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Research Article

# Experimental performance evaluation of water source heat pumps in different circumstances and comparison to air source heat pumps

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## ABSTRACT

This study aims to present a novel experimental method for studying the performance of water source heat pumps which have not received sufficient attention, although this is particularly important for hot regions with great potential of hot water sources. The experimental model has special characteristics as it allows to investigate the performance of heat pumps under different operating conditions and allows a comparison between different types of heat pumps without the need to install a ground heat exchanger. The ground heat exchanger is known to be the most expensive part of any experimental model. In addition to that, it only allows to study the performance under specific conditions. The ground heat exchanger was replaced by a secondary heat pump that allows to provide an environment that simulates the different operating conditions of different types of heat pumps. It was found that water source heat pumps are more efficient than air source heat pumps with efficiency that increases with increasing water source temperature. It was found that increasing the water source temperature from 5 to 20 °C, improved the rate of heat extracted from the water source by 11.3% and the coefficient of performance by 2.8% for each degree. Another important feature of water source heat pumps is the stability of the energy flow rates, which is a guarantee of higher seasonal performance coefficients. It can be concluded that hot regions with high potential of hot water sources has valuable opportunities to invest in the field of water source heat pumps with the consequent significant energy savings.

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## INTRODUCTION

With the increasing consumption of energy, the environmental damage resulting from the use of traditional energy sources such as coal and oil has become known to all, which has led to resorting to renewable energy resources such as geothermal energy [1]. Geothermal energy is an environmentally friendly source of energy with a wide range of applications in many fields such as power generation, air conditioning, along with agricultural and industrial applications [2]. Kanoglu et al. [3] proposed the use of geothermal energy for hydrogen production and liquefaction by

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investigating six possible models and observed that as the temperature of geothermal water increases the amount of hydrogen production and liquefaction as well as energy efficiency increase. Rahman et al. [4] presented the thermodynamic analyses for a double flash-binary based integrated geothermal power plant with useful outputs of electricity, floor heating and lithium carbonate. It was found that the overall energy and exergy efficiencies are 58.41% and 66.63%, respectively.

One well-known and important application of geothermal energy is the use of heat pumps for space heating and cooling for commercial and residential buildings. The U.S. Environmental Protection Agency recognized ground source heat pumps (*GSHPs*) as being among the most efficient and comfortable systems in heating and cooling applications [5]. In fact, heat pumps have many advantages by having competitive levels of comfort, low noise levels, reduced greenhouse gas emissions. The main advantage is their reduced electrical consumption and maintenance costs compared to conventional systems [6]. In general, technology of heat pumps with their various types, whether they are air source heat pumps (*ASHPs*), ground source heat pumps (*GSHPs*) or water source heat pumps (*WSHPs*), is an efficient energy saving technology [7,8].

Heat pump systems can be described as being air source, water source, or ground coupled, according to the source of heat derived by the evaporator coil. Furthermore, the heat pumps may be used to heat air or water. Thus, heat pumps are referred to as air-to-air, air to-water, water-to-water, ground-to-water, etc. The source itself, whether air, water, or ground, may in turn be a receiver of thermal energy, or a sink, from some other sources such as industrial, waste process or renewable sources. Many authors investigated the performance of heat pumps driven by different types of heat source such as natural gas, propane, solar-heated water, or absorption systems [9-12]. Each type have its relative merits or drawbacks. Air-source heat pumps are easier to install and maintain and have a lower first cost. Ground and water-source heat pumps are more efficient, characterized by a longer lifespan and not subject to the large variations in source temperature seen by most air-source systems. The relative stability in the source temperature permits optimization of the design and, generally, achievement of higher seasonal efficiencies. Air source heat pumps have been widely studied and applied. It is well-known that their coefficient of performance (*COP*) is directly related to the air temperature. However, the instability of air temperature and humidity negatively affect the efficiency of the pumps [13].

With the emergence of the advantages of *GSHPs*, they began to receive more attention from researchers. Many researchers directed their research in analyzing *COP* of *GSHPs* and comparing with that of *ASHPs*. Koyun et al. [14] presented a thermodynamic analysis of ground-source, water to water heat pump systems for district heating and concluded that the value of *COP* of heat pump unit and the whole system was obtained to be 2.85 and 2.64 respectively. A system using the ground water with a fresh air preconditioner, abbreviated *GWHP-FAP*, is proposed in order to improve the efficiency of *GSHPs* [15]. The results showed that energy saving of the *GWHP-FAP* is between 16.6% and 39.2% in the cooling mode compared with traditional *GSHP* system. Thermoeconomic analysis of a water-to-water heat pump is realized for different evaporator and condenser conditions [16]. Results showed that unit cost of heat delivered and of entropy generation increased with increase in the volumetric flow rate of water supplied to the condenser and with increase in the inlet temperature of water supplied to the condenser.

One type of heat pumps that has received less attention from researchers is the water source heat pump *WSHP*, although it has many advantages with respect to other types of *GSHPs* [17]. According to ASHRAE classification [18], this type is considered to be one of the types of *GSHPs*. It is well known that water is a renewable, abundant, and concentrated heat storing medium that can be used by heat pumps as an energy source [19]. The use of water as a source in *WSHPs* also provides other great advantages such as high thermal performance, low initial cost, no need to ground surface area in addition to providing a stable heat source [20,21].

The water source can be seawater, river water, ground water, or wastewater [22]. Water extracted from wells usually involves high energy efficiencies [23,24], however, several factors prevent it from being widely used. Sewage water has many advantages, such as temperature stability and representing an important source of energy saving, especially when heating and cooling are provided by heat pumps. Postrioti et al. [25] presented an experimental device to assess wastewater energy potential in civilian buildings. The results showed the potential for improving plant performance in winter conditions, in relation to combined solutions with outdoor air-water heat exchangers, once a correct control system is implemented.

It is well known that air conditioning and refrigeration systems are characterized by a high consumption of power. Space heating and cooling represent 63% of total building energy demand [19]. With the development of ocean energy exploitation, the superiority of surface water heat pumps, *SWHPs*, when used for heating and cooling, with respect to *ASHPs* at a low ambient temperature in winter was investigated [26,27]. A case study of a surface water heat pump, *SWHP*, installed in the city of Venice was presented [20]. It was found that, compared to traditional plants, lagoon water allows a saving on the energy consumption higher than 20% in heating, ventilation and air conditioning (*HVAC*) applications. The *SWHP* solution is particularly attractive in coastal areas, despite some drawbacks such as the influence of seawater temperature by the outside air and the occasional occurrence of unfavorable ocean currents. The enormous thermal inertia and vertical stratification coupled with the low freezing temperature due to salinity indicate interesting chances for this solution, especially compared to *ASHP* solution, even in severe climatic conditions [28].

The effects of water source temperature and the difference between condensing and evaporating temperatures were investigated by many researchers. Xia et al [29] presented a rule of thumb that relates the *COP* changes to the change in the difference between condensing and evaporating temperatures. It states that the *COP* of the heat pump improves by 2 to 4 percent for each degree by which the evaporation temperature raised or the condensing temperature lowered. In the case of *WSHPs*, the actual performance is a function of the water source temperature which can assume a wide range of values depending on type of water, location or seasonal conditions. An example is the lagoon water temperature, a lake in Italy, which varies from 7.5 °C to 19 °C in winter and from17 °C to 28 °C in summer [30]. The same holds true for ground temperatures. Al-Hinti et al. [31] presented an inclusive experimental data for the profile of the ground temperature at various depths in the region of Zarqa, where the university in which this study was conducted is located. It was found that the seasonal cyclic environmental effect disappears starting at a depth of 5.0 m where the temperature assumes a stable throughout the year around 21°C. The effect of wastewater temperature system on the *COP* of *WSHPs* was also investigated [32]. It was found that the *COP* of a wastewater *WSHP* system in heating mode was 3.36, 3.43 and 3.69 at temperatures 20, 30 and 40 °C respectively. The effect of water source temperature and heat exchanger structure on the *COP* was investigated by [13]. It was found that a drop of one degree in the source water temperature increases *COP* by 2.3%. An improvement in the COP was noticed also by increasing in the length of each heat exchanger branch.

