Performance analysis of a solar-assisted ground-source heat pump system for greenhouse heating: an experimental study

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Abstract

This study investigates the performance characteristics of a solar-assisted ground-source (geothermal) heat pump system (SAGSHPS) for greenhouse heating with a 50 m vertical 32 mm nominal diameter U-bend ground heat-exchanger. This system was designed and installed in the Solar Energy Institute, Ege University, Izmir (568 degree days cooling, base: 22 °C, 1226 degree days heating, base: 18 °C), Turkey. Based upon the measurements made in the heating mode from the 20th of January till 31st of March 2004, the heat extraction rate from the soil is found to be, on average, 57.78 W/m of bore depth, while the required borehole length in metre per kW of capacity is obtained as 11.92. Design practices in Turkey normally call for U-bend depths between 11 and 13 m/kW of heating. The entering water temperature to the unit ranges from 8.2 to 16.2 °C, with an average value of 14 °C. The greenhouse air has a maximum day temperature of 31.05 °C and night temperature of 14.54 °C with a relative humidity of 40.35%. The heating coefficient of performance of the heat pump (COPHP) is about 2.00 at the end of a cloudy day, while it is about 3.13 at the end of sunny day and fluctuates between these values in other times. The COP values for the whole system are also obtained to be 5–20% lower than COPHP. The clearness index during experimental period is computed as average 0.56. At the same period, Cucumis sativus cv. pandora F1 was raised, and product quality was improved with the climatic conditions in the designed SAGSHPS. However, experimental results show that monovalent central heating operation (independent of any other heating system) cannot meet the overall heat loss of the greenhouse if the ambient temperature is very low. The bivalent operation (combined with other heating system) can be suggested as the best solution in Mediterranean and Aegean regions of Turkey.

Keywords: Performance analysis; Solar-assisted ground-source heat pump; Greenhouse heating; Izmir

1. Introduction

Good plant-growth conditions can be achieved by using greenhouses. A greenhouse can be managed to protect the plants by creating a favourable environment, allowing intensive use of soil, and helping sanitary plant control. From an economic point of view, the main objective of horticultural greenhouses is to advance the normal season production or to obtain a completely out-of-season production, which corresponds with higher crop prices. Many variables must be controlled in order to provide the good environmental conditions. The most important parameters to be controlled inside a greenhouse are temperature, humidity and light [1]. Especially, the temperature at night appears as an important critical variable to be controlled. Although different species are cultivated, the requirement is
always the same: the avoidance of low temperatures at night. The conventional solution for this problem is the burning of some fossil fuel inside the greenhouse during dangerous freezing night. Since fuel prices are high, this option is very expensive, but it is necessary, considering the possibility of the complete destruction of plants. Consumed energy for greenhouses heating purposes depends on seasons and daily changing climatic conditions. In addition to solar energy gain, greenhouses should be heated during nights and cold days.

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Taking Turkey’s geothermal energy potential into account, it may be concluded that the use of geothermal energy in heating greenhouses is very low. At present, it is estimated that only 35.7 ha of greenhouses are heated by geothermal energy in Turkey, although the country has a total greenhouse area of 22,000 ha. The majority of the geothermal greenhouse applications are in the western part of Turkey and heating capacity ranges from 3.01 to 101.72 million kJ/h [6].

When the temperature variations of Mediterranean, Aegean and Marmara regions of Turkey, where greenhouse agriculture is particularly popular, are examined, it is observed that the daily average temperatures stay below 12°C especially in December, January and February [7]. That is why the greenhouses need to be heated in certain time of the year in the Mediterranean region. However, it was found that almost none of the producers have regular heating system in their greenhouses since heating cost was on average 60–80% of total operating cost. They seem to have some basic means of heating in order to prevent the crop from the frost. This can have a considerable effect on the quality, yield and cultivation time of horticultural products. In order to establish optimum growth conditions in greenhouses, renewable energy sources should be used as much as possible. These sources are mainly solar, geothermal, biomass and wind energies.

Although solar energy is free and available, practical utilization for greenhouse heating still presents technical and economical problems. Solar energy that has been absorbed inside the greenhouse during the day covers a part of heating energy that is needed during daytime. To use solar energy for heating the greenhouse during the night, two problems are to be solved [8]: (i) the conversion of global radiation into thermal energy and, (ii) the storage of thermal energy for heating purposes during night. The problems of storage and the utilization of solar energy and the low radiation in the Mediterranean region during the production period (November to May) limited its use for greenhouses heating purposes.

