EXPERIMENTAL INVESTIGATION ON THE EFFECT OF SOIL TYPE TO THE GROUND SOURCE HEAT PUMP'S PERFORMANCE AND ENERGY CONSUMPTION

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In the present study an experimental investigation is carried out to predict the performance of ground source heat pump systems with using different soil type under Karabük prevailing conditions. A series of experiments were conducted on designed and produced experimental test rig. This study examines the effect of five different soil types on the performance and energy consumption of a heat pump. The experimental analysis showed that the evaporator capacity provided by sand was 46% and 42% higher than the capacity provided by red soil at the air flow rates of 0.087 kg/s and 0.015 kg/s, respectively. In terms of the condenser capacity, sand provided 46% and 30% higher capacity than red soil at the air flow rates of 0.087 kg/s and 0.015 kg/s, respectively. On the other hand, red soil consumed 8% and 6% less energy than sand at the air flow rates of 0.087 kg/s and 0.015 kg/s. The capacities provided by other soil types and their energy consumption ranged between these values. In terms of the COP values, red soil provided 6% higher performance than sand and humus soil at the air flow rates of 0.087 kg/s and 0.015 kg/s. The performance values obtained with other soil types ranged between these values.

Keywords: *COP, Energy Consumption, Ground Source, Heat Pump, Soil Types.*

1. Introduction

Decreasing fossil fuel reserves and the harmful effects of their emissions on the environment, together with the increasing costs of energy consumption and generation due to the rapidly increasing human population necessitate the more efficient use of energy. Heat pump systems are one of the best alternatives among the heating and cooling systems due to their certain features such as using energy more efficiently and being environmentally friendly. The primary heat sources of heat pumps are air, water, soil and ground water[1]. Ground source heat pump (GSHP) systems are attractive because the ground temperature does not change much from the surface to the bottom, and it changes less throughout the year compared to air and water. Besides, the ground is warmer than air in the winter, but cooler than air in the summer, which also makes this type of heat pumps desirable. Moreover, heat pumps that use ground as a heat source provide heating and cooling with much less energy than the air-source heat pumps. Ground source heat pumps can also provide the ideal conditions for heating and cooling in the most cost-effective way by using the energy in the ground and without giving any harm to the environment [2-3]. In recent years, ground source heat pumps have begun to be used commonly due to the use of low-carbon and renewable energy resources for meeting the heating and air conditioning needs of buildings. The concept of GSHP system was first proposed in 1912 [3]. Many theoretical and experimental studies have been conducted to this date and the studies are still ongoing [4-5]. Due to the their higher efficincy [6-7] compared with the conventionel systems heat pump technologies are gerat important. To reduce the energy consumption of the heating and cooling systems, GSHP technology has been widely used in all around the world[8-9]. A ground source heat pump using R744 refirigerant was proposed by Hu et al [10] in order to decrease the earth's energy imbalance degree. Both air cooled and water cooled gas coolers performance were examined numerically in their study. The investment and operation costs of the proposed R744 system were found lower than that of the existing R134a systems. Emmi et al [11] studied the solar assisted ground source heat pump system, used to heat environments located in a cold climate. TRNSYS, one kind of energy simulation software were used in order to investigate energy efficiency of solar assisted ground source heat pumps in their study. The authors found that the heat collected by the solar thermal collectors and that rejected to the ground ranged between 80% and 95%. Furthermore it was reported that in their paper the seasonal energy efficiency of the heat pump was not affected to any degree when the total borehole length was halved with respect to the initial value. In another study which used TRNSYS tool to simulate the operation performances and feasibility of GSHP was performed by Liu et. al [12] for three cities (Qiqihaer, Shenyang and Beijing) located in cold climate zone in China. A minitype ground source heat pump (GSHP) system was designed and installed by Zhai et al. [13] to investigate the heating and cooling performance of the heat pump system. Experiments were performed in average temperature of soil as 18 °C for both heating and cooling mode which resulted heating capacity 20.9 kW with the COP of 3 and cooling capacity 17 kW with the COP of 3.2 respectively. Schibuola and Scarpa [14] studied on the observation of the operating performance of the invertible ground source heat pump with borehole heat exchangers which was installed in the ancient historical building of Venice. They pointed out that the invertible ground source heat pump gived the convincing results when it was compared with the traditional air source heat pumps. Besides, they presented that the rapid thermal rebalancing was occured in the borehole because of humidity of the soil and the underground water flow. Esen et al. [l5] experimentally studied on the horizantal GSHP constructed for space heating in Elazığ, Turkey. Performance and economic analysis of GSHP were performed in their study. The authours had compared the GSHP system with conventional heating methods such as electric resistance, fuel oil, liquid petrol gas, coal, oil and natural gas using an annualised life cycle cost method. They concluded that GSHP system offers an economic advantages for all heating methods except natural gas. Bi et al. [16] designed a vertical double spiral ground heat exchanger for the GSHP to investigate the temperature field using control volume method. Obtained results were compared with experimental data and the analytical results. They concluded that presented numerical model provide a design guidance for the GSHP systems. Ally et al. [17] designed a water-to-air GSHP system and reported the results of an analysis over a 12-month period. They indicated that the system yielded a COP value ranging between 3.49 and 3.75 during the heating mode and between 3.72 and 5.16 during the cooling mode. They also suggested that the performance optimization could be done through exergy analysis. Hagihara et al. [18] studied the usability of the existing wells of the traditional wooden dwellings in Kyoto as a heat source for the GSHP systems in order to meet the heating requirements of these dwellings during winter. Col et al. [19] coupled a variable capacity heat pump with a variable speed drive in order to obtain maximum energy efficiency. To do this, they equipped a GSHP with a variable speed compressor, a variable speed water pump and variable speed fans and conducted the performance tests on the GSHP system under different operating conditions in order to achieve the maximum COP value. Zhang et al. [20] used a hybrid GSHP system to meet the energy needs of the zero energy office buildings in a cold region of China. The simulation results they obtained show that heating, cooling and lighting demands of the buildings could be reduced down to 25 kWh/m^2 per year by using hybrid renewable energy systems and high performance envelopes. Besides, they reported that the COP of the GSHP system could reach 5.0 in cooling operation strategy.