The application of *WSHP* systems in cold climates has been extensively investigated by many researchers. Unfortunately, not much data exists for applications of *WSHP* in hot and dry climates [33]. Jordan is a country that has high potential for geothermal water energy and nevertheless, there is no evaluation of future uses. More efforts need to be done to highlight this potential and its applications [34].

The great diversity and difference in the conditions and temperatures in which heat pumps operate in general makes it important to search for a method that allows the transition between different conditions easily and at low cost. This papers aims to introduce a novel experimental method that deals with this need effectively and is easy to implement. The method allows the investigation of the performance of the different types of heat pumps including air source and water source heat pumps, which is a types of geothermal heat pumps, in different operating conditions without the need to use the ground heat exchanger. It is known that the ground heat exchanger is the most expensive part of geothermal heat pumps, and it often allows studying the performance of heat pumps in specific working conditions. The method introduced here saves the high cost and allows studying different working conditions that cannot be achieved by installing a single ground source exchanger.

The device that was used in this study allows an easy control of the water source temperature in addition to other parameters of the heat pumps. Taking advantage of the characteristics of the device used in this study, this study aims to investigate the performance of *WSHP* systems, which are a potential technology for both residential and commercial applications, in different regions with different climatic conditions of the world. It also aims to conduct a comparison between the specifications and performances of *WSHPs* a *ASHPs*. This study also highlights the importance of promoting the use of the huge potential of geothermal energy available in poor countries especially in the field of air conditioning with the significant savings involved.

## EXPERIMENTAL SETUP

### Working principle

To investigate the effect of various parameters affecting the performance of the *WSHPs* in heating mode, a test rig was designed and built with a schematic diagram presented in Figure 1. As shown in Figure, the model installed consists of two heat pump units, the main unit, Unit 1, considered for the investigation of the performance of the system and a secondary unit, Unit 2, introduced with the task of providing a suitable ambient that simulates the water source. In heating mode, the condenser of the secondary unit is immersed in a water tank, main water tank. A water loop is installed between this tank and another tank, auxiliary tank, to provide a water at a controlled temperature that simulates the water source temperature. The evaporator of the main unit is immersed in the auxiliary tank. The condenser of the main unit is located in the conditioned space to provide the required heating capacity. Once the set point of the main tank temperature, representing the water source temperature, is fixed, the rate of heat released by the condenser of the secondary unit will be controlled to equalize the rate of heat subtracted by the evaporator of the main unit. This will ensure a maintained constant temperature value in the main tank during the test. This temperature simulates the water source temperature. By changing the set point of the tank temperature another equilibrium state will be established. Thus, the water source temperature can be reproduced easily over a wide range as needed.

The proposed design includes two major advantages. The first advantage is that it allows to conduct the study without needing to design and install the water or ground heat exchanger *GHX* which constitutes the main cost source in implementing the geothermal heat pumps. The second advantage is the flexibility provided by allowing the water source temperature to be changed over a wide range easily, allowing to track the work of heat pumps in various operating conditions.

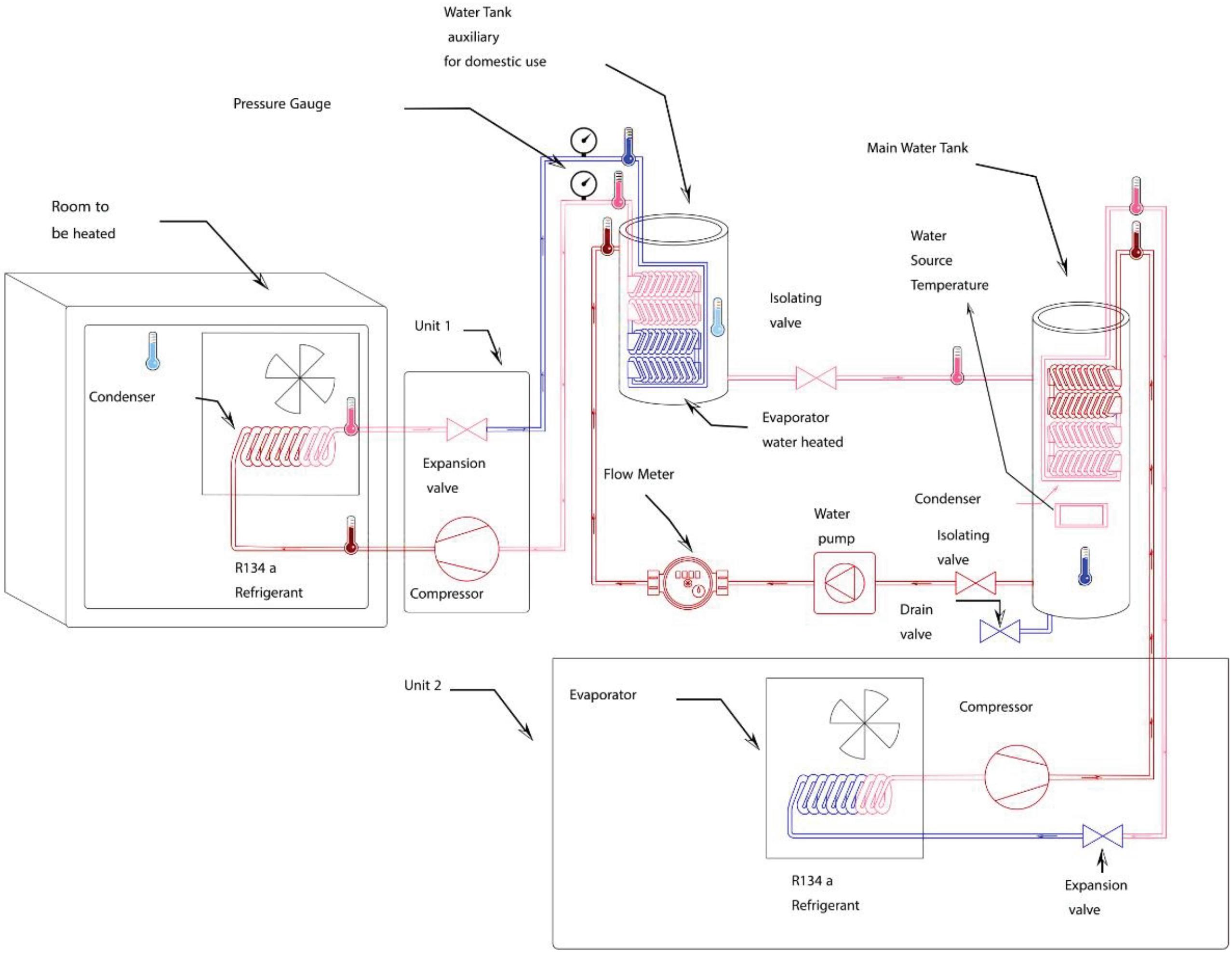


**Figure 1.** Schematic Diagram of the WSHP system in heating mode.

**Description of the System**

The experimental set-up is installed inside the laboratory of Mechanical Engineering Department at Zarqa University. The system can operate either in heating or in cooling modes. The details of the installed system are shown in Figure 2.

The refrigeration circuit is built by a closed loop copper tubes. Each of the two units consists of the same basic components of any heat pump. The compressor is reciprocating type with rated input power of 347 W. The working fluid is R-134a and the displacement volume is 8.21 $cm^3$/rev. To avoid overloading, the suction and discharge pressures of the compressor were monitored during operation of the system. The evaporator and condenser are locally-made tube and fin types. A capillary expansion valve is installed downstream from the condenser. The two water tanks designed to transfer heat between the two units are thermally insulated to reduce heat loss as much as possible. A 4-way reversing valve was installed to allow switching between heating and cooling modes.