Greenhouses also have important economical potential in Turkey’s agriculture. In addition to solar energy gain, greenhouses should be heated during nights and cold days. In order to establish optimum growth conditions in greenhouses, renewable energy sources should be utilized as much as possible. Effective use of heat pump with suitable technology in the modern greenhouses plays a leading role in Turkey in the foreseeable future. Although in greenhouses not only the possibility of heating but also the ability of cooling and dehumidification has been recognized, only a restricted number of practical applications have been realized [6].

In northern climates where the heating load is the driving design factor, supplementing the system with solar heat can reduce the required size of a closed-loop ground-coupling system. Solar collectors, designed to heat water, can be installed into the ground-coupled loop (by means of a heat exchanger or directly). The collectors provide additional heat to the heat transfer fluid. This type of variation can reduce the required size of the ground-coupled system and increase heat pump efficiency by providing a higher temperature heat transfer fluid [9].

During the last decade, a number of investigations have been conducted by some researchers in the design, modelling and testing of solar-assisted heat pump systems (SAHPSSs). These studies undertaken on a system basis may be categorized into three groups as follows: (i) SAHPSSs for water heating [10–12], (ii) SAHPSSs with storage (conventional type) for space heating [13–17], and (iii) SAHPSSs with direct expansion for space heating [18]. Among them, the study performed by Bi et al. [17] is noteworthy since it is...
closely associated with the present study. They performed theoretical and experimental studies on a so-called solar ground-source heat-pump system with a vertical double-spiral coil ground heat exchanger and concluded that the utilization of this heat pump system is feasible. Although various studies were undertaken to evaluate the performance of SAHPSs, as described previously, to the best of authors’ knowledge, no studies published on the experimental performance testing of a SAHPS with a 50 m vertical 32 mm nominal diameter U-bend ground heat exchanger (GHE) for greenhouse heating have appeared in the open literature.

The study reported here includes the performance evaluation of a vertical SAHPS with R-22 as the refrigerant in the heating mode. A flat-type solar collector is directly installed into the ground-coupled loop. An experimental set-up, described in the next section, is constructed and tested for the first time on the basis of a university study performed in the country. The coefficient of performance (COP) of the heat pump itself and the whole system is computed from the measurements.

2. System description

2.1. Experimental set-up

A schematic diagram and some photographs of the constructed experimental system are illustrated in Figs. 1 and 2, respectively, while its comprehensive description without solar collector and greenhouse is given elsewhere [19]. This system mainly consists of three separate circuits as follows: (i) the ground coupling circuit with solar collector (brine circuit or water–antifreeze solution circuit), (ii) the refrigerant circuit (or a reversible vapour compression cycle) and (iii) the fan-coil circuit for greenhouse heating (water circuit). The main characteristics of the elements of the solar-assisted

![Fig. 1. The main components and schematic of the SAGSHPS.](image1)

![Fig. 2. Various views of the SAGSHPS.](image2)
ground-source (geothermal) heat pump system (SAGSHPS) are given in Table 1, where the numbers in parentheses correspond to these elements as depicted in Fig. 1. Conversion from the heating cycle to the cooling cycle is obtained by means of a four-way valve. To avoid freezing the water under the working condition and during the winter, a 10% ethyl glycol mixture by weight was prepared. The refrigerant circuit was built on the closed-loop copper tubing. The working fluid is R-22. The SAGSHPS studied was installed at Solar Energy Institute of Ege University (latitude 38°24′N, longitude 27°50′E), Izmir, Turkey. Solar greenhouse was positioned towards the south along south–north. The greenhouse will be conditioned during the summer and winter seasons according to the type of the agricultural products to be raised in it.

