An experimental analysis of a solar-assisted heat pump (SAHP) system for heating a semisolar greenhouse

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ABSTRACT
Solar thermal is the primary source of heat and the heat pump is used as a back-up when sun is not shining with enough intensity. In this paper, theoretical and experimental studies were carried out on a solar-assisted heat pump (SAHP) under the metrological condition of Tabriz city (located in the northwest of Iran). The system mainly includes a flat-plate solar collector (totally of 1 m²), an evaporator with refrigerant R134a, a piston type compressor with a power of 3/4 hp, a condenser and a capillary tube. Tests were performed on four consecutive days from 24 to 27 December 2018. The effect of environmental parameters on the greenhouse temperature in day and night was compared with and without the heat pump using SAS9.1 software. The effect of various parameters such as solar radiation and environmental temperature, evaporation and condensation temperature, solar collector area, compressor speed, and number of collector cover has been analyzed in order to understand the coefficient of performance (COP) and efficiency of system (η). The results showed that by increasing the solar radiation and temperature, COP and collector efficiency increases with relatively direct relation. According to the results, environment temperature, radiation intensity, evaporation temperature, collector area, and collector covers have a direct relationship with COP but condensation temperature and compressor speed have inverse relationship with it. In this study, η has a direct relationship with environment temperature, evaporation temperature, compressor speed and collector covers, and the other three parameters have an inverse relationship with it. Statistical analysis showed that there was a significant difference between the temperature of the greenhouse with and without the heat pump. In addition, using the thermal screen at night was significant for keeping the storage temperature.

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Introduction
Greenhouse farming is a growing industry in many countries. All plant growth factors can be controlled and maintained at optimum level year around in the greenhouses. Greenhouse acts as a solar collector, in which heating is caused in part, by the selective transmission of solar energy and by a trapping of the infrared radiation from the soil and plants in the greenhouse by the covering material. One of the major problems encountered in greenhouses is the control of the internal climate. The lack of heating has unfavorable effects on the precocity of production (Mashonjowa et al. 2013). Unfortunately, the investment required to implement a greenhouse environmental control system can be quite high and the energy costs for heating a greenhouse can reach 70% of total production costs. The changes exerted by agriculture on...
ecosystems are characterized by the consumption of renewable and nonrenewable energy sources (Manetto et al. 2017). In order to decrease the energy costs and investigate some new solutions, several research projects are taking place on the microgeneration systems supplied by renewable energy sources for stand-alone applications. Furthermore, using of renewable energy for heating structures and greenhouses is being promoted by the European Community as highlighted by the new European Directive 2010/31/CE and the near zero energy building (NZEB) concepts. In particular, in the agricultural sector, geothermal heating systems could be the best solution for the climate control of greenhouses (Anifantis, Colantoni, and Pascuzzi 2017). Greenhouses can also be seen as energy-intensive and antiseasonal objects, and therefore it is very important to strive for minimal energy consumption (Shen, Wei, and Xu 2018). One solution that aims to reduce electricity charges is the installation of solar panels (Carreño-Ortega et al. 2017). In turn, cooling systems are needed in very warm climates to protect plants from too high temperatures. Research regarding the possibilities of cooling greenhouses using groundwater from indirect-direct evaporative cooling (IDEC) is carried out in Iraq. This research allows the temperature in a greenhouse to be lowered by about 12.1–21.6°C, and relative humidity to be increased from 8% to 62% when compared to the ambient temperature (Issam, Aljubury, and Dhia’a Ridha 2017).

The increasingly demand for petroleum-based fossil fuels has led to concerns that they may be depleted in the next two decades to the extent that would cause a major disruption in the energy supply chain. As a result, finding alternative energy sources which are cleaner as well as economical, has become a critical societal need that led to the development of renewable energy sources such as solar and wind in recent years (ASHRAE Handbook 2000).

Among different renewable energy sources, solar energy is one of the best alternatives which is readily available in many parts of the world. Consequently, research and developments have been conducted to expand application of solar systems. Solar energy, an abundant, clean and safe source, is an attractive substitute for conventional fuels for passive and active heating applications. Use of solar agricultural greenhouses has increased in agricultural production since the last two decades. During the day, excess solar heat is collected for short- or long-term storage, and it is recovered at night in order to satisfy the heating needs of greenhouses (Sethi and Sharma 2007, 2008).

In the winter period, heating greenhouses leads to heavy charges, difficult to handle by farmers because of the increase of the energy cost. Solar-assisted heat pump (SAHP) is one these applications. The idea of combining conventional heat pumps and solar systems has taken many interests. The coefficient of performance (COP) of heat pumps improves by increasing of evaporator temperature (El-Maghlany, Teamah, and Tanaka 2015).

A greenhouse heating system is used to increase heat energy storage inside the greenhouse during the day or to transfer excess heat from the inside of the greenhouse to the heat storage site (Chou et al. 2004).