A set of measuring devices were installed to measure the temperatures of water entering and leaving the main tank in addition to its flow rate. A single phase power analysis meter is used to measure the power consumed by the compressor and the whole system. Additional measurements were also recorded for the temperatures of refrigeration gas at inlets and outlets of heat exchangers, evaporator and condenser, and pressures at inlet and outlet of the compressor of the main cycle. The specifications of measuring devices are as follows:

- A single phase power analysis meter which can measure voltage, current, power factor, active and reactive power; accuracy is 0.2% more or less Full Scale; sampling rate is 8KHz/s and refresh 2 times/s.
- Thermostats type F/2000 used for controlling temperatures in refrigerators, freezers, air conditioners; operating range from -30°C to +30°C; rating voltage is 250V, 16A, 50HZ/60HZ.
- Digital LCD Temperature Thermometers with sensor; temperature range from -50 to +70°C; temperature display resolution is 0.1; temperature measurement

accuracy is ±1 degree C; temperature display resolution is 0.1.
- Water meter type winged single-jet dry-running; nominal flow rate is 2.5 $m^3$/h; maximum flow is 3 $m^3$/h; maximum working environment temperature is +50°C.
- Single manifold pressure gauge manometers compatible for R22 R12 R134a R404A; Accuracy is ± 1.6%; application is for refrigeration; measuring range is 0-500 psi; high quality aluminum alloy valve body.

**Experimental Procedure**

The experimental test was carried out in winter during the heating period from the start of January to the end of March. Many variables were measured and recorded every 15 minutes such as:
- Mass flow of water extracting heat from the main tank.
- Temperature of water entering and leaving the same tank.
- Temperature of refrigerant at inlet and outlet of the evaporator and condenser of the main cycle.
- Temperature of the water source tank.
- The power consumed by the compressor and circulating pump.

Before starting to record readings, the pump and compressor were switched on and the water source temperature was set at the desired value in the main tank. Then the conditioned space temperature was fixed. It is then waiting for a period of time sufficient so that the temperature in the auxiliary tank rises to assume the same of that in the main tank. Readings for the various values were recorded every 15 minutes. After completion, the water source set point was changed to another value and the experiment was repeated. The water source temperature values were changed over a wide range to cover different conditions at which *WSHPs*, *GSHPs* and *ASHPs* were running.

## MATHEMATICAL MODEL

**The Coefficient of Performance (*COP*)**

The coefficient of performance, *COP*, in heating mode is defined as the ratio between the heating capacity and the power consumed. It can be defined for the heat pump considering the power consumed by the compressor only and is given by equation (1):

$$COP_{HP} = \dot{Q}_H / \dot{W}_{comp} \tag{1}$$

The maximum heating coefficient of performance is that of Carnot heat pump, operating between condenser temperature $T_C$ and evaporator temperature $T_E$, and is given by equation (2) [30]:

$$COP_{HP,max} = T_C / (T_C - T_E) \tag{2}$$

The coefficient of performance can be defined also for the system by including, in addition to the power consumed by the compressor, the power consumed by other devices such as pump and fan. In this case it is given by equation (3):

$$COP_{sys} = \dot{Q}_H / \dot{W}_{sys} \tag{3}$$

**Figure 2.** Outside view of the experimental setup.

where $\dot{W}_{sys}$ is the total power consumed by the system.

**The rate of heat extracted from the water source**

The rate of heat extracted from the main tank is given by equation (4):

$$\dot{Q}_L = \dot{m}_w c_{p,w}\left(T_{o,w} - T_{i,w}\right) \tag{4}$$

where $\dot{m}_w$ is the mass flow rate of water, $c_{p,w}$ is the specific heat of water and $\left(T_{o,w} - T_{i.w}\right)$ is the temperature difference between outlet and inlet of the main tank. This simulates the rate of heat extracted from water source in case of *WSHP* or extracted from the outside air in case of *ASHP*.

Assuming negligible hea t loss through water loop and tanks, the rate of heat extracted by the water loop will be equal to the rate of heat absorbed by the refrigerant in the evaporator of the main cycle and is given by equation (5):

$$\dot{Q}_{eva} = \dot{m}_{ref}\left(h_{o,eva} - h_{i,eva}\right) \tag{5}$$

where $\dot{m}_{ref}$ is the mass flow rate of the refrigerant and $(h_{o,eva} - h_{i,eva})$ is the change in refrigerant enthalpy between evaporator outlet and inlet. The refrigerant mass flow was not measured during the test. Therefore, the rate of heat extracted will be calculated using equation (4).

**The rate of Heat Released to the Conditioned Space by the Fan Coil Unit**

The rate of heat released to the conditioned space is equal to the heating capacity $\dot{Q}_H$. From the air side, it is given by equation (6):

$$\dot{Q}_H = \rho_a \dot{v}_a c_{p,a}\left(T_{o.a} - T_{i,a}\right) \tag{6}$$

where $\rho_a$ is the density of the air, $\dot{v}_a$ is the volumetric flow rate of the air, $c_{p,a}$ is its specific, and $\left(T_{o,a} - T_{i,a}\right)$ is temperature difference between outlet and inlet through the fan coil unit. Neglecting the losses between the condenser and the fan coil unit, heating capacity can be approximated by heat rate rejected from the condenser of the main unit and is given by equation (7):

$$\dot{Q}_{con} = \dot{m}_{ref}\left(h_{i,con} - h_{o,con}\right) \tag{7}$$

where $\dot{m}_{ref}$ is mass flow is rate of the refrigerant and $(h_{i,con} - h_{o,con})$ is enthalpy change of the refrigerant between inlet and outlet of condenser.

The rate of heat rejected can be estimated also directly by considering the first law of thermodynamics and is given by equation (8):

$$\dot{Q}_{con} = \dot{Q}_{eva} + \dot{W}_{comp} \tag{8}$$

whereas, the compressor power was measured during the experiment.

**The Power of the compressor, fan, and pump**

The power consumed by the compressor is given by equation (9):

$$\dot{W}_{comp} = I_{comp}V_{comp}cos\emptyset \tag{9}$$

And that consumed by the fan is given by equation (10):

$$\dot{W}_{fan} = I_{fan}V_{fan}cos\emptyset \tag{10}$$

And that consumed by the pump is given by equation (11):

$$\dot{W}_{pump} = I_{pump}V_{pump}cos\emptyset \tag{11}$$

where $I$, $V$ and cos$\emptyset$ are current, voltage and power factor respectively. As the used device can measure the power values directly in addition to measuring the current and the voltage difference, the direct power values will be adopted.

## RESULTS AND DISCUSSION

To investigate the system performance, the recorded variables were collected and analyzed by using MATLAB software.

**Energy Flow Rates**

Changes with time of different energy flow rates are shown in Figure 3. Changes in rate of heat released to the space, rate of heat extracted from the water source and power consumed by the system are illustrated at temperature 17 °C, typical value of water source temperature in many locations of the word.

It can be noticed that there is a good level of stability with some fluctuations. These fluctuations can be attributed to surrounding temperature variations, to fluctuations in heating load and to sensitivity of control devices and sensors. The stability of the rate of heat extracted from the water source is a consequence of the stability of water source temperature. This is a very important feature of water source heat pumps with respect to air source heat pumps, as the energy stability flow is a guarantee of higher seasonal performance indicators. Another consequence that can be noticed is that the stability of the rate of heat extracted from the earth if combined with a relative stability of the rate of heat released to space will be positively reflected on the compressor load, which will also be stable and better performing. As it is known, the rate of heat released to the space is the sum of the rate of heat extracted from the water source and power consumed by the compressor.

