Turkey. In the application where drilling and injection wells are used at a depth of 80 m, the heat source cycle is designed as an open system. The heat from the condenser was transferred by fan-coil appliances. In the experimental study; measurements were taken in 9 different conditions in which the evaporator water flow, condenser water flow, and fan-coil air flow were kept constant at certain values. Energy and performance analysis of this system, which has been experimentally examined, has been carried out.

2. Experimental Method

2.1. GWHP system design

In this study which is performed experimentally in the province of Erzincan, a single storey building with a floor area of 80 m² in Erzincan Public Hospitals Union settlement was heated using GWHP system. The overview of the system, calculated as a total heat load of approximately 12.4 kW, is as shown in *Figure 1*. *Figure2* shows the schematic of the GWHP operating system, the elements on the system and the temperature measurement points.

Figure 1. Overview of the GWHP system

A scroll compressor was selected at 3 kW power which can work with R410a gas for pressurization and circulation of refrigerant in heat pump. Scroll compressors, which have become increasingly popular in recent years, are more efficient and silent than other compressors (Wang, 2000). A coaxial heat exchanger with a capacity of 15 kW is used for the condenser into which the pressurized refrigerant is transferred. In the water-cooled condenser, the heat of the refrigerant is transferred to the fan-coil line. A 12 kW coaxial heat exchanger of the same type is used as the evaporator to transfer the heat from the heat source to the refrigerant.

The temperature values were measured with a k-type thermocouple at 0.1 ^{o}C accuracy from 12 different points. It was also performed using a 12-channel datalogger. The water flow in the primary and secondary loops is measured using float flowmeters.

In the GWHP system, the primary circuit between the evaporator and the heat source is designed as an open system. To use underground water as a heat source, a suction well and an injection well have been drilled at a depth of 80 m. The static level of the drilling water is observed to be 25 m deep.

The secondary closed-loop is the heat sink that receives the heat from the condenser. While the thermal energy transferred to the secondary loop is transferred to the working area via fan-coil devices. Four of the fan-coil units with heating power of 3.5 kW in the operating temperature range of 50 ^{o}C - 45 ^{o}C were used to heat the working area with a heat load of approximately 12.4 kW.