Heat pumps could extract heat from the environment in a low to higher temperature levels. A review paper has indicated that the COP values of the SAHP systems range from 2 to 9 and that the collector/evaporator efficiencies vary between 40% and 75% under different climatic conditions experimentally (Kuang, Sumathy, and Wang 2003). The concept of SAHP systems can be dated back to the 1950s and the extensive studies on these systems began in the 1970s (Elliot 2011). In literature, numerous studies on SAHP systems were presented for heating and domestic hot water proposes (Hepbasli et al, 2009, Raisul et al. 2012). Ito, Miura, and Wang (1999) carried out a theoretical and experimental study on the thermal performance of a heat pump using a bare flat-plate collector as the evaporator. Data in simulation section agreed well with experimental results, and COP reported about 5.3. Hawladar et al. (2001), studied the thermal performance of SAHP system. The results showed that COP and average collector efficiency is about 9 and of 75%, respectively. The thermal performance of refrigerant (R134a) based SAHP was studied by Saldo et al. (2004). The results indicated that COP and solar collector efficiency is between 4 and 9 and is 60–85%, respectively. Sporn and Ambrose integrated solar energy and heat pump technology in 1955 and the field of SAHP has since been in research interests. However, solutions concerning the consumer market are still few and most the SAHPs today are custom built (Muginer et al, 2013). Chaichana, Aye, and Charters (2003), evaluated the natural refrigerants in SAHP systems and pointed out that R744 (CO₂) can reach temperature up to its critical point. Cerit et al. (2013), investigated...
the performance of different rollbond collector-evaporator system with R134a working fluid. The results showed that COP factor can be varied from 2.42 to 3.30 based on the design parameters in Turkey climate condition. Panaras, Mathioulakis, and Belessiotis (2014), proposed a method called dynamic system testing (DST) as a black box approach for test and evaluation the performance of solar thermal heat pump hot water systems. The results showed that this method can predict a long-term performance of a SAHP system with an uncertainty rate below 10% only in the short test periods. Sami Buker and Riffat (2016), conducted a review of the applications of SAHP systems for low temperature water heating. The review faithfully stated that having a variety of configurations, parameters and performance criteria may lead to a major inconsistency that increase the degree of complexity to compare and analyze the studies of different systems. Postrioti et al. (2016), realized an experimental device to evaluate the wastewater energy potential in civil buildings. Araz et al. (2017), evaluated the performance of two combined systems, the wastewater-source heat pump (WWSHP) system and building-integrated photovoltaic (BIPV) system, using both energy and exergy analysis methods. This system conserved energy better and consumed less than gas-fired boilers. Wang et al. (2017), simulated and analyzed three sorts of rotating flexible heat transfer tubes by fluent software, managing to enhance the heat transfer performance of the wastewater heat exchanger in the wastewater source heat pump system. Anifantis et al. (2018), created a mathematical model to analyze the energy efficiency of photovoltaic, hydrogen and ground source gas in a stand-alone system used for heating a greenhouse tunnel during winter. The proposed system increased the greenhouse’s temperature by 3–9°C in relation to the external air temperatures.

Based on the above literatures and lack of similar studies in agricultural greenhouses, a SAHP is investigated experimentally and theoretically for a semisolar greenhouse located in Tabriz city, Iran. This system, detailed in next sections with inclusion of the novelty items, will expect to lower the temperature of the solar absorber, thus minimizing its heat loss to surrounding and improving the solar efficiency of the system. It is significant to deal with improving the heat pump systems’ thermal performance using R134a as refrigerant in order to enhancing the energy saving in the developing countries. In this study, an attempt has been made to recover the condenser heat and utilize it in space heating with renewable heat sources. This research will help promote development and market penetration of such an innovative solar heating technology, thus leading to reduction in fossil fuel consumption in the greenhouse and carbon emission to the environment.

So this study arranged into three main parts as follows:

- SAHPs and low temperature heating applications
- Design components and configurations
- Thermal performance characteristics of SAHP

**Study background**

In this paragraph, the parameters of the examined greenhouse and the most important mathematical equations are presented. It is essential to note that the greenhouse and the heating loads are the same in all the examined cases in order to make realistic comparisons for the different heating systems. EES software was used for simulation and the weather data of Tabriz city were taken from Bureau of Meteorology.

**The examined greenhouse**

The examined greenhouse characteristics are presented in this paragraph. Many parameters were selected which have typical values that usually exist in the building simulations. Table 1 gives the main dimensions of the building and the main loads. A greenhouse door was installed in the east side of the greenhouse with an area of 1.67 m² and the ventilation compartment of 0.25 m² in the west wall. The height of the south wall was 80cm and the south roof slope was 26.56°. The total area of the greenhouse was 42.77 m², that 31.25m² was covered with glass (4mm) and the rest total area of 11.52 m² was covered with a ceramic brick wall with
two layers of cement (the north wall). Figures 1 and 2 show the main and geographical views of the greenhouse. The desired temperature was set to be 15°C in order to create high quality thermal comfort conditions. For this temperature level, the maximum heating load of the greenhouse, according to calculations, is about 3.1 kW.

**Mathematical equations**

When the radiant energy of the sun enters the greenhouse due to the accumulation of heat loss of the walls \((Q_1)\), intrusive air \((Q_2)\) and ground \((Q_3)\) and also the greenhouse heat load is calculated as follows:

\[
Q_t = (Q_1 + Q_2 + Q_3 - Q_{solar}) \times SF
\]  

(1)

Where, \(SF\) is a confidence coefficient to compensate for the error which is approximately 1.2 (Albright 1990). \(Q_{solar}\) is the solar energy and calculated as 1.61 kW (Ghasemi 2016) and \(Q_t\) is the thermal value to be provided by the solar heat pump system. The energy balance equation for the solar collector is (Koelet 1992):

\[
\frac{dE}{dt} = m_{plate}C_{plate} \frac{dT_{plate}}{dt} = Q_{solar} - Q_{rad} - Q_{conv} - Q_{evap}
\]  

(2)

