1 INTRODUCTION

Using energy-efficient and low-carbon technologies for buildings has become a global task because nearly 40% of global energy consumption is allocated to the building sector.\(^1\),\(^2\) Since 2010, \(\text{CO}_2\) emissions, electricity, and natural gas use have been developed by 1%, 1%, and 2.5% per year, respectively, despite global efforts to improve the building energy performance.\(^3\) Therefore, further research is still needed to enhance the performance of building energy systems. One of the most effective ways to reduce energy consumption in buildings and, thus, to prevent the growth
of CO₂ emission intensity, is combined heating, cooling, and power generation systems (CCHP), which recover the wasted energy. Conventional power plants convert only about 30 percent of the energy in the fuel into electricity, and much of that energy is lost as heat. In addition, much energy is wasted due to the transfer of electrical power to the desired location. Therefore, to reduce these losses, CCHP systems have been introduced, which have the potential to boost system efficiency and thus diminish the air pollution caused by the burning of fossil fuels.

Mago et al. compared primary energy consumption (PEC), current costs, and carbon dioxide emissions of a CCHP system for a large office building in Chicago with a conventional system for the same building. They reported that using CCHP reduced current costs, PEC, and carbon dioxide emissions by an average of about 2.6%, 12.1%, and 40.6% in comparison with the conventional system. Maravera et al. stated that the biomass-CCHP system may not always be better than conventional methods from the viewpoint of energy efficiency and environmentally friendly aspects. They first developed a thermodynamic model to evaluate different system sizes and primary energy saving ratio (PESR) analysis and then used a life cycle cost analysis method to examine the system from an environmental perspective. Their results showed that the heating-to-cooling ratios of small CCHP plants are environmentally feasible, while that ratio was environmentally unfeasible for large plants. Li et al. investigated the impact of using the CCHP system on two types of residential and office buildings. An air conditioning system and a heat storage tank are added to the layout of the CCHP system. Optimization based on energy savings, overall cost savings, and best environmental performance using the Genetic Algorithm showed that the combined energy storage ratios of the hybrid system are 12.9% in residential buildings and 23.68% in office buildings. The heat storage tank increased the integrated storage ratio from 22.54% to 23.66% in office buildings, while the air conditioning system increased the ratio to only 22.7%.

Using renewable energies and combining them with other units enhances system efficiency and diminishes CO₂ emissions. Solar energy as a low-temperature source for CCHP systems has been used in recent years to reduce energy consumption in buildings, and much research has been done on solar CCHP systems. Among various solar thermal technologies, parabolic trough collectors are known as sustainable and reliable units, which are widely used on large and small scales. Zhai et al. performed an energy and exergy analysis on a solar CCHP system. According to the results for off-grid remote areas of northwest China, the system’s efficiency using these parabolic trough collectors increased from 10.2% to 58% and exergy efficiency from 12.5% to 15.2%. Economic analysis of the system showed that the desired system’s payback period in the current energy cost situation is about 18 years. From the sensitivity analysis results, it was also found that increasing the rate of return to 3% or increasing energy prices by 50% would reduce the payback period to ten years. Javan et al. investigated a CCHP system that used the Rankin cycle and cooling through an ejector cycle. The main energy of this cycle was supplied by the waste heat of an internal combustion engine. Among the different operating fluids used in this system, R-11 was found to be both energy-efficient and cost-effective. Habibi et al. investigated the performance of a solar cascade Rankine cycle combined with the parabolic trough collector. It was stated that the net power output and exergy efficiency could reach 782 kW and 18.61% after optimization. Wang et al. examined the performance of a CCHP unit operating on solar energy and natural gas. The cycle energy and exergy efficiencies were obtained 62.23% and 17.56% using the absorption cooling and photovoltaic systems, respectively. Using Tibetan climate data, Su et al. simulated a CCHP system using biogas and solar energy, which was first converted to chemical energy and then used. Compared to the traditional biogas direct burning method, annual electricity generation and cooling capacity improved by 8.7% and 2.57%, respectively. Also, natural gas consumption decreased by 8.66%.

Aghaziarati and Aghdam compared three types of solar collectors, which are parabolic trough collector (PTC), linear Fresnel reflector (LFR), and parabolic dish collector for a combined cooling, heating, and power (CCHP) system. The results indicated that PTC was the best option from the energy, exergy, and exergo-economic points of view due to its higher optical efficiency and lower solar field area needs. Wu et al. studied three CCHP systems with the Rankine cycle and the solar and biomass sources to provide the energy demand for an office building and a hotel in Shanghai, China. The solar system had the potential to reduce annual costs, while the biomass system had the best environmental performance. Moaleman et al. developed a TRNSYS model to investigate the performance of a CCHP system with a concentrating photovoltaic-thermal unit energy source that was connected to a water-ammonia absorption chiller. They found that a conventional type collector could not provide the thermal energy required for the refrigeration cycle. The average annual efficiency of the collector and trigeneration unit was 12.8% and 58.01%, respectively. The proposed method cannot adequately meet the electricity needs of the building, and 6030 kWh of electricity needs to be provided by the grid. Yang and Zhai developed a mathematical model to investigate the performance of a CCHP system equipped with photovoltaic cells and thermal collectors for a hotel building in Atlanta. Using the...
particle swarm optimization method, they observed that the solar CCHP system has 8.76% energy savings and 8.94% CO₂ reduction compared to the conventional CCHP system. Due to the higher initial cost, the total annual cost of the solar system was 8.09% higher than the conventional system.

