An efficient biogas-based tri-generation of power, heating and cooling integrating inverted Brayton and ejector transcritical CO$_2$ cycles: exergoeconomic evaluation

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Abstract

In the present work, an efficient multigeneration system is proposed to meet diverse energy requirements such as power, heating, and cooling. The system consists of a biogas-fueled gas turbine cycle as the topping cycle, and a Brayton, an inverted Brayton, and also, a transcritical carbon dioxide refrigeration cycles with an ejector expansion as the bottoming cycles. Using the ejector instead of compressor results in a reduction in the power consumption. Moreover, the required power of the refrigeration cycle can meet by the inverted Brayton cycle which eliminates the need for an external power source. The thermodynamic and exergoeconomic evaluations are done for the suggested system considering the energy and exergy efficiencies and total specific cost of the system as objective functions. Also, a parametric analysis is performed to specify the effects of decision variables on the system performance. The energy and exergy efficiencies and total cost rate of the system are determined as 79%, 44.4%, and 183.4 $/h$, respectively. These values demonstrate that the energy and exergy efficiencies of the proposed system have been improved by 48.9% and 54%, respectively, compared to the gas turbine cycle. Also, the cost of produced electricity is calculated to be 52.84 $/MWh.

Keywords: gas turbine, inverted Brayton, transcritical carbon dioxide refrigeration, ejector expansion, exergoeconomics

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1. Introduction

In recent years, the growing trend of energy consumption has caused an energy crisis in the world. Therefore, energy and energy saving is one of the most serious issues in the world. Global concerns in addition to energy supply problems, include environmental problems such as ozone depletion, air pollution, acid rain, deforestation, and radioactive emissions. These problems must be considered simultaneously so that humanity can achieve a future with a definite energy with least environmental impact [1]. Pollution reduction and minimizing the fossil fuel consumption are possible through a strong focus on environmental sustainability. Cogeneration systems
fueled by renewable fuels are some of the most suitable options to achieve this goal. One of these renewable fuels is biogas, which is produced from the anaerobic decomposition of biomass [2]. This method is best solution to dissolve this obstacle and get rid of wastes. In recent years, researchers have done more research on the CCHP system based on renewable energy and waste heat recovery to decrease the required energy and emission of pollutants. Recently, in industrialized countries, more than 50% of the heat from the combustion of fuel in the engine is wasted by the coolant and exhaust [3, 4].

Darabadi et al. [2], analyzed exergy and exergoeconomic performance of a cogeneration system fueled by the blend of natural gas and biogas. They used municipal solid waste as the feedstock to the digestion. The results revealed that in the absence of biogas, the heat recovery steam generator and combustion chamber have the highest exergy destruction rate while it is highest for the anaerobic digester and combustion chamber in the absence of NG. Moreover, the exergy efficiency of the cycle in the case of pure biogas and pure natural gas was 46.94% and 50.64%, respectively.

Kun Yang et al. [5], studied thermo-economic evaluation of multiple production system of cooling, power and heating by gasification of biomass fuel. Sensitivity analysis showed that the most sensitive factor for the cost of the manufactured products is the cost of biomass. Also, the unit exergetic cost of electricity was estimated as 4.445, 4.447 and 4.447 for the cold season, warm season, and transition season, respectively. María Uris et al [6]. assessed and compared the performance of the CHP and CCHP (combined power, heating, and cooling) systems with biomass fuel in Spain. The Organic Rankine Cycle (ORC) was used in both systems. The results showed that the emission of carbon dioxide in the CHP system is more than the CCHP system.

Leonzio [7], proposed a novel cogeneration system which uses biogas as a renewable fuel. They burned 3,280 kW biogas to produce 925 kW electricity and 1875 kW heating.

Shu et al. [3], introduced CTRC (CO₂ Transcritical Rankine Cycle) including the regenerator and preheater for recovering waste heat in exhaust gas of an engine. The results show the considerable advantage of extremely recycling the engine coolant and exhaust gas. The preheater and regenerator (PR-CTRC) Upgrades the exergy efficiency and energy efficiency of the CTRC, with increases of 227% and 184%, respectively. Mohammadi [8] analyzed new multi-generation systems based on carbon dioxide refrigeration cycles. The systems are examined
considering 1 MW capacity and evaporator temperatures of $-35 \, ^\circ C$ to $-45 \, ^\circ C$. Carbon dioxide has attracted a lot of attention due to its many advantages as a working fluid in various cycles. First of all, CO$_2$ has no restriction on the temperature of thermal decomposition. Secondly, heat can be transferred directly from the exhaust gases to carbon dioxide, and due to the transcritical state of the carbon dioxide in the evaporator, a better temperature matching can be achieved. Thirdly, carbon dioxide is non-flammable and environmental friendly [9].

Yari [10], assessed the performance of a two-stage ejector transcritical refrigeration CO$_2$ cycle, which uses an intercooler and an internal heat exchanger to increase the system efficiency. The results confirm that the Coefficient of Performance (COP) and exergy efficiency for the novel two-stage ejector expansion system are on average 12.5–21% more than those for the conventional two-stage system. Domanski reported that the ejector efficiency is more important feature in the performance coefficient of the ejector expansion refrigeration cycle. The amount of COP betterment depends on the refrigerant’s thermodynamic properties and it is different from system to system [11, 12]. The ejector is a low-cost, simple and easy-to-maintain device with high capability for suction, mixing of flows and compression. Thus, ejector has high potential in exergy and energy exploitation. Nevertheless, the CCHP system can utilize the gas turbine and ejector-refrigeration cycle for multiple production of power, heat and cold.

Dai et al. [13], introduced a combined power and refrigeration cycle with carbon dioxide refrigerant which is driven by the renewable energy source or flue gas of the engine. The results of parametric study showed that condenser temperature, evaporator temperature, and turbine inlet pressure had the greatest effect on the system performance. Also, it is concluded that the steam heat exchangers, ejectors and turbines are the most destructive components from the exergy viewpoint.

Gholizadeh et al. [14], Proposed a triple generation system of cooling, fresh water and electricity by utilizing the geothermal heat source. One of the most important advantages of this proposed system was the ability to produce cooling at two different temperatures by utilizing the two ejector refrigeration cycles. The optimization results showed that the cooling effect, exergy efficiency and power generation were increased to 87%, 46.3% and 77%. Song et al. [15], studied the exergy and economic evaluation of the hybrid system. This system was a combination of supercritical carbon dioxide and ORCs. The results displayed that the total power produced is 2940 kW.
Yari [16], performed thermodynamic analysis on the ejector-vapor compression refrigeration cycle. Based on the results, COP and exergy efficiency of the novel system are about 16% more than the vapor compression cycle. Tashtoush and et al. [17], studied a novel transcritical refrigeration cycle from the thermodynamic and thermoeconomic viewpoints. The results showed that total cost of the final product and overall exergy destruction rate are 6% and 22.25% less than the conventional ejector refrigeration cycle, respectively.