Based on the above literature survey and according to the best of authors' knowledge, although there are many papers on ground source heat pumps, there have been no studies considering the GSHP performance for different types of soil. Therefore, according to the authors, the study is valuable and considerable. For this reason; the main objective of the present study is to experimentally investigate the effects of soil types on GSHP performance.

For this purpose an experimental test unit was designed and manufactured. Five different soil types from different regions of Karabük (each weighing 100 kg) were used in the GSHP system. Each soil type was tested at different air flow rates (0.015, 0.039, 0.063, 0.087 kg/s). Performance of the ground source heat pump was determined by using the data obtained from the experimental measurements and the results were depicted graphically for a detailed discussion.

2. Methodology

2.1 Experimental setup

The experimental studies was conducted in Karabük city that located in the West Black Sea Region of Turkey. The city has hot summers, but the winters are not very cold. We examined how soil type can affect the performance of a ground source heat pump in Karabük where many different soil types can be found. An experimental setup was designed and manufactured and get ready for the tests. A schematic view of the experimental test unit are given in Fig.1

In this study, a ground source heat pump was used. The GSHP system consisted of two main components, a soil tank and a ground heat exchanger. Table 1 shows the specifications of the components of this experimental system.

Table 1. Features of the experimental test unit.

Compressor	**Condenser**	**Evaporator**	**Heat exchanger**	**Recirculation pump**
Model: AE 146 220/240 V AC 160 W 50-60 Hz Piston number: 1	Model: Fin and tube Size: 0.20x 0.25x 0.05 m Outer tube diameter: 0.0048 m Internal tube diameter: 0.0038 m 230 W	Material plastic pipe Lenght: 3.8 m Operating press.: 4 bar Bursting press.: 10 bar Operat. temp.: - 5 °C /+ 60 °C outer diameter : 17 mm internal diameter: 22 mm	Model: MIT B3-095 Heat trans. area: 0.095 m^2 Design pressure:30 bar Test pressure: 45 bar Design temp.:-196/+200 °C Heat load: 30-200 kW	Model: B20 – 6A Operating voltage: 220 V Power: 40 W Frequency: 50-60 Hz Volumetric flow rate: 3 lt / min.

2.2. Test procedure

Four different parameters were measured on the experimental test unit for this study. These parameters are temperature, pressure, air velocity and energy consumption. The following temperature measurements were carried out: ambient temperature, temperature of the soil in the tank, inlet and outlet temperatures of water and compressor and condenser outlet. Type K Cr-Ni thermocouple with $\pm$0.3 °C accuracy was used for temperature measurements. Pressure was measured using an analog manometer at two different points, i.e. at the inlet and outlet of the compressor. The air velocity at the condenser outlet was measured with an anemometer. An electronic counter was used to measure the energy consumption of the system. The measurements were performed at 10-minute intervals and measurements of the parameters (the energy (Wh), temperature (°C), pressure (kPa) and air velocity (m/s)) were recorded during two hours. Based on the recorded data, the COP[21], evaporator and condenser capacities and power consumption of the system were computed for each test separately. During the tests, it was observed that the ambient temperature did not change to a considerable extent.

