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Theoretical analysis of a novel combined cooling, heating, and power (CCHP) cycle

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ABSTRACT

This study presents a theoretical analysis of a new combined cooling, heating, and power cycle by the novel integration of an organic Rankine cycle (ORC), an ejector refrigeration cycle (ERC), and a heat pump cycle (HPC) for producing cooling output, heating output, and power output simultaneously. Three different working fluids—namely R113, isobutane, and R141b—have been used in power, refrigeration, and heating sub-cycles, respectively. Energetic and exergetic analyses of the proposed cycle have been conducted to demonstrate its efficiency. The thermal and exergy efficiencies are obtained as 71.08% and 38.3%, respectively. The exergy destruction rate of each component and the overall cycle have been calculated where it is shown that among all the components, the generator has a main contribution in the cycle inefficiency. Finally, the sensitivity analysis of the different key parameters on the performance of the proposed cycle has been investigated. It has been demonstrated that the proposed cycle performs well in high generator pressure and low evaporator outlet pressure, based on the first and second laws of thermodynamics.

Keywords: Organic Rankine Cycle (ORC); Ejector Refrigeration Fycle (ERC); Heat Pump Cycle (HPC); Combined Cooling, Heating and Power (CCHP) Cycle; Working Fluid.

1. Introduction

In recent years, many research studies have been conducted to develop combined cooling, heating, and power (CCHP) cycle applications. In the competitive market of today, the utilization of CCHP systems has been highlighted on account of their wide range of usability as well as profitability. According to the prime movers, there are many CCHP technologies for performing trigeneration cycles. These technologies are gas turbine, Stirling engine, fuel cell, internal combustion engines, and so on.

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There have been many interesting pioneering studies in the cogeneration areas [1–5]. In all the relevant literature, the main purpose is to produce cooling and power, simultaneously. But since we are also interested in the heating user, trigeneration systems have come in the spotlight. Needless to say, the thermal efficiency is also enhanced compared to the cogeneration systems. Many studies have been conducted to show the profitability of the CCHP cycles based on the classical laws of thermodynamics and the economic point of view. Habibzadeh et al. [6] performed a simulation of the combined power and ejector refrigeration system by using five different working fluids—R123, R141b, R245fa, R600a and R601a— based on the first and second laws of thermodynamics. They showed that R601a has the highest thermal efficiency and the lowest exergy destruction. They also presented optimum values for the key parameters of the cycle—for example, the inlet pressure of the pump and the evaporator temperature, among others. Wang and Yang [7] combined the biomass gasification sub-system, solar-evacuated collector, internal combustion engine, and the dual-source-powered mixed-effect absorption chiller in order to construct a novel hybrid CCHP system driven by biomass and solar energies. They showed that the biomass sub-system had a larger contribution to the total energy and exergy efficiencies compared to the solar sub-system. They also demonstrated that the carbon emission can be reduced by highlighting the solar sub-system part in the hybrid system.

Sun et al. [8] presented a new combined heating and power (CHP) cycle to reduce the heating energy consumption of the previous CHP systems by recovering the waste heat of the exhausted steam from a steam turbine. They also invented a new ejector heat exchanger to decrease the temperature of the return water based on the ejector refrigeration cycle (ERC). Li et al. [9] presented the optimization of the power generation unit (PGU) capacity and the operation strategy of the CCHP system for hotels, offices, and residential buildings for three sub-models. They showed that the local climate data is an influential factor in the optimization of the CCHP system. Javan et al. [10] proposed a new CCHP cycle, and simulated and optimized the proposed cycle based on the multi-objective optimization method in terms of exergy efficiency maximization and the total cost rate minimization. They suggested R11 as the most appropriate working fluid for their CCHP system in order to ensure cooling production profitability rather than power production profitability.

