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Experimental thermal performance of a solar source heat-pump system for residential heating in cold climate region

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ABSTRACT

Solar source heat pump systems present tremendous environmental benefits when compared to the conventional systems for residential applications. In addition to not exhausting natural resources, their main advantage is, in most cases, total absence of almost any air emissions or waste products. In order to investigate the performance of a solar source and energy stored heat-pump system in the province of Erzurum, an experimental set-up was constructed, which consisted of twelve flat-plate solar collector, a sensible heat energy storage tank, a water-to-water plate heat exchanger, a liquid-to-liquid vapor compression heat pump, water circulating pumps and other measurement equipments. The experiments were carried out from January to June of 2004 and, the collector efficiency (η_c), the heat pump coefficient of performance (COP) and the system performance (COPS) were calculated. In these months performed of the experiments, the outdoor temperature range varies from −10.8 °C to 14.6 °C. This study shows that the system could be used for residential heating in the province of Erzurum having the coldest climate of Turkey.

1. Introduction

In the future the world's energy supply must become more sustainable. This means that it must meet the basic needs of the poor worldwide without using up in this process the limited natural resources to the detriment of future generations. This can be achieved by a more efficient use of energy and relying on renewable sources of energy, particularly wind, hydropower, solar and geothermal energy [1].

Solar energy arriving on earth is the most fundamental renewable energy source in nature. Solar energy occupies one of the most important places among various alternative energy sources [2]. Solar energy technologies offer a clean, renewable and domestic energy source, and are essential components of a sustainable energy future [3].

Costing of energy resources remains inequitable, as it does not include subsidies, or environmental and other consequences. Development of renewable energy, and of all energy systems for that matter, is dominated by the highly controlled, cost-unrelated, highly fluctuating and unpredictable conventional energy prices. Fuel and energy consumption in general must be significantly constrained, with due attention to prevention of the rebound effects [4].

Solar energy systems and heat pumps are two promising means of reducing the consumption of fossil energy resources (coal, petroleum etc.), and hopefully, the cost of delivered energy for residential heating. An intelligent extension is to try to combine the two to further reduce the cost of delivered energy. In general, it is widely believed that combined systems will save energy, but what is not often known is the magnitude of the possible energy saving and the value of those savings relative to the additional expense [5].

Although solar space heat is a mature technology and reliable design methods exist, the size and cost of an active solar heating system, which depend not only on the heat collected but also on the storage facilities, affect its successful utilization on a large scale. One attractive way to reduce the collection and storage requirements is to utilize a solar source heat-pump system for heat supply, including domestic hot water and space heating. The low temperature thermal requirement of a heat pump makes it an excellent match for the use of low temperature solar energy and, as such, adds the benefit of a smaller solar energy system, with its lower associated cost [6]. The idea of combining the heat pump and solar energy has been proposed and the performance of these systems has been experimentally analyzed by several researchers in the literature [7–12].

Chaturvedi et al. [13] analyzed two-stage direct expansion solar-assisted heat pump for high temperature applications. Comparisons between the two-stage and the single-stage direct expansion solar-assisted heat pump systems were performed and presented.

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Nomenclature

A_{col}	solar collector area (m^2)
COP	heat-pump coefficient of performance
COPS	all system's coefficient of performance
C	specific heat (kJ/kg.K)
I_i	Incident solar radiation (W/m^2)
$\dot{m}$	mass flow rate of water in the system (kg/s)
$\dot{Q}$	heat extracted (kW)
T	temperature (°C)
$\dot{V}$	volumetric flow rate (l/s)
$\dot{W}$	compressor power (kW)
$w_{\dot{V}}$	total uncertainty in the volumetric flow rate of the water
η_{col}	collector efficiency
ρ	density (kg/m^3)

Subscripts

aw	heat transfer fluid (ethylene glycol–water mixture in 50% by volume)
col	collector
com	compressor
con	condenser
cwo	condenser water outlet
ewi	plate heat exchanger water inlet
i	incident surface
p	circulation pump
st	storage tank
stt	top region temperature of storage tank
stu	under region temperature of storage tank
w	water

Hawlader et al. [14] examined the solar-assisted heat-pump dryer and water heater. The performance of the system has been investigated under the meteorological conditions of Singapore. Comakli et al. [15] constructed an experimental set-up in order to investigate the performance of a solar source heat pump system with energy storage for residential heating in the Black Sea region of Turkey. Yumrutas and Kaska [16] designed and constructed an experimental solar source heat pump space heating system with a daily energy storage tank, and investigated its thermal performance. Ozgener and Hepbasli [17] investigated experimental performance analysis of a solar source ground-source heat pump for greenhouse heating. Best et al. [18] carried out an experimental study of a solar-assisted heat pump system for rice drying. Experimental results have been obtained on the efficiency and drying quality of a solar-assisted heat pump drying prototype system. Kaygusuz et al. [19] investigated the solar-assisted heat pump system and energy storage for domestic heating in Turkey. They experimentally investigated and compared the solar-assisted series heat pump system with storage and parallel heat pump system with storage. Gortari and Reyes [20] performed the experiments on a solar-assisted heat pump with direct expansion of the refrigerant within the solar collector and an exergy analysis of the system. Huang and Lee [21] examined the solar-assisted heat pump water heater and derived a simple linear correlation for the performance evaluation of different solar-assisted heat pump water heater. Li et al. [22] investigated the experimental performance analysis and optimization of a direct expansion solar-assisted heat pump water heater. Scarpa et al. [23] compared a direct expansion integrated solar-assisted heat pump with a traditional flat-plate solar panel for low temperature water heating applications. Sozen et al. [24] studied on the development and testing of a prototype of absorption heat pump system operated by solar energy.

The studies related to the design of the solar-assisted heat pump systems in the open literature [25,26]. The thermodynamic analysis of the solar-assisted heat pump systems has been carried out by several researchers [27–30]. Also, the performances of the solar-assisted heat pump systems have been theoretically analyzed in the literature [31–35].

As mentioned above, a number of studies have been carried out by various researches in order to analyze the performance of solar source heat pump systems, while the studies related to the region with cold climate and with sensible heat energy storage are few in numbers. Therefore, the objective of this study is to experimentally analyze the performance of a solar source heat pump system with sensible heat energy storage in Erzurum, Turkey having a cold climate for space heating.