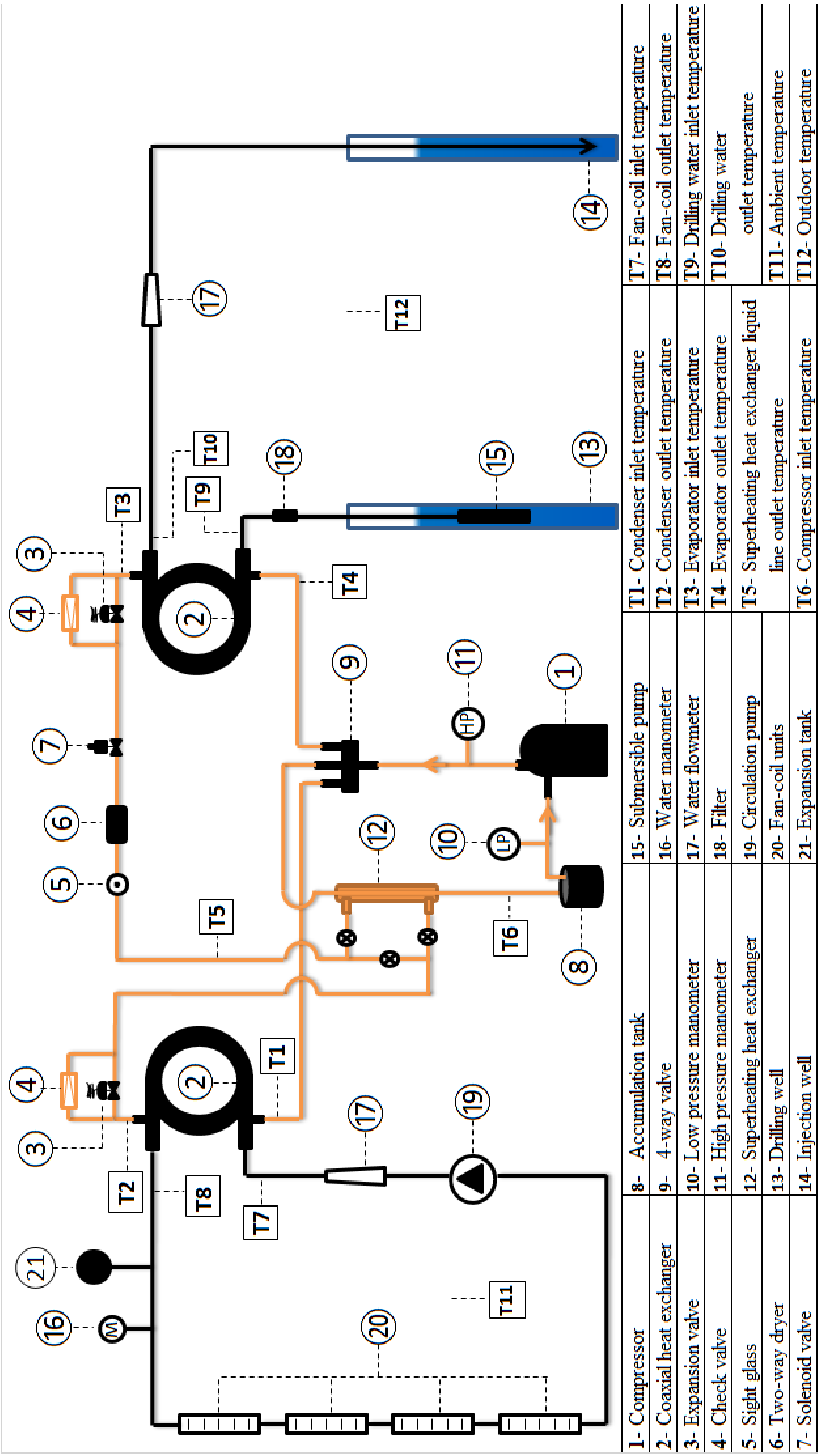


Figure 2. System scheme of the GWHP

2.2. Determination of the experimental measurements and parameters

The GWHP system, designed and installed according to Erzincan province conditions, was operated during the heating season of 2016 and subjected to experimental measurements. Since Erzincan province is in a cold climate region, the heat pump will consume more energy than cooling for heating. For this reason, experimental measurements were made only in heating mode. On the other hand, experiments were conducted between 9:00 and 17:00 hours during the day as the heated space was used as a public institution and during working hours. Climatic data for Erzincan is given in *Table 1* (Turkish State Meteorological Service, 2016).

Table 1. Climatic condition of Erzincan for long-term average values.

Climatic values	Months in cold season					
	Nov.	Dec.	Jan.	Feb.	Mar.	Yearly
Average outdoor temperature (ºC)	5.5	-0.3	-3.1	-1.3	3.9	10.9
Maximum outdoor temperature (ºC)	11.6	4.3	1.5	3.5	9.4	17.2
Minimum outdoor temperature (ºC)	0.7	-4.2	-7.2	-5.5	-0.9	4.6
Average sunshine duration (h)	4.3	2.5	3.6	4.6	5.1	81
Average relative humidity (%)	75	77	75	72	66	63.4

In the experiments carried out, the water mass flow rates of the evaporator are; 1500 L/h, 1700 L/h and 2000 L/h, respectively. However, the condenser values are; 1700 L/h, 2000 L/h and 2300 L/h. The required water mass flow rates were determined at certain evaporator and condenser loads in Eq. (1) and Eq. (2).

$$\dot{Q}_{evp} = \dot{m}_{e,w}\, c_{p,w}\, (T_{evp,wi} - T_{evp,wo}) \tag{1}$$

$$\dot{Q}_{con} = \dot{m}_{c,w}\, c_{p,w}\, (T_{con,wo} - T_{con,wi}) \tag{2}$$

Besides, fan coil devices have 3-stage fan settings. Thus, 3 different parameters were measured as 1st stage, 2nd stage and 3rd stage. Fan-coil devices consume 237 W at first stage and have 620 m3/h air flow, consuming 312 W at 2nd stage and having 850 m3/h air flow, consuming 452 W at 3rd stage and 1000 m3/h air flow. And with that measurements were made in 9 different conditions shown in *Table 2* to perform the energy analysis of the system.

