A comprehensive comparative investigation on solar heating and cooling technologies from a thermo-economic viewpoint—A dynamic simulation

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Abstract
The yearly thermo-economic performance is dynamically investigated for three solar heating and cooling systems: solar heating and absorption cooling (SHAC), solar heating and ejector cooling (SHEC), and heating and solar vapor compression cooling (HSVC). First, the effects of important design parameters on the thermo-economic performance of the systems to supply the heating and cooling loads of the building are evaluated. The systems are parametrically analyzed with the weather conditions of Tehran, Iran. The results show that the life cycle costs (LCC) of the SHAC and HSVC systems are alike and much lower than those of the SHEC system. The HSVC system exhibits the best performance from exergetic and solar fraction viewpoints. The comparative analysis shows that the energy efficiencies of the SHAC and SHEC systems are higher in colder climatic conditions. However, the collector efficiency of the HSVC system declines in colder climates, mainly due to the lower solar intensities relative to in hotter climates. Further, the solar fraction of the SHAC system is higher than the SHEC technology under all climatic conditions. Moreover, higher values of solar fractions are obtained under colder weather conditions for the SHEC and HSVC systems. The best economic performance is observed for the SHAC and HSVC technologies, having significantly lower LCCs than the SHEC system. These lower LCCs under colder climatic conditions are due to the lower cost of supplying the heating load compared to the cooling load. Furthermore, all systems exhibit enhanced exergetic performance in colder weather conditions. The yearly thermo-economic performance is dynamically investigated for three solar heating and cooling systems: SHAC, SHEC, and HSVC. In addition, the effects of important design parameters on the thermo-economic performance of the systems to supply the heating and cooling loads of the building are evaluated.

KEYWORDS
absorption chiller, dynamic simulation, ejector cooling, solar cooling and heating, thermo-economic analysis
1 | INTRODUCTION

The estimations show that about 30%-40% of worldwide energy consumption is in buildings. Suppling this high energy requirement for buildings with fossil fuels increases environmental emissions and global warming. Solar air-conditioning technologies provide an eco-friendly replacement for conventional air-conditioning systems and are highly capable of satisfying this energy demand in buildings.

Solar cooling technologies encompass systems driven by either thermal energy or electricity, namely thermally and electrically driven solar cooling systems, respectively. The core of electrically driven systems is the conventional vapor compression cycle (VCC), which provides cooling. These systems can be driven by either photovoltaic (PV) modules or solar-driven power cycles (usually an organic Rankine cycle [ORC]). A comprehensive review on thermally and electrically driven cooling technologies was reported in Sleiti et al. A VCC driven by PV (denoted PV-VCC) is the most common solar electrically driven cooling technology for small-scale applications, primarily due to its advantages such as compactness and easy maintenance. Although the cost of PV cells has declined significantly over the last few years, the high price of battery storage has restricted the application of PV-VCC to only sunny hours. Huang et al. experimentally estimated the operating probability of six PV-VCC systems with different sizes of PV panels and air conditioners at various levels of solar irradiation. An operation probability of about 98% was reported at for a solar irradiation value of more than 600 W/m². Energy storage systems have been suggested for extending the working hours of PV-VCC systems. For instance, the integration of a thermochemical reactor as an energy storage system in a PV-VCC system was studied in Ferrucci et al. The proposed system exhibited a cooling capacity of 4 kWh/day per square meter of PV panel area, which is more than that of PV-VCC systems using conventional electrochemical batteries.

Solar thermal cooling technology has drawn more attention compared to solar electricity driven cooling systems owing to its distinct advantages. These advantages include competitive energy-to-cooling efficiency, heat recovery capability, and applications where the noise of the compressor of the VCC is problematic. Generally, thermally driven cooling cycles can be classified into four main categories: absorption cycle, adsorption cycle, desiccant cycle, and ejector cycle.

Although the COP is usually lower for the ejector cooling cycle than other systems, the simple structure and low maintenance cost of this technology makes the solar ejector cooling system a viable option for building cooling. Salimpour et al, a comparative exergoeconomic study, were performed on four solar ejector cooling configurations utilizing flat plate solar collectors. A preheater and precooler were employed to enhance the performance of the ejector cooling system. The lowest total investment cost of the proposed configurations was 0.19 $/h. At a cooling load of 5 kW, it was suggested for all configurations to have evaporator and condenser temperatures at 278 and 311 K, respectively. In another study, a dynamic simulation using TRNSYS and EES (Engineering Equation Solver) software was carried out for a 7 kW solar ejector cooling system. The key variables considered were solar collector type and area, volume of storage tank and the working fluid flow rate of the cycle and the overall COP was found to range between 0.32 and 0.47. A novel solar-driven combined system comprising of solar still and ejector cooling systems was proposed by Sleiti et al. The integration of both systems resulted in a significant improvement in the productivity of the solar still and enhanced the COP of the ejector system.

Compared to other thermally driven cooling cycles, a key advantage of absorption cooling cycles is their relatively high COP. Moreover, the levelized cost of the cooling load of this cooling technology usually is competitive, especially when employed in large buildings. Numerous investigations have been performed on solar absorption cooling technologies. Khan et al. proposed two absorption refrigeration cycle configurations and performed dynamic simulations of them using TRNSYS. Their findings showed a considerable difference (up to 30%) between monthly collector efficiencies when using evacuated tube collector (ETC) or flat plate collector. The viability of using the solar absorption cooling cycle has been investigated for ten cities. A dynamic thermo-economic study was performed and highlighted the need for considering economic as well as energy factors in designing absorption cooling cycles. Bellos et al. investigated thermo-economically the effects of storage tank volume and collector area on the performance of a solar absorption cooling cycle. For the case exhibiting the best economic performance, the cycle had a 15-year payback period and a net present value of 67 000€. In another study, the performance of a 5-t absorption cooling cycle in Iran was dynamically analyzed using TRNSYS. The findings demonstrated that the system solar fraction could be improved by 28%. With the aim of lowering electricity use in an absorption solar cooling system, Nienborg et al. conducted a dynamic simulation in which controlling strategies were employed to decrease the auxiliary electricity consumption. It was found that appropriate flow rates and temperature set points resulted in electricity savings of up to 25%. Some studies have examined the use of different concentrating solar collectors, such as compound parabolic collectors, ETCs, and parabolic trough collectors in solar absorption cooling.
systems. Comparatively, ETCs appear to be a better option for building cooling using absorption cycles due to their reasonable costs and high flexibility for installation on roofs, which is not the case for PTCs. 25