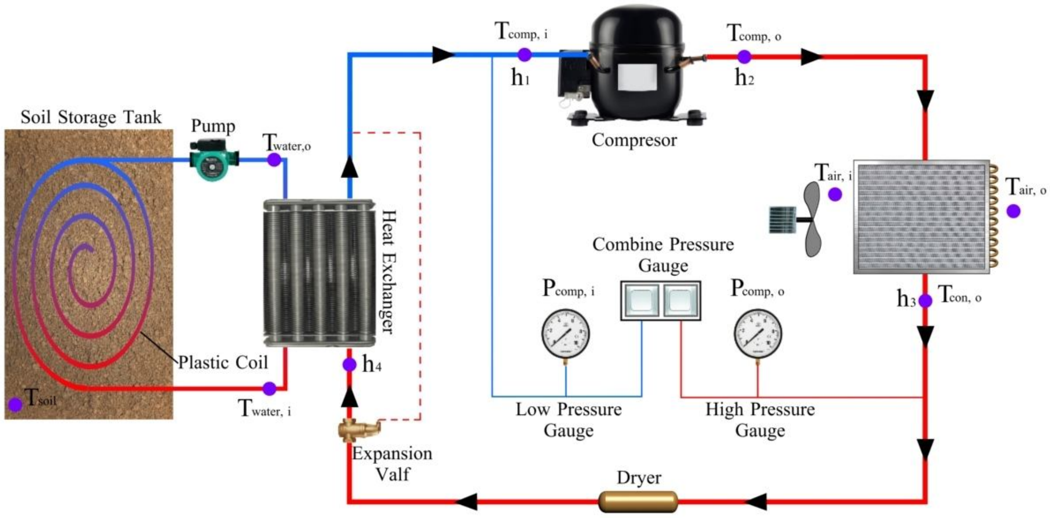

Figure 1. Schematic diagram of the GSHP system in the heating mode.

Figure 1 also shows the pressure and temperature measurement points. The cooling and heating coefficients of performance were calculated using the following equations (2.1) based on the temperature and pressure values at the measurement points.

$$COP_H = \frac{\dot{m}(h_2 - h_3)}{\dot{m}(h_2 - h_1) + \dot{W}_p + \dot{W}_f} \quad (2.1)$$

Equation (2.2) and (2.3) were used to calculate the evaporator and condenser capacities of the system.

$$\dot{Q}_E = \dot{m}c\Delta T \quad (2.2)$$

$$\dot{Q}_C = \dot{m}c\Delta T \quad (2.3)$$

where, ($\dot{m}$) indicates the mass flow rate defined as follows:

$$\dot{m} = \varrho AV \quad (2.4)$$

Table 2. Thermodynamic features of the measurement points.

Experiment	$\dot{m}$	**T_1**	**T_2**	**T_3**	**$P_2 = P_3$**	**$P_1 = P_4$**	**h_1**	**h_2**	**$h_3 = h_4$**
	kg/s		**^{0}C**			**kPa**		**kJ/kg**	
Clay Soil	0.015	36.6	104.55	69.05	1550	320	417.81	481.56	302.52
	0.039	23.35	87.1	53.60	1300	260	411.36	465.65	277.11
	0.063	21.60	77.1	48.70	1100	220	410.45	457.69	269.53
	0.087	18.50	71.55	46.65	1100	210	408.82	451.89	266.40
Mixed Soil	0.015	34.25	103.6	67.30	1500	300	416.72	459.76	299.50
	0.039	21.35	81.70	52.45	1300	260	410.32	459.76	275.32
	0.063	20.04	73.60	49.95	1200	240	409.82	452.49	271.44
	0.087	17.90	68.75	46.00	1100	220	408.50	448.90	265.41
Humus Soil	0.015	34.85	108.03	75.65	1500	300	417.00	486.31	314.34
	0.039	23.55	91.40	68.00	1500	290	411.46	467.58	300.71
	0.063	22.05	79.85	51.95	1200	240	410.68	459.24	274.54
	0.087	17.75	78.70	51.20	1100	220	408.42	459.49	273.38
Red Soil	0.015	34.75	102.80	63.35	1550	320	416.96	479.62	292.84
	0.039	19.50	79.40	52.10	1300	280	409.35	457.24	274.77
	0.063	19.05	72.40	47.25	1250	260	409.11	450.36	267.31
	0.087	13.50	65.85	45.30	1150	240	406.12	444.93	264.36
Sandy Soil	0.015	35.35	108.95	76.95	1600	340	417.24	485.86	316.76
	0.039	24.85	91.95	69.00	1550	320	412.12	467.49	302.44
	0.063	22.45	80.50	52.25	1250	260	410.89	459.20	275.01
	0.087	16.85	81.65	44.82	1150	240	414.58	461.90	275.55