In the present study, an experimental study was performed to determine the performance of a heat pump system with solar collectors and a sensible energy storage tank. The heat pump COP and all system COPS, the temperature variation of the energy storage tank and the temperatures of the heat transfer fluid in the solar collectors and energy storage tank for the heat pump system were investigated. Also, the collector efficiency, heat pump COP, all system's COPS and energy consumption of the solar source heat pump system from January to June in 2004 were calculated.

2. Description of the experimental set-up

The solar source heat pump system linked to a sensible heat energy storage tank was installed in the Energy Laboratory (approximately having an area of 175 m^2), Atatürk University, Erzurum placed on the East Anatolian Region of Turkey. A schematic overview of the system installed is given in Fig. 1.

The solar collectors used in this system were constructed by modifying flat-plate water-cooled collectors. Each collector was made of a pipe and fin type; the absorber unit consists of nine copper tubes fitted longitudinally at 0.1 m pitch across copper sheet. The copper plate had 1.64 m^2 effective absorber areas. The plate was painted with black board paint and placed inside a metal box made of aluminum sheet and insulated at the bottom and all sides with glass wool. The top of the metal box was glazed with a 4 mm thick glass cover. Twelve collectors were connected in a parallel combination as shown in Fig. 2. The mass flow rate of the heat transferring fluid through the collectors was 0.180 kg/s. The solar collectors were installed at an angle of 50° from the horizontal and faced due south. The compressor used for the solar source heat pump system was a hermetic-scroll type which was driven by a 1491 W electrical motor. The heat pump had an evaporator and condenser; both are water-cooled shall-tube type heat exchanger.

The storage tank was vertically made of sheet iron, and had a diameter of 1.00 m and 2.50 m long. The heat storage tank was linked to the heat pump by means of an evaporator for using as a heat source. The heat storage tank was insulated entirely by a 120 mm thick glass wool [36].

2.1. Weather data

The experimental solar source heat pump system was established and tested in Erzurum province having an altitude of 1869 m and the coldest climate in Turkey (yearly average lowest temperature is 4.7 °C). The climatic conditions of Erzurum for long-term average values (monthly average minimum, maximum and mean

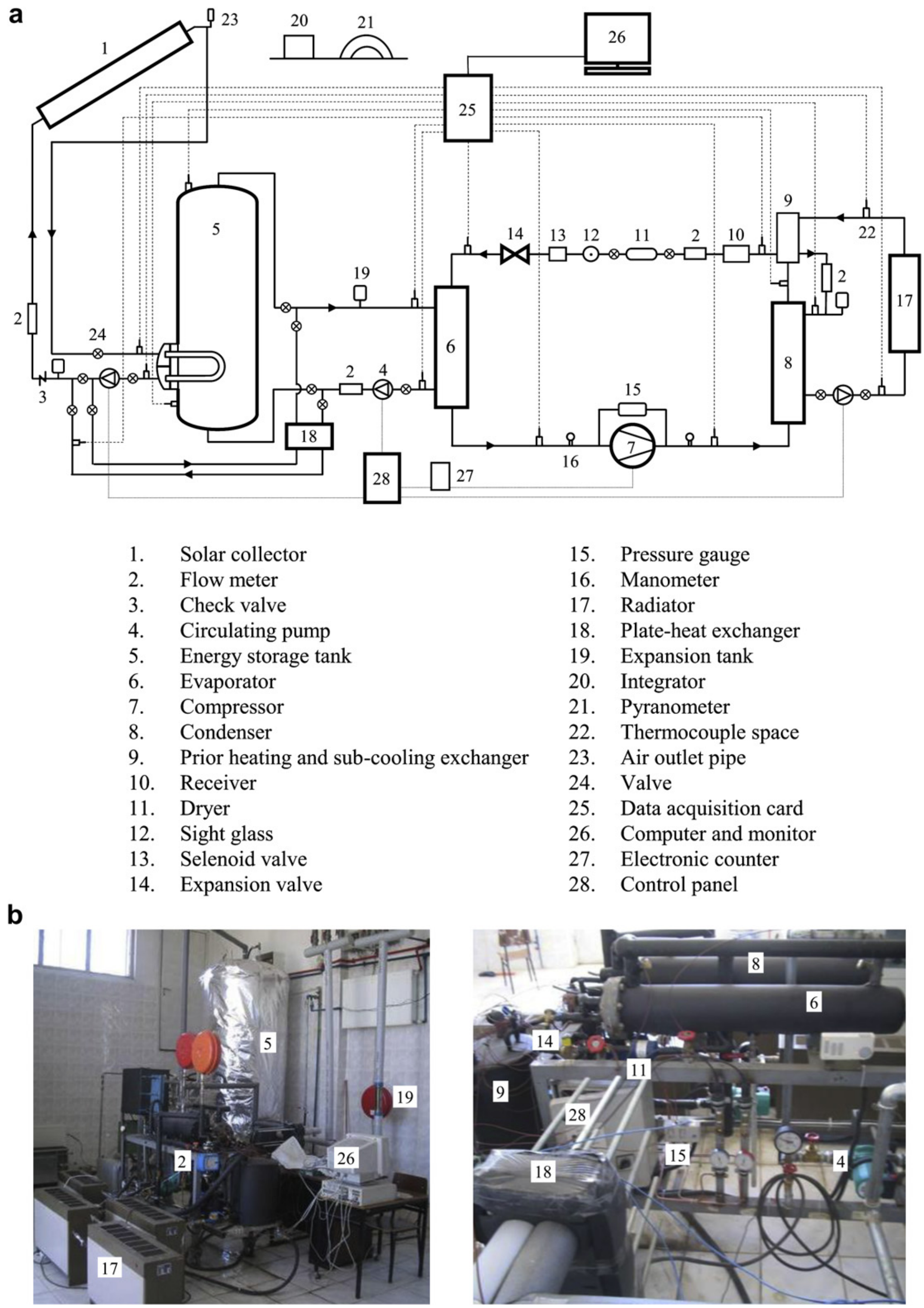

Fig. 1. a) schematic representation of the system. b) Outside view of the system.

outdoor temperature, the monthly averages of relative humidity, wind velocity, solar radiation, sunshine durations and heating degree days for the cold season) are given in Table 1. The annual heating degree days for Erzurum with a base temperature of 18 °C is 4870 [37]. This situation supply that the region has a cold climate. As seen in Table 1, the energy requirement for heating is maximum in January, while it is minimum in June for the months of the cold season (from January to June).