Wang et al. [18], simulated the cogeneration system of power and cooling from the energy, exergy and exergoeconomic viewpoints by employing of low-temperature heat sources. The results showed that the ejector has the highest irreversibility among the system components. Also, this proposed system can provide 18.52 kW of power and 23.89 kW of cooling and the efficiencies of the first and second laws of thermodynamic were 21.7% and 52.22%, respectively. In another study, Sarkhel et al. [19], investigated and optimized the polygeneration system of cooling, power and heating with solar energy. The system was analyzed from thermodynamic and thermoeconomic perspectives. The results showed that the total cost rate and the exergy efficiency at the optimal point are 0.4835 $/h and 10.06% respectively.

Sahu et al. [20], studied transcritical carbon dioxide refrigeration and combined refrigeration-power systems. They also did a sensitivity analysis on the COP and the second-law efficiency. The results revealed that the COP rises with turbine inlet temperature. Also, the exergy efficiency has a maximum value with the turbine inlet temperature. Yan Cao et al. [21], proposed new systems with biogas fuel for power and cooling, and analyzed and optimized the systems in terms of energy and exergy. The results showed that the steam generator has the highest irreversibility rate in the downstream cycle and the combustion chamber has the highest irreversibility rate in the gas turbine cycle. Also, the energy and exergy efficiencies of the system improved after optimization and reached 57.11% and 36.68%, respectively.

Chen et al. [22], introduced a power generation system combined with a carbon dioxide Brayton cycle and a carbon-dioxide air conditioning for vehicles engine utilizing waste energy of the exhaust gas. The proposed system is not attractive due to the high consumption of the compressor as most of the produced power will be used to provide part of the required power by the compressor in the refrigeration part of the system. Wang et al. [23], proposed transcritical carbon dioxide cycle for trigeneration of power, heating, and cooling, operating by solar energy. In this
proposed system, the total net power is zero or close to zero in the different operating conditions, so input power is required for cycle operation.

Sinha et al [24]. analyzed the energy and exergy of power generation systems. The results showed that by adding heat recovery and fuel cell to the system, the performance of the system increases.

Kumar et al [25]. studied the thermodynamic analysis of the integrated system of gas turbine cycle with solid oxide fuel cell and organic Rankine cycle. By a conducting a parametric study, the results showed that the maximum energy and exergy efficiencies of the proposed system are 47.34% and 56.85%, respectively.

Mohammadi et al [26]. studied the thermodynamic analysis of multiple generation system of power, cooling and heating. The results showed that gas turbine inlet temperature and pressure ratio are the most effective influencing parameters.

Considering the above discussion, it could be deduced that inventing a trigeneration system of the power, and cooling, heating with high efficiency driven by renewable fuel as the input energy, and also, the use of safe and environmentally friendly refrigerant as a coolant in the refrigeration systems is necessary. Accordingly, in the present work, to increase the thermal and exergy efficiencies, a new arrangement based on ejector CO2 transcritical cycle has been proposed. In this work, by using the ejector instead of the compressor, the power consumption is reduced. Moreover, by applying the inverted Brayton cycle in the refrigeration cycle, the generated power improves the system efficiency, as well as the required power for the refrigeration system, as the sub-cycle does not need to be supplied from another resource. Biogas is suggested as a fuel owing to its many benefits illustrated in our previous works [2, 27]. The proposed hybrid system with carbon dioxide working fluid has been analyzed in terms of thermodynamics and exergoeconomics using a developed simulation model in Engineering Equation Solver (EES). This system combines a Gas turbine, a Brayton, an inverted Brayton and a refrigeration cycles with ejector expansion. Also, to identify the influences of the various thermodynamic parameters on the system performance, a parametric assessment has been accomplished. Briefly, the novelties and main objectives of the work can be written as:

• Proposing a novel efficient multigeneration configuration to recover the waste heat of a biogas-fueled gas turbine combining a gas turbine, a Brayton, an inverted Brayton, and a refrigeration cycles with ejector expansion.

• Employing an ejector instead of the compressor to reduce the power requirement.
Using an inverted Brayton cycle to meet the required power of the refrigeration subsystem and eliminate the need for the external power source.

Evaluating the proposed system in terms of thermodynamics and exergoeconomics.

Performing a parametric study to identify the effects of decision variables on the thermodynamic and exergoeconomic parameters of the system.

2. System description

The proposed CCHP system is shown in Figure 1, that includes a biogas fueled gas turbine as the topping cycle while the bottoming cycle consists of a Brayton, an inverted Brayton and an ejector-expansion transcritical CO₂ refrigeration cycles followed by a single-pressure HRSG. The bottoming cycle utilizes the waste heat of the gas turbine exhaust from the topping cycle through a gas heater. According to Fig. 1, air enters to the air compressor at the atmospheric pressure and temperature (1) and is pressurized to the operating pressure of the gas turbine (2). The pressurized air enters to the APH until its temperature reaches to 700 K (3). Then, it is burned with the biogas in the combustion chamber. Flue gases leave the combustion chamber at 1250 K and enter the gas turbine (4) where 1 MW of power is generated. After passing the turbine, flue gas enters the preheater (5), and then, to use the dissipated heat and also supply the supercritical cycle of carbon dioxide, it enters into the gas heater to transfer heat (6). The hot gasses leaving the gas heater enter the HRSG (7). From the other side, water at the temperature of 25°C and the pressure of 3500 kPa enters the HRSG and comes out of it as saturated steam. Sub-critical carbon dioxide (22) is sent to the compressor and is compressed to a supercritical state (23). SCO₂ enters the gas heater to reach the desired temperature (8). After leaving the gas heater, the high-temperature supercritical CO₂ enters the HP and LP turbines and expands to a lower pressure to generate power (9, 10). The output carbon dioxide from the LP turbine enters the heater to provide district heating (11). After leaving the heater, carbon dioxide enters the compressor and reaches the desired pressure (12), and then loses heat in the gas cooler (13). Then, carbon dioxide enters the ejector to entrain the secondary flow (18) from the evaporator. The carbon dioxide stream enters to the separator (14) after leaving the ejector to separate the saturated vapor and saturated liquid. Saturated liquid enters the evaporator after crossing by the throttle valve (15), and also, a portion of the carbon
dioxide saturated vapor is returned to the evaporator through the throttle valve (21). It is worth noting that in Figure 2 shows the T-s diagram for the proposed CCHP subsystem.