While calculating the heating and cooling coefficients of performance for the GSHP system, the temperature data that would be the basis of enthalpy values were obtained by recording the temperatures at 10 minute-intervals during the two hours after the experimental setup operated in a steady state and by taking the average of these values for the last 20 minutes. Table 2 shows the thermodynamic features of the measurement points which formed the basis for the tests. In a similar way, evaporator and condenser capacities were calculated by taking the average of the values over the last 20 minutes. Outlet air temperature and air speed of the condenser were measured with an anemometer and the measured values were used in the calculations. The first and final readings of energy consumption were recorded and 1-hour energy consumption was calculated by subtracting the first reading from the final reading.

3. Results and Discussion

3.1. Performance analysis of the system

In the analysis, the effect of soil type on the performance and energy consumption of GSHPs was examined. Based on the data obtained from the tests, the amount of heat absorbed by the refrigerant from the soil as a heat source at the end of a two-hour period of operation was calculated using Equation (2.2) As shown in Figure 2, evaporator capacities were found to have increased with increasing air flow rate for all soil types. The reason is that the heat transferred from the soil was extracted by the condenser when the condenser's air flow increased. As shown in the figure, the highest evaporator capacity was observed with sandy soil, followed by humus soil, clay soil, mixed soil and red soil, respectively. The highest evaporator capacity was observed as 900.2 W with the sandy soil at the air flow rate of 0.087 kg/s. This may be attributed to the higher thermal conductivity of sandy soil than that of the other soil types. The lowest evaporator capacity was observed as 418.7 W with red soil at the air flow rate of 0.015 kg/s. At the air flow rate of 0.087 kg/s, the evaporator capacity was found to be 481.5 W with red soil. In terms of the evaporator capacity, sandy soil provided 46% and 42% higher evaporator capacity than red soil at the air flow rates of 0.087 kg/s and 0.015 kg/s, respectively. The capacities obtained with other soil types ranged between these values. This reveals that red soil has lower thermal conductivity than other soil types. Humus soil provided the second highest evaporator capacity after sandy soil, followed by clay and mixed soil. In this case, it would be appropriate to surround the evaporator with sandy soil in order to increase the evaporation performance.

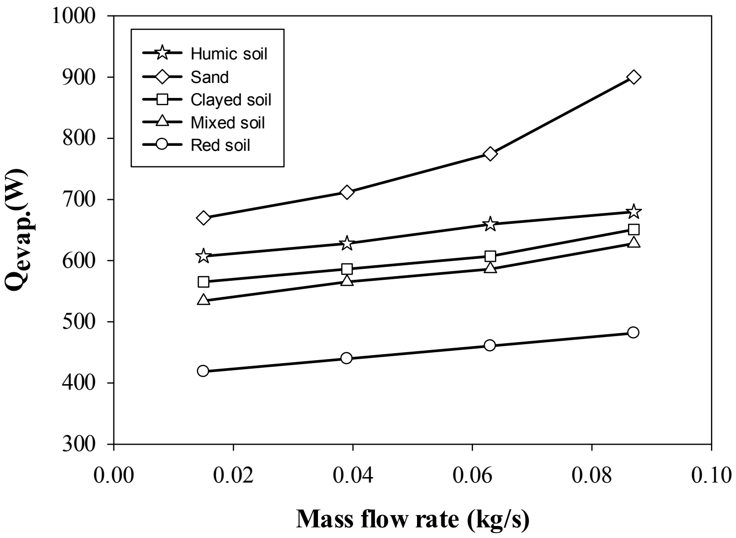

Figure 2. Evaporator capacity change versus air mass flow rate for various soil type.

Figure 3 shows the condenser capacities changing depending on increasing air flow. As can be seen in the figure, the condenser capacity increased with increasing air flow rate for all soil types. As it is known, the amount of heat transferred to air will increase as the airflow over the condenser increases. This will also decrease the condensing pressure, thus reducing the amount of energy consumed by the compressor. The heat transferred by the soil surrounding the evaporator to the water constitutes most of the heat transferred by the condenser to the ambient air.