Where \(m_{plate}\) is the mass of the solar collector (kg), \(C_{plate}\) is the specific heat capacity of the solar collector (J/kgK), \(\frac{dT_{plate}}{dt}\) is the rate of changes in temperature in the solar collector (K/s), \(Q_{solar}\) is the absorbed solar radiation (W), \(Q_{rad}\) is the radiation exchange with the sky (W), \(Q_{conv}\) is the
convection heat transfer (W), and $Q_{\text{evap}}$ is the effect of heat pump evaporator (W). The mathematical representation takes the heat transfer in and out of the collector-evaporator where the collector temperature is supposed to be uniform. The initial conditions of $T_{\text{plate}}$ is the environment air temperature. The solar energy absorbed by the system is assumed to be as:

$$Q_{\text{solar}} = \alpha G A_{\text{plate--solar}}$$  \(3\)

Where $\alpha$ is the absorptivity of the plate in a dimensionless parameter, $G$ is the incoming radiation (W/m$^2$), and $A_{\text{plate--solar}}$ is the collector area being exposed to solar radiation (m$^2$). $Q_{\text{rad}}$ can calculated by:

$$Q_{\text{rad}} = \sigma \varepsilon_{\text{plate}} (T_{\text{plate}} - T_{\text{sky}}) A_{\text{evap--solar}}$$  \(4\)

Where $\sigma$ is the Stefan-Boltzmann's constant equal to $5.67 \times 10^{-8}$ (W/m$^2$K$^4$), $\varepsilon_{\text{plate}}$ is the emissivity, and $T_{\text{plate}}$ is the temperature of the plate. So, $T_{\text{sky}} = 0.0552 T_{\text{amb}}$ (K). Convective heat transfer calculated by:

$$Q_{\text{conv}} = h_{\text{conv}} (T_{\text{plate}} - T_{\text{amb}}) A_{\text{plate}}$$  \(5\)

Where $h_{\text{conv}}$ is the convective heat transfer coefficient of the environment air and $T_{\text{amb}}$ is the environment temperature. Convection heat transfer for flat plates solar thermal collectors with less than 0.5 meters long, can be calculated by the following empirical relation (Sartori 2006):

$$h_{\text{conv}} = 5.7 + 3.8 V_{\text{wind}}$$  \(6\)

Where $V_{\text{wind}}$ is the wind speed (m/s).

**Figure 2.** A view of the geographic location of the greenhouse walls.
The evaporative effect is determined by the state points of the thermodynamic cycle and the refrigerant mass flow and implemented in the following way:

\[ Q_{\text{evap}} = m_{\text{refrig}}(h_1 - h_4) \]  

Where \( m_{\text{refrig}} \) is the refrigerant mass flow rate (m/s) and \( h_1 \) and \( h_4 \) are the thermodynamic state points for the evaporator side of the heat pump. Equally, the condenser effect is implemented as follows:

\[ Q_{\text{cond}} = m_{\text{refrig}}(h_2 - h_4) \]  

Where \( h_2 \) and \( h_4 \) are the thermodynamic state points of the condenser side of the heat pump.

Compressor effect is implemented as follows:

\[ W_{\text{comp}} = m_{\text{refrig}}(h_2 - h_1) \]  

Where \( h_1 \) and \( h_2 \) correspond to the inlet and outlet of the compressor respectively. The refrigerant mass flow rate is defined by the following equation:

\[ m_{\text{refrig}} = \frac{NV\lambda(\pi)}{v_1} \]  

Where \( N \) is the constant rotational speed of the compressor in rotations per second, (rps), \( V \) is the clearance volume of the compressor (m\(^3\)), \( \lambda(\pi) \) is the volumetric efficiency dependent on the pressure ratio \( (\pi = p_2/p_1) \), and \( v_1 \) is the specific volume at state one of the heat pump cycle (m\(^3\)/kg). \( \lambda(\pi) \) can be calculated:

\[ \lambda(\pi) = 1 - \Phi_0\left(\pi^{1/k} - 1\right) \]  

Where \( k = c_p/c_v \) for the refrigerant and \( \Phi_0 \) is clearance volumetric fraction of the compressor. It was assumed all of the heat rejected by the condenser is accumulated in the water and stone tank and further treats the water mass in the tank as a uniform body with the same temperature. So:

\[ \dot{Q}_{\text{cond}} = (m_w C_w + m_s C_s) \Delta T \]  

Where \( m_w \) is the mass of the water (kg), \( m_s \) is the mass of the stone (kg), \( C_w \) is the specific heat capacity of the water (J/kg°C), \( C_s \) is the specific heat capacity of the stone (J/kgK), \( \Delta T \) is the rate of changes in temperature in the condenser (°C).