Another way to improve energy consumption in buildings is to use the ejector cooling cycle in solar CCHP systems. The ejector has a simple structure that consumes no energy due to the lack of moving components and consequently reduces CO₂ pollution. Godfroy et al. investigated the design of a trigeneration system based on the gas engine mini-CCHP unit (5.5 kWₑ) and the ejector cooling cycle. The cycle analysis showed that the overall efficiency was 50% for co-generating heating and cooling. Using the electrical power generated by the system to produce cooling increased the cooling capacity but reduced overall system efficiency and increased CO₂ production. Wang et al. investigated the effect of hour angle and collector tilt angle on the performance of the solar CCHP system with the Rankine cycle and the ERC. Using the genetic algorithm optimization method, they found that the optimal collector tilt angle is 60°, which occurred at 10 A.M. on 12 June. At the optimum hour and tilt angle, the maximum exergy efficiency was 60.33%. Al-Sulaiman et al. designed a CCHP system to generate 500 kW of electricity using a parabolic collector and a Rankin cycle. In this study, three different modes, including solar mode, solar and storage mode, and storage mode, were considered. The highest CCHP efficiencies were 94% for solar mode, 47% for solar and storage modes, and 42% for storage mode. Ebrahimi et al. designed a CCHP system with an ejector cooling cycle to meet the energy needs of a residential building. In this paper, the effects of steam turbine inlet temperature and pressure on the cycle were investigated. It was found that the fuel-saving ratio was higher than 69% in summer and more than 25% in winter. Daily load analysis had also indicated that the CCHP could save energy compared to individual power generation, heating, and cooling systems. Wang et al. optimized the performance of a solar CCHP system using a flat plate collector, Rankine cycle, and ejector cycle. Using the genetic algorithm, they found that the optimum useful output of the system in three modes, namely the power mode, CCHP mode, and CHP mode, are averagely 6.4, 5.84, and 8.89 kW, respectively. In another work, they proposed a solar CCHP system using the Brayton cycle and ejector-based transcritical CO₂ cooling cycle. The results showed that by increasing the turbine inlet pressure and the ejector inlet temperature, the system efficiency decreased while increasing the turbine backpressure and inlet temperature resulted in higher efficiency. Besides, the increasing of the ejector backpressure led to diminishing thermal efficiency and enhancing the exergy efficiency. On the analysis of a solar ejector-based CCHP using evacuated tube solar collectors, Boyaghchi et al. found that the thermal and exergy efficiencies were 23.66% and 9.51% in summer and 48.45% and 13.76% in winter, respectively. In the next work, they used combined solar-geothermal energy as the energy source for a CCHP. They reported that R143a is the best refrigerant, which maximizes the cycle efficiency to 39.94%. Using TRNSYS modeling, Cioccolanti et al. investigated a CCHP system with a 3.5 kW organic Rankin cycle and a 17 kW absorption chiller. They also built a small prototype of the system, which is supplied by the CCHP solar collector. Based on the results, the operating temperatures of the storage tanks significantly influenced the total performance of the system. With the proper selection of these ranges, primary energy production could increase by about 6.5% compared to the base configuration with no additional investment costs. Wu et al. compared three CCHP units types: CCHP connected to the solar thermal collector, CCHP integrated with solar thermal collector and ORC, and CCHP. It was shown that the second structure could produce more power compared to other configurations. Also, it led to a decrease of 12.4% in the consumption of natural gas. Jafari Mosleh et al. dynamically investigated a solar CCHP system equipped with an ejector and hot and cold storage tanks. It was revealed that increasing the volume of the hot storage tank did not enhance system performance but could shift the system's peak load. They also observed that increasing the turbine inlet pressure from 1500 to 2500 kPa led to the overall system efficiency from 14.2% to 23.5%. Rostamzadeh et al. designed two new micro-CCHPs based on the Rankine cycle and the Kalina cycle as a topping cycle. In addition, the systems used the ERC and the heat pump cycle as the bottoming cycle. Using genetic algorithm optimization, they found that the optimum thermal efficiency of micro-CCHP was 76.54% and 77.32%, respectively.

In this paper, a dynamic simulation of a solar ejector-based CCHP has been performed for a residential complex in Tehran using a TRNSYS-EES co-simulator. Due to the rising cost of energy carriers and their environmental impacts, the waste energy recovery has been used at various points of the thermal cycle, for heating and cooling the building. Most previous studies were performed for a specific design state under steady-state conditions. Also, in the transient analyzes, simplifications have been made in different parts, especially in heat exchangers of systems, turbines, and condensers. For example, the energy conservation equations for heat exchangers or condensers are based on the first law of thermodynamics, in which the characteristics of the heat exchanger and the effect of the variable conditions on its performance have not been considered.
Besides, the impact of changing climatic parameters and off-design conditions has not been taken into account. In this research, the authors have focused on the detailed analysis of various cycle components by studying the effects of environmental factors and off-design conditions. The intended cycle comprises a Rankin cycle using water as the working fluid and an ERC using R134a as the operating fluid. The analytical equations for the ejector cycle have been written according to the available references, and the effect of the weather changes in the condenser performance has been considered. It has been attempted to use the wasted heat in the cycle to increase the system efficiency. Finally, the influence of some effective parameters has been investigated.