The amount of heat absorbed by the refrigerant from the soil as a heat source and transferred by condenser to the ambient air at the end of a two-hour period of operation was calculated using Equation (2.3) The results show that the highest condenser capacity at the air flow rate of 0.087 kg/s is

1792.48 W and 953.04 W with the sandy soil and red soil, respectively. Just like the evaporator capacity, sandy soil provided 46% higher condenser capacity compared to red soil. The lowest condenser capacity was found to be 511.79 W with red soil at the air flow rate of 0.015 kg/s. In terms of the condenser capacity, sandy soil provided 46% and 30% higher condenser capacity than red soil at the air flow rates of 0.087 kg/s and 0.015 kg/s, respectively. The capacities obtained with other soil types ranged between these values. Figure 2 and 3 show that the evaporator capacities are parallel to the condenser capacities. Depending on the soil types, condenser capacities increased with increasing evaporator capacities.

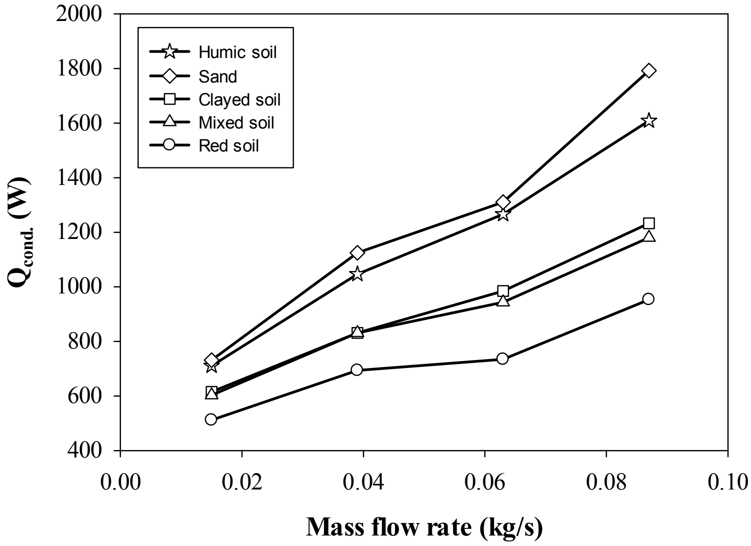

Figure 3. Condenser capacity change versus air mass flow rate for various soil type.

Figure 4 shows the energy consumption values obtained with different soil types at different air flow rates. The experimental setup was operated continuously for 2 hours and the energy consumption of the system was measured with an electronic counter. Energy consumption was found to be high with all soil types at the air flow rate of 0.015 kg/s. It was measured as 360 Wh when sandy soil was used as a heat source. The amount of energy consumed decreased with increasing air flow rate and was reduced down to 330 Wh at the air flow rate of 0.087 kg/s. With the same heat source, energy consumption was reduced by 8% with increasing air flow rate. It was also found to be 340 Wh and 309 Wh with red soil at the air flow rates of 0.015 kg/s and 0.087 kg/s, respectively. This indicates a decrease by 9%. The reason for this decrease is the increase in the airflow over the condenser. With the increasing air flow rate, the heat transfer of the condenser speeded up, thus reducing the condensing pressure in the condenser. The reduced condensing pressure caused the compressor to use less energy. Humus soil had the second highest energy consumption after sandy soil, followed by clay soil and mixed soil, respectively. The lowest energy consumption was found to be 309 Wh with red soil at the air flow rate of 0.087 kg/s. The values obtained with other soil types ranged between these values. To sum up, red soil consumed 8% and 6% less energy than sandy soil at the air flow rates of 0.087 kg/s and 0.015 kg/s, respectively.

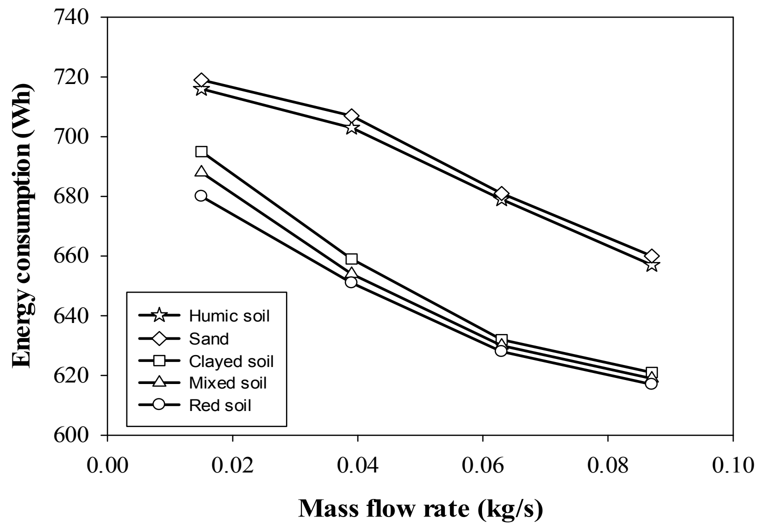

Figure 4. Changes in energy consumption versus air mass flow rate for various soil type.