Table 2. Experimental measurement conditions

Conditions	Water mass flow rate in evaporator (L/h)	Water mass flow rate in condenser (L/h)	Air mass flow rate in fan-coil (m^3/h)
1	1700	2000	620
2	1700	2000	850
3	1700	2000	1000
4	2000	2300	620
5	2000	2300	850
6	2000	2300	1000
7	2200	2500	620
8	2200	2500	850
9	2200	2500	1000

2.3. Uncertainty Analysis

Experimental errors and uncertainties are the result of factors such as instrument selection, instrument calibration, instrument condition, environment, observation and reading and test planning. Uncertainty analysis is needed to prove the accuracy of the experiments. An uncertainty analysis was performed using a method described by Holman. The uncertainty in the result is given as follows (Holman, 1994).

$$\omega_R = \left[\left(\frac{\partial R}{\partial x_1}\omega_1\right)^2 + \left(\frac{\partial R}{\partial x_2}\omega_2\right)^2 + \cdots + \left(\frac{\partial R}{\partial x_n}\omega_n\right)^2\right]^{1/2} \tag{3}$$

The measurements recorded during the experiment were carried out and their uncertainty values were calculated as follows:

a) The volumetric flow rate of the primary and secondary water loops was measured using a flowmeter (Gentek Lzs-25, 300-3000 L/h) and the uncertainty was calculated as ±4.26% based on the catalog data.

b) The inlet and outlet temperatures of water and refrigerant, as well as the ambient temperatures were measured by K-type thermocouple with ±0.75% accuracy. The measured temperatures were recorded by a 12-channel datalogger with ±0.3% accuracy.

c) Condensation and evaporation pressures of the refrigerant were measured by bourdon tube pressure gauges (Refco M2, accuracy ±1.6%)

d) Electrical power input of compressor, circulation pump, submersible pump and fan-coils were calculated by using a digital clamp multimeter (UNI-T UT200A, Current: 2A/20A/200A, Voltage: 600V, Display count: 1999, accuracy ±1.5%).

e) The total power input of the system was measured by a digital electricity meter (MAKEL Class: T510.2251, accuracy ±0.1 %).

The following equations can be used to calculate $\mathrm{COP_{hp}}$ and $\mathrm{COP_{sys}}$ uncertainty estimates (Holman, 1994).

$$\omega_{COP_{hp}} = \left[\left(\frac{\partial COP_{hp}}{\partial \dot{m}} \omega_{\dot{m}} \right)^2 + \left(\frac{\partial COP_{hp}}{\partial c_w} \omega_{c_w} \right)^2 + \left(\frac{\partial COP_{hp}}{\partial T_{con,wi}} \omega_{T_{con,wi}} \right)^2 + \left(\frac{\partial COP_{hp}}{\partial T_{con,wo}} \omega_{T_{con,wo}} \right)^2 + \left(\frac{\partial COP_{hp}}{\partial \dot{W}_c} \omega_{\dot{W}_c} \right)^2 \right]^{1/2} \tag{4}$$

$$\omega_{COP_{sys}} = \left[\left(\frac{\partial COP_{sys}}{\partial \dot{m}} \omega_{\dot{m}} \right)^2 + \left(\frac{\partial COP_{sys}}{\partial c_w} \omega_{c_w} \right)^2 + \left(\frac{\partial COP_{sys}}{\partial T_{con,wi}} \omega_{T_{con,wi}} \right)^2 + \left(\frac{\partial COP_{sys}}{\partial T_{con,wo}} \omega_{T_{con,wo}} \right)^2 + \left(\frac{\partial COP_{sys}}{\partial \dot{W}_c} \omega_{\dot{W}_c} \right)^2 + \left(\frac{\partial COP_{sys}}{\partial \dot{W}_{cp}} \omega_{\dot{W}_{cp}} \right)^2 + \left(\frac{\partial COP_{sys}}{\partial \dot{W}_{sp}} \omega_{\dot{W}_{sp}} \right)^2 + \left(\frac{\partial COP_{sys}}{\partial \dot{W}_{fc}} \omega_{\dot{W}_{fc}} \right)^2 \right]^{1/2} \tag{5}$$

The total uncertainties ratios of COP_{hp} and COP_{sys} were calculated as 4,88% and 5,16%, respectively.