The energy balance of a (loss-free) heat pump process can be written as follows (Hadorn 2015):

\[ \dot{Q}_{\text{cond}} = \dot{Q}_{\text{eva}} + P_{\text{el,comp}} \]  

Where \( \dot{Q}_{\text{cond}} \) is the useful heat output from the condenser, \( \dot{Q}_{\text{eva}} \) is the heat input into the evaporator, and \( P_{\text{el,comp}} \) is the electric power needed to drive the compressor. The COP is defined as the amount of useful heat divided by the electric input:

\[ \text{COP} = \frac{\dot{Q}_{\text{cond}}}{\dot{W}_{\text{el}}} \]  

The theoretical upper limit for the COP is defined by the Carnot efficiency of the reversible heat pump cycle:

\[ \text{COP}_{\text{lim}} = \frac{T_{\text{cond}}}{T_{\text{cond}} + T_{\text{eva}}} \]  

Where the temperatures of the sink and the source should be applied in K. The collector efficiency is defined as follows (Klein 1975):
\[ \eta_c = \frac{Q_{\text{cond}}}{ITAC} \]  

(16)

**System description**

The initial calculations were made at January, the month with higher heating load. Figure 3 focuses on the relevant energy transfer rates occurring in the SAHP device. Heat delivered from the storage tank is based on user demand:

- \( q_{\text{in}} \): thermal power emitted from the surroundings to the solar panel (solar irradiation and other convective heat transfers);
- \( q_{\text{evap}} \): heat transferred to the operating fluid, equal to \( q_{\text{in}} \) in steady state conditions;
- \( q_{\text{cond}} \): thermal power from the operating fluid to the condenser, equal to \( q_{\text{out}} \) in steady state conditions;
- \( q_{\text{out}} \): thermal power to the water and stone reservoir;
- \( q_{\text{stg}} \): heat emitted from the storage tank, based on user demand. (In this study, the compressor will start up if the water temperature in the storage tank be less than 50°C);
- \( q_{\text{d}} \): thermal power transferred to the surroundings through the tank walls.

In the thermal process of system, the vapor compression refrigeration cycle is adopted, which \( P-h \) diagram is shown in Figure 4 and the \( T-s \) diagram shown in Figure 5. The working fluid called refrigerant passes through the compressor, the condenser, the capillary tube and the collector/evaporator in turn. The corresponding thermodynamics points denoted by 1 to 4. The refrigerant left the collector/evaporator at low pressure, low temperature, superheated vapor condition and entered the compressor (state 1). Then, it left the compressor at high temperature, high pressure, superheated vapor condition (state 2), and entered the condenser of water and stone tank, where it transferred heat to water. Then, the refrigerant left the condenser at high pressure condition and entered the capillary tube where it expanded irreversibly and adiabatically (state 3). At state No. 4, it left the capillary tube at low pressure, low temperature, low quality vapor and entered the collector/evaporator directly. The R-134a refrigerant properties used in this study are shown in Table 1 (Sonntag 2013).

**The experimental system**

The system consisted of the solar collector serving as the evaporator of the heat pump, a compressor, receiver, condenser, capillary tube, thermostat and a filter-drier. The solar collector was constructed...
by two refrigerator treason plates. The solar heat pump system linked to a latent heat energy storage tank was installed at the East-West semi-solar greenhouse in the postgraduate building of Tabriz University, Iran. The experimental greenhouse with length, width and height was 4.8, 3.2 and 2.4 m, respectively. Experiments was performed in four full days (24 to 27 December 2018), From 8 am to 6 pm, the parameters of temperature and humidity of the environment, temperature and humidity of the greenhouse, solar radiation perpendicular to the surface of the collector and the wind speed of the environment, and from 6 pm to 8 am, the parameters of the temperature and humidity of the greenhouse. Table 2 shows the climatic data for Tabriz city in experimental days.

Each refrigerator treason plate has about 95 cm long and 50 cm wide covered with the copper spiral pipes of about 8 mm in diameter. The plate is painted with black board color placed inside a metal box made of 0.8 mm thick iron sheet and insulated at the bottom and all of the sides with glass wool of 6 cm thick ($k = 0.038 \text{ W/m°C}$). The top of the metal box is glazed with a 3.5mm thick glass cover. The solar collector was installed in angle of 45° to the horizontal and quite to the south. The compressor used for the solar heat pump system was a piston type with 3/4hp. The used capillary tube was low pressure type with 3.02 m long and an inside diameter of 0.055 inch. The system installed in the greenhouse is shown in Figure 6. The summary of system information is presented in Table 3 and the main parameters used in system are shown in Table 4.

Results and discussion

In this section, the effect of different parameters on the thermal performance of the system is examined using the equations presented in the previous sections. It should be noted that in the numerical analysis of this subject, the equations of EES software with successive repetitions and experimental data, and similar analytical or numerical solutions, were confirmed by their integrity,
converged to the final solution. Then, these theoretical results are evaluated by the results of experimental data for validating them. Finally, the results of the study were analyzed and presented in the form of final conclusions and provided useful suggestions for future research. The effect of various parameters such as solar radiation and environmental temperature, evaporation and condensation temperature, solar collector area, compressor speed, number of collector cover and wind speed has been analyzed in order to understand the COP and efficiency of system ($\eta$). Tests were performed on four consecutive days from 24 to 27 December 2018. The effect of environmental parameters on the greenhouse temperature was compared with and without the heat pump using SAS9.1 software. In order to compare the temperature of the greenhouse in the night, the first night was as a control, and on the second night without the system, a thermal screen made of canvas with distance of approximately 80 cm from the floor of the greenhouse was used to reduce heat loss and on the third night the thermal screen with the function of the system was considered.