2 | SYSTEM DESCRIPTION

The schematic of the proposed CCHP system is illustrated in Figure 1, which consists of a basic Rankine power cycle. In this system, a portion of the required thermal energy is provided by the solar system. It is also comprised of an ERC to generate cooling, which receives the required thermal energy as the driving force of the primary flow through a heat exchanger in the extraction path of the turbine. The operating fluid of the ERC is R134a.

The solar part includes parabolic solar collectors, a hot storage tank, a pump, and an auxiliary heater. When the solar collectors are unable to supply the cycle required energy, the auxiliary heater, located between the heat exchanger (a) and the storage tank, is turned on. The temperature of oil, which is DOWTHERM Q oil, increases, passing through the collector (point 22), and it enters the storage tank. Then, the oil goes to the auxiliary heater from the hot part of the tank, and its temperature reaches the required level (point 20). Finally, the oil transfers the heat to the working fluid of the CCHP cycle (water) (point 26) in the heat exchanger (a) and returns to the tank again (point 25).

The working fluid of the power cycle enters the turbine at point 1 after passing the heat exchanger (a) and reaching the desired temperature. The steam flow in the turbine is divided into two parts, in which one part is extracted (point 2). After passing through the heat exchanger (b), the extracted working fluid of the power cycle enters the deaerator (point 3) and combines with the inlet flow from the power cycle condenser (point 17). The second part of the steam in the turbine enters the power cycle condenser...
and is converted to saturated liquid after transferring heat to the environment. After the condenser, the working fluid is pumped to the deaerator. Before entering the power cycle condenser, the working fluid stream has a suitable temperature for supplying the building’s heating load (point 15); therefore, a heat exchanger (d) is installed before the condenser to generate the heat. The outlet condenser flow and the extracted flow are combined in the deaerator and then the mixed stream is pumped to the heat exchanger (a).

The ejector cycle is connected to the power cycle through the heat exchanger (b). The main components of this cycle include an ejector, a generator, a condenser, an evaporator and, an expansion valve. When the cooling load is required, the ejector cycle receives the required thermal energy for the primary flow through the heat exchanger (b), in which the refrigerant of the ejector cycle exchanges heat with the extracted steam and reaches the desired pressure and temperature (point 5). The heat exchanger (b) acts as the generator of the ejector cooling cycle. At point 5, the superheated refrigerant enters the Converging Diverging Nozzle of the ejector and creates a very low-pressure region at the nozzle outlet. The secondary flow (the refrigerant flow in the evaporator (point 9)) is also drawn into the ejector due to the pressure difference between the evaporator and the nozzle outlet. The mixing of two flows occurs after the choking of the secondary flow. Then a shock wave of approximately zero thickness occurs because of the presence of a high-pressure region downstream of the mixing section. Finally, the refrigerant flow exits the ejector at point 6, which can be used for producing hot water for the building; therefore, a heat exchanger (c) is used to supply part of the building’s hot water consumption (point 10). Then, the refrigerant enters into the cooling cycle condenser, which cools by the water flow of the cooling tower. The refrigerant flows out of the condenser under saturated liquid conditions (points 7 and 11) and is divided into two parts. A part of it enters the pump for increasing the pressure and then enters the heat exchanger (b) at point 4. The other part enters the evaporator (heat exchanger (e)) after passing through the expansion valve (point 8). The water from the cold storage tank (point 12) enters the evaporator and its temperature is reduced. The controller turns on the cooling system if the outlet temperature of the cold storage tank (point 12) reaches above 16°C and the cooling load is needed. On the other side, the outlet flow from the cold storage tank (point 31) reaches the consumer and returns back to the tank after its temperature rises (point 32).

In the heating section, which is supplied by the heat exchanger (c) and (d), if the produced heating is not sufficient to provide the required hot water in hot seasons and the required hot water and space heating in cold seasons, an auxiliary boiler (point 28) is used.

## 3 SYSTEM MODELING

### 3.1 Modeling tools

In this study, TRNSYS\(^{39}\) and EES\(^{40}\) software are used to simulate and analyze the proposed CCHP system. TRNSYS is a transient simulator software with a modular structure, which is used to simulate various systems such as air conditioning systems, solar systems, and other renewable energy systems. EES software has also been used to model some subsystems of the CCHP system, such as the ejector and the heat exchangers, which are not available in TRNSYS. Moreover, METEONORM software information is used for the weather data.

### 3.2 Methodology

Various types of TRNSYS components have been used to model the cycle, which are listed in Table 1. The parameters and assumptions considered for each component are also given in Table 1. Figure 2 indicates the TRNSYS model of the proposed CCHP system.

### 3.3 Ejector refrigeration cycle

Since the ejector is not defined in TRNSYS software, the mathematical modeling of this component is performed using EES software. The schematic of the modeled ejector is shown in Figure 3. In this paper, a one-dimensional constant pressure analysis of the ejector is performed based on the work done by Ref. 41, in which the modeling flowchart is depicted in Figure 4. Due to the constant ejector geometry and the given temperature and pressure \(P_e\) and \(T_e\) of the secondary flow, the pressure of the primary flow is assumed to calculate the outlet temperature and pressure of the ejector \((P_1\) and \(T_1\)). The pressure of the primary flow should be higher than the condenser pressure; therefore, the first guess for the pressure of the primary flow is considered as the condenser pressure, and then it is modified until the outlet pressure of the ejector is equal to the condenser pressure.