3. Solar source heat pump system with storage

As shown in Fig. 1, the system consists of conventional flat-plate water cooled solar collectors, a sensible energy storage tank, a heat pump with water-to-refrigerant heat exchanger, a water-cooled evaporator and condenser, a water-to-water plate heat exchanger, a water circulating pump and other conventional equipments. In this system, the hot water which comes from the collectors, first

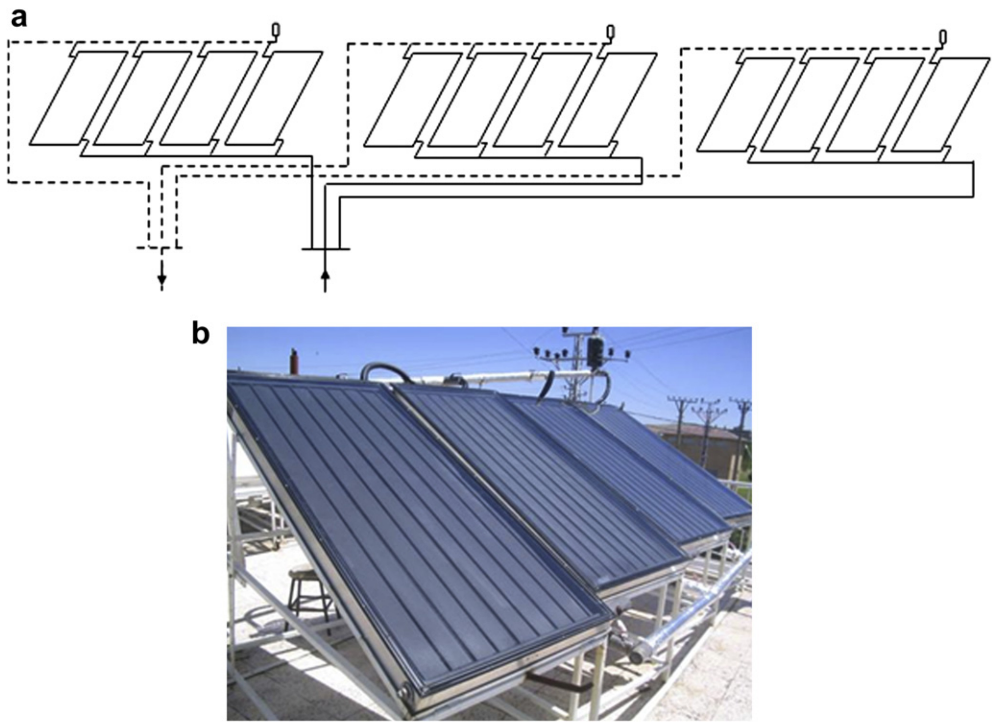

Fig. 2. a) Connection schematic of the solar collector group. b) Outside view of a solar collection group.

goes to the energy storage tank where it releases some energy to the storage, then, it is passed from the plate-heat exchanger, and thus is used as a heat source by the water-source evaporator in the daytime. Finally, the heat transfer fluid is sent to the solar collectors by a water-circulating pump. However, at nights and cloudy days, water with lower temperature that comes from the evaporator of the heat pump is sent to the energy storage tank instead of the solar collectors. The cold water extracts energy from the energy storage tank, and flows to the evaporator for using as a heat source.

4. Calculation of the experimental results

The COP of the heat pump is calculated as:

$$COP = \frac{\dot{m}_{con} C_w (T_{cwo} - T_{ewi})}{\dot{W}_{com}} \tag{1}$$

The COPS of the whole system are given by

$$COPS = \frac{\dot{m}_{con} C_w (T_{cwo} - T_{ewi})}{\dot{W}_{com} + \dot{W}_{\Sigma p}} \tag{2}$$

Also the instantaneous collector efficiency is given as follows:

$$\eta_{col} = \frac{\dot{Q}_{col}}{A_{col} I_i} \tag{3}$$

where $\dot{Q}_{col}$ is the useful heat received from the collector and transferred to heat transfer fluid (ethylene glycol–water mixture in 50% by volume) and can be calculated as:

$$\dot{Q}_{col} = \dot{m}_{col} C_{aw} (T_2 - T_1) \tag{4}$$

where T_1 is the temperature of the collector inlet and T_2 is the temperature of the collector outlet of heat transfer fluid. The useful heat obtained from the condenser $\dot{Q}_{con}$ is also calculated as:

$$\dot{Q}_{con} = \dot{m}_{con} C_w (T_{cwo} - T_{ewi}) \tag{5}$$

5. Uncertainty analysis

Experimental errors and uncertainties can result from instrument selection, condition and calibration of the instrument as well as environmental conditions and reading errors. Uncertainty

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

Latitude: 39.55 °N; longitude: 41.16 °E							
Climatic values	Yearly	Jan.	Feb.	Mar.	Apr.	May	Jun.
Average outdoor temperature (°C)	4.7	−10.8	−10.1	−3.7	5.2	10.3	14.6
Minimum outdoor temperature (°C)	−2.8	−16.9	−16.7	−9.8	−0.9	2.7	5.5
Maximum outdoor temperature (°C)	12.2	−4.4	−3.1	2.6	11.8	17.3	22.3
Average relative humidity (%)	64.6	77.5	73.1	75.0	56.7	60.7	54.9
Average wind velocity (m/s)	2.7	2.3	2.4	2.8	3.3	3.1	3.0
Average solar radiation(MJ/m^2.day)	15.6	8.9	12.6	16.0	17.0	19.9	23.2
Average sunshine duration (h)	6.4	3.0	3.8	4.4	5.9	7.6	9.9
Average degree days (for 18 °C base temp.)	4870	889	784	674	385	236	103

analysis is needed to prove the accuracy of the experiments [17]. The uncertainty analysis can be performed using a method proposed by Holman [38]. In the present study, the temperatures, flow rates, pressure drops, voltages and currents were measured by appropriate instruments explained below:

- Measurement of mass flow rates of the ethylene glycol–water mixture (approximately 50% by volume) by a rotameter.
- Measurement of the mass flow rates of the refrigeration in liquid phase by a flowmeter.
- Measurement of the temperature of the water–ethylene glycol solution entering and leaving the solar collector by copper-constantan thermocouples (by assisted data acquisition card) mounted on the unit of water inlet and outlet lines.
- Measurement of the condenser and evaporator pressures by Bourdon-type manometers.
- Measurement of the outdoor air temperatures and humidity by Meteorological Station.
- Measurement of the electrical power input to the circulating pump by a wattmeter.
- Measurement of the inlet water temperature to and exit water temperature from heating unit by copper-constantan thermocouples.
- Measurement of the solar flux by a pyranometer in the meteorological station installed in the Energy Laboratory of Engineering Faculty of Atatürk University.
- Measurement of the instantaneous power consumptions of the compressor by an electronic counter.