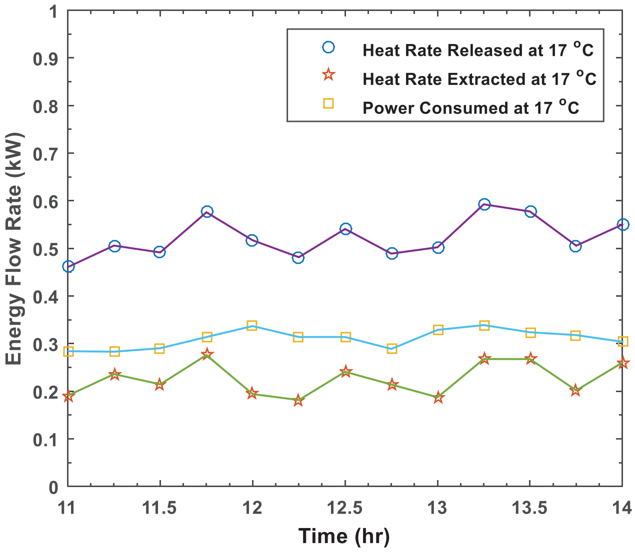


**Figure 3.** Changes of energy flow rates with time.

### Effect of Water Source Temperature on Energy Flow Rates

The source temperature is perhaps the most representative parameter of heat pumps. The effect of water source temperature on the various rates of energy flows is shown in Figure 4. Two values of temperature are considered, 17 and 5 °C. The first was chosen to represent the operation of *WSHPs* and the second was chosen to represent the operation of *ASHPs*. It can be noticed that the *WSHP* extracts more than twice the power extracted by *ASHP*. Also it can be seen that the ability of the *WSHP* to meet the heating load has also doubled, while the increase in consumed power was relatively low. Since the *COP* of the system is proportional to the heating capacity and the power consumed, it will be greatly improved. This clearly reflects the higher ability of *WSHP* to subtract more energy from the water source. Usually the temperature of water sources is ranging from 10 to 20 °C all year around according to location and season. But as changes due to seasons are limited, *WSHPs* remain consistently efficient throughout the year. It can also be noted that there is a stability in the energy flow rates even at the low value of temperature, 5 °C that represents the operating condition of *ASHPs*; In fact, the case here does not represent the real operating conditions of *ASHPs*, which is subject to many fluctuations in the source conditions, especially in the winter season. This is in fact one of the main reasons for the deterioration of the performance of *ASHPs*. This leads to another conclusion that

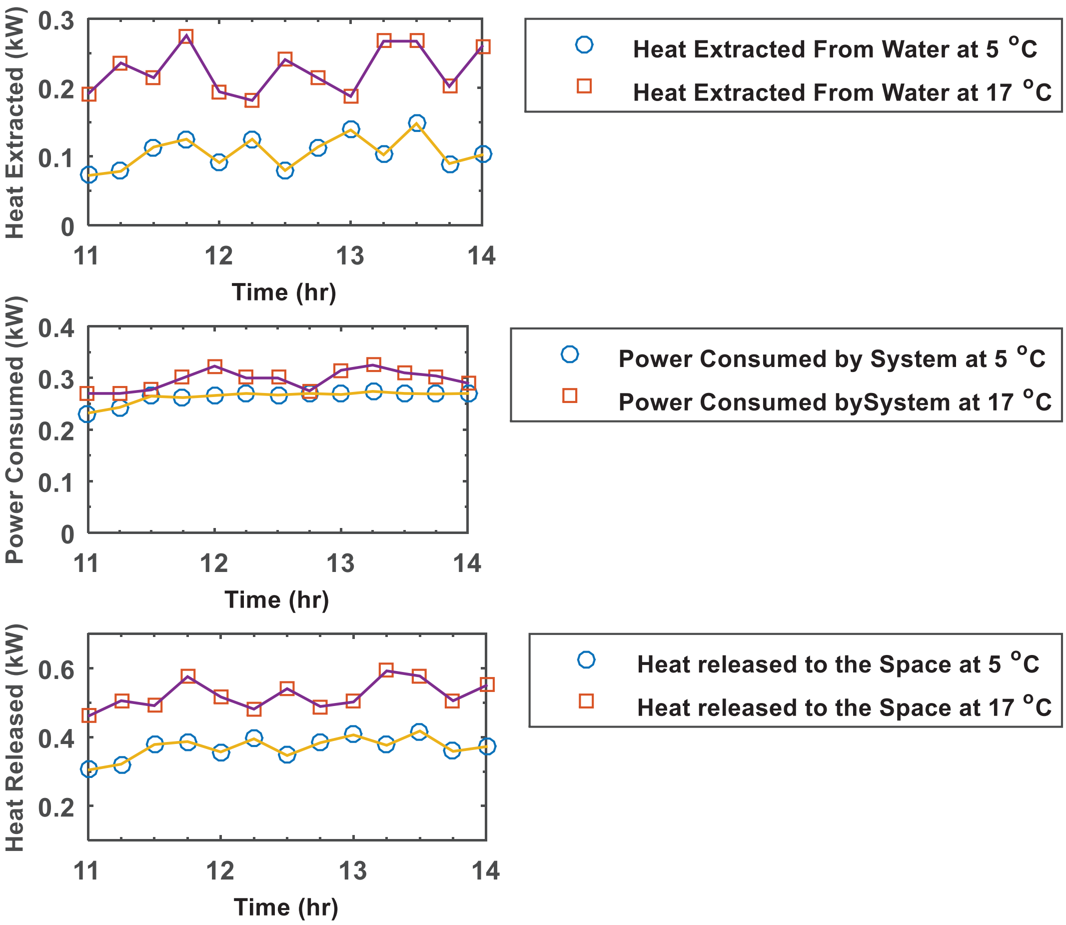


**Figure 4.** Effect of water source temperature on different energy flow rates.

*WSHPs* have better performance even if they work under the same conditions of the *ASHPs*.

### Effect of Water Source Temperature on the Daily Average Energy Flow Rates

By changing the water source temperature over a wide range as shown in Figure 5, it can be seen that both the daily average rates of extracted heat from the source and that released to the conditioned space increase steadily. As the water source temperature increases from 5 to 20 °C, the heat released to conditioned space increases from about 0.37 kW to about 0.59 kW with an increase of 59.5%. This corresponds to an average of 3.97% increase in the rate of heat released for each degree that increases in water source temperature. On the other hand, the rate of heat extracted increases from 0.106 to 0.286 with an increase of 169.8%. This corresponds to an average of 11.3% increase in the rate of heat extracted for each degree increase in water source temperature. This result is important as it shows the significant increase in the extraction potential with the increase in the water source temperature. On the other hand, the daily average value of the power consumed by the compressor is showing a slight increase. In fact, with the increase in the temperature of the source from 5 to 20 °C, the power consumed by the compressor has increased from 0.264 to 0.304 kW, an increase that does not exceed 15%. This corresponds to an average of 1% increase in the rate of heat extracted for each degree that increases in water source temperature. This result is important, as it shows that the change in the water source temperature does not have a significant effect on the compressor load. These results give a distinct advantage to regions that contain high temperature water sources in terms of possibility to establish more efficient *WSHPs*.

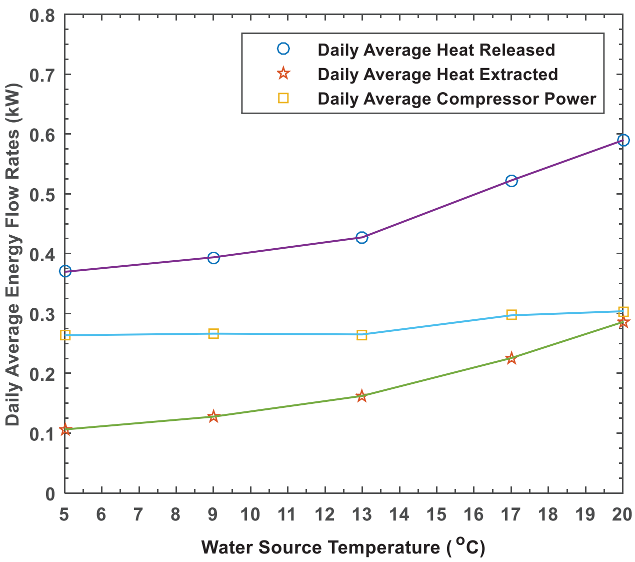


**Figure 5.** Effect of water source temperature on the daily average energy flow rates.

### Effect of Water Source Temperature on the Coefficient of Performance

The assumed values by the coefficient of performance of the system with time at the two representative values of water source temperatures, 17 and 5 °C, are shown in Figure 6. The assumed values by *COP* at 17 °C, representative of the operation of *WSHPs* range between 1.5 and 1.8 while the assumed values by *COP* at 5 °C, representative of the operation of *ASHPs* range between 1.2 and 1.4. Accordingly, the superiority of *WSHPs* with respect to *ASHPs* is clearly evident. Once again it is noted that fluctuations in *COP* values of the coefficient of performance are small and this is an expected result due to small fluctuations in the values of heating capacity and power consumed. This stability, as mentioned previously, is due to the stability at the water source temperature. The stability of the *COP* values of the system means the stability of the operation and control of the system in general. Also, it can be noticed that the fluctuations in the real working conditions of the *ASHPs* will be more significant than those appear in the figure for the reasons mentioned earlier.