The assumptions for modeling the one-dimensional ejector are given as follows:

1. The operating fluid is an ideal gas and the specific heat capacity at constant pressure \(C_p\) and the ratio
| Model                     | Features                                                                                                                                                                                                 |
|---------------------------|--------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------|
| Weather information       | Type 109 reads climate data from the weather data file for computing the solar radiation in different directions  
Weather information is obtained from METEONORM software  
Type 69 is used for calculating sky temperature  
Psychrometric properties such as dew point temperature, relative humidity, etc. are calculated by Type 33  
Geographic location: Tehran, Longitude: 54.3, Latitude: 35.68  
Annual air temperature variation is shown in Figure 5 |
| Solar collectors          | Type 536 is used to model parabolic collectors  
Collector area: 2000–2500–3000 m²  
Collector flowrate: 100,000 kg/h  
Collector heat transfer fluid: DOWTHERM Q  
Sun tracking system: Single-axis tracking, North-south axis with east-west tracking  
Concentrating ratio: 14  
Intercept efficiency: 0.9 |
| Hot storage tank          | The hot storage tank is modeled by Type 4  
Storage tank volume: 100–150–200 m³  
Heat transfer coefficient of tank: 0.277 W/m²K  
Number of nodes: 6 |
| Circulating pump          | Pumps are modeled by Type 3b  
Power coefficient is 0.5  
Conversion coefficient: 0.05  
Maximum power: 70 kW |
| Auxiliary heater          | Type 6 is used to model the auxiliary heater  
The set-point temperature of the heater is considered 190–200–210°C  
The heater efficiency is 90% |
| Ejector refrigeration cycle (modeled in EES) | Type 60 is used to model the ejector, which uses EES software  
Refrigerant: R134a  
Operating time: regarding the cooling load of the buildings  
Evaporator pressure: 340 kPa  
Evaporator temperature: 4.5°C  
T-S diagram of an instantaneous performance is shown in Figure 3 |
| Heat exchangers (modeled in EES) | Type 60 is used to model the heat exchangers, which use EES  
The ε-NTU method is used for modeling the heat exchangers |
| Steam turbine             | The type 318 is used for modeling the turbine based on Stodola's law  
Internal efficiency: 80%  
Generator efficiency: 98%  
Turbine inlet design pressure: 5 bar  
Turbine outlet design pressure: 0.05 bar  
Turbine design flowrate: 5350 kg/h  
Turbine inlet design temperature: 190–200–210°C  
Turbine extraction design temperature: 134°C  
Turbine extraction design pressure: 300 kPa |
| Cooling tower of the steam cycle | Type: wet, counter flow  
Number of tower cells: 4  
Maximum cell flow rate: 15,000 m³/h  
Fan power at max flow: 6 kW |
| Cooling tower of ejector cycle | Type: wet, counter flow  
Number of tower cells: 4  
Maximum cell flow rate: 5000 m³/h  
Fan power at max flow: 5 kW |
TABLE 1  (Continued)

| Model               | Features                                                                 |
|---------------------|--------------------------------------------------------------------------|
| Deaerator           | Type 384 of TRNSYS is used for modelling                               |
|                     | The type of deaerator is mixed-flow                                      |
|                     | The quality of outlet flow is zero ($x = 0$)                             |
| Feedwater pump      | Type 300 is used for modeling the Feedwater pump                        |
|                     | Power coefficient: 0.5                                                  |
|                     | Maximum power: 140 kW                                                   |
| Steam cycle evaporator | Type 316                        |
|                     | Counter flow                                                            |
|                     | Overall heat transfer coefficient in design condition: 137.5 kW/K       |
|                     | Pressure drop: 0.1 bar                                                   |
| Steam cycle economizer | Type 315                        |
|                     | Counter flow                                                            |
|                     | Overall heat transfer coefficient in design condition: 108.3 kW/K        |
|                     | Pressure drop: 0.1 bar                                                   |
| Superheater         | Type 315                    |
|                     | Counter flow                                                            |
|                     | Overall heat transfer coefficient in design condition: 27.2 kW/K         |
|                     | Pressure drop: 0.1 bar                                                   |
| Steam cycle condenser | Type 383                      |
|                     | Temperature increase of cooling fluid: 10°C                             |
| Controller          | Type 2                     |
|                     | Controller type: Feedback controller                                    |
|                     | Controller 1, which controls the collector flow rate, will be turned off when the temperature difference between collector inlet and outlet is <1°C. It will turn on when the temperature difference exceeds 2°C |
|                     | Controller 2 is used to turn the cooling system on and off if required   |
| Flow separator      | Type 389                   |

![FIGURE 2](TRNSYS model of the proposed CCHP system)
of specific heat capacity \( (k) \) is variable across the ejector.
2. The flow inside the ejector is stable and one-dimensional.
3. The kinetic (velocity) term in the energy equation in the ejector input and output is ignored.
4. An isentropic condition is used, but the effects of friction are considered by several constant coefficients.
5. It is assumed that at the y intersection, mixing takes place and \( P_{my} = P_{ey} \).
6. The ejector wall is assumed adiabatic.
7. The ejector is constant pressure and its constant dimensions are as follows: \( D_d = 12 \) cm, \( D_t = 4 \) cm and \( D_n = 5 \) cm.