The tests were conducted on the solar source heat pump system in the heating mode over the period from January to June 2004. Daily average values of 57 measurements from 9.00 a.m. to 23.00 p.m. with an interval of 15 min were taken. The temperatures measured by the thermocouples were monitored in a computer and recorded by data acquisition card in every second, which was later used for analysis.

The total uncertainty in the measurement of the volumetric flow rate of the water in the radiators by using a rotameter and the uncertainty arising in calculating the mass flow rate of the water are given in the following. Besides, the uncertainties in the other measured and calculated parameters are determined in a similar fashion.

The total uncertainty in the measurement of the volumetric flow rate of the water $w_{\dot{V}}$ may be calculated as follows [39–41]:

$$w_{\dot{V}} = \left(w_{ro}^2 + w_{sl}^2 + w_{td}^2\right)^{1/2} \tag{6}$$

where w_{ro} is the uncertainty in the rotameter reading (%), w_{sl} is the uncertainty associated with the system leakages (%) which is the value of the accuracy (±5.00%) given in the catalog of the rotameter used in the experimental system, w_{td} is the uncertainty associated with the temperature differences (density differences) (%) given as:

$$w_{\dot{V}} = \left(1.25^2 + 5.00^2 + 0.75^2\right)^{1/2} = 5.208\% \tag{7}$$

The uncertainties arising in the calculating results (such as w_R) due to several independent variables are given in Ref. [38] as

$$W_R = \left[\left(\frac{\partial R}{\partial x_1}w_1\right)^2 + \left(\frac{\partial R}{\partial x_2}w_2\right)^2 + \cdots + \left(\frac{\partial R}{\partial x_n}w_n\right)^2\right]^{1/2} \tag{8}$$

where the result R is a given function of the independent variables $x_1, x_2, \ldots, x_n$ and $w_1, w_2, \ldots, w_n$ are the uncertainties in the independent variables.

Table 2
Results obtained in the experimental study (from January to June).

Calculated values	Months in cold season					
	Jan.	Feb.	Mar.	Apr.	May	Jun.
Temperature of condenser water outlet (°C)	32–43	35–48	42–51	42–53	42–53	43–58
COP (average)	3.7	3.8	3.8	3.5	3.4	3.3
COPS (average)	2.7	2.9	2.8	2.7	2.6	2.5
η_{col} (average)	0.40	0.38	0.44	0.48	0.54	0.60

The uncertainty in calculating the mass flow rate $w_{\dot{m}}$ may be found as follows:

$$\dot{m} = \rho\dot{V} \tag{9}$$

$$w_{\dot{m}} = \left[\left(\frac{\partial \dot{m}}{\partial \rho}\right)^2 w_\rho^2 + \left(\frac{\partial \dot{m}}{\partial \dot{V}}\right)^2 w_{\dot{V}}{}^2\right]^{1/2} \tag{10}$$

Taking into account an uncertainty value of ±0.20% in the thermophysical properties [39,40] and inserting the numerical values of uncertainty yields as follow:

$$\frac{w_{\dot{m}}}{\dot{m}} = \left[(0.2)^2 + (5.208)^2\right]^{1/2} = 5.212\% \tag{11}$$

6. Results and discussions

We have analyzed the performance of the solar source heat pump system with a sensible heat energy storage in Erzurum, Turkey experimentally. The experiments were performed under clear sky conditions so that a quasi-steady state could be realized. For most runs, the system was started well ahead of solar noon and the quasi-steady state of the operation was usually achieved around noon. The data were obtained for global solar radiation on the plane of the collector from 300 to 1050 W/m^2. The condenser outlet temperature is about 32–58 °C. In this case, floor heating should be preferred to radiator since it is more suitable than radiator for supply temperatures of 40–45 °C. Therefore, the system is appropriate for floor heating.

In the experimental study performed from January to June of 2004, the temperature of the condenser water outlet, the mean value of the heat pump COP, the COPS of the whole system and the average collector efficiencies have been deduced from the experimental data, and given in Table 2. Fig. 3 shows the variation of total solar radiation on the tilted-surface with time of day. As shown in

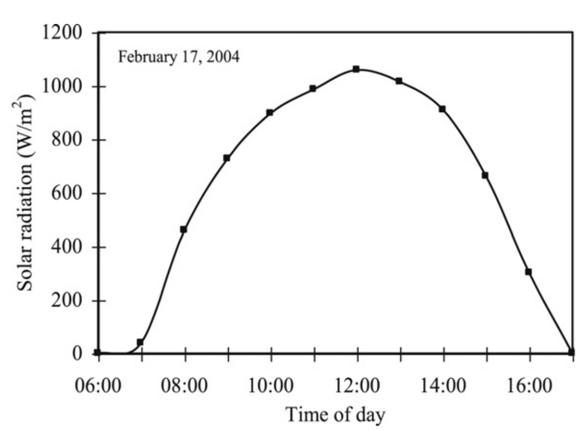

Fig. 3. Solar radiation I_i (W/m^2) versus time of day.

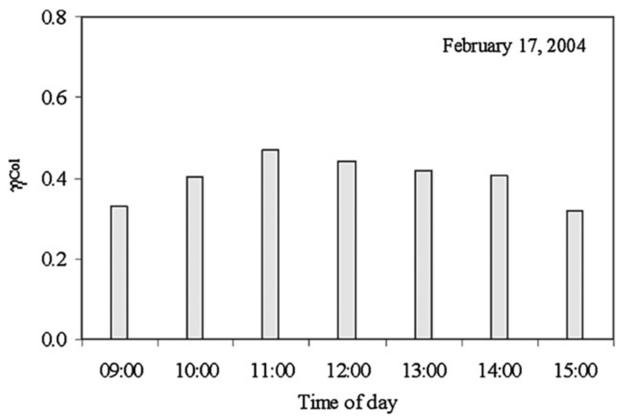

Fig. 4. Collector efficiency versus time of day.