Solar heating and cooling (SHC) systems have the privilege of supplying both heating and cooling loads for buildings in year-round operation. In spite of this great advantage of the SHC systems over the solar cooling systems, a limited number of investigations have been performed on the analysis and comparison of different SHC systems. Moreover, the studied SHC systems have consisted mainly of absorption cycles. 27-29 For instance, Delač et al. 30 dynamically simulated a SHC system in which waste heat from the absorption cooling subcycle was utilized for preheating domestic hot water. It was shown that, although only 8% of the heat released from the condenser and the absorber was recovered, the proposed SHC system achieved up to 53% of heat recovery when used in appropriate scenarios. A comprehensive study and a multi-objective optimization of a SHC system were conducted by Shirazi et al. 31 from energetic, economic, and environmental perspectives. Various absorption cycle configurations driven by different solar collector types were investigated and compared. The optimization results showed that the double-effect absorption chiller exhibited the best economic performance of the evaluated options, with a levelized total annual cost of 0.7–0.9 M$. 32

According to this literature review, few investigations have been conducted on solar heating and cooling (SHC) systems compared to solar cooling technologies. Moreover, the majority of SHC investigations have focused only on the employment of absorption cooling cycles, and the possibility of using PV-VCC as well as solar ejector cooling systems to supply both cooling and heating demands of buildings has received less attention. As far as the authors know, a comprehensive comparative investigation of electrically driven and thermally driven SHC technologies has not been conducted heretofore. Therefore, this study using TRNSYS presents a year-round dynamic simulation of three SHC systems, namely solar heating and absorption cooling (SHAC), solar heating and ejector cooling (SHEC), and heating and solar vapor compression cooling (HSVC). The annual thermo-economic performances of these SHC systems are evaluated for supplying both cooling and heating requirements of a building where is located in several cities in Iran (Tehran, Tabriz, Hamedan, Isfahan, Bushehr, and Kerman). The effects of key design parameters on the thermo-economic criteria are also determined for each SHC system to identify the best-operating condition. To compare the thermo-economic performances of SHC systems, a comparison is also made at the best-operating condition, which lays the groundwork for selecting the best SHC system for different climatic conditions.

2 | DESCRIPTION OF SYSTEMS

2.1 | Thermally driven systems (SHAC and SHEC systems)

Schematic diagrams of SHAC and SHEC systems are provided in Figure 1A,B, respectively. The only difference between these two thermally driven SHC systems is the cooling cycle. An absorption chiller is used to provide cooling for the SHAC technology, while an ejector cooling cycle is utilized for the SHEC system. The SHAC and SHEC systems are each comprised of six main components: solar thermal collectors, storage tank, auxiliary heaters, cooling cycle (absorption chiller or ejector cooling cycle), circulating pumps, and wet cooling tower. In the solar subsystem, ETCs are utilized, and a storage tank is employed to store the solar energy and stabilize the heat source temperature. In the case of instabilities in solar energy, two auxiliary heaters, with different temperature set points for the cooling and heating modes, supply the extra energy requirement so as to ensure the system satisfies the heating and cooling loads. Water is selected as the heat transfer fluid for both the SHAC and SHEC systems, while lithium bromide-water (LiBr-H$_2$O) and R134a are considered to be the working fluids for the absorption chiller and ejector cooling cycles, respectively.

2.2 | Electrically driven systems (HSVC system)

The HSVC system considered is presented in Figure 1C. It consists of seven main components: PV panels, inverter, battery, vapor compression chiller, wet cooling tower, heater, and circulating pumps. A heater is utilized to provide the heating load of the building. Furthermore, a vapor compression chiller powered by PV panels meets the cooling load. Since the generated electricity by the PV panels is DC, an inverter is employed to convert it to AC before supplying it to the vapor compression chiller. Note that grid-connected PV panels are used to ensure steady operation of the vapor compression chiller when the generated electricity by PV panels is insufficient due to instabilities in solar energy. In addition, when the generated electricity by PV exceeds the chiller consumption, the surplus is sold to the grid.

3 | METHODOLOGY AND ASSUMPTIONS

3.1 | System simulations

In this study, annual dynamic simulations of the proposed SHC technologies are performed using TRNSYS. 32 The basic system components utilized in the three SHC systems are selected from predefined types in TRNSYS. Also, EES
is employed to simulate the components not available in TRNSYS like the ejector cooling system. The system components and simulation assumptions of the three SHC technologies are listed in Tables A.1 and A.2 of Appendix A. For the SHEC system simulation, an ejector of fixed dimensions identical to the one used by Pridasawas and Lundqvist is employed, and the ε-NTU method is applied to determine the heat transfer rates in the heat exchangers inside the ejector chiller (generator, evaporator, and condenser). For more information about the simulation of the ejector chiller, the reader is referred to the authors’ previous work. For better functioning of the proposed SHC systems, a number of controllers are employed. For the SHAC and SHEC systems, a controller is utilized in the solar subcycle whereby the circulating pump functions only when the temperature difference of the inlet and outlet of ETC is more than 2°C, and the circulating pump stops functioning when this temperature difference is under 1°C. Furthermore, the operation of auxiliary heaters employed in both the SHAC and SHEC systems is dictated by controllers. To attain the fixed temperature set points, the controllers are switched off or on in the auxiliary heaters to ensure the heating and cooling loads are satisfied. Note that all components for supplying the cooling loads in both thermally driven technologies (ie, auxiliary heater, absorption chiller and cooling tower in SHAC and auxiliary heater, ejector chiller, and cooling tower in SHEC) are switched on or off simultaneously according to the cooling load. For the SHVC system, a controller is employed to switch on the heater when there is a need for heating, and another controller is utilized to switch on the vapor compression chiller and cooling tower so as to satisfy the cooling demand. The utilization of grid-connected PV panels not only ensures the steady operation of the vapor compression chiller but also enables electricity sales to the power grid when there exists excess electricity is generated.