3. Modeling

Energy and exergy and exergoeconomic modeling of the proposed CCHP system is done using the developed simulation code in Engineering Equation Solver (EES). More attention is paid to the combustion chamber and ejector, since they are important components of the systems. For this reason, modeling of the combustion chamber and ejector are described in sections 3.5 and 3.4, respectively.

3.1. Energy Analysis

In this study, with assuming steady state condition and neglecting the changes of potential and kinetic energies mass balance and the first law of thermodynamics equations are written as follows [2, 27]:

\[ \sum \dot{m} = \sum \dot{m} \]  
\[ \sum \dot{Q} + \sum \dot{m} h = \sum \dot{W} + \sum \dot{m} h \]  

3.2 Exergy Analysis

The exergy equation for a control volume is defined as below [28]:

\[ \sum \dot{E}_i = \sum \dot{E}_i + \dot{I} \]  

where, $\dot{E}_i$ and $\dot{E}_j$ are the input and output exergy rates of the component, respectively. Also, $\dot{I}$ is the exergy destruction rate of the component.

Exergy is divided into the chemical and thermo-mechanical parts. In the absence of the surface tension, magnetic, electrical and nuclear effects, the total kinetic, potential and physical exergies are defined as thermo-mechanical exergy.

$$E_{tot} = E_{th} + E_{ch}$$

$$E_{th} = E_{ph} + E_{pt} + E_{kn}$$

(4)

(5)

The potential and kinetic exergies are negligible, so thermo-mechanical exergy is defined as [29]:

$$\dot{E}_{th} = \dot{E}_{ph} = \dot{m}(h - h_0 - T_0(s - s_0))$$

(6)

here, $s$, $T$, $h$ and are specific entropy, temperature and specific enthalpy, respectively. Chemical exergy is defined as [29]:

$$E^{CH} = \sum x_i e^{CH} + \overline{RT}_0 \sum x_i \ln x_i$$

(7)

Which $\overline{R}$ and $x_i$ are the universal gas constant and the mole fraction of the $i$th gas, respectively.

The assumed values for the simulation of the proposed CCHP system are given in Table A1 in the Appendix.

To thermodynamic modeling, the made assumptions in this study are as follows:

- The compressors and turbines isentropic efficiencies are 86% [1].
- The pressure drops in the APH and the combustion chamber are 5% and 4%, respectively.
- All processes are steady state ($\frac{d}{dt} = 0$) [29].
- The streams through the throttle valves are constant enthalpy.
- The exiting flow from the evaporator is saturated vapor.
3.4 Ejector

The structure of the ejector that is one of the most important components of the CCHP system consists of four parts: suction, mixing, constant-area and diffuser sections as seen in Figure 3. The gas cooler output stream as a high-pressure motive stream enters the ejector which has an insignificant velocity and entrains secondary stream from evaporator, which has less pressure than the primary fluid. The entrainment ratio is one of the important characteristics of ejectors which is written as the ratio of the secondary to primary mass flow rates. Then the streams are blended in the mixing section. If the mixed fluid is supersonic, a shock wave will occur in the diffuser inlet, that causes the refrigerant velocity to reach from the supersonic to the subsonic state, and consequently, pressure will be increased by the diffuser to separator pressure. It is worth noting that the ejector can be assortment into two kinds based on the situation of the nozzle: constant-pressure mixing and constant-area mixing [23]. It is assumed in this study that the mixing of ejector takes place at the constant pressure. The details of the mathematical model (energy, momentum and mass) for the ejector can be written as below:

In the nozzle section, the energy equation for the primary current is defined as follows:

$$h_{pr,n1} + \frac{v_{pr,n1}^2}{2} = h_{pr,n2} + \frac{v_{pr,n2}^2}{2}$$  \hspace{1cm} (8)

In comparison with the exit velocity of primary stream $h_{pr,n2}$, the inlet velocity of primary stream $h_{pr,n1}$ could be insignificant.

$$v_{pr,n2} = \sqrt{2*(h_{pr,n1} - h_{pr,n2})}$$  \hspace{1cm} (9)

Nozzle efficiency is defined as follows:

$$\eta_{nuzzle} = \frac{h_{pr,n1} - h_{pr,n2}}{h_{pr,n1} - h_{pr,n2,s}}$$  \hspace{1cm} (10)

Hence, by combining equations 10 and 9, equation 11 is obtained:
The entrainment ratio of the ejector can be defined as follows:

\[ \mu = \frac{m_{se}}{m_{pr}} \]  

(12)

Also, the energy equation for the secondly stream is expressed as:

\[ h_{se,n1} + \frac{v_{se,n1}^2}{2} = h_{se,n2} + \frac{v_{se,n2}^2}{2} \]  

(13)

In comparison with the exit velocity of secondly stream \( h_{se,n2} \), the inlet velocity of secondly stream \( h_{se,n1} \) could be insignificant.

\[ v_{se,n2} = \sqrt{2(h_{se,n1} - h_{se,n2})} \]  

(14)

\[ \eta_{nuzzle} = \frac{h_{se,n1} - h_{se,n2}}{h_{se,n1} - h_{se,n2,s}} \]  

(15)

By combining equations 15 and 14, equation 16 is obtained:

\[ v_{se,n2} = \sqrt{2 \eta_{nuzzle} (h_{se,n1} - h_{se,n2,s})} \]  

(16)

As well as in the mixing section, the momentum conservation equation can be written as below:

\[ m_{pr,n2} \cdot v_{pr,n2} + m_{se,n2} \cdot v_{se,n2} = (m_{pr,n2} + m_{se,n2}) \cdot v_{mix} \]  

(17)

Hence, by combining equations 12 and 17, one can write:
\[ v_{\text{mix,s}} = \frac{v_{pr,n2} + \mu v_{se,n2}}{1 + \mu} \]  

The mixing efficiency is given as:

\[ \eta_m = \frac{v_{\text{mix,m}}^2}{v_{\text{mix,s}}^2} \]  

\[ v_{\text{mix,m}} = \sqrt{\eta_m} \frac{v_{pr,n2} + \mu v_{se,n2}}{1 + \mu} \]  

The energy equation for the mixing section is defined as follows

\[ m_{pr} \left( h_{pr,n2} + \frac{v_{pr,n2}^2}{2} \right) + m_{se} \left( h_{se,n2} + \frac{v_{se,n2}^2}{2} \right) = (m_{pr} + m_{se}) \left( h_{\text{mix}} + \frac{v_{\text{mix,m}}^2}{2} \right) \]  