Fig. 3, the total solar radiation is maximum at around solar noon, which is approximately 1050 W/m^2. In this study, solar radiation incident on the collector aperture is measured with a pyranometer.

The collector efficiency shown in Fig. 4 varies from 33 to 47 percent around the day. Fig. 5 illustrates the temperature change in the energy storage tank versus time of day. Fig. 6 also shows the maximum energy storage temperature versus months during the experiments. As shown in Fig. 6, the maximum energy storage temperature is increasing steadily from January to June due to the total solar radiation increased over this period (from January to June). The technical details of the experimental set-up are given in Table 3. Daily average values of 57 measurements taken from 9.00 a.m. to 23.00 p.m. with an interval of 15 min are also given in Table 4.

Fig. 7 demonstrates the temperatures of the collector inlet (T_1), collector outlet (T_2) and heat exchanger inlet (T_3) of heat-transported fluid (water–ethylene glycol) versus time of day. As shown in Fig. 7, all the temperatures (T_1, T_2 and T_3) are maximum at around solar noon. Also, the temperature of T_2 is equal to T_3 at local time of 16:00 when the sun set and the circulating pump for the collector loop was closed. This means that the incident solar energy on the collector surface has decreased very fast. After this local time, the storage outlet temperature becomes higher than the storage inlet temperature. Thus, the solar energy charging period has been completed.

Fig. 8 shows the temperatures of the evaporator inlet (T_4) and condenser outlet (T_{cwo}) of water versus time of day. As shown in

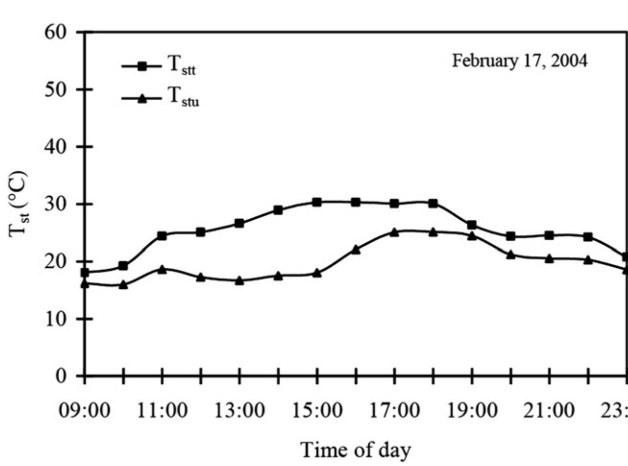

Fig. 5. The temperature change in the energy storage tank versus time of day.

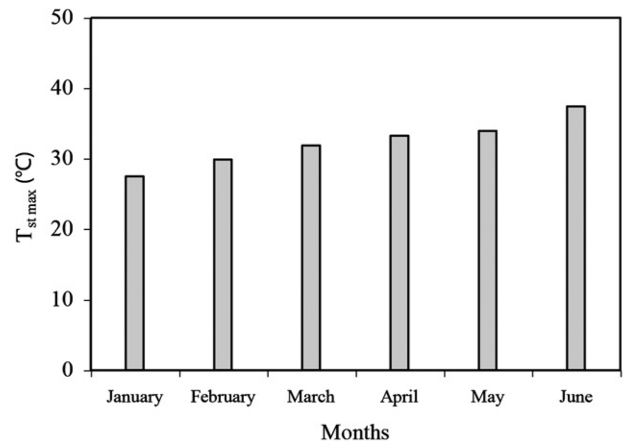

Fig. 6. Maximum energy storage tank temperature versus months.

Fig. 8, the temperature of condenser outlet (T_{cwo}) is varying steadily between 35 and 48 °C. But the temperature of the evaporator inlet is at a maximum value for a local time of 16:00 because the sun set between the local times of 15:00 and 16:00 and the collector loop was closed. After that time, the evaporator has received energy from the storage, so again T_4 increased at the local time of 17:00.

Table 3
Technical specification of the experimental set-up.

Main element	Technical specification
Pyranometer information	
Measuring range (W/m^2)	0 to 1800
Resolution and units (W/m^2)	1
Collector information	
Type (copper tube and fin)	Flat-plate
Glass number	Single
Collector area (m^2)	1.64
Collector number	12
Capacity (l)	3.5
Water–ethylene glycol mass flow rate in collectors (l/h)	600
Energy storage tank information	
Type	Cylindrical
Volume of water in store (l)	2000
Wall thickness (mm)	7
Diameter (mm)	1000
Surface area of serpentine (m^2)	4
Flowmeters	
Measuring range (water–ethylene glycol) (l/h)	150–1500
Measuring range (refrigerant) (l/h)	25–250
Measuring range (water in condenser and evaporator) (l/min)	5–35
Circulation pump(water–ethylene glycol and water circuit)	
Type (three-stage variable speed, power supply; 220–240 V/1–50 Hz)	TOP– S30/10
Heat pump information	
Compressor type (power supply: 220–240 V/1–50 Hz)	Hermetic-scroll
Evaporator type	Shall-and-tube
Condenser type	Shall-and-tube
Compressor power input (kW)	1.47
Compressor displacement (m^3/h)	5.34
Compressor rotation speed (rpm)	2900
Water mass flow rate in evaporator (l/h)	450
Water mass flow rate in condenser (l/h)	600
Refrigerant type	R-134a

Table 4
Measured parameters and experimental results in average.

Item	Value	Unit	Total uncertainty (%)
Measured parameters			
Solar radiation	696	W m^{-2}	±4.00
Evaporation pressure	0.46	Mpa	±2.05
Condensation pressure	1.98	Mpa	±2.05
Condensing temperature	67.04	°C	±1.01
Evaporating temperature	12.93	°C	±1.01
Temperature of heat transfer fluid at energy storage tank inlet	37.22	°C	±1.01
Temperature of heat transfer fluid at energy storage tank outlet	30.18	°C	±1.01
Temperature of energy storage tank	30.26	°C	±1.01
Supply water temperature of heating unit	43.15	°C	±1.01
Return water temperature of heating unit	39.23	°C	±1.01
Flow rate of water–ethylene glycol solution in solar collector	0.180	kg s^{-1}	±1.78
Flow rate of water in evaporator	0.125	kg s^{-1}	±5.21
Flow rate of water in heating unit or condenser	0.167	kg s^{-1}	±5.21
Flow rate of refrigerant in heat pump unit	0.016	kg s^{-1}	±2.17
Outdoor air temperature	−10.86	°C	±0.97
Indoor air temperature	18.03	°C	±1.01
Current of circulating pump at solar collector	0.70	A	±2.00
Current of circulating pump at heating unit side	0.68	A	±2.00
Two-phase voltage	220.0	V	±2.00
Temperature of water at evaporator inlet	24.49	°C	±1.01
Temperature of water at evaporator outlet	20.30	°C	±1.01
Power input to the compressor	0.968	kW	±1.00
Calculated parameters			
Power input to the heat transfer fluid circulating pump	0.129	kW	±3.00
Power input to the water circulating pumps	0.129	kW	±3.00
Heating load of condenser	3.801	kW	±5.32
Cooling load of evaporatör	2.191	kW	±5.32
Useful heat received from solar collectors	6.76	kW	±2.07
Instantaneous collector efficiency	0.38	–	±4.75
Heating COP of the heat pump	3.805	–	±5.41
Heating COPS of the whole system	2.860	–	±5.39