**Table 2.** Climatic condition of Tabriz city in experimental days.

<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Average environment temperature (°C)</td>
<td>3.72</td>
<td>2.28</td>
<td>4.52</td>
<td>6.05</td>
</tr>
<tr>
<td>Minimum environment temperature (°C)</td>
<td>-0.8</td>
<td>-3.2</td>
<td>2.7</td>
<td>1.6</td>
</tr>
<tr>
<td>Maximum environment temperature (°C)</td>
<td>6.9</td>
<td>7.1</td>
<td>7.3</td>
<td>9.3</td>
</tr>
<tr>
<td>Average relative humidity (%)</td>
<td>47.7</td>
<td>39.26</td>
<td>25.54</td>
<td>34.79</td>
</tr>
<tr>
<td>Average wind velocity (m/s)</td>
<td>0.06</td>
<td>0.04</td>
<td>0.05</td>
<td>0.48</td>
</tr>
<tr>
<td>Average solar radiation (W/m$^2$.day)</td>
<td>116.14</td>
<td>121.48</td>
<td>232.04</td>
<td>159.16</td>
</tr>
</tbody>
</table>

**Figure 6.** The system installed in the greenhouse.

**Table 3.** A sample of information about system concepts.

<table>
<thead>
<tr>
<th>Solar collector type</th>
<th>Solar heat sink</th>
<th>Heat sources for heat pump</th>
<th>Refrigerant</th>
<th>Heat pump sink</th>
<th>Storage concept</th>
<th>Additional heating (backup)</th>
<th>Application</th>
</tr>
</thead>
<tbody>
<tr>
<td>Flat plate (covered)</td>
<td>Heat pump evaporator</td>
<td>Solar collector (direct evaporation)</td>
<td>R134a</td>
<td>Space heating</td>
<td>Water and stone</td>
<td>Electricity (direct)</td>
<td>Space heating-air</td>
</tr>
</tbody>
</table>
The solar radiation and daily temperature

Figure 7 shows the average values of solar radiation and daily temperature from 8 am to 18 pm in experimental days in Tabriz city, where data were taken as an average hourly for 4 days and every 10 min of data were taken.

The values of coefficient of performance and collector efficiency during the test hours

Figure 8 shows that by increasing the amount of solar radiation and consequently increasing the temperature, COP and collector efficiency increases with relatively direct relation. These results were approved by Cai et al. (2017), that a novel dual source multifunctional heat pump is studied by theoretical and experimental methods and Alexandros Sotirios et al. (2018).

Table 4. Main parameters used in the system.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Compressor power</td>
<td>0.75 hp</td>
</tr>
<tr>
<td>Total solar collector area</td>
<td>About 1 m²</td>
</tr>
<tr>
<td>Collector plate (refrigerator treason) number</td>
<td>2</td>
</tr>
<tr>
<td>Flow path number of each plate</td>
<td>2</td>
</tr>
<tr>
<td>Pipe length of each plate</td>
<td>14.9 m</td>
</tr>
<tr>
<td>Thickness of collector plate</td>
<td>4 mm</td>
</tr>
<tr>
<td>Thermal conductivity of collector plate</td>
<td>336 W/m.°K</td>
</tr>
<tr>
<td>Internal diameter of the tube in solar collector plate</td>
<td>8.6 mm</td>
</tr>
<tr>
<td>External diameter of the tube in solar collector plate</td>
<td>9.4 mm</td>
</tr>
<tr>
<td>Absorptivity of collector plate (α)</td>
<td>0.9</td>
</tr>
<tr>
<td>Emissivity of collector plate (εp)</td>
<td>0.1</td>
</tr>
<tr>
<td>Emissivity of glass (εg)</td>
<td>0.88</td>
</tr>
<tr>
<td>Water volume of hot water tank</td>
<td>60 L</td>
</tr>
<tr>
<td>Pipe length of condenser</td>
<td>30 m</td>
</tr>
<tr>
<td>Internal diameter of the condenser pipe</td>
<td>8.4 mm</td>
</tr>
<tr>
<td>External diameter of the condenser pipe</td>
<td>9.9 mm</td>
</tr>
<tr>
<td>Initial water temperature</td>
<td>10.5°C</td>
</tr>
<tr>
<td>Final water temperature in</td>
<td>60°C</td>
</tr>
<tr>
<td>Refrigerant type</td>
<td>R134a</td>
</tr>
<tr>
<td>Refrigerant charge</td>
<td>About 2 kg</td>
</tr>
<tr>
<td>Solar radiation</td>
<td>1500 W/m²</td>
</tr>
<tr>
<td>Environment air temperature</td>
<td>−3.2°C</td>
</tr>
<tr>
<td>Wind speed</td>
<td>0.2 m/s</td>
</tr>
<tr>
<td>Optimum tilted angle of collector</td>
<td>45°</td>
</tr>
</tbody>
</table>

Figure 7. Variations of solar radiation and daily temperature during test hours.
COP and collector efficiency based on environment temperature changes

Figure 9 shows that by increasing the environment temperature, the surface temperature increased. As a result, the evaporation temperature increased but the amount of heat loss from the collector surface decreased. Thus, the collector efficiency and also the COP of the system increased. By increasing the average of environment temperature from 1.6 to 6.5°C, with constant solar radiation and compressor speed and wind speed, considering a specific area and collector glass cover, COP and collector efficiency increased from 3.54 to 5.46 and 51 to 94%, respectively, for example Gorozabel Chata, Chaturvedi, and Almogbel (2005); Molinaroli, M-Joppolo, and De Antonellis (2014); and Alexandros Sotirios et al. (2018) were studies in case of space heating with heat pump and these are good confirmation of this study.

Figure 8. COP and collector efficiency during the test hours.