3.4 | Heat exchangers

All the heat exchangers (a)–(e) are simulated in EES software. Since the heat exchanger with the two-phase flow has not been defined in TRNSYS software, the mathematical modeling of the heat exchanger is performed in EES software. In this paper, the \( \varepsilon \)-NTU method is used for modeling the counter-flow heat exchangers. In this method, the efficiency of the heat exchanger is defined as\(^{42} \):

\[
\varepsilon = \frac{q}{q_{\text{max}}}
\]

(1)

where:

\[
q_{\text{max}} = C_{\text{min}} (T_{h,i} - T_{c,i})
\]

(2)

\[
C_{\text{min}} = \min (C_c, C_h)
\]

(3)

\[
C = mc
\]

(4)

In the case of the one-phase flow heat exchanger, the efficiency is given by:

\[
\varepsilon = \frac{1 - \exp (-NTU (1 - c_r))}{1 - c_r \exp (-NTU (1 - c_r))} c_r < 1
\]

(5)

\[
c_r = \frac{C_{\text{min}}}{C_{\text{max}}}
\]

(6)

In the case of the two-phase flow heat exchanger, the efficiency is given by:

\[
\varepsilon = 1 - \exp (-NTU)
\]

(8)

\[
c_r = 0
\]

(9)

The number of the transfer units (NTU) is obtained using the following relation:

\[
NTU = \frac{UA}{C_{\text{min}}}
\]

(10)

3.5 | Thermodynamic diagram of CCHP system

In Figure 5A, the T-S diagram of the ERC at a sample operating condition is shown. The dotted lines show the water temperature changes in the generator (heat exchanger (b)) and evaporator (heat exchanger (e)). The T-S diagram of the power and heating cycle at a sample operating state is illustrated in Figure 5B. It should be noted that the turbine outlet pressure varies by the condenser conditions. In other words, the turbine outlet pressure is dependent on the condenser pressure, which changes by the weather conditions.

4 | CASE STUDY BUILDING

The case study building of this study is a residential building with 14 apartments located in Tehran. The thermal loads of an apartment unit of this building are calculated by TRNSYS software (Type 56a), according to the thermal properties of the building (listed in Table 2). Based on the comfort conditions, the inside temperatures are assumed 26 and 20°C in summer and winter, respectively. For the thermal load calculation, information such as building geometry, type of windows, materials, and insulation used in building walls are required. Internal heating loads, including heat gain from people, which has been determined by the ISO7730 standard, and appliances such as TVs, refrigerators, etc., are also considered. The typical meteorological year of Tehran is used for weather information such as sunlight, wind speed, ambient temperature, humidity, etc. The window area on the south side is 20% of the south wall area and 10% on the west side. The air infiltration rate has been determined according to ASHRAE 90.1 and using the values in Table 2. It should be noted that the effect of apartments
on each other and the effect of their floor number have been neglected.

Indoor and outdoor air temperatures, as well as the heating and cooling loads of an apartment, are shown in Figure 6. The heating load represents the thermal energy required for hot water production and space heating of an apartment unit. The hot water consumed per person per day is considered 200 L. The peak heating and cooling loads per each apartment unit of the building are 8.47 and 9.83 kW, respectively. The yearly heating, cooling, and domestic hot water loads per each apartment unit are 8897, 13430, and 3017 kWh, respectively. The maximum energy consumption for heating and cooling also occurs in January (3200 kWh) and July (3600 kWh), respectively (Figure 7).

5 | PERFORMANCE INDICATORS

Different indicators have been used to evaluate system performance. The solar fraction (SF) is defined as the ratio of the amount of solar energy received to the total energy required by the system:

\[
SF = \frac{Q_{\text{Solar}}}{Q_{\text{Solar}} + Q_{\text{Aux}} + Q_{\text{Boiler}}} \tag{11}
\]

where \(Q_{\text{Solar}}\) is the solar useful energy, \(Q_{\text{Aux}}\) is the auxiliary energy in the solar cycle and \(Q_{\text{Boiler}}\) is the boiler energy required for supplying the extra domestic hot water and heating.
The overall system efficiency is defined as:

\[ \eta_{\text{total}} = \frac{W_T - W_P + Q_{\text{cooling}} + Q_{\text{heating}}}{Q_{\text{solar}} + Q_{\text{aux}} + Q_{\text{boiler}}} \]  

(12)

where \( W_T \) is the produced work by the turbine, \( W_P \) is the consumed work by pumps, \( Q_{\text{cooling}} \) is cooling load produced by the ejector cooling system and \( Q_{\text{heating}} \) is the produced heating.

The efficiency of the Rankine cycle is calculated by the following relation:

\[ \eta_{\text{Rankine power cycle}} = \frac{W_T - W_P + Q_{\text{cooling}} + Q_{\text{heating}}}{Q_{\text{hex, a}} + Q_{\text{boiler}}} \]  

(13)

Moreover, the efficiency of the ejector cooling system is computed using the following relation:

\[ \eta_{\text{ejector - cycle}} = \frac{Q_{\text{cooling}} + Q_{\text{Recover}}}{Q_{\text{Gen}}} \]  

(14)

\( Q_{\text{Recover}} \) is the amount of heat recovered in the heat exchanger (c) and \( Q_{\text{Gen}} \) is the transferred heat at the generator of the ejector cycle (heat exchanger (b)).