By combining equations 12 and 21, equation 22 is obtained:

\[ h_{\text{mix}} = \frac{1}{1 + \mu} \left( h_{pr,n2} + \frac{v_{pr,n2}^2}{2} \right) + \frac{\mu}{1 + \mu} \left( h_{se,n2} + \frac{v_{se,n2}^2}{2} \right) - \frac{v_{\text{mix,m}}^2}{2} \]  

Also, the output enthalpy and temperature of diffuser section is calculated as (By developing a simulation code in EES):

\[ h_{di,s} = f(S_{di,s}, P_{14}) \]  

Diffuser efficiency is defined as follows:

\[ \eta_{\text{diff}} = \frac{h_{di,s} - h_{\text{mix}}}{h_{di,a} - h_{\text{mix}}} \]  

From relation 24, relation 25 is obtained:
Finally, the ejector outlet temperature is calculated as (By writing code in EES program):

\[ T_{14} = f(h_{\text{dia}}, P_{14}) \]  \hspace{1cm} (26)

3.5 Combustion chamber

It is considered that the combustion process happens in the combustion chamber with the excess air. As example for one mole of fuel, reaction equation is written as follows:

\[
0.6CH_4 + 0.4CO_2 + n_3(0.2059O_2 + 0.7748N_2 + 0.0003CO_2 + 0.019H_2O) \rightarrow \]

\[
(1 + 0.0003n_3)CO_2 + (1.2 + 0.019n_3)H_2O + 0.7748n_3N_2 + (0.2059n_3 - 1.2)O_2
\]  \hspace{1cm} (27)

Where \( n_3 \) is the air mole numbers of combustion chamber. As mentioned above, the equation of the first law for the combustion chamber is written as follows [2, 30]:

\[
\dot{Q} - \dot{W} = \dot{n} \cdot \bar{h}_{pr} - \dot{n}_{\text{react}} \cdot \bar{h}_{\text{react}}
\]  \hspace{1cm} (28)

here, heat transfer to the environment and the enthalpy of the products are calculated as follows:

\[
\dot{Q} = 0.02 \cdot \dot{n}_f \cdot LHV_F
\]  \hspace{1cm} (29)

\[
\bar{h}_{pr} = X_{CO_2} \bar{h}_{CO_2} + X_{H_2O} \bar{h}_{H_2O} + X_{N_2} \bar{h}_{N_2} + X_{O_2} \bar{h}_{O_2}
\]  \hspace{1cm} (30)

where, \( X_l \) is the molar fraction of combustion products components.

\[
X_{CO_2} = \frac{1 + 0.0003n_3}{1 + n_3} \hspace{1cm} X_{H_2O} = \frac{1.2 + 0.019n_3}{1 + n_3} \hspace{1cm} X_{N_2} = \frac{0.7748n_3}{1 + n_3} \hspace{1cm} X_{O_2} = \frac{0.2059n_3 - 1.2}{1 + n_3}
\]

3.6 Exergoeconomic analysis
Exergoeconomics is a powerful technique for evaluation and optimization of energy systems which is the combination of economic principles and the second law of thermodynamics. This method provides useful information that exergy or economic cannot handle separately. In this field, many approaches have been presented, including the methods of average cost approach, the exergy cost theory and the specific exergy costing (SPECO). In this study, exergoeconomic analysis has been employed based on SPECO method due to the easy procedure and lower calculation time. To start the analysis in this method, in the first stage, all input and output flows are identified in the system boundary and are introduced as product or fuel. Then, the cost balance is expanded for components of the system as:

\[
\sum_{j} \dot{C}_{j,m} + \dot{C}_{w,m} = \dot{C}_{q,m} + \sum_{i} \dot{C}_{i,m} + \dot{Z}_m
\]  

(31)

Here, the subscripts i and j indicate input and output streams, respectively. \(\dot{Z}\) expresses sum of the cost rate of Capital Investment (CI) and cost rate interdependent with Operation and Maintenance (OM), and \(\dot{C}\) is the cost rate. Also, \(\dot{C}_{q,k}\) and \(\dot{C}_{w,k}\) represent the cost rate associated with heat transfer and work, respectively, and are written as follows:

\[
\dot{C}_i = c_i \dot{E}_i
\]  

(32)

\[
\dot{C}_j = c_j \dot{E}_j
\]  

(33)

\[
\dot{C}_q = c_q \dot{E}_q
\]  

(34)

\[
\dot{C}_w = c_w \dot{E}_w
\]  

(35)

\[
\dot{Z}_m = \dot{Z}_{ci}^{\text{om}} + \dot{Z}_{m}^{\text{om}} = CRF \frac{\Phi_r}{N} \* PEC_m
\]  

(36)

where, N indicates the annual operating hours, \(\Phi_r\) and \(PEC_k\) are the maintenance factor and the components’ purchased cost in US dollar, respectively. Also, CRF depends on Beneficial Life (BL) and interest rate (i) and it is defined as:
\[ CRF = \frac{i(1+i)^{BL}}{(1+i)^{BL} - 1} \]  

(37)

Which CRF is Capital Recovery Factor.

In this study, \(i\) and BL are supposed to be 0.12 and 10 years, respectively. Also, the PEC of the components is given in Table A2 in the Appendix.

In the equation of 29, auxiliary equations are needed as the number of unknown flow cost equations is more than the number of cost balance equations. Whenever the number of output exergy currents of a component is more than one \((k>1)\), the number of \((k-1)\) auxiliary equations will be needed. To develop the auxiliary equations, the P and F principles can be utilized. Using this method, a set of linear equations \(\left[\dot{E}_k\right]\left[c_k\right]=\left[\dot{Z}_k\right]\) is written for components of the system. Thus the cost rate of all streams is obtained. Here , \([\dot{Z}_k]\) and \([\dot{E}_k]\) are obtained by Eq. (34) and exergy analysis, respectively, and so, \([c_k]\) which is the matrix of the specific cost can be appraised.