Figure 5 shows the changing COP_{hp} values for different soil types used as a heat source when the experimental setup was in the heating mode at different air flow rates. While calculating the COP_{hp} values, pressure and temperature were measured from the points shown on the experimental setup for each soil type and each air flow rate, and the enthalpy values of the R134a refrigerant was found by using the software package "SOLKANE". The values obtained were used in Equation (2.2) to find the COP_{hp} values. As can be seen in the figure, the highest COP_{hp} value is 4.65 at the air flow rate of 0.087 kg/s with red soil used as a heat source. A decrease was observed in the COP_{hp} values with decreasing air flow rate. The COP_{hp} value was found to be 2.98 at the air flow rate of 0.015 kg/s with red soil used as a heat source. The reason for this decrease in the COP_{hp} value is the reduced airflow over the condenser with decreasing rotational speed of the fan. The decrease in the air flow rate made it difficult for the condenser to transfer heat to the ambient air, which gradually increased both the condensing pressure and the energy consumption. The second highest COP_{hp} value after the one obtained with red soil was observed in the experimental setup in which mixed soil was used as a heat source. The third highest COP_{hp} value was obtained with clay soil. The COP_{hp} values were found to be lowest and almost identical in the experimental setups where sandy and humus soil was used. To sum up, red soil provided 8% and 6% higher performance than sandy and humus soil at the air flow rates of 0.015 kg/s and 0.087 kg/s, respectively.

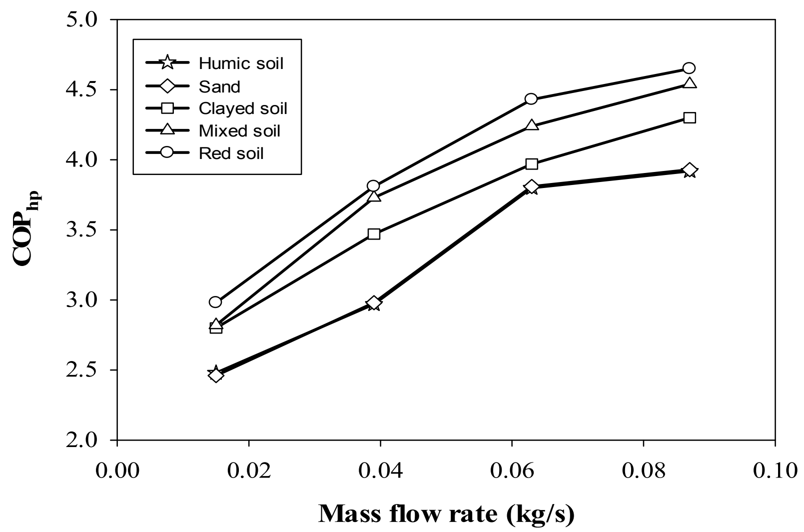

Figure 5. Variations of COP_{hp} versus air mass flow rate for various soil type.

3.2 Pressure and temperature analysis of the system

Pressure was measured with an analog manometer from two different points of the model system, i.e. discharge line and suction line. Figure 6 shows the arithmetic mean of the data obtained in the last 20 minutes of two-hour operation. Figure 5 shows that suction pressure decreases for all soil types with increasing air flow over the condenser. The reason for this decrease is the faster heat transfer from the condenser due to the increasing air flow. Another reason is the decrease in the condensing pressure. Evaporation pressure also decreases depending on the condensing pressure. As it is known, the lower the difference between the evaporation and condensing pressure of a cooling system, the more the performance of the system is. The highest evaporation pressure was observed at the air flow rate of 0.015 kg/s in the evaporator in which clay soil and red soil were used. At the same air flow rate, the lowest evaporation pressure was obtained with humus and mixed soils. Condensing pressure decreased with increasing air flow rate. In parallel with this decrease, a decrease was also observed in the evaporation pressure. At the air flow rate of 0.087 kg/s, the lowest evaporation pressure was obtained as 2.2 bar with clay and mixed soil, while the highest one was obtained as 2.4 bar with red soil. Similar results were obtained at other air flow rates. The evaporation pressure values are almost the same for all soil types. We found that soil type did not have much effect on evaporation pressure.