Recent research has been less focused on the application of the heat pump cycle (HPC) in the CCHP system as a heating sub-cycle. The main objective of this study is to introduce and highlight this concept in the CCHP system for thermal design purposes. The objectives of this paper are multi-fold and can be summarized as below:

- To propose a novel CCHP system
- To analyze the proposed system based on the first and second laws of thermodynamics
- To perform a parametric study of the different key elements on the performance of the proposed system

2. Cycle description

Figure 1 shows the schematic of the thermodynamic cycle for the basic CCHP cycle. This cycle is brought together by the novel integration of an organic Rankine cycle (ORC), an ERC, and an HPC to produce power output, cooling output, and heating output simultaneously. The cycle operation of this system is as follows: The saturated vapor leaves the vapor generator at Point 1 and enters the turbine by expanding the operated working fluid into the superheated state at Point 2. This superheated working fluid passes through Heat Exchanger 1 (HE1) by exchanging a specific heat with the ERC through an isobar process. In practical terms, there are large heat losses in this heat exchanger. This heat exchanger acts as the condenser of the ORC and the generator of the ERC simultaneously. The saturated liquid (Point 3) pumps back through Pump 1 and enters the vapor generator in a sub-cooled state. On the other hand, the required transferred heat for the ERC is obtained by cold and hot working fluid interaction in HE1. The high-pressure vapor from HE1 (primary fluid) enters the ejector and draws the low-pressure superheated vapor of the evaporator (secondary fluid) into the ejector. These two fluids are then mixed in the mixing chamber (Point 7) and they enter Heat Exchanger 2 (HE2), where the condensation process occurs by rejecting a specific amount of the HE2 heat into the HPC (Point 8). This heat exchanger also acts as a condenser for ERC and an evaporator for HPC. The liquid from HE2 itself is divided into two parts. One part goes through Expansion Valve 1 (EV1) at Point 10 and then enters the evaporator, which produces a cooling capacity ($\dot{Q}_e$). The rest of the liquid is pumped back into HE1 by the means of Pump 2, thus completing the ejector refrigeration sub-cycle. The saturated vapor at Point 13 is entered into the compressor and compressed into the superheated state at Point 14. This superheated working fluid is then entered into the condenser by producing the required heating capacity ($\dot{Q}_c$). The saturated liquid (Point 15) is then expanded into a two-phase flow state across the EV2, experiencing an isenthalpic process; then, it is entered into

HE2, thus completing the HPC operation.

3. Thermodynamic analyses

3.1. Thermodynamic assumptions

A suitable simulation code has been developed in the Engineering Equation Solver (EES) which has been constructed based on some thermodynamic assumptions. These thermodynamic assumptions are as follows:

- The reference pressure and temperature are 0.101 MPa and 280 K, respectively.
- The flow across the expansion valves is assumed to be isenthalpic.
- Kinetic energy at the inlet and outlet of all the components is neglected.
- The systems reach a steady state.
- There are no pressure drops through all the components and ducts, except in the heat exchangers. The efficiency of the heat exchangers is assumed to be 60%, which is reasonable in many practical researches.
- The isentropic efficiency of the turbine, pumps and compressor are assumed to be 90%, 95%, and 75%, respectively.
- Only the inlet and outlet states of the ejector have been analyzed. In other words, the ejector is treated as a black-box model.
- The flow inside the ejector is considered as one-dimensional and homogeneous.
- A constant mixing pressure assumption has been taken into the consideration in the mixing region of the ejector.
- The outlet states of the condenser and HE1 in ORC and HE2 in ERC are at saturated liquid state.
- The outlet states of the HE1 in ERC, vapor generator, and HE2 in HPC are assumed to be at the saturated vapor state.
- The outlet temperature of the evaporator has been superheated.
- The kinetic and potential exergy rates are neglected, since the system and its components are at rest relative to the environment [11].
- The rate of chemical exergy is neglected compared with the physical exergy rate [11].
- All outer surface of the system is at constant reference temperature. So, the rate of exergy losses is neglected [11].

Considering these assumptions, we also need some input parameters for the thermodynamic simulation of the proposed CCHP system, which are classified in Table 1.