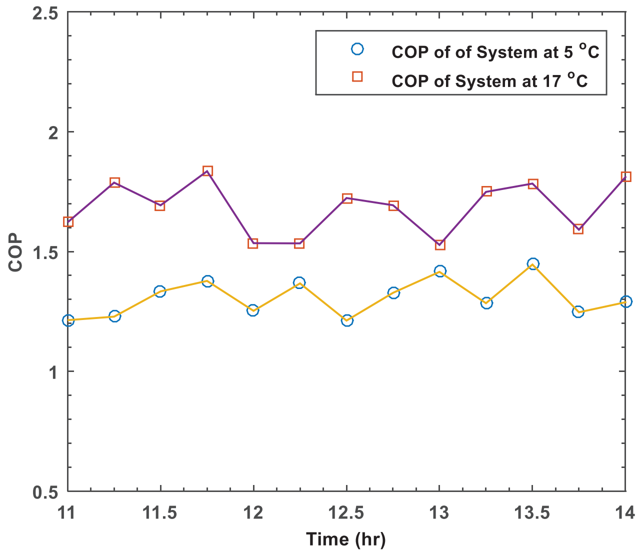


**Figure 6.** Effect of water source temperature on the coefficient of performance.

### Effect of Water Source Temperature on the Daily Average Coefficient of Performance

As shown in Figure 7, the daily average values of *COP* of the system increases with the water source temperature. The same trend is taken by *COP* of the heat pump. The daily average *COP* of the system at 5 °C is around 1.3 and increases with heat source temperature up to reach 1.68 at 17 °C. This corresponds to an improvement in performance that exceeded 29%. These results are in good agreement with the results provided by Büyükalaca et al. [35] who investigated the effect of using the Seyhan River and dam lake as a heat source/sink on the heat pump performance.

They used laboratory-scale heat pump with a dimension of 4.3×5.8×2.2 $m^3$ and a water tank of 1000 L for modeling the lake water. Their results showed that the *COP* of the *WSHP* is about 15–40 % and 35–40 % higher than that of the *ASHP* in heating and cooling mode, respectively.

It can be noticed that if the source water temperature considered is 20 °C instead of 17 °C, the system *COP* increases up to reach 1.85 with an improvement of more than 42%. This means an average of 2.8% improvement in the coefficient of performance for each degree increase in water source temperature. It can be noticed that the rate of increase in the system *COP* increases with the increase in the water source temperature. Also, it can be observed that the difference between the *COP* of the system and that of the pump tends to converge slightly with the water source temperature; this means the negative impact of the pumping power tends to diminish gradually with increasing of water source temperature.

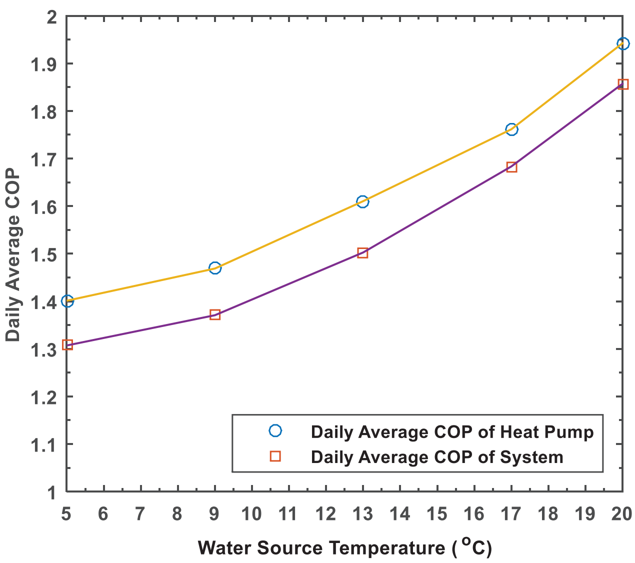


**Figure 7.** Effect of water source temperature on the daily average coefficient of performance.

## Pumping Energy Effect on the Performance

The performance of heat pumps is affected by the required pumping energy. Kavanaugh and Rafferty [36] have presented guidelines proposed as a benchmark for judging the effectiveness of pumping and piping system design. The performance was evaluated based on the ratio of the pump input power to heating capacity. In this study this ratio is decreasing with water source temperature as shown in Figure 8. This ratio at water source temperature of 5 °C is equal to 0.508 and drops to reach 0.0226 at 20 °C. According to Kavanaugh and Rafferty classification, the system is inefficient with grade E-bad at 5 °C but is getting better as becoming acceptable with grade C-medium at 20 °C. It can be concluded that the increase in the temperature of water source affects positively the performance of the heat pump by reducing the influence of pumping energy. Other researchers have reported that pumping energy should range between 6% and 7% of the total system energy used [37,38]. In this study, as can be noticed in Figure 8, this ratio is 6.7% and 4.4% at 5 °C and 17 °C respectively. So according to this criterion as well, the increased water source temperature reduces the negative effect of pumping energy on the performance of the heat pump. Also, it can be noticed that the pumping power was almost not affected by the rise in the temperature of the source, while the heating capacity and the power extracted from the water have been increased. This confirms the previously reached result that increasing the temperature of the source leads to a decrease in the negative impact of the pumping power.

## Effect of Pumping Energy on the Coefficient of Performance

Figure 9 shows the change with time of *COP* and energy flow rates of the system and the heat pump at 17 °C. The difference is due to the difference in consumed power by the whole system and the compressor alone. Obviously, the largest part of the consumption is due to the compressor. Fluctuations in the *COP* values are a predictable consequence of the fluctuations in heat flow rates and in consumed power as shown in Figure 3. The daily average *COP* values of the heat pump unit $COP_{HP}$ and the whole system $COP_{sys}$ at 17 °C, as shown in Figure 7, are 1.76 and 1.68 respectively. The percentage drop in the *COP* due to the presence of pumping energy is 4.5%. The limited fluctuations that appear in the figure, whether with regard to the energy consumed by the compressor, or the energy consumed by the system as a whole confirms the great advantage of *WSHPs* distinguished by the stability of the loads for the system as a whole and for all its components, compared to *ASHPs* that are subject to unexpected fluctuations that negatively affect their performance.