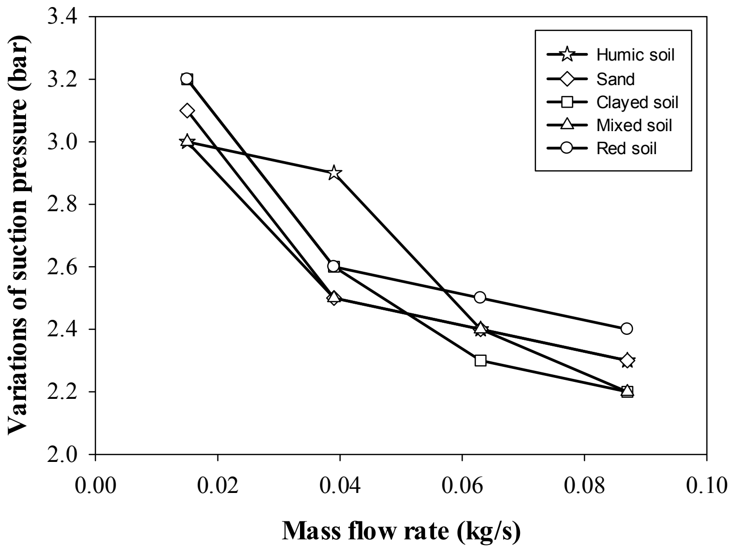

Figure 6. Changes in the suction pressure of the model system based on air flow rate and soil type.

Figure 7 shows that the discharge line condensing pressure of the experimental setup changes depending on the air flow rate and the soil type used as a heat source. As can be clearly seen in the figure, the highest discharge line pressure was observed as 16 bar at the air flow rate of 0.15 kg/s with sandy soil. At the same air flow rate, the lowest discharge line pressure was obtained as 15 bar with clay and mixed soil. Condensing pressure decreased with increasing airflow over the condenser. The main reason for this decrease is the increased amount of air over the condenser with increasing speed of the fan, thus the increased amount of heat transferred by the condenser to the ambient air. At the air flow rate of 0.087 kg/s, the lowest discharge line pressure was obtained as 10.5 bar with clay soil, while the highest pressure was obtained as 11.5 bar with sandy soil. The discharge line pressure values obtained during the tests with other soil types ranged between these values. Just like in the suction line pressure, soil type does not have much effect on the discharge line pressure. The airflow over the condenser was found to be the factor that had the most effect.

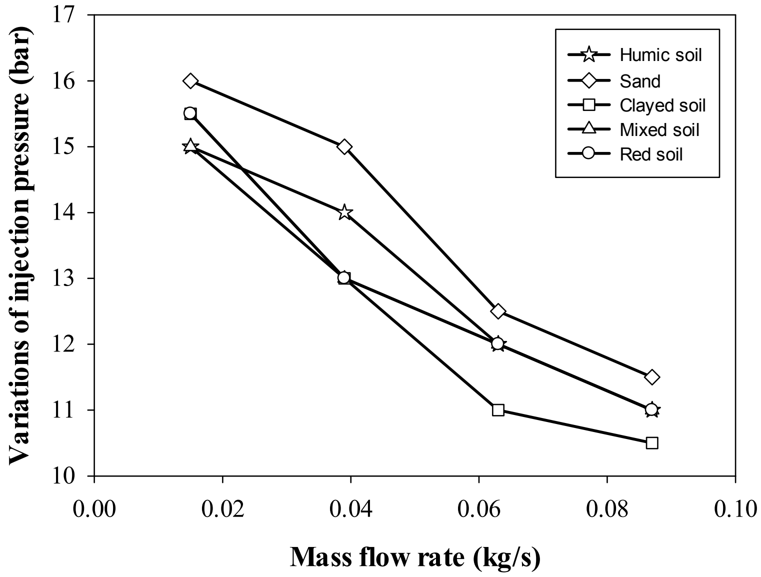

Figure 7. Changes in the discharge pressure of the model system based on air flow rate and soil type

In the model system, the water was circulated in the evaporation coils and heat was transferred from the soil to the water. Then the heat transferred to the water was moved to the refrigerant by means of an exchanger. Equal amounts of different types of soils were placed in the tank and attention was paid to maintain the soils at the same temperature. Figure 8 shows the differences in water temperature that form the basis for the amount of heat transferred from the soil to the water. At the air flow rate of 0.015 kg/s, the highest water temperature difference was found to be 3.2 ^{0}C in the experimental setup where sandy soil was used as a heat source, while the lowest temperature difference was observed as 2 ^{0}C with red soil. Water temperature difference increased with increasing air flow rate. At the air flow rate of 0.087 kg/s, the highest water temperature difference was observed as 4.3 ^{0}C again with sandy soil, while the lowest temperature difference was observed as 2.3 ^{0}C with red soil. The values obtained with other soil types ranged between these values. The highest water temperature difference was observed with humus soil, followed by clay and mixed soils. In conclusion, it can be said that more heat transfer can be provided if the area where the evaporation pipes are located is covered with sand. As it is known, sandy soil is superior to other soil types in terms of thermal conductivity.