Figure 9. The effect of environment temperature variation on the COP and collector efficiency.
**Coefficient of performance and collector efficiency values based on solar radiation variations**

Figure 10 shows that by increasing the solar radiation intensity, the evaporative temperature of the refrigerant in the collector, increased, which increased the amount of heat from the collector and reduced the work done by the compressor and thus increased the COP. Also with increasing the solar radiation, the absorbent surface temperature is increased and, as a result, the difference of the absorbent surface temperature and the environment temperature increased, which increasing the heat loss of the collector and decreased its efficiency. It should be noted that in low radiation levels, the absorbent surface temperature is less than the environment temperature and the collector in addition to the solar radiation absorption, absorbs some useful heat caused by the temperature difference between absorbent plate and environment temperature that it would increase the collector efficiency. By increasing the average of solar radiation from 67.16 to 235.19 W/m², with constant environment temperature and compressor speed and wind speed, considering a specific area and collector glass cover, the COP increased from 3.74 to 5.35 and the collector efficiency decreased from 88 to 36%, for example Molinaroli, M-Joppolo, and De Antonellis (2014) and Moreno-Rodriguez et al. (2013), that his study aims to present an experimental validation of a theoretical model for a direct-expansion SAHP applied to heating.

**Coefficient of performance and collector efficiency values based on evaporation temperature changes**

Figure 11 shows that by increasing the evaporation temperature, the amount of thermal loss in the collector surface decreased, which increased its efficiency. Also, an increase in the temperature of evaporation, increase the heat generated from the collector and therefore the COP increase. So with the evaporation temperature changes from 10 to 40°C, the COP and collector efficiency rise from 3.58 to 4.91 and 57% to 89%. Morad Ali and Jafar Kazemi (2013) examined the performance of a direct solar expansion heat pump system for water heating in Tehran’s climatic conditions and compared the COPs of this system with COP of a typical heat pump. Their results also confirmed this.

![Figure 10](image-url)  
*Figure 10. Effect of solar radiation intensity variations on the COP and collector efficiency.*
Coefficient of performance and collector efficiency values based on condensation temperature variations

According to Figure 12, it can be seen that the effectiveness of the COP and collector efficiency is maximum when the condensation temperature is low with increasing the condensation temperature, the compressor work increase, and reduce the COP of the system. Also with increasing the condensation temperature, refrigerant enthalpy increase when entering the solar collector and the refrigerant capacity for removing heat from the environment is less than it as a result, reduces the collector efficiency. As it can be seen, by increasing the condensation temperature from 40 to 70°C, the COP and collector efficiency decreases from 4.76 to 3.51 and 83% to 72%. This is confirmed by Gomri and Boulkamah (2011); Kong et al. (2011); and Tagliafico, Scarpa, and Valsuani (2014). They presented an approach for the steady state analysis of SAHPs. Their models were based on the inverse Carnot cycle and described the behavior of system.

Figure 11. Effect of evaporation temperature changes on the COP and collector efficiency.

Figure 12. Effect of condensation temperature variations on the COP and collector efficiency.
COP and collector efficiency values based on changes in solar collector area

Solar collector area is a very important structural parameter for optimization the system. In this study, six collector levels were tested with approximate dimensions of 0.1, 0.2, 0.3, 0.4, 0.5, and 0.6 m\(^2\). Figure 13 shows the effect of solar collector area on the system performance. It can be clearly seen that the COP increases and collector efficiency decreases sharply with increasing the solar collector area. This is mainly because of three reasons: (1) increasing the collector area enables enhancement of the useful energy gain of the bare flat-plate collector, consequently results in a higher COP system; (2) for the given environment air temperature and solar radiation intensity, the larger collector area is, the higher outlet temperature of the collector can be, and the electrical power consumption of compressor decreases; and (3) increasing the collector outlet refrigerant temperature changes the temperature difference between the collector plate temperature and environment temperature. When the collector plate temperature is lower than the environment temperature, the collector could obtain useful energy gain from the surroundings, which prove that the collector efficiency can exceed 1 with smaller collector area. The higher collector plate temperature enables temperature difference between collector plate temperature and environment temperature to decrease, even that the collector plate temperature is higher than environment temperature, and hence the collector efficiency decreases due to the change of heat transfer between the collector plate and environment temperature. With increasing the area from 0.1 to 0.6 m\(^2\), at steady environment temperature, compressor speed and solar radiation, the COP rises from 3.58 to 4.63 and the collector efficiency decreases from 90% to 62%, for example Paradeshi, Srinivas, and Jayaraj (2016) and Gomri and Boulkamah (2011), were carried out the theoretical and experimental studies on a direct expansion SAHP (DX-SAHP) under the different metrological condition.

Coefficient of performance and collector efficiency values based on changes in compressor speed

Figure 14 shows that the effect of compressor speed changes in the values of the COP and collector efficiency at steady environment temperature and solar radiation. With increasing the speed, the compressor works more and also due to the increasing in temperature of refrigerant discharge, the compressor losses increased, and this reduce the COP of the system. However, by increasing the compressor speed and consequently increasing the flow rate of the refrigerant in the collector, its surface temperature decreases, which reduces the thermal losses from the collector surface and increases its efficiency. The behavioral

![Figure 13. Effect of solar collector area on the COP and collector efficiency.](image-url)
The mismatch of the COP and collector efficiency with compressor speed variations indicate that there is an optimal amount of compressor speed for the constant level of the collector in the constant solar radiation and environment temperature which the system design is based on this value (approved by studies conducted in the field of heat pump systems by Zhang et al. 2014, Tzivanidis et al. 2016, and Zhongchao et al. 2018). With rising the compressor speed from 1500 to 3500 rpm, the COP reduces from 4.91 to 3.76 and collector efficiency increases from 61% to 87%.