3.2. Energy analysis

The general forms of the conservation equations for the energy analysis of the proposed cycle are as follows:

$$\sum \dot{m}_{in} - \sum \dot{m}_{out} = 0 \qquad (1)$$

$$\sum(\dot{m}h)_{in} - \sum(\dot{m}h)_{out} = \sum\dot{Q}_{out} - \sum\dot{Q}_{in} - \dot{W} \qquad (2)$$

Once we impose these equations on each component in the proposed cycle, the energy balance equations can be obtained which are tabulated in Table 2. For the analysis of each he mass entrainment ratio of the ejector is another essential parameter in the ERCs. This parameter is defined as the mass flow rate of the secondary flow ($\dot{m}_{sf}$) to the primary flow ($\dot{m}_{pf}$):

$$U = \dot{m}_{sf}/\dot{m}_{pf} \qquad (3)$$

in which both $\dot{m}_{sf}$ and $\dot{m}_{pf}$ are in kg/s.

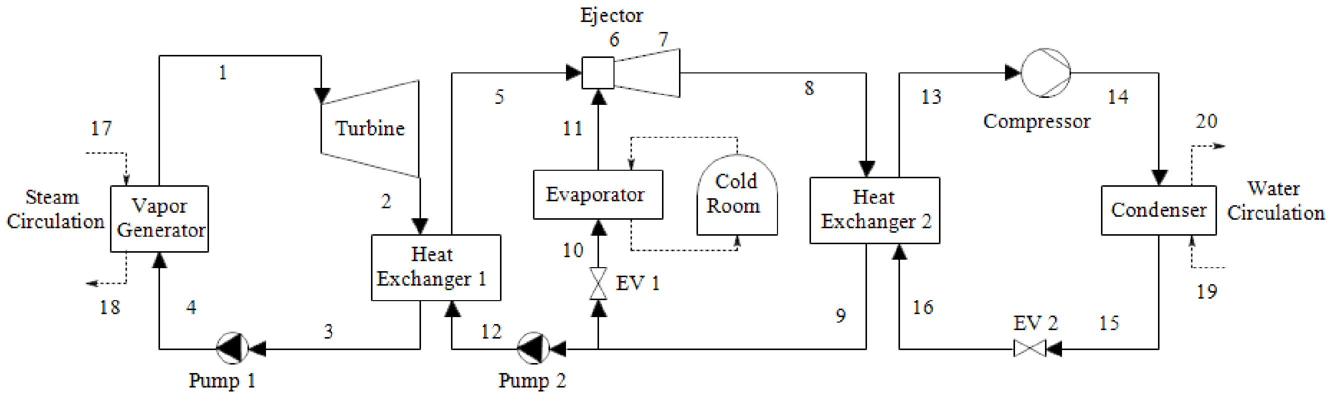

Fig. 1 Schematic of thermodynamic cycle of the proposed CCHP system

Table 1. Thermodynamic input parameters for the proposed CCHP system.

Input Parameter	Value
Generator pressure $\mathbf{P_g}$ $(\mathbf{MPa})$	2.5
Evaporator outlet pressure $\mathbf{P_e}$ $(\mathbf{MPa})$	0.22
Evaporator saturated temperature $\mathbf{T_{es}}$ $(\mathbf{K})$	288.5
Condenser temperature $\mathbf{T_c}$ $(\mathbf{K})$	333
Cold room temperature $\mathbf{T_{room}}$ $(\mathbf{K})$	277
Mass flow rate of vapor $\dot{\mathbf{m}}_{\mathbf{v}}$ $(\mathbf{kg.\,s^{-1}})$	0.258
Mass entrainment ratio **U**	0.32
Pinch point temperature difference($^{\circ}$**C**)	10

Table 2. Some balance equations for the energy analysis of the proposed CCHP system