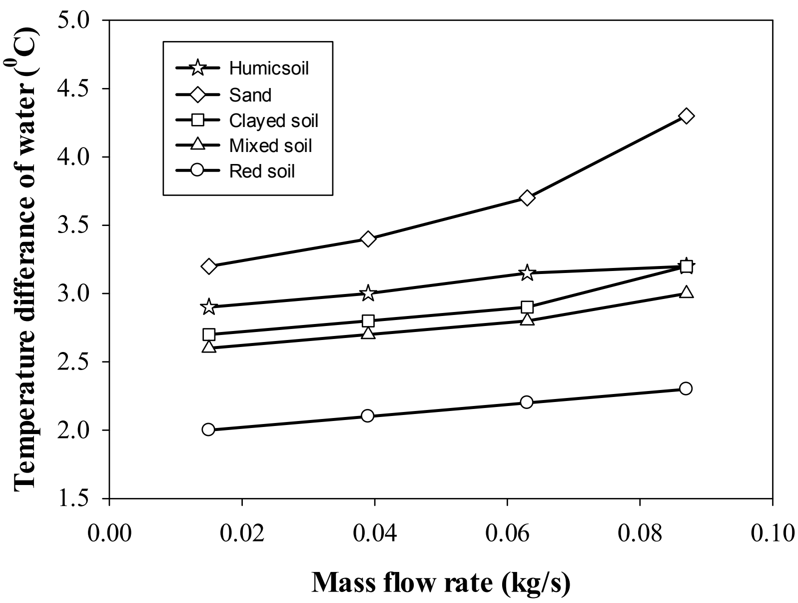

Figure 8. Water inlet-outlet temperature differences based on air flow rate and soil type.

4. Conclusions

This study experimentally examined the effects of the same amount of five different soil types obtained from different regions of Karabük on the performance and energy consumption of a ground source heat pump at four different air flow rates. The data obtained from the experimental part of the study were used and the basic parameters such as evaporator and condenser inlet-outlet temperatures, COP_{hp} values and energy consumption rates that have effect on the performance of the system were taken as basis to present the results in charts and discuss them in detail. As a result, the following can be concluded from the study:

- Condenser and evaporator capacities and the COP_{hp} values of the GSHP has increased with increase in air mass flow rate.
- Based on the soil types used as a heat source, the highest condenser and evaporator capacities at all air flow rates were obtained in the experimental setup where sandy soil was used. The lowest condenser and evaporator capacities were observed with red soil.
- At all air flow rates, the highest energy consumption was observed in the experimental setup where sandy soil was used as a heat source, while the lowest energy consumption was obtained with red soil. Energy consumption is decreasing with the increase in air mass flow rate from 0.015 to 0.087 kg/s. These decrease has occurred about 6.94% for sandy soil whereas it was found as 8.82% for red soil.
- Based on the COP_{hp} values, the highest performance at all air flow rates was obtained with red soil, while the lowest performance was obtained with humus soil and sandy soil. To sum up, red soil provided 8% and 6% higher performance than sandy and humus soil at the air flow rates of 0.015 kg/s and 0.087 kg/s, respectively.

Nomenclature

Q	- Heat obtained from a heat source and transferred to the environment desired to be heated (W)
P	- Pressure (bar)
T	- Temperature ($^{\circ}$C)
V	- Velocity (m/s)
h	- Enthalpy, heat transfer coefficient (kJ/kg)
k	- Thermal conductivity coefficient (W/m^{2}K)
$\dot{m}$	- Mass flow rate (kg/s)
$\dot{W}_p$	- Pump power (W)
$\dot{W}_f$	- Fan power (W)
c	- Specific heat (kJ/kgK)
ρ	- Density (kg/m^3)
ΔT	- Temperature difference ($^{\circ}$C)

Abbreviations

GSHP	- Ground source heat pump
GHE	- Ground heat exchanger
COP_H	- Heating coefficient of performance
$\dot{Q}_E$	- Evaporator capacity
$\dot{Q}_C$	- Condenser capacity
Comp	- Compressor
Cond	- Condenser
i	- inlet
o	- outlet

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