2.2. Measurements

The following data were regularly recorded with a time interval of 15 min:

(a) Measurement of mass flow rates of the water/antifreeze solution by a rotameter.
(b) Measurement of mass flow rates of the refrigeration by a flowmeter.
(c) Measurement of temperature of the water/antifreeze solution entering and leaving the GHE and solar collector by copper–constantan thermocouples mounted on the unit water inlet and outlet lines.
(d) Measurement of condenser and evaporator pressures by Bourdon-type manometers.
(e) Measurement of ambient atmospheric pressure by a barometer.
(f) Measurement of outdoor and greenhouse air temperatures and humidities by using multi-channel cable-free thermo-hygrometer.
(g) Measurement of electrical power input to the compressor and circulating pump by a wattmeter.
(h) Measurement of inlet water temperature to and exit water temperature from fan-coil unit by copper–constantan thermocouples.
(i) Measurement of surface temperatures of glass reinforced plastics (GRP) greenhouse by inferred thermometer.
(j) Measurement of wind velocities at the height of 12 m by the Vantage Pro Meteorological Station installed in the Solar Energy Institute of Ege University, respectively.
(k) Measurement of solar flux inside and outside greenhouse by an Eppley Black and White pyranometer and the Vantage Pro Meteorological Station installed in the Solar Energy Institute of Ege University, respectively.
(l) Measurement and monitoring on a LCD display of instantaneous power consumptions of the heat pump compressor, the pumps and all electrical parameters by using electronic energy analyser.

Table 1
Main characteristics of the SAGSHP

<table>
<thead>
<tr>
<th>Main circuit</th>
<th>Element</th>
<th>Technical specification</th>
</tr>
</thead>
<tbody>
<tr>
<td>Refrigerant circuit</td>
<td>Compressor (I)</td>
<td>Type: hermetic; reciprocating; manufacturer: Tecumseh; model: TFH 4524 F; volumetric flow rate: 7.5 m³/h; speed: 2900 rpm; the rated power of electric motor driving: 2 HP (1.4 kW); refrigerant: R-22; capacity: 4.134 kW (at evaporating/condensing temperatures of 0/45°C);</td>
</tr>
<tr>
<td></td>
<td>Heat exchanger (II)</td>
<td>Manufacturer: Alfa Laval; model: CB 26-24; capacity: 6.66 kW; heat transfer surface: 0.55 m²</td>
</tr>
<tr>
<td></td>
<td>● Condenser for heating</td>
<td></td>
</tr>
<tr>
<td></td>
<td>● Evaporator for cooling</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Capillary tube (III)</td>
<td>Copper capillary tube; 1.5 m long; inside diameter: 1.5 mm</td>
</tr>
<tr>
<td></td>
<td>Heat exchanger (IV)</td>
<td>Manufacturer: Alfa Laval; model: CB 26-34; capacity: 8.2 kW; heat transfer surface: 0.80 m²</td>
</tr>
<tr>
<td></td>
<td>● Evaporator for heating</td>
<td></td>
</tr>
<tr>
<td></td>
<td>● Condenser for cooling</td>
<td></td>
</tr>
<tr>
<td>Ground coupling circuit</td>
<td>Ground heat exchanger (V)</td>
<td>Vertical-single U-bend type; bore diameter: 105 mm; Diameter of U-bends: 32 mm; of a bore diameter with a boring depth of 50 m; boring depth: 50 m; material: polyethylene</td>
</tr>
<tr>
<td></td>
<td>Brine circulating pump (VI)</td>
<td>Manufacturer: Marina; type: KPM 50; range of volumetric flow rate: 0.36–2.4 m³/h; pressure head: 41–8 m of water column, power: 0.37 kW; speed: 2800 rpm</td>
</tr>
<tr>
<td></td>
<td>Expansion tank (VII)</td>
<td>Manufacturer: Zimmet; type: 541/L; capacity: 12 l; precharge: 1 bar</td>
</tr>
<tr>
<td></td>
<td>Solar collector (VIII)</td>
<td>1.82 m², flat-type</td>
</tr>
<tr>
<td>Fan-coil circuit</td>
<td>Water circulating pump (IX)</td>
<td>Manufacturer: Marina; type: KPM 50; range of volumetric flow rate: 0.36–2.4 m³/h; pressure head: 41–8 m of water column; power: 0.37 kW; speed: 2800 rpm</td>
</tr>
<tr>
<td></td>
<td>Fan-coil unit (X)</td>
<td>Manufacturer: Aldag; type: SAS 28; Cooling/heating capacities: 3.25/9.3 kW; air flow rate: 600 m³/h</td>
</tr>
<tr>
<td></td>
<td>Greenhouse (XI)</td>
<td>GRP surface area: 48.51 m²</td>
</tr>
</tbody>
</table>
The tests were conducted on the SAGSHPS under steady-state conditions in the heating mode over the period from 20th of January till 31st of March 2004. Daily average values of 37 measurements from 8.30 a.m. to 4.00 p.m. with an interval of 15 min are given in Table 2.