Component	Equation	Component	Equation
Turbine power $(\dot{W}_t)$	$\dot{W}_t=\dot{m}_{ORC}(h_1-h_2)$	HE2 duty in HPC $(\dot{Q}_{HE2,HPC})$	$\dot{Q}_{HE2,HPC}=\dot{m}_{13}(h_{13}-h_{16})$
Pump 1 power $(\dot{W}_{p1})$	$\dot{W}_{p1}=\dot{m}_{ORC}(h_4-h_3)$	Generator duty $(\dot{Q}_g)$	$\dot{Q}_g=\dot{m}_{ORC}(h_1-h_4)$
Pump 2 power $(\dot{W}_{p2})$	$\dot{W}_{p2}=\dot{m}_{12}(h_{12}-h_9)$	Heating capacity $(\dot{Q}_c)$	$\dot{Q}_c=\dot{m}_{15}(h_{14}-h_{15})$
Compressor power $(\dot{\mathbf{W}}_{\mathbf{com}})$	$\dot{W}_{com}=\dot{m}_{13}(h_{14}-h_{13})$	Cooling capacity $(\dot{Q}_e)$	$\dot{Q}_e=\dot{m}_{10}(h_{11}-h_{10})$
Net produced power $(\dot{W}_{net})$	$\dot{W}_{net}=\dot{W}_t-\dot{W}_{p1}-\dot{W}_{p2}-\dot{W}_{com}$	COP of heating $(COP_{heating})$	$COP_{heating}=\dot{Q}_c/\dot{W}_{com}$
HE1 duty in ORC $(\dot{Q}_{HE1,ORC})$	$\dot{Q}_{HE1,ORC}=\dot{m}_{ORC}(h_2-h_3)$	COP of cooling $(COP_{cooling})$	$COP_{cooling}=\dot{Q}_e/(\dot{W}_{p2}+\dot{Q}_{HE1,ERC})$
HE1 duty in ERC $(\dot{Q}_{HE1,ERC})$	$\dot{Q}_{HE1,ERC}=\dot{m}_5(h_5-h_{12})$	Power efficiency (η_{power})	$\eta_{power}=(\dot{W}_t-\dot{W}_{p1})/\dot{Q}_g$
HE2 duty in ERC $(\dot{Q}_{HE2,ERC})$	$\dot{Q}_{HE2,ERC}=\dot{m}_8(h_8-h_9)$	Thermal efficiency (η_{th})	$\eta_{th}=(\dot{W}_{net}+\dot{Q}_e+\dot{Q}_c)/\dot{Q}_g$

3.3. Exergy analysis

The exergy of a system is defined as the maximum theoretical useful work that can be obtained as the system interacts with the equilibrium [11]. In the absence of the magnetic, electrical, nuclear, and surface tension effects, the rate of the total exergy of the system ($\dot{E}_{total}$) can be divided into four components: physical exergy rate ($\dot{E}_{PH}$), kinetic exergy rate ($\dot{E}_{KN}$), potential exergy rate ($\dot{E}_{PT}$), and chemical exergy rate ($\dot{E}_{CH}$) [11]:

$$\dot{E}_{total}=\dot{E}_{PH}+\dot{E}_{KN}+\dot{E}_{PT}+\dot{E}_{CH} \tag{4}$$

The rate of the physical exergy of a closed system for different working fluids can be calculated from the following equation:

$$\dot{E}_{PH}=\dot{m}\left(h-h_0-T_0(s-s_0)\right) \tag{5}$$

in which h and s are specific enthalpy and entropy of the substance, respectively, and h_0 and s_0 are those parameters at a reference state of known pressure and temperature of P_0 and T_0.

In an exergetic analysis, we need to introduce two useful concepts: fuel and product. According to their definitions, product represents the desired produced results and fuel represents the expended resources to generate the product. Both these concepts can be expressed in terms of exergy. Considering that, let us express the exergy rate balance for the element i of a system as [11]:

$$\dot{E}_F^i=\dot{E}_P^i+\dot{E}_D^i \tag{6}$$

in which $\dot{E}_P^i$ and $\dot{E}_F^i$ are the rate of the generated product and the supplied fuel of element i, respectively. On the other hand, $\dot{E}_D^i$ is the rate of exergy destruction of element i. The same equation for the total of a system can be written as:

$$\dot{E}_F^{total} = \dot{E}_P^{total} + \dot{E}_D^{total} \quad (7)$$

whereas the components are the corresponding ones in a system.