3.2 | Building simulation

To examine the thermo-economic performance of the SHC systems under various climatic conditions, a residential
building located in various cities of Iran was simulated in TRNSYS using the multi-zone building model (Type 56a). Set-point temperatures for the heating and cooling modes were considered 20 and 26°C, respectively. The parameters used for calculating the building thermal loads are presented in Table 1.

In order to simulate the building loads by Type 56a, a variety of data were specified, such as weather information, specifications of windows, and geometrical properties of the building. Climatic information for the three studied cities was implemented by Type 109. Heat gains by occupants were determined by ISO7730 standard, and the heat generated from equipment (computers, refrigerators, washing machines, etc.) was estimated based on the default values in TNRSYS. The Rate of Infiltration and exchange rates of air were taken from the ASHRAE 90.1.

| TABLE 1 | Parameters used for determining the building thermal loads |
|---|---|---|
| **Title** | **Unit** | **Value/Level** |
| Infrastructure area | m² | 200 |
| Number of occupants | — | 4 |
| Occupants’ activity | — | Light work- Seated |
| Exchange rate of air | AC/h | 1 |
| Rate of infiltration | AC/h | 0.16 |
| Level of lighting | W/m² | 5 |
| Maximum solar irradiation | W/m² | Tehran:1038 Tabriz: 1036 Kerman: 1083 Isfahan: 1093 Bushehr: 1117 Hamedan: 1085 Mazandaran:1043 Mashhad: 1059 |
| Maximum summer temperature | °C | Tehran: 41 Tabriz: 37 Kerman: 42 Isfahan:41 Bushehr: 45 Hamedan: 39 Mazandaran:35 Mashhad: 37 |
| Minimum winter temperature | °C | Tehran: −8 Tabriz: −14 Kerman: −11 Isfahan: −12 Bushehr: 3 Hamedan: −20 Mazandaran: 0 Mashhad: −12.5 |
| Design temperature in cooling mode | °C | 26 |
| Design temperature in heating mode | °C | 20 |

### 3.3 Energetic performance criteria

#### 3.3.1 Thermally driven systems (SHAC and SHEC systems)

The following performance indicators are applied to the thermally driven systems and used for comparisons:

**Collector efficiency**

The collector efficiency is calculated by Equation 1:

\[
\eta_{\text{collector}} = \frac{\dot{Q}_{\text{solar}}}{\dot{Q}_{\text{in}}}
\]

Here, \(\dot{Q}_{\text{solar}}\) and \(\dot{Q}_{\text{in}}\) denote the rate of gained solar energy by the solar collectors and the rate of total solar energy incident on the collector, respectively.

**Solar fraction**

The solar fraction (SF) is defined as the ratio of the solar energy provided to the total energy input to the system and can be expressed as follows:

\[
\text{SF} = \frac{\dot{Q}_{\text{solar}}}{\dot{Q}_{\text{solar}} + \dot{Q}_{\text{auxiliary}}}
\]

where \(\dot{Q}_{\text{Auxiliary}}\) denotes the required auxiliary energy rate which is supplied by auxiliary heaters.

**Energy efficiency**

The energy efficiency of the system is written as follows:

\[
\eta_{\text{energy}} = \frac{\dot{Q}_{\text{cooling}} + \dot{Q}_{\text{heating}}}{\dot{Q}_{\text{solar}} + \dot{Q}_{\text{auxiliary}} + W_{\text{pumps}} + W_{\text{fan}}}
\]

where \(\dot{Q}_{\text{cooling}}, \dot{Q}_{\text{heating}}, W_{\text{pumps}},\) and \(W_{\text{fan}}\) denote building cooling load, building heating load, pump power, and energy use rate of the cooling tower fan, respectively.

#### 3.3.2 Electrically driven system (HSVC system)

The following indicators are used for investigating the performance of the HSVC system:

**Collector efficiency**

The of PV panels efficiency can be expressed as follows:

\[
\eta_{\text{collector}} = \frac{P_{\text{generated}}}{\dot{Q}_{\text{in}}}
\]

Here, \(P_{\text{generated}}\) is the power generated by the PV panels and \(\dot{Q}_{\text{in}}\) is the rate of total solar energy incident on the collector, respectively.
where $P_{\text{generated}}$ and $\dot{Q}_d$ denote the generated power by PV panels and the solar energy rate incident on the PV panels, respectively.

### 3.3.3 Electrical efficiency

The electrical efficiency is formulated as follows:

$$\eta_{\text{electrical}} = \frac{P_{\text{chiller}} + P_{\text{sell}}}{P_{\text{generated}} + P_{\text{purchase}}}$$  \hspace{1cm} (5)

Here $P_{\text{chiller}}$, $P_{\text{sell}}$, and $P_{\text{purchase}}$ denote chiller power consumption, power sold to the grid, and power purchased from the grid, respectively.

### Solar fraction

The solar fraction is expressed as follows:

$$\text{SF} = \frac{P_{\text{generated}}}{P_{\text{generated}} + P_{\text{purchase}}}$$  \hspace{1cm} (6)

### 3.4 Exergetic performance criteria

Considering each component as a control volume and neglecting potential and kinetic energies, a general exergy balance can be expressed by Equation 7:

$$\sum_i m_i \dot{e}_{\text{in}} + \dot{E}_Q = \sum_{\text{out}} m_{\text{out}} \dot{e}_{\text{out}} + \dot{E}_{W} + \dot{I}$$  \hspace{1cm} (7)

Equation 7 indicates that the input exergy rate to a control volume (exergy rates of the input stream and heat) equals the sum of the outlet flow exergy rates (including the exergy rate of the outlet stream and the work rate), and exergy destruction rate.

### 3.5 Cost analysis

The life cycle cost (LCC) is selected as the economic index for this study. The future costs and today’s costs can be compared with this approach. The present worth factor (PWF) is formulated as:

$$\text{PWF}(N, i, e) = \frac{1}{(e-i)} \left[ 1 - \left( \frac{1+i}{1+e} \right)^N \right]$$  \hspace{1cm} (8)

In which $e$ is the market discount rate, $i$ denotes the energy inflation rate, and $N$ is the periods (in years).