### 6 | Exergy Analysis

The irreversibility of the cycle components is obtained using the exergy analysis by the following relations:
| Parameter                                           | Value | Unit     |
|----------------------------------------------------|-------|----------|
| Overall heat transfer coefficient of external walls | 0.04  | W/(m²K)  |
| Overall heat transfer coefficient of roof          | 0.141 | W/(m²K)  |
| Overall heat transfer coefficient of windows       | 1.27  | W/(m²K)  |
| Inside set-point temperature in warm months        | 26    | °C       |
| Inside set-point temperature in cold months        | 20    | °C       |
| Area of each apartment unit                        | 200   | m²       |
| Number of apartment units                          | 14    |          |
| Occupants per each apartment unit                  | 3     | Person   |
| Maximum ambient air temperature                    | 42    | °C       |
| Minimum ambient air temperature                    | −8    | °C       |
| Occupant activity level                            | Seated, light work |          |
| Natural ventilation                                | 1     | AC/h     |
| Infiltration                                       | 0.6   | AC/h     |
| Artificial lighting                                | 5     | W/m²     |
| Yearly power consumption of each apartment         | 2.54  | MWh      |

**FIGURE 6** Hourly variations of indoor and outdoor air temperatures and cooling and heating loads of an apartment

**FIGURE 7** Monthly variations of heating and cooling loads of the building
TABLE 3 Exergy efficiency and destruction rates of the cycle components

| System components                  | Exergy destruction rate                                                                 | Exergy efficiency |
|------------------------------------|----------------------------------------------------------------------------------------|-------------------|
| Solar collector                     | \( E_{\text{X}_\text{Solar}, \text{Collector}} = E_{X_2} - E_{X_2} + E_{X_{\text{Solar}}} \) | \( - \frac{E_{X_2} + E_{X_{\text{Solar}}}}{E_{X_{\text{Solar}}}} \) |
| Heat exchanger (a)                 | \( E_{\text{X}_\text{Exchanger-a}} = E_{X_{10}} + E_{X_{19}} - E_{X_{20}} - E_{X_1} \)  | \( E_{X_1} - E_{X_{20}} \) |
| Heat exchanger (b)                 | \( E_{\text{X}_\text{Exchanger-b}} = E_{X_2} - E_{X_2} - E_{X_1} - E_{X_3} \)            | \( E_{X_1} - E_{X_{20}} \) |
| Heat exchanger (c)                 | \( E_{\text{X}_\text{Exchanger-c}} = E_{X_1} - E_{X_{20}} - E_{X_{20}} - E_{X_{20}} \)    | \( E_{X_{20}} - E_{X_{20}} \) |
| Heat exchanger (d)                 | \( E_{\text{X}_\text{Exchanger-d}} = E_{X_{11}} - E_{X_{11}} - E_{X_{11}} - E_{X_{11}} \) | \( E_{X_{11}} - E_{X_{11}} \) |
| Heat exchanger (e) (evaporator)    | \( E_{\text{X}_\text{Exchanger-e}} = E_{X_{12}} - E_{X_{12}} - E_{X_{12}} - E_{X_{12}} \) | \( E_{X_{12}} - E_{X_{12}} \) |
| Turbine                            | \( E_{\text{X}_\text{Turbine}} = E_{X_1} - E_{X_1} - E_{X_1} - E_{X_1} + W_{\text{Turbine}} \) | \( \frac{E_{X_1}}{E_{X_1}} \) |
| Condenser of ejector cycle         | \( E_{\text{X}_{\text{Cond-ejector}}} = E_{X_{16}} - E_{X_{16}} - E_{X_{16}} - E_{X_{16}} \) | \( E_{X_{16}} - E_{X_{16}} \) |
| Condenser of steam cycle           | \( E_{\text{X}_{\text{Cond-steam cycle}}} = E_{X_{14}} - E_{X_{14}} - E_{X_{14}} - E_{X_{14}} \) | \( E_{X_{14}} - E_{X_{14}} \) |
| Expansion valve                    | \( E_{\text{X}_\text{Expansion valve}} = E_{X_1} - E_{X_1} - E_{X_1} - E_{X_1} \)        | \( E_{X_1} - E_{X_1} \) |
| Auxiliary heater                   | \( E_{\text{X}_\text{Aux-heater}} = E_{X_{11}} - E_{X_{11}} + E_{X_{11}} + E_{X_{11}} \)   | \( E_{X_{11}} - E_{X_{11}} \) |
| Ejector                            | \( E_{\text{X}_\text{Ejector}} = E_{X_1} - E_{X_1} - E_{X_1} - E_{X_1} \)                | \( E_{X_1} - E_{X_1} \) |
| Deaerator                          | \( E_{\text{X}_\text{Deaerator}} = E_{X_1} - E_{X_1} - E_{X_1} - E_{X_1} \)              | \( E_{X_1} - E_{X_1} \) |
| Auxiliary boiler                   | \( E_{\text{X}_\text{Aux-boiler}} = E_{X_{16}} - E_{X_{16}} + E_{X_{16}} + E_{X_{16}} \)   | \( E_{X_{16}} - E_{X_{16}} \) |
| Pump (f)                           | \( E_{\text{X}_\text{Pump-f}} = E_{X_{16}} - E_{X_{16}} - E_{X_{16}} + W_{\text{Pump}} \)  | \( E_{X_{16}} - E_{X_{16}} \) |
| Pump (c)                           | \( E_{\text{X}_\text{Pump-c}} = E_{X_{18}} - E_{X_{18}} + E_{X_{18}} + E_{X_{18}} \)       | \( E_{X_{18}} - E_{X_{18}} \) |
| Hot storage tank                   | \( E_{\text{X}_\text{Hot-stORAGE}} = E_{X_{22}} + E_{X_{22}} - E_{X_{22}} - E_{X_{22}} \)  | \( E_{X_{22}} - E_{X_{22}} \) |
| Cold storage tank                  | \( E_{\text{X}_\text{Cold-stORAGE}} = E_{X_{12}} + E_{X_{12}} - E_{X_{12}} - E_{X_{12}} \) | \( E_{X_{12}} - E_{X_{12}} \) |