3. Results and discussions

The experimental measurements of the GWHP system in the Erzincan province were carried out during the heating season of 2016. As already mentioned, the COP values of the heat pump and the whole system are calculated by making measurements according to the fixed flow rates held in 9 different conditions. The variation of COP values of the heat pump and the system over the hours in the day for 9 different conditions is shown in *Figures 3 and 4*. COP_{hp} varies between 3.03 and 4.51 for January 14 to January 22. On the other hand COP_{sys} varies between 2.12 and 2.98 for January 14 to January 22. The change in the heating load of condenser for the 9 different conditions is shown in *Figure 5.*

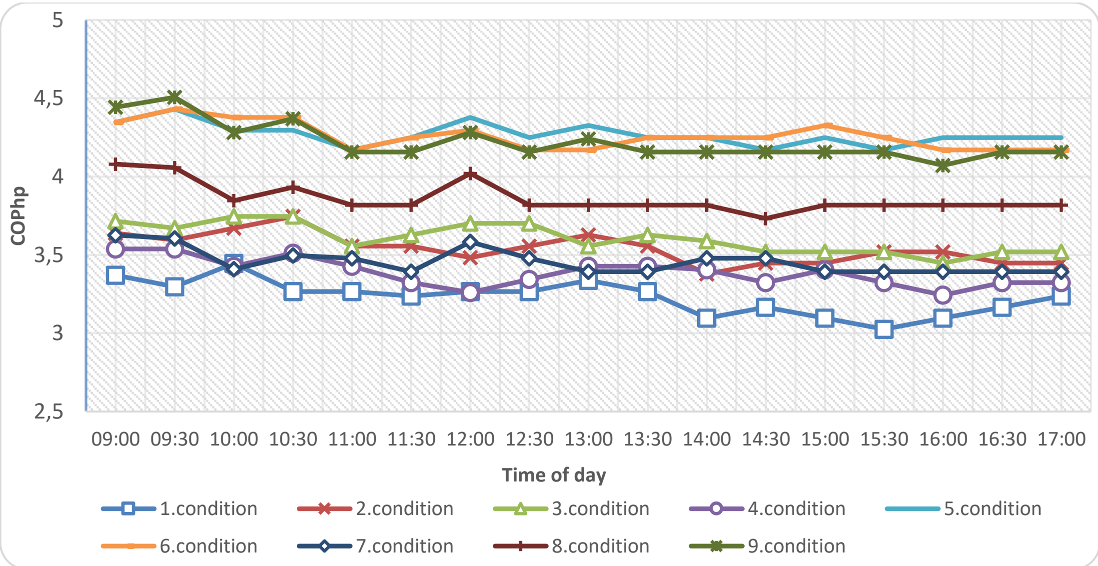