LCC is defined as follows:

$$\text{LCC} = C_{\text{initial}} \left[ 1 + f_{\text{om}} \text{ PWF} - f_{\text{salv}} \left( \frac{1+i}{1+e} \right)^N \right] + Q_{\text{annual}} \cdot \frac{\text{Price}}{\eta_{\text{eff}}} \cdot \text{PWF}$$  \hspace{1cm} (9)

In Equation 9, $C_{\text{initial}}$, $f_{\text{om}}$, $f_{\text{salv}}$, Price, and denote the initial cost of the system, the operating and maintenance fraction, salvage fraction, energy price, system efficiency, and the annual energy consumption, respectively. Note that the initial cost of components is calculated based on the equations in Table 2.

### 4 RESULTS AND DISCUSSION

The results are discussed in two sections. In the first section, the results from the parametric analysis of three SHC systems for the climatic conditions of Tehran are described. This parametric analysis highlights the effects of key variable parameters on the thermo-economic performance of each SHC system. In the second section, outcomes regarding the thermo-economic performance of the SHC systems at their best-operating conditions and under various climates are reported. The results obtained from the comparison of the

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**TABLE 2** Equations used for calculating initial cost of components

| Component               | Price function |
|-------------------------|----------------|
| Solar collector         | 221 ($/m^2$)   |
| PV panel                | 2000 ($/kW$)   |
| Storage tank            | $Z_{\text{tank}} = 297.36 \times V + 140.85$ ($) |
| Inverter                | 470 ($/kW$)    |
| Pump                    | $Z_{\text{pump}} = 705.48 \times W^{0.71} \times \left( 1 + \frac{0.2}{1 - \eta_{\text{pump}}} \right)$ ($) |
| Vapor compression chiller | $Z = (30.763) + 6739$ ($) |
| Cooling tower           | $Z = 746.749 \times (\dot{m}_{CT})^{0.79} \times (\Delta T)^{0.57} \times (T_{CT} - T_{WT, out})^{-0.9924} \times (0.022T_{WT, out} + 0.39)^{2.447}$ ($) |
| Absorption chiller      | 440 ($/kW$)    |
| Ejector chiller         | $Z = (4.9263 \times \dot{Q}_{HT} + 438630) / 7000$ ($) |
| Auxiliary heater        | 6 ($/kW$)      |
three SHC systems operating under different climates lay the foundation for the selection of suitable SHC systems for the corresponding climatic condition.

### 4.1 Parametric analysis results

The key variable parameters for each of the SHC systems are listed in Table 3. To perform the parametric study, when a parameter changes, the other parameters remain unchanged as given in Table 3 (base case). As stated before, the climate of Tehran was chosen to perform the parametric analysis.

#### 4.1.1 Solar heating and absorption cooling

Figure 2 shows the effect of the set-point temperature (SPT) of the heating auxiliary heater (HAH) on the thermo-economic performance of the SHAC system. It can be seen that with an increase in SPT of the HAH from 60 to 80°C, the SF, energy efficiency, and collector efficiency decrease by 4.3%, 0.9%, and 2%, respectively. Because when the SPT increases, the required auxiliary heat rises, leading to an increase in the tank temperature. Therefore, the solar collector inlet temperature rises, resulting in a decline in collector efficiency. In fact, the heat absorbed by the solar collectors decreases while the energy consumption in the HAH increases. Therefore, SF, energy efficiency, and collector efficiency each experience a downward trend with increasing the HAH set-point temperature. It is evident that the exergy destruction and LCC remain relatively unchanged with an increase in SPT of the HAH. This is due to the fact that the main part of the required heating load is provided by solar energy, and the energy consumption in the HAH remains roughly constant at a value much lower than the solar energy provided.

The effect of the SPT of the cooling auxiliary heater (CAH) on the thermo-economic performance of the SHAC system is illustrated in Figure 3. An increase in set-point temperatures of the heating and cooling auxiliary heaters exhibited similar impacts on the SF and the energy and collector efficiencies. As can be seen from Figure 3, with an increase in SPT of the CAH from 80°C to 110°C, the SF and the energy and collector efficiencies decrease by 21.2%, 4.6%, and 9.7%, respectively. Note that the SPT of the CAH had a much greater effect on the SF and the energy and collector efficiencies because of the higher energy consumption in the CAH compared to the HAH. With increasing the SPT of the CAH from 80 to 110°C, the exergy

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### Table 3: Base case conditions

| SHAC and SHEC | HSVC |
|---------------|------|
| Parameter | Value | Parameter | Value |
| Auxiliary heater set point for heating system (°C) | 60 | Auxiliary heater set point for heating system (°C) | 60 |
| Auxiliary heater set point for cooling system (°C) | 80 | Chilled water set-point temperature (°C) | 9 |
| Chilled water set-point temperature (°C) | 9 | PV panel area (m²) | 20 |
| Hot storage tank volume (m³) | 2 | Collector area (m²) | 20 |

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**FIGURE 2** Effect of set-point temperature of heating auxiliary heater on thermo-economic performance of the SHAC system

**FIGURE 3** Effect of set-point temperatures of auxiliary heaters on the thermoeconomic performance of the SHAC system.
destruction declined by 5.3%. This can be attributed to improved performance of the absorption chiller as a direct result of an increase in the generator temperature. Note that the figures for LCC remained unchanged with increasing SPT of the CAH.

The variation of thermo-economic indicators with the SPT of the absorption chiller is illustrated in Figure 4. As the SPT of the absorption chiller increases, the performance of the absorption chiller is enhanced, causing the SF and the energy efficiency of the SHAC system to increase. For an increase in SPT of the absorption chiller from 5 to 11°C, the SF and energy efficiency increase by 2.5% and 4.5%, respectively. Also, the collector efficiency decreases marginally with an increase in the SPT of the chiller. This is due to an insignificant augmentation in the storage tank (ST) temperature. Another important point in Figure 4 is that life cost cycle and exergy destruction remained unchanged with increasing chiller set-point temperature. The main reason for this observation is that the energy provided by the solar collector is much more than the increase in energy consumption of the CAH, which is as a direct result of increasing the SPT of the absorption chiller.