4. Performance parameters

To appraise the proficiency of a CCHP system, some indicators are determined. The thermal and exergy efficiencies of the CCHP system are defined as below [2]:

\[ \eta_{th} = \frac{\dot{w}_{GT} + \dot{w}_{T1} + \dot{w}_{re} - \dot{w}_{C1} + \dot{Q}_{HRS} + \dot{Q}_{Eva} + \dot{Q}_{Heater}}{m_{bio} \cdot LHV_{bio}} \]  

(38)

\[ \varepsilon = \frac{\dot{w}_{GT} + \dot{w}_{BC} + \dot{w}_{IB} + \dot{E}_{26} - \dot{E}_{25} + \dot{E}_{eva} + \dot{Q}_{Heat} \cdot (1 - \frac{T_0}{T_{heat}})}{\dot{E}_{1} + \dot{E}_{bio}} \]  

(39)
The $\dot{W}_{GT}$, $\dot{W}_{IBC}$ are the production power of the gas turbine cycle, inverted Brayton cycle, respectively, and they are defined as:

\[
\dot{W}_{GT} = \dot{W}_T - \dot{W}_C
\]  
\[\text{(40)}\]
\[
\dot{W}_{IBC} = \dot{W}_{T, re} - \dot{W}_{C, re}
\]  
\[\text{(41)}\]

$\dot{E}_{eva}$ is the exergy associated with the refrigeration output, which is calculated as follows:

\[
\dot{E}_{eva} = \dot{m}_{eva} \ast ((h_{eva, in} - h_{eva, out}) - T_0 \ast (S_{eva, in} - S_{eva, out}))
\]  
\[\text{(42)}\]

And the total cost rate of the system, $\dot{C}_{\text{system}}$, is defined as:

\[
\dot{C}_{\text{system}} = \dot{C}_F + \sum_i \dot{C}_i + \sum_i \dot{C}_{D,i} + \dot{C}_L
\]  
\[\text{(43)}\]

Which $\dot{C}_{D,i}$ is the exergy destruction cost rate.

The average costs per the exergy unit are described as:

\[
c_{p,k} = \frac{\dot{C}_{p,k}}{\dot{E}_{p,k}}
\]  
\[\text{(44)}\]
\[
c_{f,k} = \frac{\dot{C}_{f,k}}{\dot{E}_{f,k}}
\]  
\[\text{(45)}\]
\[
c_{D,k} = \frac{\dot{C}_{D,k}}{\dot{E}_{D,k}}
\]  
\[\text{(46)}\]

Relative cost difference ($r_k$) and also exergoeconomic factor ($f_k$) are defined as (45) and (46), respectively [27]:

\[
r_k = \frac{c_{p,k} - c_{f,k}}{c_{f,k}}
\]  
\[\text{(47)}\]
5. Results and discussion

A simulation code is developed using Engineering Equation Solver (EES) to run the set of equations derived from the thermoeconomic modeling of each component. Fig. 4 shows the solving methodology flowchart.

To ensure the accuracy of the simulation, the results are compared with references [23, 30]. The comparison for the gas turbine and transcritical CO$_2$ cycle are given in Table 1 and Figure 5, respectively, which shows a good agreement. The maximum relative difference between the results is about 2\%.

The schematic figure of Wang et al. [23] is shown in Figure 6.

5.1 Energy and exergy analysis results

In this section, the achieved results from energy, exergy, and exergoeconomic simulation and as well as the design parameters effects on the system performance are reported. Based on the described analysis in the previous section, a simulation program was developed in EES for the integration of a gas turbine cycle with a production capacity of 1000 kW, a Brayton cycle, an inverted Brayton cycle and a transcritical carbon dioxide refrigeration system with ejector. The thermodynamic properties of each stream for the CCHP system are given in Table A3 in the Appendix. Also, Figures. 7a and 7b show the exergy destruction contribution of each component of topping and bottoming cycles. In the topping cycle, the combustor and in the bottoming cycle, the ejector have the highest irreversibility due to the combustion and mixing of fluids, respectively. By applying the inverted Brayton cycle, the total power of the ejector refrigeration cycle will be positive, and the best possible amount of back pressure of the inverted Brayton turbine (LP turbine) will be 5300 kPa, resulting in a power production of 5 kW. Also, the values demonstrate that the energy and exergy efficiencies will improve by 48.9\% and 54\%, respectively, compared to the gas turbine cycle, as well as energy and exergy efficiencies of the proposed system will increase
by 49.0% and 57%, respectively, compared to the basic cycle [23]. Table 2 shows the evaluation criteria of the proposed system from the thermodynamic viewpoint.

### 5.2 Exergoeconomic analysis results

The exergoeconomic results for the CCHP system are listed in Table 3. As can be seen from the results, the HRSG and combustion chamber have the maximum amount of $\dot{Z} + \dot{C}_D + \dot{C}_L$ among all the components. Thus, these components are the most important components in terms of thermo-economics. Also, the heater and gas heater have the highest amount of relative cost difference, which shows the sharp increase in the product cost of these components. $f_k$ which is the correlation between the exergy destruction and capital investment cost rates. Lower values of this parameter in the ejector and combustion chamber demonstrate that these are components with a higher cost rate associated with exergy destruction. Thus, exergy destruction cost rate should be decreased to enhance exergoeconomic performance. Moreover, the higher amount of the exergoeconomic factor in the system components shows that the capital cost rate is superior to the exergy destruction cost rate. According to Table 3, the exergoeconomic factor of the gas turbine of inverted Brayton cycle has a maximum value among the components.

Based on the obtained results, total cost rate of the system and cost of produced electricity are 183 $/h and 52.84 $/MWh, respectively.

### 5.3 Parametric study

The selected parameters for the parametric study were air compressor pressure ratio, ejector back pressure, turbine inlet temperature, air preheater outlet temperature, HP-turbine back pressure, ejector inlet temperature, HP-turbine inlet temperature and back pressure of the inverted Brayton turbine (LP turbine). To evaluate the effect of every parameter, the other parameters were kept fixed.

Figures 8a and 8b depict the influences of the air compressor pressure ratio on the system performance parameters. Because of the net power of gas turbine cycle is constant, the compressor input power and the gas turbine produced power rise by increasing this parameter. Thus, air flow rate first reduces and then increases. The enthalpy of the outlet and inlet streams of the combustion chamber are getting fixed. Thus, the biogas consumption
first decreases and then increases. Thus, exergy and energy input to the system, and also, water stream and heat recovery of HRSG have minimum values with the air compressor pressure ratio. These trends will lead a minimum value for the energy efficiency of the system. Moreover, increasing pressure ratio, reduces the total cost rate of the system because of reduction in the fuel mass flow rate. However, for the pressure ratios greater than 10, by increasing this parameter, the investment cost rates and also exergy destruction cost rate of the air compressor and combustion chamber are elevated. Thus, the net effect is a minimum value for the total cost rate.

Figures 9a and 9b demonstrate the effects of air preheater outlet temperature on the system performance. By rising this parameter, the fuel mass rate decreases that leads to a reduction in the input energy and exergy into the system. Also, by reducing the fuel mass flow rate, the temperature of the gas turbine exhaust reduces causing a reduction in the mass flow rate, and consequently, the exergy rate of the obtained steam in the HRSG. These variations lead to a reduction in the energy and exergy efficiencies of the system. Also, fuel and investment cost rates and irreversibility reduce with this parameter. As a result, the total cost rate of the system will be reduced with the output temperature of the air preheater.