Uncertainty analysis is needed to prove the accuracy of the experiments. An uncertainty analysis was performed using the method described by Holman [20]. In the present study, the temperatures, flow rates, pressure drops, voltages and amperes were measured with appropriate instruments explained previously. Total uncertainties of these measured parameters are presented in Table 2.

3. Analysis

When designing a heating system for a greenhouse, the first step is calculating how much heat energy must be supplied. The amount of heat necessary to keep the greenhouse warm can be determined by adding up the amount of heat that is lost and must be replaced. The calculation is made for heat loss under the average highest heat loss conditions. The average winter low temperature for the coldest month is used. The second step is selecting an economical and efficient heating system to supply the heat energy [21].

The present work is geared towards studying the performance of a ground-source heat pump coupled to a model-sized greenhouse to meet its heating, cooling and dehumidification requirements.

3.1. Useful solar energy

The solar radiation equations given below are taken from Duffie and Beckman [22,23]. Ratio of the instantaneous direct solar radiation on a tilted surface to the instantaneous direct solar radiation on a horizontal surface is calculated by the equation

\[ R_b = \frac{\cos(\phi - \beta) \cos \delta \cos \omega + \sin(\phi - \beta) \sin \delta}{\cos \phi \cos \delta \cos \omega + \sin \phi \sin \delta}. \]

Table 2
Measured parameters in average and their total uncertainties in average

<table>
<thead>
<tr>
<th>Item</th>
<th>Nominal value</th>
<th>Unit</th>
<th>Total uncertainty (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Average energy consumption of all systems</td>
<td>1.335</td>
<td>kW</td>
<td>± 1.02</td>
</tr>
<tr>
<td>Average power input to the compressor</td>
<td>0.806</td>
<td>kW</td>
<td>± 1.02</td>
</tr>
<tr>
<td>Power input to the brine circulating pump</td>
<td>0.059</td>
<td>kW</td>
<td>± 1.02</td>
</tr>
<tr>
<td>Power input to the water circulating pump</td>
<td>0.059</td>
<td>kW</td>
<td>± 1.02</td>
</tr>
<tr>
<td>Total power input to the fan of the fan-coil unit</td>
<td>0.048</td>
<td>kW</td>
<td>± 1.02</td>
</tr>
<tr>
<td>Current of antifreeze solution circulating pump at ground heat exchanger (GHE) side</td>
<td>0.33</td>
<td>A</td>
<td>± 1.02</td>
</tr>
<tr>
<td>Current of water circulating pump at fan-coil side</td>
<td>0.33</td>
<td>A</td>
<td>± 1.02</td>
</tr>
<tr>
<td>Maximum phase to phase voltage ($V_{LL}$)</td>
<td>407</td>
<td>V</td>
<td>± 1.02</td>
</tr>
<tr>
<td>Maximum phase voltages ($V_{LN}$)</td>
<td>232</td>
<td>V</td>
<td>± 1.02</td>
</tr>
<tr>
<td>Average phase voltage ($V$)</td>
<td>220</td>
<td>V</td>
<td>± 1.02</td>
</tr>
<tr>
<td>Average phase to phase voltage</td>
<td>380</td>
<td>V</td>
<td>± 1.02</td>
</tr>
<tr>
<td>Total maximum current ($I_A$)</td>
<td>4.93</td>
<td>A</td>
<td>± 1.02</td>
</tr>
<tr>
<td>Frequency</td>
<td>50</td>
<td>Hz</td>
<td>± 1.02</td>
</tr>
<tr>
<td>Average power factor (cos $\Psi$)</td>
<td>0.80</td>
<td>Dimensionless</td>
<td>± 1.02</td>
</tr>
<tr>
<td>Evaporation (low) pressure</td>
<td>0.425</td>
<td>MPa</td>
<td>± 3.32</td>
</tr>
<tr>
<td>Condensation (high) pressure</td>
<td>2.8</td>
<td>MPa</td>
<td>± 3.32</td>
</tr>
<tr>
<td>Condensing temperature</td>
<td>66.75</td>
<td>°C</td>
<td>± 3.33</td>
</tr>
<tr>
<td>Evaporating temperature</td>
<td>-4.74</td>
<td>°C</td>
<td>± 3.33</td>
</tr>
<tr>
<td>Temperature of water at GHE inlet</td>
<td>14</td>
<td>°C</td>
<td>± 1.59</td>
</tr>
<tr>
<td>Temperature of water at GHE outlet (solar collector inlet)</td>
<td>17.2</td>
<td>°C</td>
<td>± 1.59</td>
</tr>
<tr>
<td>Temperature of water at solar collector outlet</td>
<td>17.7</td>
<td>°C</td>
<td>± 1.59</td>
</tr>
<tr>
<td>Supply water temperature of fan-coil unit</td>
<td>52</td>
<td>°C</td>
<td>± 1.59</td>
</tr>
<tr>
<td>Return water temperature of fan-coil unit</td>
<td>42</td>
<td>°C</td>
<td>± 1.59</td>
</tr>
<tr>
<td>Volumetric flow rate of brine</td>
<td>0.0002</td>
<td>m$^3$/s</td>
<td>± 3.01</td>
</tr>
<tr>
<td>Volumetric flow rate of refrigerant</td>
<td>0.0186 x 10$^{-3}$</td>
<td>m$^3$/s</td>
<td>± 1.50</td>
</tr>
<tr>
<td>Outdoor temperature</td>
<td>13.34</td>
<td>°C</td>
<td>± 1.59</td>
</tr>
<tr>
<td>Outdoor relative humidity</td>
<td>60.47</td>
<td>%</td>
<td>± 1.02</td>
</tr>
<tr>
<td>Greenhouse inside design relative humidity</td>
<td>43</td>
<td>%</td>
<td>± 1.02</td>
</tr>
<tr>
<td>Solar radiation inside the greenhouse</td>
<td>110.87</td>
<td>W/m</td>
<td>± 1.81</td>
</tr>
<tr>
<td>Solar radiation outside the greenhouse</td>
<td>138.59</td>
<td>W/m</td>
<td>± 0.80</td>
</tr>
<tr>
<td>Wind velocity at a height of 12 m</td>
<td>3.39</td>
<td>m/s</td>
<td>± 2.00</td>
</tr>
<tr>
<td>Length (width) of the collector</td>
<td>1.94 (0.94)</td>
<td>m</td>
<td>± 1.51</td>
</tr>
</tbody>
</table>
The total irradiance on a tilted surface under clear sky conditions is computed from the following equation:

\[ I_T = R_b I_b + I_d \left( \frac{1 + \cos \beta}{2} \right) + \rho I \left( \frac{1 - \cos \beta}{2} \right). \]  

(2)

Ratio of the total radiation on a tilted surface to that on a horizontal surface is expressed as

\[ R = \frac{I_T}{I} = R_b \left( 1 - \frac{I_d}{I} \right) + I_d \left( \frac{1 + \cos \beta}{2} \right) + \rho \left( \frac{1 - \cos \beta}{2} \right). \]

(3)

A flat-plate solar collector was used in the experimental set-up. It was positioned towards the south along south–north at a tilt angle of latitude plus 15° (53°), which is the optimum angle for winter application in Izmir. Optimum tilt angle is given as \( \phi \pm 15° \) in the northern hemisphere by Duffie and Backman [23]. However, different recommendations such as \( \phi \pm 10° \) have been made for the optimum tilt angle in the heating season depending on the latitude [24,25].

The performance of SAGSHPS is studied during the cold periods. However, only clear skies and monthly average clearness index base equations are given in the present study. Monthly average clearness index \( (K_T) \) is the ratio of monthly average daily radiation on a horizontal surface \( (H) \) to the monthly average daily extraterrestrial radiation \( (H_0) \), that is, [23,26]

\[ K_T = \frac{H}{H_0}. \]

(4)

Cloudy or partially cloudy skies effects take into account. Average clearness index was found as about 0.56 for this experimental period.