Figure 5 shows the variation of thermo-economic performance criteria with the ST volume. As the ST volume increases from 1 to 4 m³, the collector efficiency and SF increase by 20.8% and 13.1%, respectively. This is due to the fact that enhancement of the ST volume enhances the stratification of the ST and therefore raises the SF and collector efficiency. In other words, increasing the ST volume leads to an increase in the tank height, which enhances the stratification of the ST. This results in lowering the temperature of the inlet flow to the solar collectors and therefore an increase in collector efficiency (ie, the energy loss of the collector decreases). Enhancing the collector efficiency results in an increase in the input energy provided by the solar collectors, so the system SF is raised. Note that there exists an optimum value for energy efficiency at a ST volume of 2 m³. This is due to the fact that, by enhancing the ST volume, the performance of the solar collector improves, while the energy loss from the ST increases. By increasing the ST size, its
energy loss is increased and therefore the exergy destruction is greater in bigger tanks. Regarding the LCC, an increase in the ST volume raises the initial cost. Consequently, the LCC experiences a slight increase.

The changes of the thermo-economic performance criteria of the SHAC system with variations in collector area are shown in Figure 6. It is evident that, with increasing the collector area, the SF increases significantly, and the energy efficiency rises due to a decline in the energy supplied by the auxiliary heaters. For an increase in collector area from 20 to 50 m², the collector efficiency decreases by 22.9% due to a rise in the solar collector inlet temperature. It can be seen that larger collector areas exhibit higher exergy destructions, mainly because of the large contribution of the solar collector to the exergy destruction. The LCC increases steadily with increasing the collector area which is due to the high capital cost of the solar collectors, and the capital cost rise is larger than the decline in the cost associated with energy consumption.

### 4.1.2 Solar heating and ejector cooling

The effect of the SPT of the HAH on the thermo-economic performance of the SHEC system is illustrated in Figure 7. As for the SHAC system, a rise in SPT of the HAH leads to a decrease in SF, energy efficiency, and collector efficiency. But the LCC and exergy destruction remain relatively constant as the temperature of the HAH varies.

Figure 8 depicts the variations in thermo-economic criteria with changes in the SPT of the CAH. The SF, energy efficiency, and collector efficiency all decrease as the SPT of CAH rises. This is due to an increase in the energy required by the CAH as well as higher temperature values of the fluid entering the solar collector. With increasing the SPT of the CAH, the exergy destruction declines steadily, due to the enhanced performance of the ejector chiller. Moreover, the system LCC rises with the SPT of the CAH.

Figure 9 shows the effect of the SPT of the ejector chiller on the thermo-economic performance criteria. By

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**FIGURE 5** Effect of storage tank volume on thermo-economic performance of the SHAC system

**FIGURE 6** Effect of collector area on thermo-economic performance of the SHAC system
increasing the SPT of the ejector chiller, SF and energy efficiency rise due to the improved performance of the ejector chiller, while the collector efficiency reaches a peak and then experience a downward trend. Another important feature of is that the values for LCC and exergy destruction remain almost unchanged with increasing the SPT of the
ejector chiller. As for the absorption chiller, increasing the SPT of the ejector chiller exhibits no effect on the LCC and exergy destruction.

Figure 10 illustrates the variation in thermo-economic criteria of the SHEC system with the ST volume. By increasing the ST volume from 1 to 4 m³, the SF and collector efficiency rise by 27% and 4.5%, respectively. As for the SHAC system, the stratification of the ST is enhanced as a direct result of increasing the ST volume, resulting in higher values of SF and collector efficiency. It is evident that there is a slight increase in energy efficiency with increasing volume of the ST. Furthermore, the exergy destruction rises by almost 6.4% because of increasing energy loss from the bigger tanks. Regarding the LCC criterion, larger tanks have higher initial costs and therefore the LCC rises as the ST volume increases.

The effect of the collector area on the thermo-economic performance of the SHEC system is shown in Figure 11. With incrementing the collector surface area from 20 to 40 m², the SF rises considerably, from 20% to 36.5%, while the energy efficiency increases slightly and the collector efficiency decreases, mainly due to a corresponding rise in the temperature of the fluid entering the solar collector. Larger collector areas lead to higher exergy destructions, mainly because a significant share of the total exergy destruction of the system is due to the solar collectors. Also, increasing the collector area results in higher capital costs, thus increasing the LCC.

4.1.3 Heating and solar vapor compression cooling

In Figure 12, the variations in thermo-economic performance criteria with respect to the SPT of the heating heater are illustrated. It can be seen that almost all the criteria remain constant as SPT varies in the HSVC system. The main reason variation why varying the SPT of the heating heater has no effect on the thermo-economic performance of the system is
that the heating cycle in the HSVC system is separate from the rest of the system. Therefore, increasing the temperature of heater only raises the energy and exergy losses from the heater and has no effect on the performance criteria.

The effect of the SPT of the vapor compression chiller on the thermo-economic performance of the HSVC system is depicted in Figure 13. As the SPT of the vapor compression chiller is increased, the SF rises while the electrical efficiency remains almost unchanged. With increasing the SPT from 5 to 11°C, the SF is increased by 5%, and electrical efficiency is decreased by 1%. With an increase in the SPT of the chiller, electricity consumption by the chiller rises, and a higher amount of electrical power is sold to the grid. Therefore, the electrical efficiency remains relatively unchanged as the SPT of the chiller increases. It can be seen for all the set-point temperatures that the collector efficiencies remain constant at 12%. As for the absorption chiller, the variation in the SPT of the vapor compression chiller has no effect on the exergetic performance and LCC of the HSVC system.

Figure 14 demonstrates the impact of the solar collector area on the thermo-economic performance of the HSVC system. With increasing the collector surface area from 12 to 31 m², the SF increases considerably, by 47.8%, while the electrical efficiency decreases from 84.8% to 79.4%. With increasing the collector area, the electrical power generated by the PV panels and the amount of power sold to the grid both rise, while a lower amount of power is purchased from the grid. Consequently, the electrical efficiency decreases as the collector area is increased. Another important indication of Figure 14 is that collector efficiency is unaffected by the collector area. It can be seen that, due to the large contribution of the solar collectors to the overall exergy destruction, the exergy destruction increases steadily as the collector area rises. Moreover, it is clear that the LCC increases with solar collector area, mainly due to a larger capital cost increase in the solar collectors compared to a decrease in energy consumption.