**COP and collector efficiency values based on the number of collector cover**

Figure 15 shows the values of the COP and collector efficiency for without cover and with one to three glassy covers. The cover layer acts on the surface of the absorbent plate as a thermal insulation that cause to capture solar radiation at a specific wavelength. This subject causes reduction of the heat transfer from the absorbent surface and also decrease the heat loss from the collector surface and increases the collector efficiency. By reducing the heat transfer of the collector surface, the temperature of the absorbent surface and the temperature of the refrigerant evaporative increases. This increases the COP system. As shown in Figure 14, the collector with one cover relative to no cover, rises the COP and collector efficiency from 4.1 to 4.7 and 77% to 84%. But with using more than one glass cover, the COP and collector efficiency increases slightly and, on the other hand, increases the cost of the system, and thus has no economic justification. In study of Tagliafico, Scarpa, and Valsuani (2014), also was referred to this issue.

**Statistical analysis**

All data recorded daily basis and then averaged over the data that were recorded at the same time in 4 days of the experiment. Due to different weather conditions in day and night, data were divided into two parts of the day (8 am–6 pm) and night (6 pm–8 am). In order to easily make comparisons in software and according to the same conditions of the first and second days, the average of these days as Day1 and the average of the data of days 3 and 4 was considered as Day2. The first night was Night1, the second night was Night2, and the average of the third and fourth night data was introduced according to the same test conditions as Night3. All relevant charts drawn using the mean of the data and relationships presented in this study in Excel 2010 software. Data analysis performed using SAS9.1 software. Table 5 shows that statistics of the factors investigated in the Day1
and Day2 were tested as well as their differences. The information from the left-to-right table is the name of the variable, the sample code (Day1, Day2), number of observations per sample, the mean, standard deviations, and the standard error. Variables are included environment temperature \( (T_o) \), environment humidity \( (RH_o) \), environment wind \( (W_o) \), sun radiation \( (I_o) \), greenhouse temperature \( (T_{in}) \), and greenhouse humidity \( (RH_{in}) \).

In Table 6, the result of T test for the mean comparison of the investigated factors showed that the difference between the mean of all factors in Day1 and Day2 with a probability error of less than 0.05 was significant. For example Artur, Magdalena, and Klaudia (2018) was conducted a case study to analyze the possibility of using a heat pump for greenhouse heating in Poland which is in good agreement with the results of this study.

To obtain the regression relationship, using stepwise regression, all of the variables under consideration were analyzed and in the process of calculating an independent variable that has no significant effect on the greenhouse temperature, it is eliminated and is not included in the equation.

Figure 15. Effect of number of collector covers on the COP and collector efficiency.

<table>
<thead>
<tr>
<th>Variable</th>
<th>Day</th>
<th>N</th>
<th>Mean</th>
<th>Std Dev</th>
<th>Std Err</th>
</tr>
</thead>
<tbody>
<tr>
<td>( T_o )</td>
<td>1</td>
<td>20</td>
<td>3.003</td>
<td>1.4555</td>
<td>0.3255</td>
</tr>
<tr>
<td>( T_o )</td>
<td>2</td>
<td>20</td>
<td>5.2855</td>
<td>2.2047</td>
<td>0.493</td>
</tr>
<tr>
<td>( T_o )</td>
<td>Diff (1–2)</td>
<td>-</td>
<td>−2.283</td>
<td>1.868</td>
<td>0.5907</td>
</tr>
<tr>
<td>( RH_o )</td>
<td>1</td>
<td>20</td>
<td>43.478</td>
<td>14.222</td>
<td>3.1801</td>
</tr>
<tr>
<td>( RH_o )</td>
<td>2</td>
<td>20</td>
<td>30.165</td>
<td>12.133</td>
<td>2.713</td>
</tr>
<tr>
<td>( RH_o )</td>
<td>Diff (1–2)</td>
<td>-</td>
<td>13.314</td>
<td>13.219</td>
<td>4.1801</td>
</tr>
<tr>
<td>( W_o )</td>
<td>1</td>
<td>20</td>
<td>0.048</td>
<td>0.0244</td>
<td>0.0055</td>
</tr>
<tr>
<td>( W_o )</td>
<td>2</td>
<td>20</td>
<td>0.265</td>
<td>0.3276</td>
<td>0.0732</td>
</tr>
<tr>
<td>( W_o )</td>
<td>Diff (1–2)</td>
<td>-</td>
<td>−0.217</td>
<td>0.2323</td>
<td>0.0734</td>
</tr>
<tr>
<td>( I_o )</td>
<td>1</td>
<td>20</td>
<td>118.81</td>
<td>46.97</td>
<td>10.503</td>
</tr>
<tr>
<td>( I_o )</td>
<td>2</td>
<td>20</td>
<td>195.6</td>
<td>96.274</td>
<td>21.527</td>
</tr>
<tr>
<td>( I_o )</td>
<td>Diff (1–2)</td>
<td>-</td>
<td>−76.79</td>
<td>75.746</td>
<td>23.953</td>
</tr>
<tr>
<td>( T_{in} )</td>
<td>1</td>
<td>20</td>
<td>8.027</td>
<td>4.2036</td>
<td>0.9399</td>
</tr>
<tr>
<td>( T_{in} )</td>
<td>2</td>
<td>20</td>
<td>29.48</td>
<td>12.033</td>
<td>2.6906</td>
</tr>
<tr>
<td>( T_{in} )</td>
<td>Diff (1–2)</td>
<td>-</td>
<td>21.45</td>
<td>9.0127</td>
<td>2.8501</td>
</tr>
<tr>
<td>( RH_{in} )</td>
<td>1</td>
<td>20</td>
<td>50.67</td>
<td>15.973</td>
<td>3.5717</td>
</tr>
<tr>
<td>( RH_{in} )</td>
<td>2</td>
<td>20</td>
<td>40.068</td>
<td>16.067</td>
<td>3.5927</td>
</tr>
<tr>
<td>( RH_{in} )</td>
<td>Diff (1–2)</td>
<td>-</td>
<td>10.603</td>
<td>16.02</td>
<td>5.066</td>
</tr>
</tbody>
</table>
This is confirmed by Artur, Magdalena, and Klaudia (2018). According to Table 7, the regression relation is as follows:

\[
T_{in} = -7.59261 + 4.1717T_o + 0.05762I_o
\]  

To analyze the significant difference between greenhouse temperature and humidity in three nights experiment, analysis of variance was used. According to Table 8, it can be concluded that there is a significant difference between the two nights in terms of average greenhouse temperature.

The comparison of average greenhouse temperature at nights of experiment, assessed using Duncan’s test. According to Table 9, the opposite alphabet indicates a significant difference at the error level of 0.05. The average of greenhouse temperature at nights 1, 2, and 3 had significant difference and the mean greenhouse temperature in the third night was significantly higher than the second and first night. In Figures 16 and 17, the differences in greenhouse temperatures were shown on the first and second day as well as at three-night experiments.

| Table 6. The results of T-test analysis in this study. |
|---------------------------------|-----------------|-----------------|
| Variable | $Pr > F$ | Variances | $Pr > |t|$ |
| $T_o$   | 0.0781 | Equal | 0.0004** |
| $RH_o$ | 0.4952 | Equal | 0.0029** |
| $W_o$   | >0.0001 | Equal | 0.0081** |
| $I_o$   | 0.0030 | Equal | 0.0034** |
| $T_{in}$ | >0.0001 | Equal | >0.0001** |
| $RH_{in}$ | 0.9798 | Equal | 0.0431* |

| Table 7. The REG procedure. |
|-------------------------------|-----------------|-----------------|
| Variable | Parameter estimate | $Pr > F$ |
| Intercept | -7.59261 | 0.0033 |
| $T_o$ | 4.1717 | >0.0001** |
| $I_o$ | 0.05762 | 0.0001** |

| Table 8. The GLM procedure in this study. |
|---------------------------------|-----------------|-----------------|
| Dependent variable | Source | $Pr > F$ |
| $T_{in}$ | Model | >0.0001** |
| | Night | >0.0001** |
| $RH_{in}$ | Model | 0.3580 ns |
| | Night | 0.3580 ns |

| Table 9. Duncan’s multiple range test for greenhouse temperature. |
|--------------------------------|-----------------|-----------------|
| Night | $N$ | Mean | Duncan grouping |
| 3 | 28 | 8.5361 | A |
| 2 | 14 | 3.9964 | B |
| 1 | 14 | 2.2700 | C |
Conclusions

Based on the experiment and developed system simulation model, the thermal performance of direct expansion SAHP system has been studied under metrological condition of Tabriz in Iran. It is found that COP to be vary from 3.64 to 4.48. The experimental values agree well with the simulation results with an average error of 2%. According to the simulations, the thermal performance of the SAHP system for different parameters are studied, which is shows that system performance affected considerably by variation of solar collector/evaporator area, solar insolation, environment temperature and followed by the effect of collector covers.

In the current study, a SAHP system for low-temperature water heating applications has been presented. The results of simulation and thermodynamic analysis of a SAHP water heater with a 60-l water and stone tank were presented. Effective parameters on the performance of system were investigated. According to calculation, increasing solar irradiance and environment temperature, reducing water and stone tank temperature and compressor speed, all enhance COP.

By the above experiments and the collection of data, we can find out that the use of SAHP in greenhouse is feasible and has a certain effect. The solar collector area of the experimental system is 1 m² and the room area is 15.36 m². In the case of Tabriz region sunshine, the rise of the greenhouse temperature can reach 16°C and thus the 1 m² solar collectors as an auxiliary heating can heat 15 m² room. By the experiment, conclusions are drawn as follows:

(1) SAHP can meet the temperature requirements of the greenhouse. The average temperature of the greenhouse reached to 30°C in day and 9°C in night.
(2) During the experiments, the minimum of storage tank temperature was 21°C and its maximum temperature was 60°C, which was turned off by the thermostat’s command when it reached this temperature.

(3) Tabriz locates in an area with abundant solar energy resources. The solar heating is both environmentally friendly and energy conservation, research should be focused.

(4) By utilizing the greater number of additional solar collectors, the total amount of thermal loss from the solar collector surface is reduced very slightly. The application of a coating layer increased the performance coefficient by 11% and increased the efficiency of 12% by collecting, but using more than one coating layer, the slope of increasing the COP and collector efficiency declined and made economically feasible and the use of more than one layer of coating for the collector is not recommended.

(5) By increasing wind speed, increases the total thermal loss coefficient from the collector surface.

(6) The increase in wind speed compared to other factors has a very small effect on the thermal performance of the system, and at a very low level, increases the thermal performance of the system and reduces the collector efficiency.

(7) The system in summer months, which has an environment temperature as well as a higher initial water temperature in hot water tank, has fewer working hours and increases in winter months during the working hours of the system.

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