Figure 3. Variation in the value of COP_{hp} according to time of day

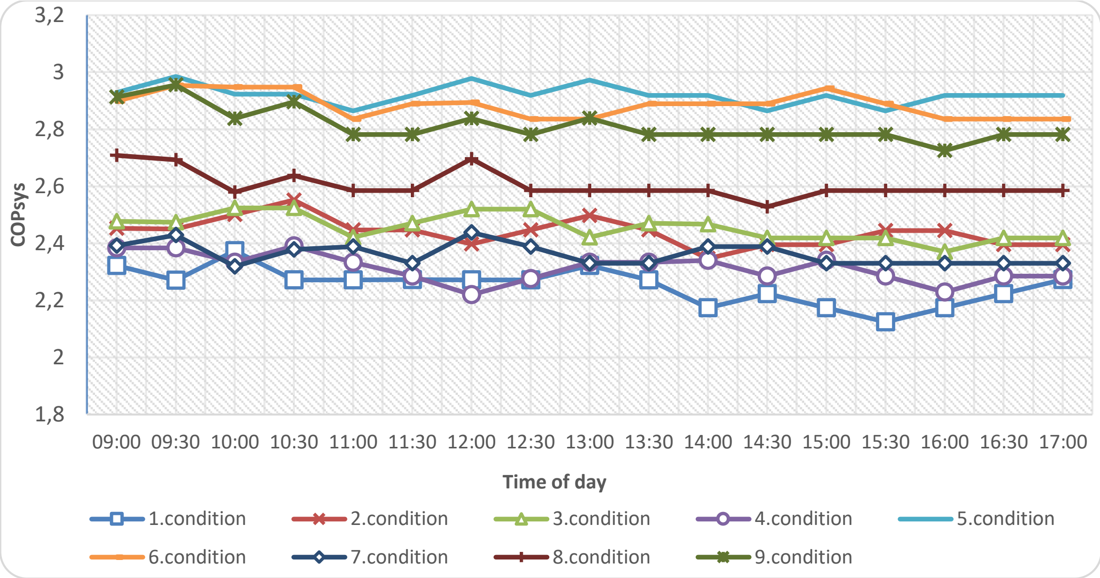


Figure 4. Variation in the value of COP_{sys} according to time of day

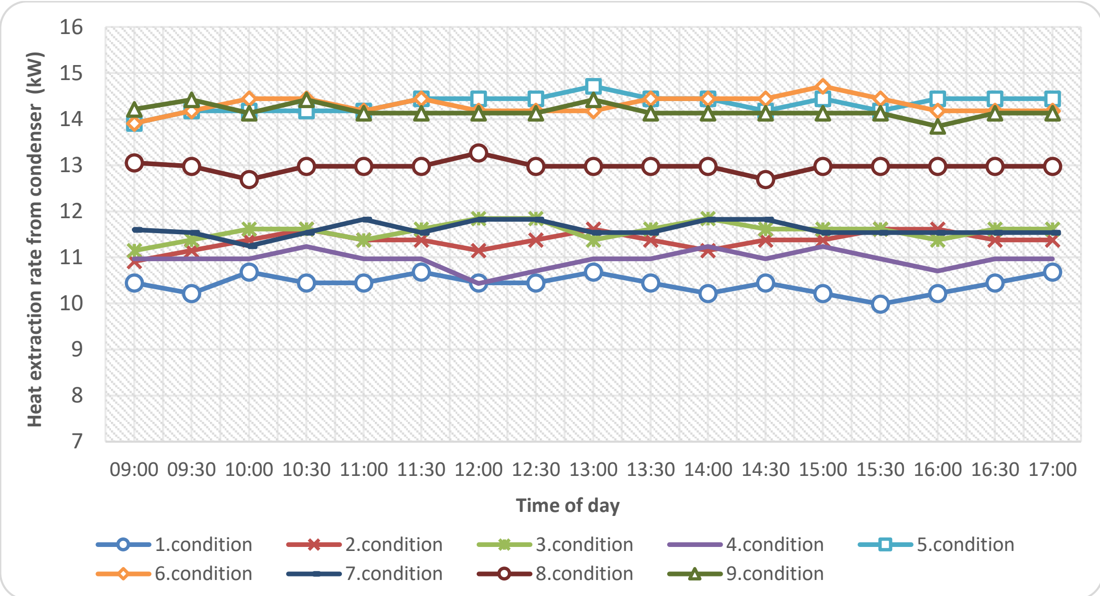


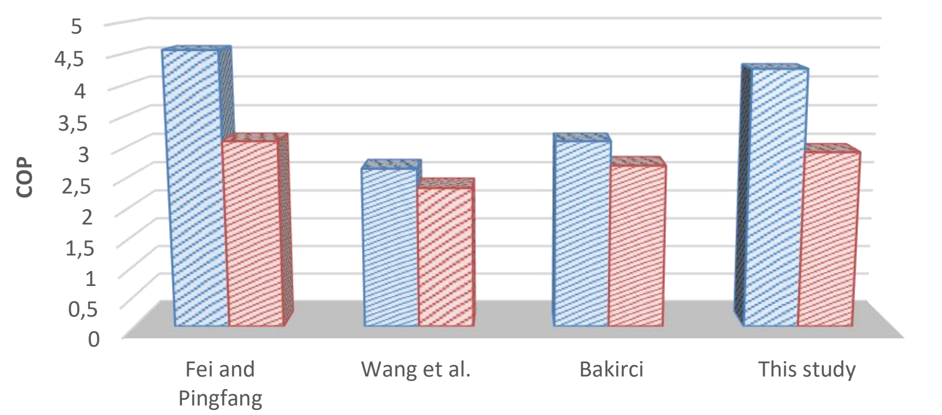
Figure 5. Variation of the amount of discarded heat from the condenser according to time

In the experimental measurements made, the most efficient result was obtained in the 5th condition. The average COP_{hp} and COP_{sys} were found to be 4.27 and 2.92, respectively. The average amount of heat drawn from the condenser is calculated as 14.3 kW. The average values of the measured parameters are shown in *Table 3* in the 5th condition that yields the most efficient results.