According to Fig. 10, the energy and exergy efficiencies of the CCHP system improve with the combustion chamber outlet temperature. An increase in the combustion chamber outlet temperature reduces the air and fuel mass flow rates, which causes a decrease in the input energy to the CCHP system. But on the other hand, by rising this parameter, temperature of the combustion products at the HRSG inlet, and also, water flow rate of HRSG will rise. As a result, the recovered heat in the HRSG increases that leads to an improvement in the energy and exergy efficiencies.

By reduction of the flue mass flow rate with rising the combustion chamber outlet temperature, the cost rates related to initial investment and the total exergy destruction increase but fuel cost rate reduces. These changes cause a minimum value for the system total cost rate at the higher temperatures.

Figure 11a and 11b show the variations of performance parameters with the gas heater outlet temperature. By increasing this parameter, the heat input to the bottoming cycle will certainly decrease, and consequently, the mass flow rate of carbon dioxide, bottoming cycle output power and the amount of refrigeration effect will
increase. But on the other side, the amount of produced steam in the HRSG (state 24) increases, which leads to an increase in the amount of obtained energy and exergy. The reduction in the power and refrigeration of the bottom cycle is less than the rise in the steam exergy. Thus, sum of these variations makes a rise in the energy and the exergy efficiencies. Moreover, by reduction of the mass flow rate of working fluid (CO2), irreversibility and its related cost rate and investment cost rates of the system components reduce. Thus, the total cost rate of the CCHP system decreases by rising of the temperature of the flue gases at the gas heater outlet (state 7).

Figures 12a and 12b show the effects of ejector back pressure on the system performance. As can be seen, with increasing this parameter decreases the HP-compressor power due to the reduction in the pressure ratio. Owing to the enhancing HP-compressor inlet pressure and since the HP-compressor outlet pressure is constant, the HP-compressor outlet temperature drops. Considering a constant heat transfer, the temperature difference of the refrigerant in the gas heater increases leading to a reduction in the mass flow rate. With the reduction of the mass flow rate, the transferred heat in the heater and the amount of produced cooling effect by the evaporator will certainly reduce. On the other hand, by reducing the power consumption of the HP-pressure compressor, the net output power of the bottoming cycle increases. All in all, with rising the ejector back pressure, the energy efficiency decreases while the exergy efficiency increases. Also, with increasing the ejector back pressure, the total exergy destruction rate and investment cost rate of S-CO2 components including ejector, evaporator, turbines and compressors decrease leading to a reduction in the total cost rate of the system.

The effects of high pressure turbine inlet temperature on the system performance is depicted in Figures 13a and 13b. As the inlet temperature of the turbine increases, the temperature difference of the refrigerant in the gas heater cold side rises. Because of the amount of heat transfer is constant, it will reduce the mass flow of refrigerant in the cycle and as a result, the power consumption of the high pressure compressor and cooling effect in the evaporator decrease. On the other hand, by increasing this parameter, the produced power of high pressure turbine and the inverted Brayton cycle increases. All in all, these changes will lead to increasing the energy and exergy efficiencies. Also, reduction of the mass flow rate causes a lower exergy destruction and investment cost rates of SCO2 cycle components and as a result, the total cost rate of the system will reduce.

Figures 14a and 14b show the impacts on the system performance of the HP-turbine back pressure. As the turbine back pressure increases, due to the reduction of the pressure ratio, and consequently, decreasing the enthalpy
difference in HP-turbine, the produced power by the HP turbine will decrease while the produced power by the LP turbine will increase due to the increasing the pressure ratio. Also, the amount of heat exchanged in the heater reduces due to the decrease in the heater inlet temperature. On the other hand, the input power of the LP compressor will increase significantly. With increasing HP turbine back pressure, the pressure of primary input stream to the ejector increases and the velocity of the primary output stream from the nozzle section increases, which leads to an enhancement in the ratio of the secondary to primary mass stream rate in the ejector. Also, the quality of carbon dioxide leaving the ejector decreases, and as a result, the quality of inlet carbon dioxide to the evaporator will decrease causing an increase in the refrigerant enthalpy difference in the evaporator. This leads to an increase in the produced cooling effect by the evaporator. Considering all these changes, by increasing the HP turbine back pressure, the energy efficiency will increase and the exergy efficiency will drop slightly. As \( P_9 \) increases, the irreversibility and its associated costs and also investment cost rate increase for the S-CO\(_2\) cycle causing an increase in the total cost of the system.

**Figure 15a and 15b** depict the effects of the ejector inlet temperature on the system performance. As this parameter increases, the quality of the outlet carbon dioxide from the ejector increases, which leads to an increase in the quality of inlet carbon dioxide to the evaporator. Thus, the refrigerant enthalpy difference in the evaporator reduces, and consequently, the amount of the cooling effect decreases which causes a reduction in both thermal and exergy efficiencies of the system. The most important reason for the increase in the system total cost rate is increasing the exergy destruction and its associated costs in the S-CO\(_2\) cycle components.

**Figure 16** shows the effects of the turbine back pressure of the inverted Brayton cycle on the output power of the ejector refrigeration sub cycle. As can be seen from this figure, the best amount of pressure to expel carbon dioxide from the LP- turbine is 5300 kPa. If it is lower than 5300 kPa, the input power of the LP-compressor will increase and as a result, the total power will decrease. On the other hand, if it is more than 5300 kPa, it causes carbon dioxide to convert to compressed liquid. Therefore, the output pressure of the LP- turbine should not exceed 5300 kPa.
6 Conclusions

In the present work, the thermodynamic and exergoeconomic simulation of the proposed novel system was performed. The system is a combination of the gas turbine cycle, Brayton cycle, inverted Brayton cycle and the transcritical carbon dioxide refrigeration cycle with ejector-expansion. The most important results of the study can be summarized as:

- The highest exergy destruction rate occurs in the combustion chamber, heat recovery steam generator and gas turbine, respectively.
- By applying the inverted Brayton cycle, the total power of the ejector refrigeration cycle will be positive, and the best possible amount of back pressure of the inverted Brayton cycle turbine (LP turbine) will be 5300 kPa, which leads to a production power of 5 kW in the bottoming cycle.
- The energy and exergy efficiencies will improve by 48.9% and 54%, respectively, compared to the gas turbine cycle, and also efficiencies of this proposed system will increase by 49.0% and 57%, respectively, compared to the basic cycle.
- The total cost rate of the system and cost of produced electricity are calculated as 183 $/h and 52.84 $/MWh, respectively.
- Increasing the inlet temperature to the turbine and gas turbine will be beneficial for the CCHP system in terms of exergo-economics and thermodynamics.
- Increasing the inlet temperature to the ejector and air preheater outlet temperature causes a reduction in the system performance while results in a maximum value for the total cost rate.
- A rising in the air compressor pressure ratio causes a minimum value for the energy efficiency and a maximum value for the exergy efficiency while the total cost rate increases with this parameter.