Fig. 9 shows the amount of energy received from the condenser ($\dot{Q}_{con}$) and given to compressor ($\dot{W}_{com}$) versus time of day. As shown in Fig. 9, the value of the $\dot{Q}_{con}$ is varying from 2.8 to 4.4 with time of day while the $\dot{W}_{com}$ is approximately kept constant. On the other hand, Fig. 10 illustrates the heat pump's COP and all system's COPS versus time of day for two different days of the year. In Fig. 10, the heat pump COP is observed to be higher than that of the whole system (COPS) as expected. The COP varies between 3.4 and 4.2 for February 17 and 3.5 to 4.3 for March 16 while the COPSvary between 2.2 and 3.3 for February 17 and 2.5 to 3.4 for March 16. The water outlet temperatures in the energy storage tank versus time of the day are graphically shown in Fig. 11 versus time of day. Consequently, the temperatures of the evaporator inlet (T_4) and condenser outlet (T_{cwo}) of water as well as the values of the COP and COPS increase because of increasing of solar radiation during the day.

The energy storage tank is used for night. However, at night, water with the lower temperature that comes from the evaporator of the heat pump is sent to the energy storage tank instead of the solar collectors. In the experiments carried out on the 17th of February, 2004, the amount of the heat supplied by the store is about 2.365 kW per hour, while it is 1.905 kW per hour in average from the heat exchanger. Also during the day, the amount of heat supplied from the solar panels to the storage tank and the evaporator of the heat pump ranges from 1.461 to 9.234 kW.

For a comparison, Bi et al. [42] reported that the COP in the cold season was found to be 2.73 for the solar energy-source heat-pump system. Yumrutas and Kaska [16] demonstrated the COP values for an experimental solar source heat pump space heating system with a daily energy storage tank as follows: the COP of a heat pump is about 2.5 for a lower source temperature at the end of the cloudy days and is about 3.5 for a higher supply temperature at the end of

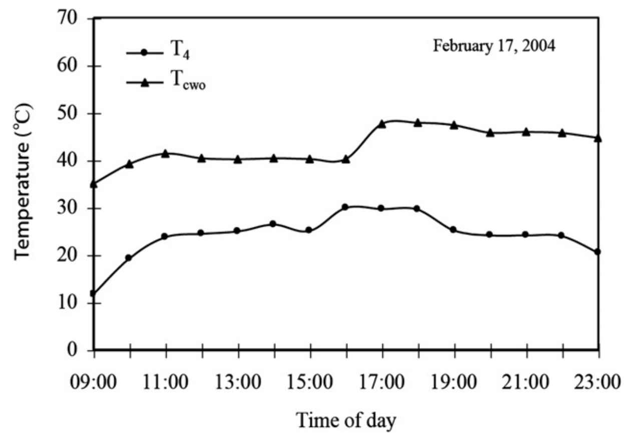

Fig. 8. Temperatures of evaporator inlet (T_4) and condenser outlet (T_{cwo}) of water versus time of day.

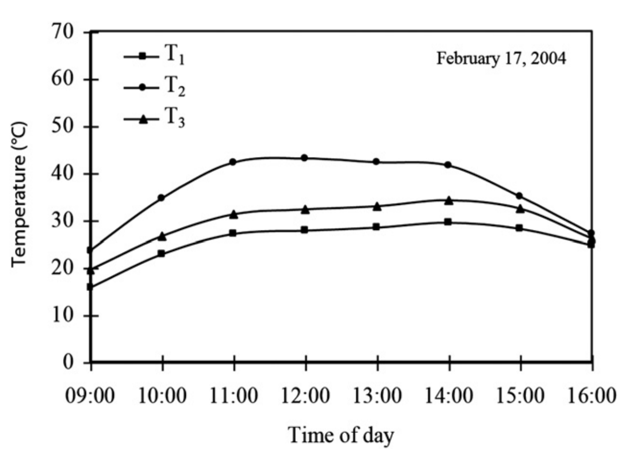

Fig. 7. Temperatures of collector inlet (T_1), collector outlet (T_2) and heat exchanger inlet (T_3) of heat transported fluid versus time of day.

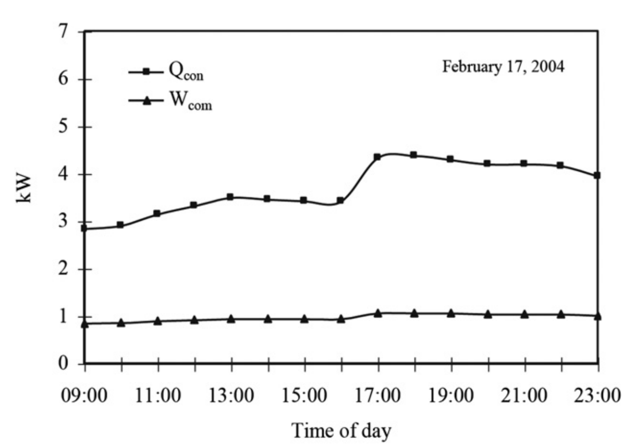

Fig. 9. Amounts of energy received from condenser ($\dot{Q}_{con}$) and given to compressor ($\dot{W}_{com}$) versus time of day.