The instantaneous usable energy collected by solar collector is calculated by the following equation:

\[ Q_u = F_R A_c [I_T(z\phi) - U_L(T_i - T_a)] = m_c C_{p,wa}(T_o - T_i). \]

(5)

The collector’s instantaneous efficiency is computed as follows:

\[ \eta_c = \frac{Q_u}{A_c I_T}. \]

(6)

3.2. Solar greenhouse

A heat loss calculation is the first step in determining heating system capacity before selecting the system and its various components. The heating system should be properly sized to needs of greenhouse under extreme weather conditions. The rate of heat loss from the greenhouse is calculated by the following equation [27]:

\[ \dot{Q}_{GRP} = \left[ \frac{A_1}{R_1} + \frac{A_2}{R_2} + \cdots \right] (T_1 - T_o)(f_c)(f_a)(f_s). \]

(7)

Using Eq. (6) and assuming that the construction type factor \( (f_c) \), the system factor \( (f_s) \), and the wind factor \( (f_w) \) are 1.08, 1.00, and 1.13, respectively, the average heating load of the prototype solar greenhouse considered is obtained to be 7.4 kW at design conditions. In this calculation, GRP surface area of greenhouses is 48.51 m², the thermal resistance of GRP is 0.16 m²°C/W and the temperature difference between greenhouse inside and outdoor temperatures \( (T_1 - T_o) \) is 20°C.

3.3. Ground-source heat pump system

During the cycle calculations for the GSHPs, the following assumptions were made: (i) the volumetric efficiency of the compressor was taken to be 85%; (ii) compressor isentropic efficiency was taken to be 71%, and (iii) there were no pressure losses in the cycle.

The rate of heat extracted (absorbed) by the unit in the heating mode (GHE load) \( \dot{Q}_c \) is calculated from the following equation:

\[ \dot{Q}_c = m_{wa} C_{p,wa}(T_o,wa - T_{i,wa}), \]

(8)

where \( C_{p,wa} \) is the specific heat of the water–antifreeze solution, \( m_{wa} \) is the mass flow rate of the water/antifreeze solution and \( (T_o,wa - T_{i,wa}) \) is the temperature difference between the outlet and inlet of the GHE.

The heat rejection rate in the condenser is calculated by

\[ \dot{Q}_{co} = m_c (h_2 - h_3). \]

(9)

The heat transfer rate in the evaporator is

\[ \dot{Q}_{ev} = m_c (h_1 - h_4). \]

(10)

The work input rate to the compressor is

\[ W_{comp} = \frac{m_c (h_{2_s} - h_1)}{\eta_{ie} \eta_{mc}}. \]

(11)

Hence, the COP of the GSHP can be calculated as

\[ \text{COP}_{HP} = \frac{\dot{Q}_{co}}{W_{\text{comp}}}. \]

(12)

The coefficient of performance of the overall heating system \( (\text{COP}_s) \), which is the ratio of the condenser load to total work consumptions of the compressor, the brine and water circulation pumps, and the fan-coil unit, is computed by the following equation:

\[ \text{COP}_s = \frac{\dot{Q}_{co}}{W_{\text{comp}} + W_{\text{pumps}} + W_{fc}}. \]

(13)
4. Results and discussion

In the present study, the results obtained from the experiments over the heating period from 20th of January till 31st of March 2004 were evaluated to determine the performance characteristics of the system studied. Figs. 3–7 show the variation of daily average values of outside air and greenhouse temperatures, relative humidity, solar radiation data on the horizontal surface, wind velocity at a height of 12 m and the greenhouse level for this period, and ground surface temperatures outside and inside greenhouse. Total uncertainties of the calculated parameters are illustrated in Table 3. The ambient air temperatures varied from \(-3.32\) to \(21.05\) °C, the greenhouse average temperatures from \(6.68\) to \(31.05\) °C, and the solar insolation incident on the horizontal surface at the inside and outside of the greenhouse from 0 to 199.10 W/m\(^2\) and from 0 to 248.88 W/m\(^2\), respectively. The average values of the temperature and the relative humidity for the ambient air and the greenhouse are obtained to be \(10\) °C and 60.47%, and \(20\) °C and 35.74%, respectively. The ground surface temperatures at the inside and outside of the greenhouse ranged from 4.68 to 26.35 °C, and from \(-5.32\) to 16.33 °C, respectively.

The pumping brine flow rate was found to be 0.185 m\(^3\)/h per kW of heating capacity, with a pumping power of 14.06 W/kW of heating. Table 4 illustrates benchmarks for GSHP system pumping efficiency required pump power to cooling capacity \([28,29]\). It is clear from this table that the circulator wattage for the closed loop can be categorized as acceptable systems with a good grade.