The exergetic efficiency of element i (η_{Ex}^i) is defined as the ratio of the product exergy of element i ($\dot{E}_P^i$) to the fuel exergy of the same element ($\dot{E}_F^i$):

$$\eta_{Ex}^i = \dot{E}_P^i / \dot{E}_F^i \quad (8)$$

Obviously, the higher the exergetic efficiency we have, the better will be the performance observed. The total exergetic efficiency of the system can be expressed as in Eq. (8):

$$\eta_{Ex}^{total} = \dot{E}_P^{total} / \dot{E}_F^{total} \quad (9)$$

in which $\dot{E}_P^{total}$ and $\dot{E}_F^{total}$ are the total exergy of the product and fuel rate, respectively.

Table 3 gives some of the exergy balance equations for the different components of the proposed cycle based on the aforementioned equations.

4. Results and discussion

Table 4 shows some properties of the utilized working fluids that are applied in this study. It is shown in the referenced works that there has been a suitable working fluid for each of the studied sub-cycles, which are considered here, too. We have used R113, isobutane, and R141b as the working fluids of ORC, ERC, and HPC, respectively.

Table 5 shows the obtained results of the energy analysis for the proposed novel CCHP system. The thermal efficiency of the CCHP system, the power efficiency, the COP of heating, and the COP of cooling have been obtained as 71.08%, 19.68%, 5.763, and 0.294, respectively. In order to ensure a more comprehensible analysis from the proposed cycle, the exergy analysis is also conducted, leading to the determination of the main source of irreversibility as well as the overall exergy destruction of the cycle. Table 6 gives some of the calculated key parameters of the exergy analysis—i.e. the exergy rate of fuel, the exergy rate of product, the overall exergy rate of destruction, and the exergy efficiency for the different components and the whole cycle. The overall exergy efficiency and exergy destruction for the proposed cycle is calculated to be 38.7% and 29.42 kW, respectively.

5. Parametric study

Figure 2 has been plotted to show the effect of the generator pressure on the thermal efficiency, exergy efficiency, and overall exergy destruction rate for the proposed CCHP system using R113 in the power sub-cycle, isobutane in the refrigeration sub-cycle, and R141b in the heating sub-cycle, respectively. Since an increase in the generator pressure increases the output power

Table 3. Some balance equations for the exergy analysis of the proposed CCHP system