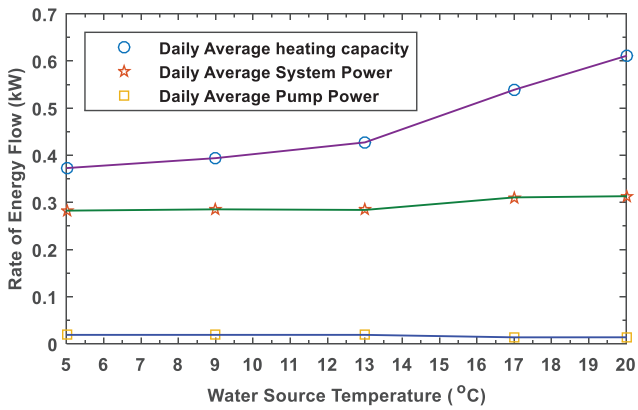


**Figure 8.** Variation of the ratio of the pump input power to heating capacity with water source temperature.

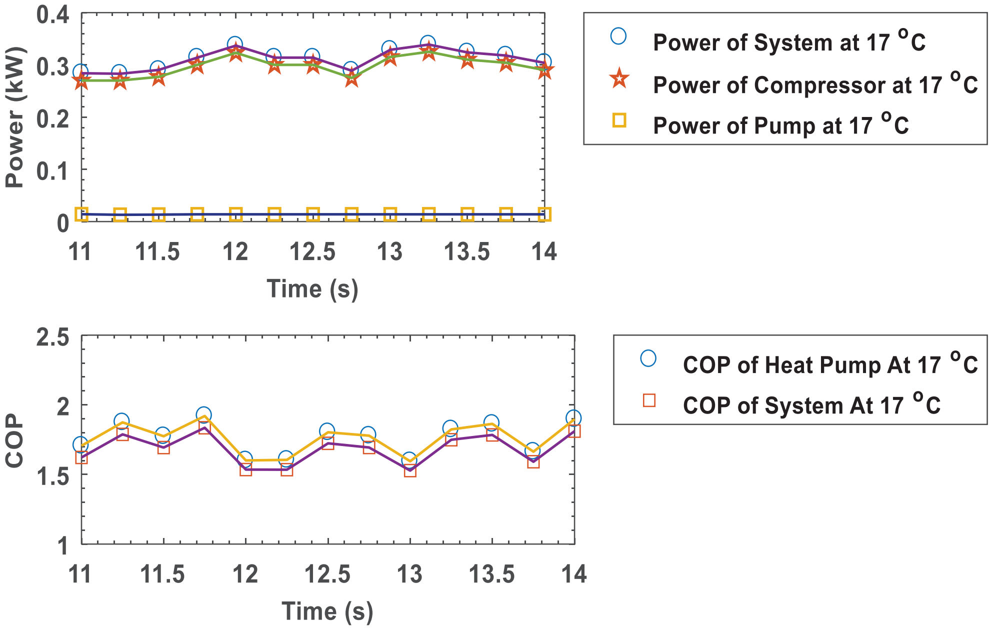


**Figure 9.** Effect of pumping energy on the coefficient of performance.

**Effect of Temperature Difference Between the Condenser and Evaporator on Performance**

It is obvious from ideal refrigeration Carnot cycle equation, that *COP* is inversely proportional with the temperature difference between the condenser and the evaporator. The obtained results in this study are consistent with the theoretical model. As can be noticed in Figure 10, the evaporator temperature increases with the water source temperature while the condenser temperature undergoes a slight variation. Thus there is a decrease in the temperature difference, and this decrease is accompanied by an increase in the *COP* of the system.

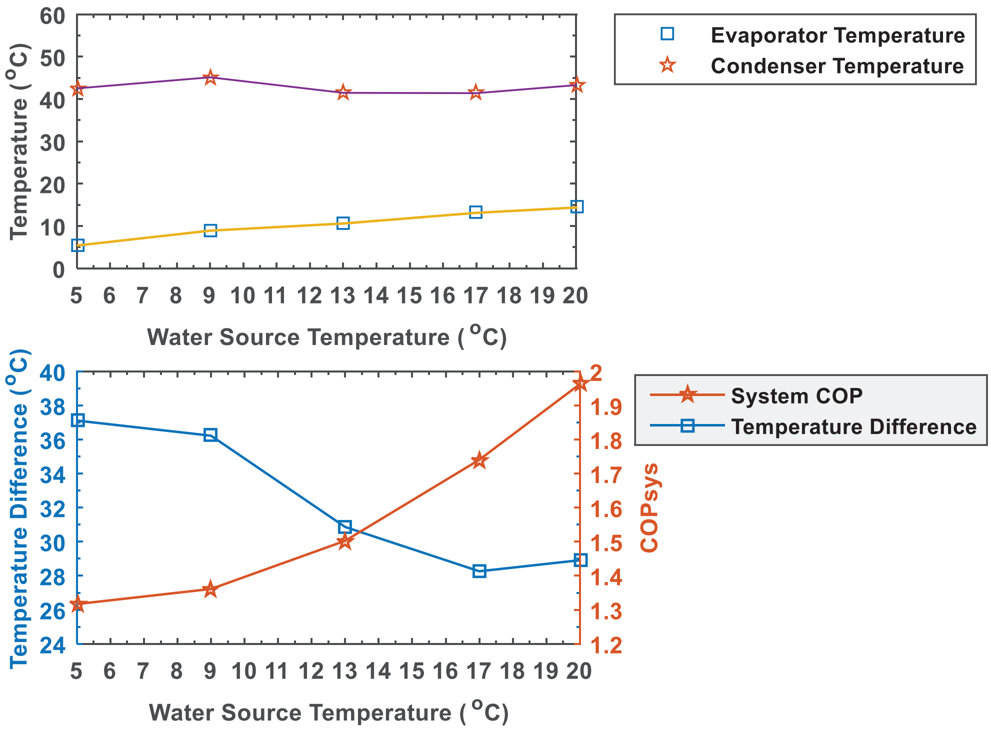


**Figure 10. Effect off temperature difference between the condenser and evaporator on system performance.**

At the two typical temperatures 5 and 17 °C, which were chosen to represent the operation of *ASHPs* and *WSHPs*, It can be noticed that the difference between the temperature of the condenser and the evaporator are 37.1 and 28.3 °C, and the corresponding system *COP* values are 1.32 and 1.74 respectively. This means a decrease of 8.8 degrees in temperature difference corresponds to 32% increase in system *COP*. Comparing these results, obtained by the actual system, with the theoretical model represented by equation (2), which provides the maximum theoretical efficiency, it can noticed that *COP* assumes at temperatures 5 and 17 °C the value 8.5 and 11.12 respectively. This means a decrease of 8.8 degrees in temperature difference corresponds to 30.8% increase in system *COP*. This result is an interesting one since the improvement of the system *COP* with the temperature difference in the ideal model is very close to the improvement in the experimental model.

Another important aspect that can be noticed is the percentage ratio between the actual value of *COP* and the maximum theoretical one, or $COP/COP_{HP.max}$%, which represents the theoretical energy potential of the system. By evaluating the energy potential at temperatures 5 and 20 degrees, it assumes the values 15.5% and 17.7% respectively. Therefore, it can also be concluded that the *WSHPs* are characterized by higher theoretical energy potential and this feature increases with increasing water source temperature.

**Effect of Heating Capacity on the Water Coil Characteristics**

An increase in heating capacity due to increase in water source temperature was accompanied by an increase in both the temperature difference between inlet and outlet of water loop and mass flow rate as can be shown in Figure 11. It can be noticed that mass flow rate increasing with a decreasing rate while temperature difference is increasing with an increasing rate. An increase of heating capacity from 0.37 to 0.61 kW corresponds to an increase in mass flow rate from 0.27 to 0.65 kg/s and an increase in temperature difference from 0.85 to 1.15 °C. Also, it can be noticed that the maximum value of temperature difference is 1.15 °C, which is very low. This can be explained by noting the