The entering water temperature (EWT) to the unit (the temperature of the water/antifreeze solution leaving the earth coil) will be lower than the normal temperature of the earth. This is due to the heat extraction from the earth to the circulating water \([30]\). The EWT is perhaps the single most representative parameter of the ground coupling effectiveness and heat pump loading. In other

![Fig. 3. Daily average hourly values of ambient and greenhouse temperatures.](image)

![Fig. 4. Daily average hourly values of ambient and greenhouse humidity.](image)

![Fig. 5. Daily average hourly solar radiation on a horizontal surface ambient and greenhouse inside.](image)

![Fig. 6. Daily average hourly wind velocity values of greenhouse level and at 12 m.](image)

![Fig. 7. Daily minimal hourly values of ground surface temperatures at the inside and outside of the greenhouse.](image)
words, the actual performance of the equipment is a function of the water temperature produced by the GHE. The average EWT over this period was approximately 14°C. The average temperature difference of water–antifreeze solution between the inlet and outlet of the GHE was obtained to be approximately 3.2°C.

Heat extraction rate is the key parameter for the GHE layout is the specific performance, i.e. the heat extraction rate in Watt per metre of borehole length. During the heating season, the rate at which heat is extracted from the ground (GHE load) was found to be on average 2.89 kW from Eq. (7). This corresponds to a heat extraction rate of 57.78 W/m of bore depth. By comparison, Sanner [31] reported that the heat rejection rate ranges from 40 to 100 W/m with a typical average of 55–70 W/m in Mid Europe. This obviously represents that the values of the heat extraction rate obtained from this study remain the range reported by Sanner.

The heating capacity of the heat pump system was obtained to be 4.194 kW. The required borehole lengths in metre per kW of heating capacity were found to be 11.92. By comparison, based on the installations in Istanbul, Turkey (1933 degree days heating, base: 18°C; 135 days cooling, base: 22°C) for horizontal loop systems, the heat exchanger circuit pipe length ranged from 39.5 to 46.7 m/kW with an average of 43.1 m/kW [32,33] as confirmed by Healy and Ugursal [34] with values of 45–55 m/kW of heat pump capacity. In addition, based on a study that covered 22 systems and was conducted by Bloomquist [35] in the United Sates, for closed-loop systems, the heat exchanger circuit pipe length ranged from 20.5 to 52.0 m/kW, with an average of 39.3 m/kW. Of those with vertical bores, the range was 14.4–17.7 m of bore per kW. As indicated above, the heat exchanger length used may maintain the heating capacity needed. However, the heat pump design should be carefully examined in order to achieve this.

Fig. 8. shows the collector efficiency as a function of the ratio of the difference between the fluid inlet and outside air temperatures to the solar insolation incident on the solar collector surface. It is obvious from this figure that the collector efficiency varied between about 0.35 and 0.70. The daily variations of COPHP and COPs for the period studied are illustrated in Fig. 9. The values for COPHP and COPs were found to be in a range among 2.00–3.13 and 1.7–2.6, respectively, while the maximum uncertainties associated with COP HP and COPs were 7.5.83% and 7.5.81%, respectively.

By comparison, in a study performed by Bi et al. [17], the total average performance in the heating season is found as follows: the heating load is 2334 W and the COP is 2.73 for the solar energy-source heat-pump system; the heating load is 2298 W and the COP is 2.83 for the ground-source heat-pump system; and the heating load is 2316 W and the COP is 2.78 for the solar ground-source heat-pump system. It may be concluded that the COP values obtained from the present study are fairly close to those reported by Bi et al. [17].

5. Conclusions

An experimental system was installed for investigating the thermal performance of a SAGSHPS for greenhouse heating. It has been satisfactorily operated without any