4.2 | Comparison results

The ranges in variation of LCC, I, and SF for the three studied SHC systems under climatic conditions of Tehran are compared in Figure 15. As can be seen, the LCC for the
SHAC and HSVC systems are much lower than that for the SHEC system, and SHAC technology has the lowest LCC among the three technologies. Although the initial cost is lower for the SHEC than other technologies, the lower thermal performance of the ejector cooling system in comparison with the absorption system results in a higher value of LCC for the SHEC system. Regarding irreversibility, the HSVC system exhibits the lowest irreversibility, highlighting the advantageous performance of this system from the exergetic viewpoint. Moreover, it is evident that the SHEC system has the highest exergy destruction among the three technologies. The main reason for this is that, for supplying cooling load with the SHEC system, a higher amount of energy is required due to its lower thermal performance compared to the SHAC system. Therefore, a significant amount of this energy is dissipated in the condenser of the cooling cycle, thereby increasing the irreversibility in the SHEC system. Furthermore, the HSVC system has the maximum mean value of SF; however, the highest SF is achieved by the SHAC technology.

Note that SFs are heavily dependent on the solar collector areas. Nonetheless, the SHEC system has the lowest SF compared to other SHC technologies. The main reason for this result is that the auxiliary heaters of the SHEC system utilize more energy than those of the absorption system due to the greater amount of energy required in the generator of the ejector cooling system compared to the absorption technology. As a result, the SHEC system exhibits a lower SF than the other technologies under the condition of identical solar collector area.

Table 4 illustrates the thermo-economic performance of the SHC systems under various climatic conditions for six cities located in Iran. To draw a suitable comparison, the thermo-economic performance of the SHC systems is compared under various climatic conditions. The following cities are considered: Bushehr as a hot and humid city, Kerman and Isfahan as hot and dry cities, Tehran as a moderate city, Mazandaran as a humid and moderate city, and Hamedan, Tabriz, and Mashhad as cold cities. Furthermore, the working conditions of the SHC systems are identical for the comparison, as follows: SPT of the HAH = 60°C, SPT of the CAH = 80°C, volume of ST = 2 m³, SPT of chiller = 11°C, and collector area = 40 m², PV panel area = 12 m². Note that electrical efficiencies were reported in Table 4 as energetic efficiencies for the HSVC technology, since they are the same. It is evident that the energy efficiency is much higher for the SHAC system than the SHEC system for all cities, regardless of climatic conditions. Moreover, the electrical efficiency of
**TABLE 4** Thermo-economic performance of the SHC systems for selected cities in Iran, representative of various climatic conditions

| City        | Max. and min. ambient temperatures (°C) | Heating load (kWh) | Cooling load (kWh) | Type of cooling system | η<sub>energy</sub>(%) | SF (%) | η<sub>collector</sub>(%) | LCC ($/kWh) | I (GWh) |
|-------------|----------------------------------------|--------------------|--------------------|------------------------|-----------------------|--------|----------------------|-------------|---------|
| Bushehr     | 45 and 3                               | 3391               | 22 052             | SHAC                   | 53.2                  | 62.0   | 51.0                 | 0.1720      | 0.0550  |
|             |                                        |                    |                    | SHEC                   | 12.7                  | 22.2   | 85.8                 | 0.2053      | 0.1890  |
|             |                                        |                    |                    | HSVC                   | 88.4                  | 40.5   | 12                   | 0.149       | 0.0263  |
| Kerman      | 42 and -11                             | 9351               | 11 403             | SHAC                   | 65.5                  | 84.1   | 42.3                 | 0.1929      | 0.0577  |
|             |                                        |                    |                    | SHEC                   | 19.3                  | 36.2   | 72.6                 | 0.2356      | 0.1198  |
|             |                                        |                    |                    | HSVC                   | 83.3                  | 64.8   | 11.8                 | 0.1685      | 0.0262  |
| Isfahan     | 41 and -12                             | 11 267             | 11 620             | SHAC                   | 66.9                  | 81.9   | 44.9                 | 0.1771      | 0.0569  |
|             |                                        |                    |                    | SHEC                   | 20.6                  | 37.2   | 77.8                 | 0.2141      | 0.1203  |
|             |                                        |                    |                    | HSVC                   | 83.1                  | 65.1   | 11.6                 | 0.1539      | 0.0267  |
| Tehran      | 41 and -8                              | 11 959             | 13 428             | SHAC                   | 63.4                  | 77.5   | 49.2                 | 0.1667      | 0.0572  |
|             |                                        |                    |                    | SHEC                   | 19.7                  | 32.2   | 83.4                 | 0.1920      | 0.1288  |
|             |                                        |                    |                    | HSVC                   | 84.8                  | 56.7   | 11.8                 | 0.146       | 0.0270  |
| Hamedan     | 39 and -20                             | 21 247             | 6334               | SHAC                   | 75.9                  | 61.8   | 41.0                 | 0.1401      | 0.0493  |
|             |                                        |                    |                    | SHEC                   | 36.3                  | 43.4   | 71.5                 | 0.1725      | 0.08045 |
|             |                                        |                    |                    | HSVC                   | 80.7                  | 77.9   | 11.5                 | 0.1226      | 0.0233  |
| Tabriz      | 37 and -14                             | 19 524             | 7640               | SHAC                   | 74.3                  | 65.3   | 41.5                 | 0.1440      | 0.0519  |
|             |                                        |                    |                    | SHEC                   | 31.8                  | 40.6   | 72.3                 | 0.1764      | 0.0902  |
|             |                                        |                    |                    | HSVC                   | 81.2                  | 74.8   | 11.5                 | 0.1265      | 0.0240  |
| Mazandaran  | 35 and 0                               | 5611               | 9583               | SHAC                   | 61.2                  | 78.1   | 48.3                 | 0.1761      | 0.0574  |
|             |                                        |                    |                    | SHEC                   | 19.6                  | 34.2   | 78.5                 | 0.2123      | 0.1232  |
|             |                                        |                    |                    | HSVC                   | 82.1                  | 60.6   | 11.6                 | 0.156       | 0.0268  |
| Mashhad     | 37 and -12.5                           | 13 611             | 8416               | SHAC                   | 74.9                  | 63.1   | 43.2                 | 0.1420      | 0.0508  |
|             |                                        |                    |                    | SHEC                   | 33.7                  | 41.3   | 73.4                 | 0.1735      | 0.0850  |
|             |                                        |                    |                    | HSVC                   | 81.0                  | 75.1   | 11.5                 | 0.1242      | 0.0237  |
the HSVC system is higher than 80% under different climatic conditions. Another important point indicated in Table 4 is that the energy efficiencies of thermally driven SHC systems (SHAC and SHEC) tend to be higher under colder conditions, mainly because the efficiency of the heating system is more than the efficiency of cooling systems (absorption and ejector cooling cycles). Therefore, both thermally driven systems exhibit higher values of energy efficiencies under cold conditions, when the demand is higher for heating than cooling. The electrical efficiency of the HSVC system tends to be higher for hotter climates due to the higher solar intensity in these climates. The SF is much higher for the SHAC system than the SHEC system. It can be inferred that the SF of the SHEC system increases under colder conditions. The main reason for this observation is that the solar intensity is lower in those cities than hotter ones, whereas a lower cooling load exists, and a higher amount of cooling load is supplied by solar energy. Likewise, higher values of solar fractions for the HSVC system are achieved for colder cities.