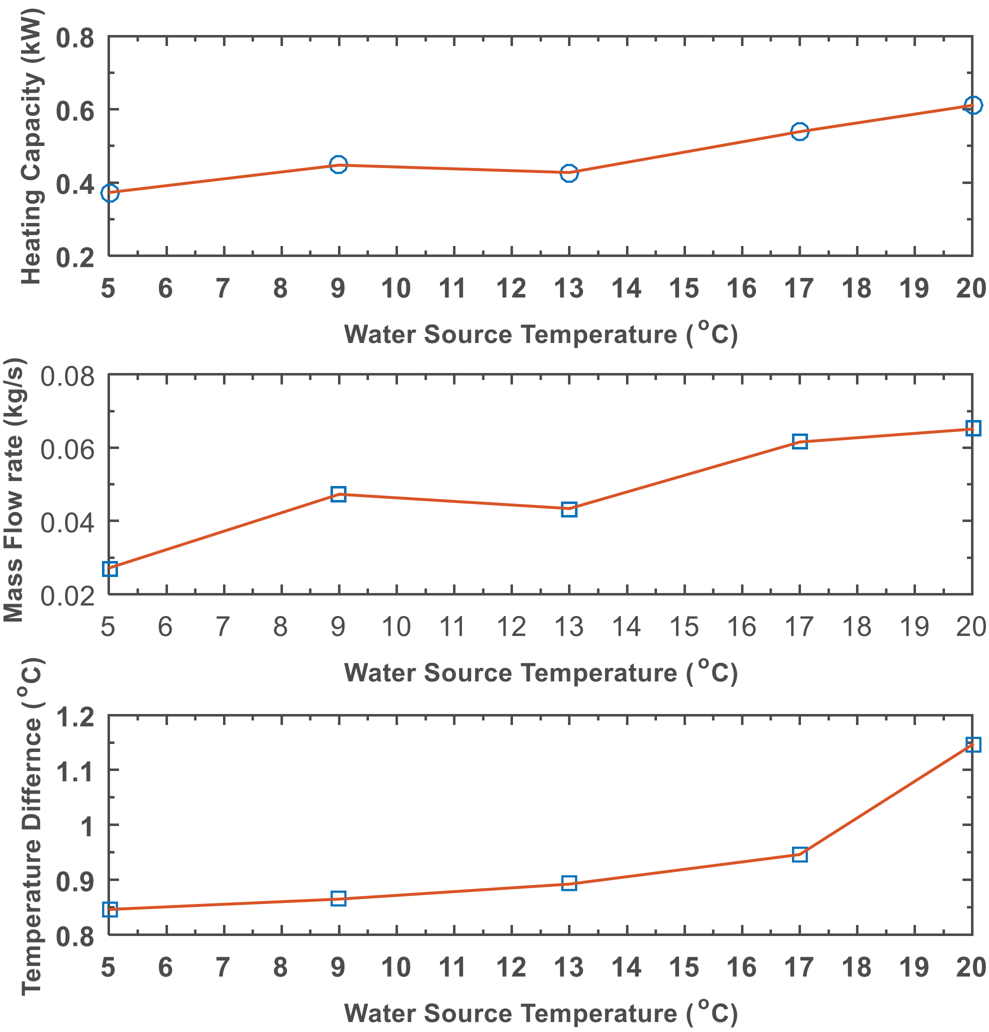


**Figure 11.** Effect of heating capacity on temperature difference and the flow rate of water loop.

short length of the water loop and the short time available for heat exchange inside the tank. This also explains the low values of the *COP* that were reached compared with values practically reachable.

Usually in the actual plants, the mass flow rate depends on the available thermal capacity of the source. Since the source temperature can be considered constant at certain condition, the power extracted depends on the mass flow rate, and therefore, the mass flow rate can be easily controlled to meet the load. As can be shown in figure, if the operating conditions change due to a change in water source temperature, the heating capacity and the temperature difference between ends of water loop change, then the function of the control system becomes the control of the mass flow rate accordingly.

## CONCLUSION

A special novel experimental model of water source heat pump system was built and tested. The model has the advantage that it allows to conduct detailed investigations of water source heat pumps in different operating conditions and compare them with other types of heat pumps without the need to install a ground heat exchanger with the consequent significant savings. The ground heat exchanger was replaced with a secondary heat pump which allows to provide a simulated flexible water source which is adaptable to fit the operating conditions of different types of heat pumps included the air source heat pumps. In light of the conducted investigation, the following conclusions can be presented:

- *WSHP* systems are a promising energy technology due to the abundance of hot water sources in many locations especially in hot regions distinguished by high water potential.
- A comparison between *ASHP* systems and *WSHP* was conducted by assigning a value of 5 °C to represent the operation of the *ASHPs* and a value of 17 °C to represent the *WSHPs*.
- It was found by increasing the water source temperature from 5 to 20 °C:
  - The power extracting potential from the water source increased by 11.3% for each degree Celsius.
  - The daily average *COP* of the system improved significantly by 2.8% for each degree Celsius.
  - The heat released to conditioned space increased by 3.97% for each degree Celsius.
- The ratio of the pump input power to heating capacity was 0.508 at 5 °C and dropped to 0.0226 at 20 °C reducing the negative impact of pumping energy on the performance of the system.
- The ratio of pumping power to the total system power improved from 6.7% to 4.4% as the water source temperature increased from 5 °C and 17 °C.
- An increase in water source temperature from 5 °C to 17 °C contributed to a decrease of 8.8 between the condensing and evaporating temperatures, an improvement by 32% in the actual value of *COP* and an improvement of 30.8% in the maximum theoretical value of *COP*.
- *WSHPs* are characterized by higher theoretical energy potential which increased from 15.5% to 17.7% with increasing water source temperature from 5 to 20 °C.
- There was a good level of stability in different energy flow rates and the performance of *WSHP* systems with some fluctuations despite major changes in environmental conditions.

## NOMENCLATURE

| | |
|---|---|
| *ASHP* | Air source heat pump |
| $c_{p,w}$ | Specific heat of water (kJ/kg.K) |
| $c_{p,a}$ | Specific heat of air (kJ/kg.K) |
| *COP* | Coefficient of performance |
| $COP_{HP}$ | Heat pump coefficient of performance |
| $COP_{sys}$ | System coefficient of performance |
| $COP_{HP,max}$ | Carnot coefficient of performance |
| $cos\emptyset$ | Power factor |
| *GHX* | Ground heat exchanger |
| *GSHP* | round source heat pump |
| *GWHP* | Ground water heat pump |
| *GWHP-FAP* | Ground water heat pump with a fresh air preconditioner |
| $h_{i,con}$ | Enthalpy at condenser inlet (kJ/kg) |
| $h_{o,con}$ | Enthalpy at condenser outlet (kJ/kg) |
| $h_{i,eva}$ | Enthalpy at evaporator inlet (kJ/kg) |
| $h_{o,eva}$ | Enthalpy at evaporator outlet (kJ/kg) |
| $I_{comp}$ | Current of compressor (A) |
| $I_{fan}$ | Current of fan (A) |
| $I_{pump}$ | Current of pump (A) |
| $\dot{m}_{ref}$ | Mass flow rate of refrigerant (kg/s) |
| $\dot{m}_w$ | Mass flow rate of water (kg/s) |
| $\dot{Q}_{con}$ | Heat rate rejected by condenser (kW) |
| $\dot{Q}_{eva}$ | Heat rate absorbed in evaporator (kW) |
| $\dot{Q}_H$ | Heating capacity (kW) |
| $\dot{Q}_L$ | Rate of heat extracted from main tank (kW) |
| *SWHP* | Surface water heat pump |
| $T_C$ | Condenser temperature (°C) |
| $T_E$ | Evaporator temperature (°C) |
| $T_{i.a}$ | Temperature at inlet of fan coil unit (°C) |
| $T_{o,a}$ | Temperature at outlet of fan coil unit (°C) |
| $T_{i.w}$ | Temperature at inlet of water tank (°C) |
| $T_{o,w}$ | Temperature at outlet of water tank (°C) |
| $V_{comp}$ | Voltage of compressor (V) |
| $V_{fan}$ | Voltage of fan (V) |
| $V_{pump}$ | Voltage of pump (V) |
| $\dot{v}_a$ | Volumetric flow rate of the air ($m^3$/s) |
| *WSHP* | Water source heat pump |
| $\dot{W}_{comp}$ | Power input to compressor (kW) |
| $\dot{W}_{fan}$ | Power input to fan (kW) |
| $\dot{W}_{pump}$ | Power input to pump (kW) |
| $\dot{W}_{sys}$ | Total power consumed by the system (kW) |

Greek symbols
$\rho_a$ Density of the air (kg/m$^3$)

## AUTHORSHIP CONTRIBUTIONS

Authors equally contributed to this work.