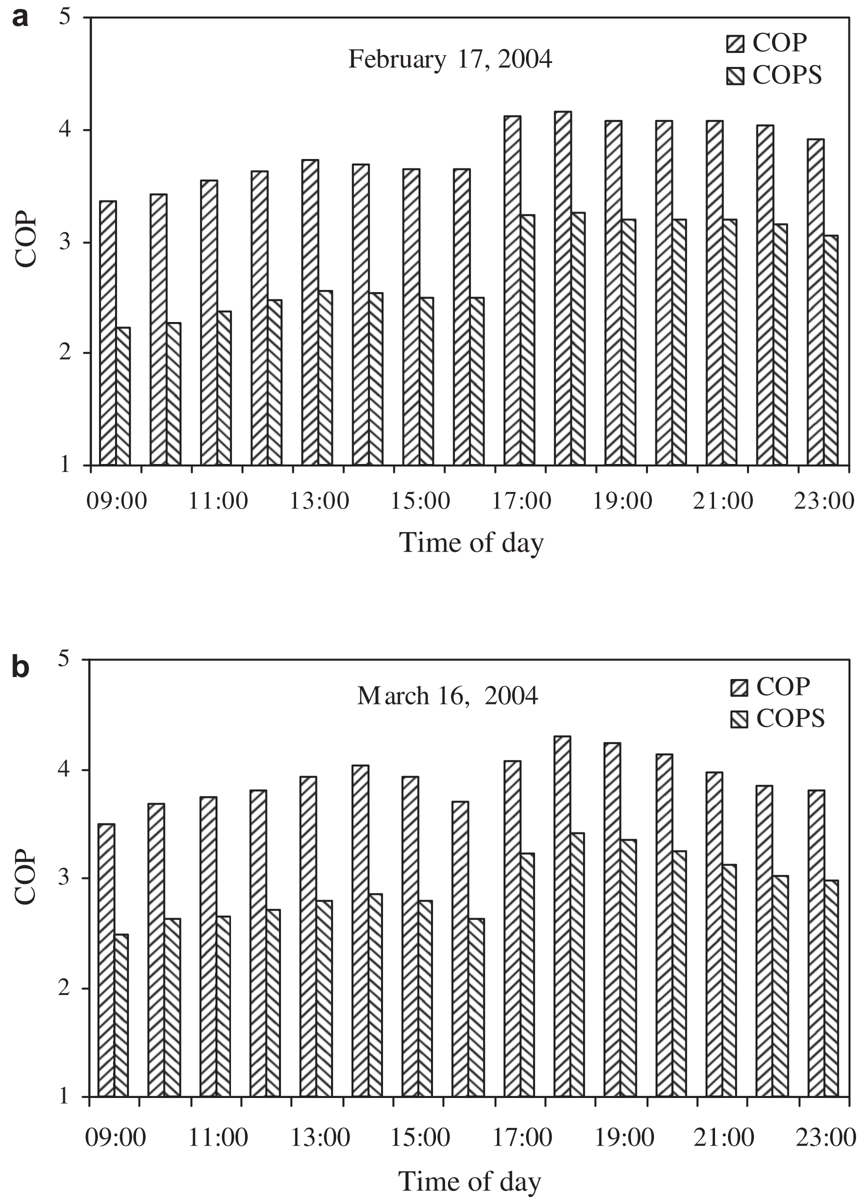

Fig. 10. Performance of coefficient of heat pump (COP) and whole system (COPS) versus time of day (a) February 17, 2004 (b) March 16, 2004.

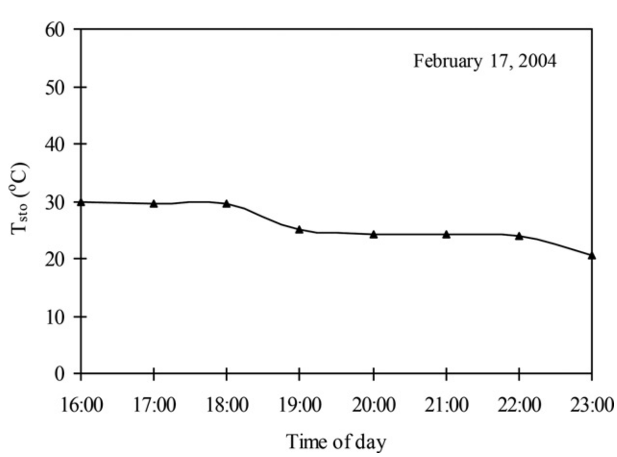

Fig. 11. The water outlet temperature in the energy storage tank versus time of day.

the sunny days. The COPS of the whole system is approximately 15–20% lower than the COP of heat pump [16]. Huang and Chyng [43] also derived that the COP for a solar source heat pump built in their study lies in the range of 2.5–3.7. It may be concluded that the COP values obtained from the present study are fairly close to those reported by Bi et al. [42], Yumrutas and Kaska [16] and Huang and Chyng [43].

6.1. Environmental and economical benefits

Solar source heat pumps work with the environment to provide clean, efficient, and energy saving heating and cooling year round. They use less energy than alternative heating and cooling systems, helping to conserve our natural resources. They are quiet, pollution free and do not damage the surrounding landscape. Solar source heat pumps may be expensive to install when compared to some natural gas, oil or electric heating units, but they are very

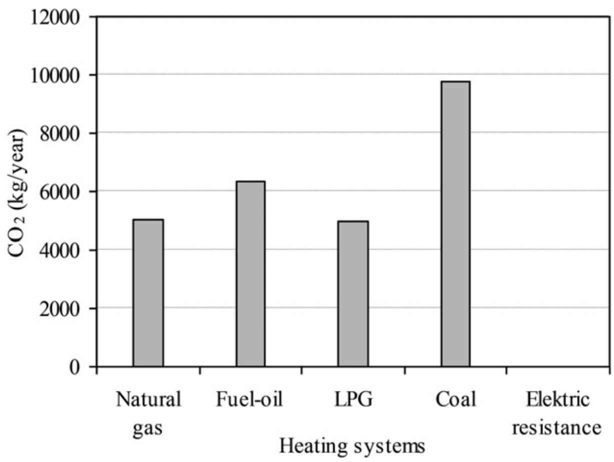

Fig. 12. CO_2 ratios for various heating systems.

Table 6
Calculations for the payback period of systems.