Finally, it should be noted that, based-on the results, some parameters have opposite effects on the thermodynamic and economic performances of the system so that increasing one parameter improves the thermodynamic performance of the system, but increases the total cost rate. This trend is sometimes even seen in
In some cases, it may be inevitable to increase the costs to meet the expectations from the system. Moreover, changing some parameters may improve the thermodynamic and economic parameters, but it is not technically possible. In these cases, the designer can give priority to the technical issues.

Reference

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   https://doi.org/10.1016/j.apenergy.2012.11.075
   https://doi.org/10.1016/j.enconman.2018.05.089
   https://doi.org/10.1016/j.energy.2017.05.160
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    https://doi.org/10.1016/j.cjche.2017.11.006


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Mortaza yari has conducted a multitude of research studies focusing on energy, exergy, and environmental analysis of systems and optimization. As a distinguished Professor within the Mechanical Engineering Department at the University of Tabriz, his contributions to the field have been significant. He has been the recipient of numerous prestigious awards in recognition of his substantial scientific services.

Farzad Mohammadkhani has conducted extensive research in the fields of energy, exergy, environmental analysis of systems, and optimization. Serving as an Assistant Professor at the Engineering Faculty of Khoy, Urmia University, he has made notable contributions through the publication of numerous research papers in esteemed international journals within this domain.

Table 1. Comparison of performance characteristics of gas turbine cycle in the present study and reference [21, 30].

<table>
<thead>
<tr>
<th>System performance characteristics</th>
<th>References</th>
<th>Present work</th>
</tr>
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<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>_value 1</td>
<td>_value 2</td>
</tr>
<tr>
<td>--------------------------</td>
<td>---------</td>
<td>---------</td>
</tr>
<tr>
<td>Gas temperature at turbine outlet (K)</td>
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<td>820</td>
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<tr>
<td>Air temperature at compressor outlet (K)</td>
<td>596</td>
<td>593.9</td>
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<td>fuel to air ratio, λ</td>
<td>0.04341</td>
<td>0.04342</td>
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<tr>
<td>air preheater outlet temperature (K)</td>
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<td>724.2</td>
</tr>
<tr>
<td>Compressor power (kW)</td>
<td>1439</td>
<td>1404</td>
</tr>
<tr>
<td>Turbine power (kW)</td>
<td>2439</td>
<td>2404</td>
</tr>
<tr>
<td>energy efficiency (%)</td>
<td>32.6</td>
<td>31.5</td>
</tr>
<tr>
<td>exergy efficiency (%)</td>
<td>30.91</td>
<td>29.8</td>
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### Table 2. Results of system evaluation parameters

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<th>Parameter (unit)</th>
<th>value</th>
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<tr>
<td>Energy efficiency (%)</td>
<td>78.36</td>
</tr>
<tr>
<td>Exergy efficiency (%)</td>
<td>62.96</td>
</tr>
<tr>
<td>Refrigeration output (kW)</td>
<td>7.122</td>
</tr>
<tr>
<td>Heat recovered in the steam generator (kW)</td>
<td>1430</td>
</tr>
<tr>
<td>Inverted Brayton cycle output power (kW)</td>
<td>5</td>
</tr>
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</table>

### Table 3. Exergoeconomic analysis results of the components

<table>
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<tr>
<th>Components</th>
<th>$c_f$ ($/GJ$)</th>
<th>$c_p$ ($/GJ$)</th>
<th>$\hat{C}_D$ ($$/h$$)</th>
<th>$\hat{Z}$ ($$/h$$)</th>
<th>$\hat{Z} + \hat{C}_D$ ($$/h$$)</th>
<th>r (%)</th>
<th>f (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>LP-compressor</td>
<td>83.01</td>
<td>92.04</td>
<td>0.2</td>
<td>-</td>
<td>0.0093</td>
<td>0.21</td>
<td>10.87</td>
</tr>
<tr>
<td>Heater</td>
<td>62.91</td>
<td>314.9</td>
<td>1.72</td>
<td>-</td>
<td>0.1412</td>
<td>1.86</td>
<td>400.5</td>
</tr>
<tr>
<td>HP-compressor</td>
<td>81.23</td>
<td>93.15</td>
<td>0.815</td>
<td>-</td>
<td>0.049</td>
<td>0.86</td>
<td>14.67</td>
</tr>
<tr>
<td>LP- Turbine</td>
<td>62.91</td>
<td>83.01</td>
<td>0.306</td>
<td>-</td>
<td>0.575</td>
<td>0.88</td>
<td>31.95</td>
</tr>
<tr>
<td>HP- Turbine</td>
<td>62.91</td>
<td>81.23</td>
<td>0.684</td>
<td>-</td>
<td>0.943</td>
<td>1.63</td>
<td>29.12</td>
</tr>
<tr>
<td>Combustion chamber</td>
<td>10.25</td>
<td>13.55</td>
<td>44.43</td>
<td>-</td>
<td>0.192</td>
<td>44.62</td>
<td>32.19</td>
</tr>
<tr>
<td>Equipment</td>
<td>HRSG</td>
<td>Air Preheater</td>
<td>Air Compressor</td>
<td>Gas Turbine</td>
<td>Ejector</td>
<td>Gas Cooler</td>
<td>Evaporator</td>
</tr>
<tr>
<td>-------------------</td>
<td>------</td>
<td>---------------</td>
<td>----------------</td>
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<td>13.55</td>
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<td>7.63</td>
<td>8.3</td>
<td>3.503</td>
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<td>20.42</td>
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<td>-</td>
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<td>-</td>
<td>5.84</td>
<td>11.12</td>
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<td></td>
<td>65.69</td>
<td>67.07</td>
<td>0.74</td>
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<td></td>
<td>64.48</td>
<td>250</td>
<td>1.43</td>
<td>-</td>
<td>0.233</td>
<td>1.67</td>
<td>287</td>
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<td></td>
<td>70.17</td>
<td>113.9</td>
<td>0.045</td>
<td>-</td>
<td>0.007</td>
<td>0.052</td>
<td>62.32</td>
</tr>
<tr>
<td></td>
<td>13.55</td>
<td>26.92</td>
<td>1.44</td>
<td>-</td>
<td>0.018</td>
<td>1.46</td>
<td>98.67</td>
</tr>
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</table>
Fig. 1. Schematic diagram of the CCHP system integrating inverted Brayton cycle and ejector transcritical CO₂ driven by biogas combustion.
Fig. 2. T-s diagram of the CCHP subsystem with transcritical CO2.