Component	Exergy rate of product ($\dot{E}_P^i$)	Exergy rate of fuel ($\dot{E}_F^i$)	Exergy rate of destruction ($\dot{E}_D^i$)
Pump 1	$\dot{E}_P^{p1}=\dot{E}_4-\dot{E}_3$	$\dot{E}_F^{p1}=\dot{W}_{p1}$	$\dot{E}_D^{p1}=\dot{E}_F^{p1}-\dot{E}_P^{p1}$
Pump 2	$\dot{E}_P^{p2}=\dot{m}_{12}(e_{12}-e_9)$	$\dot{E}_F^{p2}=\dot{W}_{p2}$	$\dot{E}_D^{p2}=\dot{E}_F^{p2}-\dot{E}_P^{p2}$
Generator	$\dot{E}_P^{g}=\dot{E}_1-\dot{E}_4$	$\dot{E}_F^{g}=\dot{E}_{13}-\dot{E}_{14}$	$\dot{E}_D^{g}=\dot{E}_F^{g}-\dot{E}_P^{g}$
Evaporator	$\dot{E}_P^{e}=\dot{Q}_e(T_0/T_{room}-1)$	$\dot{E}_F^{e}=\dot{E}_{17}-\dot{E}_{18}$	$\dot{E}_D^{e}=\dot{E}_F^{e}-\dot{E}_P^{e}$
Ejector	$\dot{E}_P^{ej}=\dot{m}_{11}(e_8-e_{11})$	$\dot{E}_F^{ej}=\dot{m}_5(e_5-e_8)$	$\dot{E}_D^{ej}=\dot{E}_F^{ej}-\dot{E}_P^{ej}$
HE1	$\dot{E}_P^{HE1}=\dot{E}_5-\dot{E}_{12}$	$\dot{E}_F^{HE1}=\dot{E}_2-\dot{E}_3$	$\dot{E}_D^{HE1}=\dot{E}_F^{HE1}-\dot{E}_P^{HE1}$
HE2	$\dot{E}_P^{HE2}=\dot{E}_{13}-\dot{E}_{16}$	$\dot{E}_F^{HE2}=\dot{E}_8-\dot{E}_9$	$\dot{E}_D^{HE2}=\dot{E}_F^{HE2}-\dot{E}_P^{HE2}$
Turbine	$\dot{E}_P^{t}=\dot{W}_t$	$\dot{E}_F^{t}=\dot{E}_1-\dot{E}_2$	$\dot{E}_D^{t}=\dot{E}_F^{t}-\dot{E}_P^{t}$
Condenser	$\dot{E}_P^{c}=\dot{E}_{20}-\dot{E}_{19}$	$\dot{E}_F^{c}=\dot{E}_{14}-\dot{E}_{15}$	$\dot{E}_D^{c}=\dot{E}_F^{c}-\dot{E}_P^{c}$
Compressor	$\dot{E}_P^{com}=\dot{E}_{14}-\dot{E}_{13}$	$\dot{E}_F^{com}=\dot{W}_{com}$	$\dot{E}_D^{com}=\dot{E}_F^{com}-\dot{E}_P^{com}$
EV1	$\dot{E}_P^{EV1}=\dot{E}_{10}$	$\dot{E}_F^{EV1}=\dot{m}_{10}e_9$	$\dot{E}_D^{EV1}=\dot{E}_F^{EV1}-\dot{E}_P^{EV1}$
EV2	$\dot{E}_P^{EV2}=\dot{E}_{16}$	$\dot{E}_F^{EV2}=\dot{E}_{15}$	$\dot{E}_D^{EV2}=\dot{E}_F^{EV2}-\dot{E}_P^{EV2}$
Total system	$\dot{E}_P^{total}=\dot{E}_P^{t}-\dot{E}_P^{p1}-\dot{E}_P^{p2}-\dot{E}_P^{com}+\dot{E}_P^{e}+\dot{E}_P^{c}$	$\dot{E}_F^{total}=\dot{E}_F^{g}$	$\dot{E}_D^{total}=\sum_{i=1}^{n}\dot{E}_D^{i}$

Table 5. Calculated flow properties for the proposed CCHP system

Flow parameter	Value	Flow parameter	Value
Duty of generator $\dot{Q}_g(kW)$	102.7	Compressor power $\dot{W}_{com}(kW)$	8.027
Cooling capacity $\dot{Q}_e(kW)$	14.6	Net produced power $\dot{W}_{net}(kW)$	12.16
Heating capacity $\dot{Q}_c(kW)$	46.26	COP of heating $COP_{heating}$	5.763
Pump 1 power $\dot{W}_{p1}(kW)$	1.331	COP of cooling $COP_{cooling}$	0.294
Pump 2 power $\dot{W}_{p2}(kW)$	0.034	Power efficiency $\eta_{power}(\%)$	19.68
Turbine power $\dot{W}_t(kW)$	21.55	Thermal efficiency $\eta_{th}(\%)$	71.08

Table 6. Calculated exergy properties for the different components of the proposed CCHP system