## DATA AVAILABILITY STATEMENT

The authors confirm that the data that supports the findings of this study are available within the article. Raw data that support the finding of this study are available from the corresponding author, upon reasonable request.

## CONFLICT OF INTEREST

The author declared no potential conflicts of interest with respect to the research, authorship, and/or publication of this article.

## ETHICS

There are no ethical issues with the publication of this manuscript.

## REFERENCES


[1] Ratlamwala TAH, Waseem S, Salman Y, Bham AA. Geothermal and solar energy-based multigeneration system for a district. Int J Energy Res 2019;43:1−22. [CrossRef]

[2] Chiasson A. Geothermal heat pump and heat engine systems. New York: John Wiley & Sons; 2016. [CrossRef]

[3] Kanoglu M, Ceyhun Y, Abusoglu A. Geothermal energy use in hydrogen production. J Therm Eng 2016;2:699−708. [CrossRef]

[4] Abdel Rahman A, Dincer I. Analysis and assessment of a geothermal based cogeneration system and lithium extraction. Int J Energy Res 2020;44:9586−9597. [CrossRef]

[5] ENERGY STAR Most Efficient 2021 - Geothermal Heat Pumps. Available at: https://www.energystar.gov/products/energy_star_most_efficient_2020/geothermal_heat_pumps Accessed on Jul 18, 2023.

[6] Urchueguia JF, Zacares M, Corberan JM, Montero A, Martos J, Witte H. Comparison between the energy performance of a ground coupled water to water heat pump system and an air to water heat pump system for heating and cooling in typical conditions of the European Mediterranean coast. Energy Convers Manag 2008;49:2917−2923. [CrossRef]

[7] Congedo PM, Colangelo G, Starace G. CFD simulations of horizontal ground heat exchangers: A comparison among different configurations. Appl Therm Eng 2012;33-34:24−32. [CrossRef]

[8] D'Arpa S, Petrosillo I, Uricchio V, Zurlini G, Colangelo G, Starace G. Heating requirements in greenhouses farming in South of Italy: evaluation of water source heat pump utilization. Energy Effic 2016;9:1065−1085. [CrossRef]

[9] Saleh A, Al-Nimr MA. A novel vacuum wastewater treatment plant integrated with a solar bsorption system. Int J Energy Res 2020;44:1685−1697. [CrossRef]

[10] Tarique S, Siddiqui MA. Performance and economic study of the combined absorption/compression heat pump. Energy Convers Manag 1999;40:575−591. [CrossRef]

[11] Azhar M, Siddiqui MA. Optimization of operating temperatures in the gas operated single to triple effect vapour absorption refrigeration cycles. Int J Refrig 2017;82:401−425. [CrossRef]

[12] Azhar Md, Siddiqui MA. Exergy analysis of single to triple effect lithium bromide-water vapour absorption cycles and optimization of the operating parameters. Energy Convers Manag 2019;180;1225−1246. [CrossRef]

[13] Zhao Z, Zhang Y, Mi H, Zhou Y, Zhang Y. Experimental research of a water-source heat pump water heater system. Energies 2018;11:1205. [CrossRef]

[14] Demir H, Koyun A, Temir G. Heat transfer of horizontal parallel pipe ground heat exchanger and experimental verification. Appl Therm Eng 2009;29:224−233. [CrossRef]

[15] Wang Z, Wang L, Ma A, Liang K, Song Z, Feng L. Performance evaluation of ground water-source heat pump system with a fresh air preconditioner using ground water. Energy Conver Manag 2019;188:250−261. [CrossRef]

[16] Aksu1B, Uysal C, Kurt H. Thermoeconomic analysis of a water to water heat pump under different condenser and evaporator conditions. J Therm Eng 2019;5:198−209. [CrossRef]

[17] Lv N, Zhang Q, Wu D, Chen Z. Surface water source heat pump air conditioning system simulation and operation performance analysis. Procedia Eng 2015;121:1880−1886. [CrossRef]

[18] ASHRAE. American Society of Heating, Refrigerating and Air Conditioning Engineers Handbook, HVAC Applications, Atlanta 2011.

[19] Valancius R, Singh RM, Jurelionis A, Vaiciunas J. A review of heat pump systems and applications in cold climates: Evidence from Lithuania. Energies 2019;12:4331. [CrossRef]

[20] Schibuol L, Scarpa M. Experimental analysis of the performances of a surface water source heat pump. Energy Build 2016;113:182−188. [CrossRef]

[21] Yu S. Introduction of water source heat pump system. In: Wang R, Zhai X (Eds.). Handbook of Energy Systems in Green Buildings. Berlin, Heidelberg: Springer; 2018. p. 473−519. [CrossRef]

[22] Sarbu I, Sebarchievici C. General review of ground-source heat pump systems for heating and cooling of buildings. Energy Build 2014;70:441–454. [CrossRef]
[23] Zhu N, Hu P, Wang W, Yu J, Lei F. Performance analysis of ground water-source heat pump system with improved control strategies for building retrofit. Renewable Energy 2015;80:324–330. [CrossRef]
[24] Sciacovelli A, Guelpa E, Verda V. Multi-scale modeling of the environmental impact and energy performance of open-loop groundwater heat pumps in urban areas. Appl Therm Eng 2014;71:780–789. [CrossRef]
[25] Postrioti L, Baldinelli G, Bianchi F, Buitoni G, Maria FD, Asdrubali F. An experimental setup for the analysis of an energy recovery system from wastewater for heat pumps in civil buildings. Appl Therm Eng 2016;102:961–971. [CrossRef]
[26] Han SK, Chae KH, Hwang DK. A design case study on sea and river water source heat pump. In Proceedings of the Society of Air-Conditioning and Refrigerating Engineers of Korea (SAREK) Summer Annual Conference, Pyeongchang, Korea, 6-8 July;2011:1212–1217.
[27] Zou S, Xie X. Simplified model for coefficient of performance calculation of surface water source heat pump. Appl Therm Eng 2017;112 (Supplement C):201–207. [CrossRef]
[28] Zheng W, Ye T, You S, Zhang H. The thermal performance of seawater-source heat pump systems in areas of severe cold during winter. Energy Convers Manag 2015;90:166–174. [CrossRef]
[29] Xia L, Ma Z, McLauchlan C, Wang S. Experimental investigation and control optimization of a water source heat pump system. Appl Therm Eng 2017;127 (Supplement C):70–80. [CrossRef]
[30] Cengel YA, Boles MA. Thermodynamics: An Engineering Approach. 8th ed. New York: McGraw-Hill; 2015. p:609–610.
[31] Al-Hinti I, Al-Muhtady A, Al-Kouz W. Measurement and modelling of the ground temperature profile in Zarqa, Jordan for geothermal heat pump applications. Appl Therm Eng 2017;123:131–137. [CrossRef]
[32] Kahraman A, Çelebi A. investigation of the performance of a heat pump using waste water as a heat source. Energies 2009;2:697–713. [CrossRef]
[33] Alshehri F, Beck S, Ingham D, Ma L, Pourkashanian M. Techno-economic analysis of ground and air source heat pumps in hot dry climates. J Build Eng 2019;26:100825. [CrossRef]
[34] Al-Zyoud S. Geothermal energy utilization in Jordanian deserts. Int J Geosci2019;10:906–918. [CrossRef]
[35] Büyükalaca O, Ekinci F, Yılmaz T. Experimental investigation of Seyhan River and dam lake as heat source-sink for a heat pump. Energy 2003;28:157–169. [CrossRef]
[36] Hepbasli A, Akdemir O, Hancioglu E. Experimental study of a closed loop vertical ground source heat pump system. Energy Convers Manag 2003;44:527–548. [CrossRef]
[37] Kavanaugh SP. Field test of vertical ground-coupled heat pump in Alabama. ASHRAE Trans 1992;98:607–616.
[38] Sulatisky M, van der Kamp G. Ground-source heat pumps in the Canadian prairies. ASHRAE Trans 1991;97:374–385.