\[
\sum E_{X_{\text{in}}} = \sum E_{X_{\text{out}}} + E_{X_D} \quad (15)
\]

\[
i \sum m_i e_i + \dot{E}_{X_Q} = e \sum m_e e_e + \dot{E}_{X_W} + \dot{E}_{X_D} \quad (16)
\]

where \( i \) and \( e \) indicate the inlet and outlet exergy flow from the control volume, \( \dot{E}_{X_W} \) and \( \dot{E}_{X_Q} \) are the exergy transfer by work and heat, respectively and \( \dot{E}_{X_D} \) is the exergy destruction. The rate of the exergy efficiency and destruction for each cycle component are obtained based on Table 3.

7 | VALIDATION

To verify the modeling of the one-dimensional ejector, its output results are compared with the experimental data of Ref. 47. The input parameters, which are used to validate the proposed model, are mentioned in Table 4. The theoretical and experimental results of different conditions are compared in Table 5. It can be seen that there is a good agreement between them.

8 | RESULTS AND DISCUSSION

In the following, the results of the annual simulation of the solar ejector-based CCHP system are discussed. The effect of various parameters on the system performance is also investigated. For the base case, the collector surface area, the hot storage tank volume, the inlet turbine temperature, and pressure are taken 3000 m², 100 m³, 210°C, and 5 bar,
respectively. Due to the system’s required cooling load and the ejector performance, ten parallel ejectors are used.

Figure 8 indicates the monthly incident solar radiation on the collector surface, as well as the collector’s useful energy gain and efficiency. It should be noted that the maximum collector efficiency (74.7%) occurs in July. The hourly variation of the collector inlet and outlet temperature, the collector flow rate, and the
auxiliary heater inlet temperature are shown for the last ten days of July in Figure 9. As can be seen, the solar collector pump is switched off during the night to prevent energy loss.

In Figure 10, the monthly variation of the heat transfer values of the heat exchangers, the turbine power generation, the collector useful energy gain, the energy consumption of the auxiliary heater and boiler are plotted. Figure 10A shows that the energy consumption of the ejector cycle generator (heat exchanger (b)) increases in the warm months with increasing the cooling load of the building. Moreover, the backpressure of the turbine increases in comparison with that in the cold months. Consequently, the turbine power generation decreases in the warm months.

The energy consumption of the auxiliary boiler is also higher in the cold months, as the heating load increases. According to Figure 10B, the collector’s useful energy gain increases in the warm months, and consequently, the energy consumption of the auxiliary heater decreases. As it is shown in Figure 10B, the amount of thermal storage losses is too low, so the sum of collector useful energy gain and auxiliary heater is approximately equal to the heat transfer in the heat exchanger (a).

Due to the excess waste heat recovery of the system, it can be sold to nearby buildings. In general, the number of additional apartments, which can be supported for the heating and hot water loads by this cycle, varies during the months. Its minimum occurs in January with 96 apartments, which is the coldest month of the year. On the other hand, its maximum happens in July with 1505 apartments, which is the hottest month of the year and only hot water load is required. According to the annual analysis, the system has the ability to provide heating load and hot water for 346 apartments, in addition to the 14 mentioned apartments.

Figure 11 shows the monthly variation of SF and the efficiency of the various components of the cycle. It is found that the maximum overall cycle efficiency occurs in January and decreases slightly during the warm months. The efficiency of the Rankine cycle is also the highest in the cold months (January), with a slight decrease in the warm months. However, the ejector cycle efficiency, which operates only in the warm months, is almost constant over the
months of April to October. The highest SF is also obtained in June and July. The results show that the collector's useful gain increases in the warm months while the heater's energy consumption decreases. The energy consumption of the auxiliary boiler decreases due to the reduction of the heating load in the warm months and also, the turbine power generation decreases as the condenser pressure and required energy of the ejector cycle generator increase; therefore, the overall cycle and steam cycle efficiencies reduce. The annual efficiency and SF are 32.6% and 9.5%, respectively. Furthermore, the annual energy consumption of the auxiliary heater, the collector useful gain, the produced cooling and heating, the turbine power generation, and the auxiliary boiler energy consumption are 31,565, 3452, 190, 4300, 5619, 1930 MWh, respectively.