The solar collector efficiency of the SHEC system is more than the SHAC system under all weather conditions. The main reason for this is that the ejector cycle generator in the SHEC system uses more energy than the absorption system, so the temperature of the storage tank of the SHEC system decreases. Therefore, the inlet temperature to the solar collector drops, increasing the solar collector efficiency. Moreover, the collector efficiency of the HSVC system remains relatively constant under different climatic conditions, and only slight variations are observed due to the variations in solar intensity in different cities.

With respect to economics, the SHAC and HSVC systems have lower LCCs than the SHEC system. It can be seen that LCC is lower for systems functioning under cold rather than hot cities. This can be attributed to the lower cost of supplying the heating load compared to the cooling load. Furthermore, SHC systems working under cold conditions exhibit enhanced performance compared to those located in hot cities from an exergetic viewpoint.

The thermo-economic performances of the three solar heating and cooling systems for supplying both heating and cooling loads of a residential building are comprehensively investigated. Parametric studies are conducted to evaluate the effects of key design parameters on the thermo-economic performances of the systems. Moreover, the thermo-economic performance of each system is investigated under various climatic conditions. The following can be concluded:

- The variations in design parameters show conflicting effects on thermo-economic criteria, which highlight the importance of multi-objective optimization for achieving the best design points for further study.
- The HSVC system exhibits the best performance from an exergetic viewpoint as well as the SF criterion. Moreover, the parametric analysis shows that the life cycle costs of the SHAC system and the HSVC system are similar. However, the LCC is much higher for the SHEC system than the other technologies.
- The energy efficiencies of the SHAC system and the SHEC system are higher under colder conditions. However, the collector efficiency of the HSVC system decreases in colder climates.
- The SHAC system exhibits higher values of solar fractions compared to the SHEC system under all climatic conditions. Moreover, the SHEC system and the HSVC system attain higher solar fractions in colder cities.
- The best economic performance is observed for the SHAC system and the HSVC system. The LCC of the SHEC system is higher than for other technologies. Furthermore, the performances of the systems are enhanced under colder conditions from the economic and exergetic viewpoints.

Nomenclature

- \( C \) Initial cost ($)
- \( CAH \) cooling auxiliary heater
- \( e \) market discount rate (%)
- \( ETC \) evacuated tube collector
- \( ex \) specific exergy (kJ/kg)
- \( S \) exergy flow rate (kW)
- \( f \) fraction
- \( HAH \) heating auxiliary heater
- \( HSVC \) heating and solar vapor compression cooling
- \( i \) inflation rate (%)
- \( İ \) exergy destruction rate (GWh)
- \( LCC \) life cycle cost ($/kWh)
- \( m \) mass flow rate (kg/s)
- \( N \) system lifetime (year)
- \( ORC \) organic Rankine cycle
- \( P \) Power (kW)
- \( PTC \) parabolic trough collector
- \( PWF \) present worth factor
- \( Q \) heat transfer rate (kW)
- \( SF \) solar fraction (%)
- \( SHAC \) solar heating and absorption cooling
- \( SHC \) solar heating and cooling
- \( SHEC \) solar heating and ejector cooling
- \( SPT \) set-point temperature
- \( ST \) storage tank
- \( V \) volume
- \( VCC \) vapor compression cycle
- \( W \) work rate (kW)
Z  capital investment cost ($)\text{Greek letters}
\eta  \text{efficiency}\text{Subscripts}
in  \text{inlet}
om  \text{operating and maintenance}
out  \text{outlet}
salv  \text{salvage}

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APPENDIX A

System components and assumptions for simulation of the three SHC technologies

| Component         | TRNSYS type | Assumptions                                                                 |
|-------------------|-------------|-----------------------------------------------------------------------------|
| Solar collector   | Type 71     | ● Collector area: 40 m²  
● Intercept efficiency: 0.7  
● Flowrate: 10 kg/h.m²  
● Collector working fluid: water  
● Azimuth surface: 0°  
● Slope of surface: 35° |
|                   | (ETC)       |                                                                            |
| Storage tank      | Type 4      | ● Volume: 3 m³  
● Coefficient of heat loss: 0.277 W/m²K |
| Pump              | Type3b      | ● Cp: 4.19 kJ/kg.K  
● Nominal power: 1 kW  
● Pump power to fluid thermal energy fraction: 0.05 |
| Weather data      | Type 109    | ● Type 109 reads weather information from the weather data file and calculates the solar energy in different directions |
|                   | Type 69b    | ● Weather information is produced by METEONORM software  
● Type 69b calculates the effective sky temperature |
|                   | Type 33e    | ● Type 33e calculates psychometric properties  
● Geographic location (°): Tehran: Longitude: 54.3, Latitude: 35.68  
● Tabriz: Longitude:46.2, Latitude:38.1  
● Kerman: Longitude:57.1, Latitude:30.2  
● Isfahan: Longitude: 51.6, Latitude: 32.6  
● Bushehr: Longitude: 50.8, Latitude: 28.9  
● Hamedan: Longitude: 48.5, Latitude: 34.8  
● Mazandaran: Longitude:53, Latitude: 36.3  
● Mashhad: Longitude:59.5, Latitude: 36.3  
● Sky model for diffuse radiation: Perez model |