Fig. 3. The Schematic of an ejector
Fig. 4. Solving methodology flowchart.

Fig. 5. Comparison of compressor power, turbine power, first law efficiency and cooling effect for the present work and Wang et al. [23] by increasing the ejector back pressure.
Fig. 6. Schematic diagram of Wang et al. [23]

Fig. 7a. Exergy destruction contribution of components of the topping cycle
Fig. 7b. Exergy destruction contribution of components of the refrigeration system

Fig. 8a Effects of air compressor pressure ratio on the energy and exergy efficiencies.

Fig. 8b Effects of air compressor pressure ratio on the heat generated in the HRSG and the total cost rate of the system.
**Fig. 9a** Effects of air preheater outlet temperature on the energy and the exergy efficiencies

**Fig. 9b** Effects of air preheater outlet temperature on heat generated in the HRSG and the total cost rate
**Fig. 10.** The effect of gas turbine inlet temperature on the energy and exergy efficiencies and the total cost rate of the system.

**Fig. 11a** Effects of gas heater outlet temperature on the energy and exergy efficiencies and the bottoming cycle produced power.
Fig. 11b. The effects of gas heater outlet temperature on heat generated in the HRSG and the total cost rate of the system.

Fig. 12a Effects of ejector back pressure on energy and exergy efficiencies.
Fig. 12b Effects of ejector back pressure on the refrigeration effect and the total cost rate.

Fig. 13a. Effects of high pressure turbine inlet temperature on the refrigeration effect and mass rate of carbon dioxide.
**Fig. 13b.** Effects of high pressure turbine inlet temperature on the energy efficiency and total cost rate of the system

**Fig. 14a.** Effects of HP-turbine back pressure on the energy and the exergy efficiencies
Fig. 14b Effects of HP-turbine back pressure on the refrigeration output in the evaporator and total cost rate of the system.

Fig. 15a Effects of the ejector inlet temperature on the energy and the exergy efficiencies.
Fig 15b Effects of the ejector inlet temperature on the refrigeration output in the evaporator and total cost rate of the system.

Fig. 16 Effects of the turbine back pressure of the inverted Brayton cycle (LP turbine) on the power produced of bottoming cycle

Appendix A

Table A1. Input parameters of the modeling [27, 29,30]

<table>
<thead>
<tr>
<th>Design parameters</th>
<th>Value</th>
</tr>
</thead>
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<tr>
<td>Component</td>
<td>Purchased equipment cost</td>
</tr>
<tr>
<td>-----------------------------------</td>
<td>------------------------------------------------</td>
</tr>
<tr>
<td>Air preheater</td>
<td>$pec_{APH} = 4122^*\left(\frac{\dot{m}<em>g \cdot \Delta h</em>{APH}}{u \cdot \Delta T_{lm,APH}}\right)^{0.6}$</td>
</tr>
<tr>
<td>Air compressor</td>
<td>$pec_{AC} = \left(\frac{71.10 \dot{m}<em>{air}}{0.9 - \eta</em>{is,c}}\right)\left(\frac{P_{out}}{P_{in}}\right)\ln\left(\frac{P_{out}}{P_{in}}\right)$</td>
</tr>
<tr>
<td>Heat recovery steam generator (HRSG)</td>
<td>$pec_{HREG} = 6570^<em>\left(\frac{\dot{Q}<em>{ec}}{\Delta T</em>{lm,ec}}\right)^{0.8} + \left(\frac{\dot{Q}<em>{ev}}{\Delta T</em>{lm,ev}}\right)^{0.8} + 21276^</em>\dot{m}_w + 1184.4^*\dot{m}_g^{1.2}$</td>
</tr>
<tr>
<td>Combustion chamber</td>
<td>$pec_{CC} = \frac{46.08^*\dot{m}_{air}}{0.995 - \frac{P_4}{P_3}} [1 + \exp(0.018T_4 - 26.4)]$</td>
</tr>
<tr>
<td>Gas turbine</td>
<td>$pec_{GT} = \frac{479.34^*\dot{m}<em>g}{0.92 - \eta</em>{GT}} \ln\left(\frac{P_4}{P_3}\right) [1 + \exp(0.036T_4 - 54.4)]$</td>
</tr>
<tr>
<td>Gas heater</td>
<td>$pec_{gasheater} = 1.3^*(190 + 310^*A_{HE})$</td>
</tr>
<tr>
<td>Evaporator</td>
<td>$pec_{evap} = 6000^*\left(\frac{A_{evap}}{100}\right)^{0.7}$</td>
</tr>
<tr>
<td>Gas cooler</td>
<td>$pec_{Gascooler} = 130^*\left(\frac{A_{Gascooler}}{0.093}\right)^{0.78}$</td>
</tr>
</tbody>
</table>

Table A2. Investment cost for the components [2, 29]

Table A3. Thermodynamic properties of the state points
<table>
<thead>
<tr>
<th>State no.</th>
<th>Working fluid</th>
<th>T (K)</th>
<th>P (kPa)</th>
<th>$m$ (kg/s)</th>
<th>$h$ (kmol/kJ)</th>
<th>$s$ (kmol/kJ)</th>
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<tbody>
<tr>
<td>1</td>
<td>Air</td>
<td>290</td>
<td>101.3</td>
<td>4.439</td>
<td>-4950</td>
<td>193.4</td>
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<tr>
<td>2</td>
<td>Air</td>
<td>593.9</td>
<td>1013</td>
<td>4.439</td>
<td>4110</td>
<td>195.6</td>
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<tr>
<td>3</td>
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<td>700</td>
<td>962.4</td>
<td>4.439</td>
<td>7402</td>
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<td>4</td>
<td>Combustion gases</td>
<td>1250</td>
<td>923.9</td>
<td>4.622</td>
<td>-1728</td>
<td>221.8</td>
</tr>
<tr>
<td>5</td>
<td>Combustion gases</td>
<td>820.2</td>
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<td>4.622</td>
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<td>6</td>
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<td>110.9</td>
<td>4.622</td>
<td>-19752</td>
<td>220.8</td>
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<tr>
<td>7</td>
<td>Combustion gases</td>
<td>704.8</td>
<td>110.9</td>
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<td>8</td>
<td>CO₂</td>
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<td>CO₂</td>
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