Table 3. The average values of the measured parameters in the 5. condition

Measurement parameters	**Unit**	**Value**
Condenser inlet temperature of the refrigerant ($\mathbf{T}_{con,ri}$)	ºC	77,9
Condenser outlet temperature of the refrigerant ($\mathbf{T}_{con,ro}$)	ºC	46,6
Evaporator inlet temperature of the refrigerant ($\mathbf{T}_{evp,ri}$)	ºC	0,1
Evaporator outlet temperature of the refrigerant ($\mathbf{T}_{evp,ro}$)	ºC	8,2
Superheating heat exchanger outlet temperature of the refrigerant ($\mathbf{T}_{hex,ro}$)	ºC	43,5
Compressor inlet temperature of the refrigerant ($\mathbf{T}_{c,ri}$)	ºC	13,8
Condenser inlet temperature of the water ($\mathbf{T}_{con,wi}$)	ºC	41,7
Condenser outlet temperature of the water ($\mathbf{T}_{con,wo}$)	ºC	47,0
Evaporator inlet temperature of the water ($\mathbf{T}_{evp,wi}$)	ºC	13,0
Evaporator outlet temperature of the water ($\mathbf{T}_{evp,wo}$)	ºC	8,1
Evaporator pressure ($\mathbf{P}_{evp}$)	bar	6,0
Condenser pressure ($\mathbf{P}_{con}$)	bar	29,5
Current of the compressor ($\mathbf{A}_{c}$)	A	6,3
Current of the submersible pump ($\mathbf{A}_{sp}$)	A	6,5
Current of the fan-coils ($\mathbf{A}_{fc}$)	A	1,45
Current of the circulation pump ($\mathbf{A}_{cp}$)	A	0,68
Voltage of the compressor ($\mathbf{V}_{c}$)	V	384
Voltage of the submersible pump ($\mathbf{V}_{sp}$)	V	220
Voltage of the fan-coils ($\mathbf{V}_{fc}$)	V	220
Voltage of the circulation pump ($\mathbf{V}_{cp}$)	V	220
Power input to the compressor ($\dot{\mathbf{W}}_{c}$)	kW	3,36
Power input to the system ($\dot{\mathbf{W}}_{sys}$)	kW	4,91
Heat extraction rate from condenser ($\dot{\mathbf{Q}}_{con}$)	kW	14,3
$\mathbf{COP}_{hp}$	-	4,27
$\mathbf{COP}_{sys}$	-	2,92

GWHP is more efficient and economical than other types of heat pumps as long as it has sufficient drilling water level and water quality. Because the ground-water temperature varies very little over the year. Since the primary-loop is the open system, the system is always fed with fresh water. Observed in this study that the average ground-water temperature was maintained at 13 ºC. In GSHP systems, the water/antifreeze temperature entering the evaporator is both low and the water/antifreeze temperature is gradually decreasing when the soil does not supply enough heat. This leads to inefficient operation of the system (Ozyurt and Ekinci, 2011). It is seen that GWHP system can be applied in Erzincan province when some similar studies in the literature are compared with this study.



	Fei and Pingfang	Wang et al.	Bakirci	This study
COPhp	4,57	2,65	3,1	4,27
COPsys	3,1	2,33	2,7	2,92

Figure 6. A comparison of COP values of this study and some studies in the literature

4. Conclusion

In this study, a model space with a floor area of 80 m² was heated by a GWHP system in Erzincan. In this experimental work, the energy and performance analysis of the heat pump and the entire system were carried out. As a result of the experiments performed under the conditions of 9 different water and air flow rates, the highest yield was obtained at the 5th measurement. Average COP_{hp} and COP_{sys} are 4.27 and 2.92, respectively. The average amount of heat extracted from the condenser is 14.3 kW. Condenser outlet temperature of the water ($T_{con,wo}$) was recorded as 54.1 ^{o}C maximum in condition 1. In the 5th condition where the maximum efficiency is obtained, the condenser water temperature has worked in the regime range of 47.0 - 41.7 ^{o}C on average. No significant change in the ground-water temperature was observed during the measurement period. The measured minimum and maximum ground-water temperatures were recorded as 12.7 ^{o}C and 13.0 ^{o}C, respectively. It was also observed that the sand particles in the ground-water occasionally clogged the filter in the evaporator water inlet and lowered the water flow. Therefore, it has been fixed that the filter should be cleaned in certain periods. As a result of the experimental study, the operating parameters of the GWHP system for Erzincan province have been determined.

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