In Figure 12, the effect of the solar collector area on the system performance is investigated using three different areas of 2000, 2500, and 3000 m². The simulation was carried out at a turbine inlet temperature and pressure of 210°C and 5 bar, respectively. It is observed that the SF increases 1.6% and 3.2% by increasing the collector area from 2000 to 2500 m² and 3000 m², whereas it does not have a significant impact on the overall cycle efficiency. By increasing the collector area, the collector useful gain increases; thus, the energy consumption of the heater decreases. As a result, the total input energy to the system does not change, which results in no variation in the overall cycle efficiency.

The effect of the turbine inlet temperature on the ejector cycle efficiency, the Rankine cycle efficiency, and the overall cycle efficiency is shown in Figure 13. Based on the obtained results, the overall cycle and steam cycle efficiencies increase with increasing turbine inlet temperature, while the ejector cycle efficiency change insignificantly. Increasing the turbine inlet temperature from 190 to 210°C resulted in an increase of 1% and 2% in the Rankine cycle and the overall cycle efficiencies, respectively. Since the increase in the turbine inlet temperature did not have a significant effect on the temperature of the turbine
extracted flow, the temperature of the generator’s primary flow does not change considerably, and the ejector cycle efficiency remains constant. As the turbine inlet temperature increases, the inlet enthalpy of the turbine slightly increases, which, in turn, increases the steam cycle efficiency; thus, the overall cycle efficiency increases.

Figure 14 illustrates the maximum cooling, heating and power generation, as well as the maximum SF versus different turbine inlet temperatures. The highest values are obtained at the turbine inlet temperatures of 210°C, which are 185.46, 598.65, 680.49 kW, and 70%, respectively.

In this paper, the effect of storage tank volume on the performance of the CCHP cycle is also investigated using three different volumes of 100, 150, and 200 m³. It is observed that changing the storage tank volume has no effect on the cycle performance criteria.

The values of the annual exergy efficiency and destruction rates for each cycle component are given in Table 6. Moreover, the portion of main components on annual exergy destruction is illustrated in Figure 15. As can be seen, the most annual exergy destruction is related to the solar collector, which comprises 27% of total exergy destruction. The expansion valve and auxiliary heater attained high annual exergy efficiency with 97.7% and 97.27%, respectively. On the other hand, the ejector achieved the lowest annual exergy efficiency with 25.9% among various components.

The annual exergy efficiency of the proposed system has been calculated, which is 40%. Furthermore, the exergy efficiencies of the solar cycle, steam Rankine cycle, and ejector cooling cycle are obtained 74.58%, 49.61%, and 25.23%, respectively.

9 | CONCLUSION

In this paper, the performance of a solar ejector-based CCHP was dynamically simulated using the TRNSYS- EES co-simulator. The findings can be summarized as follows:

- The turbine power generation decreases in the warm months as the turbine backpressure increases in comparison with that in the cold months.
- The maximum Rankine cycle efficiency and the maximum overall cycle efficiency occur in the cold months (January) and decrease slightly during the warm
months. The ejector cycle efficiency is almost constant over the months of April to October.

- Although the overall cycle efficiency does not change significantly by the collector area, the solar fraction increases by 1.6% and 3.2% when the collector area changes from 2000 to 2500 m² and 3000 m².
- By increasing the turbine inlet temperature, the Rankine cycle and the overall cycle efficiency increase, whereas the ejector cycle efficiency remains constant. The maximum cooling, heating, and power generation, as well as the maximum solar fraction, are obtained at the turbine inlet temperatures of 210°C, which are 185.46, 598.65, 680.49 kW, and 70%, respectively.
- The results show that changing the cycle performance criteria is not affected by the storage tank volume.
- The energy consumption of the auxiliary heater, the collector useful gain, the produced cooling and heating, the turbine power generation, and the auxiliary heater energy consumption over a year are 31,565, 3451, 189, 4299, 5618, 1929 MWh, respectively. The annual overall cycle efficiency is obtained 32.6%.
- The solar collector has obtained the most annual exergy destruction, comprising 27% of total exergy destruction. Also, the ejector achieved the lowest annual exergy efficiency with 25.9%. The annual system exergy efficiency has obtained 39.99%.

NOMENCLATURE

Symbols

| Symbol | Description |
|--------|-------------|
| P      | Pressure    |
| T      | Temperature |
| C_p    | Specific heat at constant pressure |
| k      | Ratio of specific heat at constant pressure to constant volume |
| D      | Area        |
| A      | Mach number |
| R      | Gas constant |
| M      | Heat transfer rate |
| Q      | Mass flow rate |
| NTU   | Number of transfer units |
| C      | Heat capacity |
| U      | Heat transfer coefficient |
| S      | Entropy |
| SF     | Solar fraction |
| h      | Hot stream |
| c      | Cold stream |
| i      | Inlet       |
| r      | Ratio       |
| aux    | Auxiliary   |
| T      | Turbine     |
| P      | Pump        |
| hex    | Heat exchanger |

Subscript

| Subscript | Description |
|-----------|-------------|
| e         | Secondary flow |
| f         | Diffuser exit |
| m         | Motive flow |
| y         | Flow mixing start |
| t         | Motive nozzle throat |
| n         | Motive nozzle exit |
| d         | Diffuser inlet |
| max       | Maximum |
| min       | Minimum |
| amb       | Ambient |
| Su        | Sun surface |
| cond      | Condenser |
| ap        | Collector aperture area |

Greek

| Greek | Description |
|-------|-------------|
| ε     | Heat exchanger efficiency |
| ω     | Entrainment ratio |

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