Systems	Investment cost (€)	Operating cost (€/year)	Heat pump (COP = 2.7)	Heat pump (COP = 5.0)
			Payback periods (year)	Payback periods (year)
Natural gas	1148	685.8	N/A	19.6
Coal	905	1019.4	34.9	7.8
Fuel oil	905	1974.3	3.9	2.8
LPG	1148	3751.7	1.4	1.2
Electric	682	2431.5	2.9	2.2
Heat pump (COP = 2.7)	5050	900.6	0.0	0.0
Heat pump (COP = 5.0)	5050	486.3	0.0	0.0

competitive with any type of combination heating/cooling system. For this reason, heat pumps are most attractive for applications requiring both heating and cooling [44]. Furthermore, their annual operating costs are lower than other conventional heating/cooling systems [45].

6.2. Energy-saving ratio

The 180 days of the year in Erzurum, Turkey are clearly sky. Also, the heating season is seven months (from 15 October to 15 May) and, the residential heating duration in a day is approximately 18 h. The heating required in the seven months is hourly 3780 h ($7 \times 30 \times 18$) and, the solar source heat pump system can be used in the 50% of this value. In this situation, the energy-saving ratio is 18 900 kWh/year for heating load of 10 kWh, when solar source heat pump system is used in Erzurum, Turkey. Besides, this supplies to decreasing of the CO_2 emission increasing in heating season.

6.3. CO_2-reduction ratio

Significant emission reductions are available through the application of heat pump system (HPS) in both residential and commercial buildings. Residential fossil fuel heating systems produced anywhere from 1.2 to 36 times the equivalent CO_2 emissions of HPS. CO_2 emission reductions from 15% to 77% were achieved through the use of HPS [46].

The CO_2 ratios of various heating systems for residential heating of 18 900 kWh/year are calculated and given in Fig. 12. As seen from Fig. 12, the system with coal, fuel oil, LPG and natural gas are released to environment the CO_2 emission in the values of 9779, 6338, 4961 and 5051 kg/year, respectively. However, the CO_2 emission of the electric resistance and heat pump system is zero when the green electricity is used. The chemical formulas of the fuels used in the study are received from the literature [47,48].

6.4. Payback periods

The required data including investment cost, average heating value, efficiency or performance, and unit price of fuels for various heating systems are given in Table 5. The payback periods of the heating systems can be computed as follow:

$$\text{Payback period} = (A_{ic} - B_{ic})/(B_{oc} - A_{oc}) \quad (12)$$

where, A_{ic} and B_{ic} are the investment cost of the heat pump (A_{ic}) and a system (such as system with LPG, B_{ic}) compared to it, respectively. A_{oc} and B_{oc} are the operating cost of the heat pump (A_{oc}) and a system (such as system with LPG, B_{oc}) compared to it, respectively. An example for the calculation of payback period in the heat pump (COP = 2.7) and system with LPG can be given as payback period = (5050-1148)/(3751.7−900.6) = 1.4 year. The calculations performed for the payback period of other systems are given in Table 6. Here, the payback periods for the commercial systems which the COP of the heat pump is about 5.0 are also calculated. It is rather high for the system with coal while the payback period of the heat pump system (for COPS = 2.7) is absent due to relatively low price of natural gas. Moreover the payback periods of the heat pump system according to the LPG, electric and fuel oil are 1.4, 2.9 and 3.9 year, respectively.

The natural gas heating is widely used heating system in other big cities of Turkey due to relatively low price of natural gas. The solar source heat pump (SSHP) systems will be more cost effective than the all other heating systems, if the investment cost of SSHP system with series productions is decreased and, a low price for electricity is provided. The main disadvantage of SSHP is the high initial investment cost, almost four times of the natural gas and five times of coal

Table 5
Data for various heating systems.

System	Investment cost (€)	Lower heating value	Efficiency or COP	Unit price of fuel	Unit price of energy (€/kWh)
Natural gas	1148	34541 kJ/m^3	0.93	0.372 €/m^3	0.0363
Coal	905	29308 kJ/kg	0.65	0.285 €/kg	0.0539
Fuel oil	905	41345 kJ/kg	0.80	0.960 €/kg	0.1045
LPG	1148	46473 kJ/kg	0.92	2.336 €/kg	0.1985
Electric	682	3601 kJ/kWh	0.99	0.127 €/kWh	0.1287
Heat pump	5050	3601 kJ/kWh	2.70	0.047 €/kWh	0.0476
Heat pump	5050	3601 kJ/kWh	5.00	0.025 €/kWh	0.0257

heating systems. The other important point is the sensitivity of SSHP systems to the electric price, and the order among the SSHP, natural gas and coal systems may change depending on electric price. However, the SSHP systems are an important economic alternative over the other conventional heating methods such as fuel oil, LPG and electric in Erzurum, Turkey. Also, the SSHP system can significantly reduce primary energy use for residential heating and cooling. This property of the SSHP system supplies an important contribution for decreasing of high investment cost.

7. Conclusions

A solar source heat pump system was investigated experimentally. The experimental results indicate that the collector efficiencies ranging from 33 to 47% can be realized with 20 m^2 flat-plate water-cooled collectors over the experimental period for the solar source heat pump with energy storage. The average values of the COP of the heat pump and the COPS of the whole system are 3.8 and 2.9 for heat pump systems, respectively. The following conclusions can also be drawn from this study:

1. The selection of a solar source heat pump system mainly depends on the operating conditions, economic viability, environmental impacts, etc.,
2. Substantial energy savings can be realized by taking advantage of solar combined heat pump system when implementing techniques such as using waste heat, reducing electrical demand charges and avoiding heating equipment purchases.
3. From the experimental studies, it was concluded that heat storage is an important component for solar source heat pump systems in the East Anatolia region of Turkey and the technical and economical feasibility must be made by the designer to select a storage material.
4. The experimental results obtained in this study show that the solar source and energy stored heat pump system can be used for residential heating in the East Anatolia region of Turkey having a cold climate.
5. The heat pump systems are environmentally friendly. The initial costs of the heat pump systems are higher, but they have low operating, maintenance, and life cycle costs and a longer life expectancy than most conventional systems. Their overall economic benefit depends primarily on the relative costs of electricity and fuels, which are highly variable over time and across the world. Also, heat pump systems provide heating, cooling and hot water.
6. Solar source heat pump systems present tremendous environmental benefits when compared to the conventional systems. In addition to not exhausting natural resources, their main advantage is, in most cases, total absence of almost any air emissions or waste products. Therefore, these systems can be used to minimize environmental impacts and air emission. This study also shows that the system could be used for residential heating in the province of Erzurum which is region with cold climate of Turkey.

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