(Continues)
| Component                          | TRNSYS type | Assumptions                                                                 |
|-----------------------------------|-------------|-----------------------------------------------------------------------------|
| Absorption chiller                | Type 680    | • Type: hot water-fired single-effect absorption chiller                     |
|                                   |             | • Nominal capacity: 15 kW                                                   |
|                                   |             | • Specific heat of hot water: 4.19 kJ/kg.K                                  |
|                                   |             | • Chilled water (CHW) specific heat: 4.19 kJ/kg.K                           |
|                                   |             | • Cooling water (CW) specific heat: 4.19 kJ/kg.K                            |
|                                   |             | • CHW set point: 11                                                         |
|                                   |             | • Flow rate of CHW: 2320 kg/h                                               |
|                                   |             | • Hot water flow rate: 1000 kg/hr                                            |
|                                   |             | • CW flow rate: 2790 kg/hr                                                  |
|                                   |             | • Operating time: regarding the cooling load of the building                |
| Ejector chiller                   | Type 66     | • Refrigerant: R134a                                                        |
|                                   | (EES model) | • CHW set point: 11                                                         |
|                                   |             | • Specific heat of hot water: 4.19 kJ/kg.K                                  |
|                                   |             | • CHW specific heat: 4.19 kJ/kg.K                                           |
|                                   |             | • CW specific heat: 4.19 kJ/kg.K                                            |
|                                   |             | • CHW flow rate: 3230 kg/h                                                  |
|                                   |             | • Hot water flow rate: 3000 kg/h                                            |
|                                   |             | • CW flow rate: 6000 kg/h                                                   |
|                                   |             | • Operating time: regarding the cooling load of the building                |
| Building                          | Type 56     | • Refer to Table 1.                                                         |
| Cooling tower                     | Type 51a    | • Type: wet, counter flow                                                   |
|                                   |             | • Number of tower cells: 4                                                  |
|                                   |             | • Maximum cell flow rate: 10 000 m³/h                                       |
|                                   |             | • Fan power at max flowrate: 1 kW                                           |
| Heating and Cooling Loads (Radiators) | Type 682   | • Working fluid specific heat: 4.19 kJ/kg.K                                 |
| Controller                        | Type 2d     | • Controller type: ON/OFF Differential Controller                           |
|                                   |             | • Controller 1 is used to control the collector flow rate. If the temperature difference between collector inlet and outlet is less than 1°C, the pump of collector cycle is turned off. If the pump is off, the pump turns on if the temperature difference exceeds 2°C. |
|                                   |             | • Controller 2 is used to turn the heating system on and off if required (according to heating load) |
|                                   |             | • Controller 3 is used to turn the cooling system on and off if required (according to cooling load) |
| Auxiliary heater                  | Type 6      | • Heater 1 is used for heating system                                       |
|                                   |             | • Set-point temperature of the heater 1 is considered to be 60°C            |
|                                   |             | • Heater 2 is used for cooling system                                       |
|                                   |             | • Set-point temperature of the heater 1 is considered to be 80°C            |
|                                   |             | • Heater efficiency: 79%                                                   |
| Flow collector                    | Type 11h    | –                                                                           |
| Flow separator                    | Type 11f    | –                                                                           |
| Component                        | TRNSYS Type | Assumptions                                                                                                                                 |
|---------------------------------|-------------|--------------------------------------------------------------------------------------------------------------------------------------------|
| Solar collector                 | Type 194    | • Module short-circuit current at reference conditions: 6.5 A  
• Module open-circuit voltage at reference conditions: 21.6 V  
• Module voltage at max power point and reference conditions: 17 V  
• Module current at max power point and reference conditions: 5.9 A  
• Number of cells wired in series: 36  
• Number of modules in series: 7  
• Number of modules in parallel: 2-3-4  
• Module area: 0.89 m²  
• Load voltage: 220 V  
• Array slope: 35°  
• Array azimuth: 0°                                                                                                                                 |
| Inverter                        | Type 48b    | • Regulator efficiency: 0.78  
• Inverter efficiency: 0.96                                                                                                                                 |
| Battery                         | Type 47a    | • Cell energy capacity: 200 Wh  
• Charging efficiency: 90%  
• Cells in series: 6  
• Cells in parallel: 1                                                                                                                                 |
| Vapor compression chiller       | Type 666    | • Type: water cooled vapor compression chiller  
• Rated capacity: 15 kW  
• CHW specific heat: 4.19 kJ/kg.K  
• CW specific heat: 4.19 kJ/kg.K  
• CHW set point: 11  
• CHW flow rate: 2320 kg/h  
• CW flow rate: 2790 kg/h  
• Operating time: regarding the cooling load of the building                                                                                                                                 |
| Cooling tower                   | Type 51a    | • Type: wet, counter flow  
• Number of tower cells: 4  
• Maximum cell flow rate: 10 000 m³/h  
• Fan power at max flow: 1 kW                                                                                                                                 |
| Pump                            | Type 3b     | • Fluid specific heat: 4.19 kJ/kg.K  
• Maximum power: 1 kW  
• Fraction of pump power converted to fluid thermal energy: 0.05                                                                                                                                 |
| Weather data                    | Type 109    | • Type 109 reads weather information from the weather data file and calculates the solar energy in different directions  
• Weather information is produced by METEONORM software  
• Type 69b calculates the effective sky temperature  
• Type 33e calculates psychometric properties  
• Geographic locations (°): Tehran: Longitude: 54.3, Latitude: 35.68  
• Tabriz: Longitude: 46.2, Latitude: 38.1  
• Kerman: Longitude: 57.1, Latitude: 30.2  
• Isfahan: Longitude: 51.6, Latitude: 32.6  
• Bushehr: Longitude: 50.8, Latitude: 28.9  
• Hamedan: Longitude: 48.5, Latitude: 34.8  
• Mazandaran: Longitude: 53, Latitude: 36.3  
• Mashhad: Longitude: 59.5, Latitude: 36.3  
• Sky model for diffuse radiation: Perez model                                                                                                           |
| Heating Load (Radiator)         | Type 682    | • Working fluid specific heat: 4.19 kJ/kg.K                                                                                                                                 |
| Auxiliary heater                | Type 6      | • Heater 1 is used for heating system  
• Set-point temperature of the heater 1 is considered 60°C  
• Heater efficiency: 79%                                                                                                                                 |
| Building                        | Type 56     | • Refer to Table 1                                                                                                                                 |