<table>
<thead>
<tr>
<th>Item</th>
<th>Nominal value</th>
<th>Unit</th>
<th>Total uncertainty (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mass flow rate of refrigerant (R-22)</td>
<td>0.02</td>
<td>kg/s</td>
<td>∓1.51</td>
</tr>
<tr>
<td>Mass flow rate of brine</td>
<td>0.21</td>
<td>kg/s</td>
<td>∓3.02</td>
</tr>
<tr>
<td>Solar collector efficiency</td>
<td>57.31</td>
<td>%</td>
<td>∓4.01</td>
</tr>
<tr>
<td>Pumping flow rate of the brine circulator</td>
<td>14.06</td>
<td>W/kW</td>
<td>∓3.57</td>
</tr>
<tr>
<td>Heating load of the greenhouse tested</td>
<td>4.19</td>
<td>kW</td>
<td>∓3.42</td>
</tr>
<tr>
<td>Heat extraction rate from the ground</td>
<td>2.89</td>
<td>kW</td>
<td>∓3.42</td>
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<tr>
<td>Heat extraction rate per meter of bore</td>
<td>57.78</td>
<td>W/m</td>
<td>∓3.74</td>
</tr>
<tr>
<td>length in meter per kW of heating</td>
<td>11.92</td>
<td>m/kW</td>
<td>∓3.74</td>
</tr>
<tr>
<td>Heating COP range of the heat pump</td>
<td>2.00–3.13</td>
<td>Dimensionless</td>
<td>∓3.57</td>
</tr>
<tr>
<td>Overall heating COP range of the system</td>
<td>1.70–2.60</td>
<td>Dimensionless</td>
<td>∓3.42</td>
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<tr>
<td>Collector area</td>
<td>1.82</td>
<td>m²</td>
<td>∓2.14</td>
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<tr>
<td>Uncertainty in reading values of table</td>
<td></td>
<td></td>
<td>∓0.20</td>
</tr>
</tbody>
</table>

Table 4
Benchmarks for GSHP system pumping efficiency required pump power to cooling capacity [25,26]

<table>
<thead>
<tr>
<th>Watts input</th>
<th>Performance</th>
<th>Grade</th>
</tr>
</thead>
<tbody>
<tr>
<td>Per ton</td>
<td>Per kW</td>
<td>Efficiency</td>
</tr>
<tr>
<td>≤50</td>
<td>≤14</td>
<td>Efficient systems</td>
</tr>
<tr>
<td>50–75</td>
<td>14–21</td>
<td>Acceptable systems</td>
</tr>
<tr>
<td>75–100</td>
<td>21–28</td>
<td>Acceptable systems</td>
</tr>
<tr>
<td>100–150</td>
<td>28–42</td>
<td>Inefficient systems</td>
</tr>
<tr>
<td>&gt;150</td>
<td>&gt;42</td>
<td>Inefficient systems</td>
</tr>
</tbody>
</table>


1048

1049
serious defects in the (2003/2004) heating season. The results obtained during the heating period of 20th of January till 31st of March 2004 were given and discussed. The effects of climatic conditions and operating parameters on the system performance parameters were also investigated. The experimental results indicate that this SAGSHPS can be used for greenhouse heating in the Mediterranean and Aegean regions of Turkey. The conclusions drawn from the present study may be summarized as follows:

1. The energy performance of a GSHP system is influenced by three primary factors: (a) the heat pump unit, (b) the circulating pump or well pumps, and (c) the GHE.

(a) The heat pump unit: The values for COP_{HP} varied from 2.00 to 3.13, while those for COP_{s} were approximately 5–20% lower than COP_{HP}. Using a scroll compressor instead of an hermetic compressor in this study will lead to the increase in the COP values obtained.

(b) The circulating pump: The pumping brine flow rate was found to be 0.185 m\(^3\)/h per kW of heating capacity. Kavanaugh suggests that the optimum pumping rates for the circulating pump should range from 0.162 to 0.192 m\(^3\)/h per kW of heating capacity [33]. It may be concluded that the pump selected falls into the acceptable limits with a good grade.

(c) The GHE: Design practices in Turkey normally call for U-bend depths between 11 and 13 m/kW of heating, while the required borehole length in metre per kW of heating capacity was obtained to be 11.92. These values are close to the lower limit quoted by Kavanaugh [36, 37], with a range of 15–25 m/kW of cooling.

2. Experimental results show that monovalent central heating operation (independent of any other heating system) cannot be met overall heat loss of greenhouse if ambient temperature is very low. The bivalent operation (combined with other heating system) can be suggested as the best solution in Mediterranean and Aegean region in Turkey, if peak load heating can be easily controlled.

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References


Fig. 8. Effect of collector fluid temperature on the collector efficiency.

Fig. 9. Daily average values of COP_{HP} and COP_{s} over a period between 20 January and 31 March 2004.
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