Component	E_F^i (kW)	E_P^i (kW)	E_D^i (kW)	η_{ex}^i (%)	Component	E_F^i (kW)	E_P^i (kW)	E_D^i (kW)	η_{ex}^i (%)
Pump 1	1.331	1.153	0.177	86.63	Turbine	23.42	21.55	1.861	92.05
Pump 2	0.034	0.029	0.004	86.06	Compressor	8.027	6.373	1.654	79.4
Generator	44.46	32.24	12.22	72.52	HE1	9.976	4.747	5.229	47.58
Evaporator	0.164	0.158	0.006	96.34	HE2	4.447	1.671	2.775	37.59
Condenser	7.397	3.048	4.349	41.21	EV1	1.309	1.235	0.074	94.33
Ejector	1.407	0.988	0.419	70.21	EV2	1.15	0.5026	0.647	43.71
					Total system	44.46	17.21	29.42	38.7

of the turbine, the thermal efficiency of the system is increased by increasing the generator pressure. Interestingly enough, the heating output and the cooling output are decreased by increasing the generator pressure, but the order of these reductions are much lower compared to the increase in the output power. On the other hand, an increase in the generator pressure increases the maximum theoretical work of the whole cycle, which will result in an increase in the exergy efficiency too. So, this will cause the overall exergy destruction rate to increase as the generator pressure increases.

Figure 3 shows the effect of the evaporator outlet pressure on some key performance parameters. In order to improve the performance of the proposed system based on the first law of thermodynamics, it is sufficient to decrease the evaporator outlet pressure, which causes a reduction in the net produced power of the whole cycle. Needless to say, the cooling and heating outputs are constant and, hence, have no effect on the system performances when the variation of the evaporator outlet pressure is taken into consideration. Because of this behavior of the system, the maximum produced theoretical work is also decreased as the evaporator outlet pressure is increased and thus the irreversibility of the overall system is increased.

Figure 4 shows the variation of the thermal efficiency, the exergy efficiency, and the overall exergy destruction rate with the ejector mass entrainment ratio. An increase in the mass entrainment ratio increases the cooling capacity of the system considerably. This will increase the rejecting energy from the ERC into the HPC, causing an increase in the produced heating capacity in the heat pump sub-cycle too. However, the power output is almost constant through the variation of the ejector mass entrainment ratio. So, these overall variations will increase the thermal efficiency when the ejector mass entrainment ratio is increased. Obviously, throughout the variation of the ejector mass entrainment ratio the utilized and produced maximum theoretical works do not vary, thus resulting in a constant exergy efficiency and overall exergy destruction rate.

In order to increase the exergy efficiency of the proposed CCHP system, one can also decrease the heating capacity of the condenser in the proposed cycle (Fig. 5). This treatment does not affect the thermal efficiency of the system and, hence, can be an applicable treatment in the energy-saving industry.

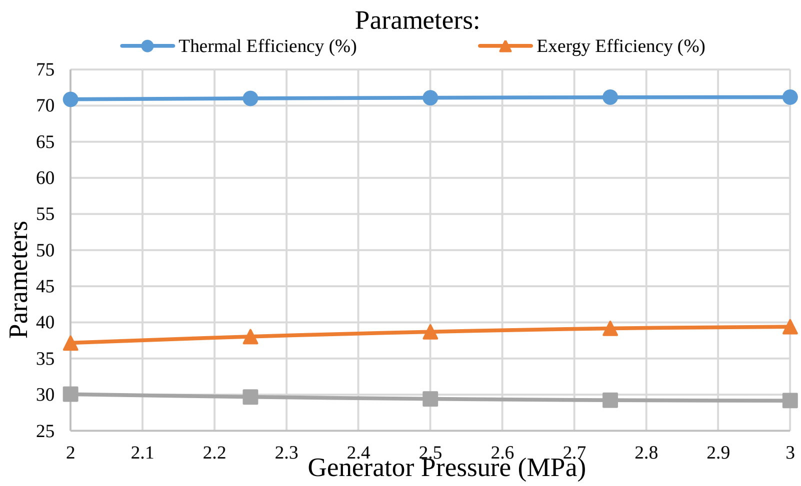

Fig. 2. Effect of the generator pressure on the thermal efficiency, exergy efficiency, and overall exergy destruction ratio

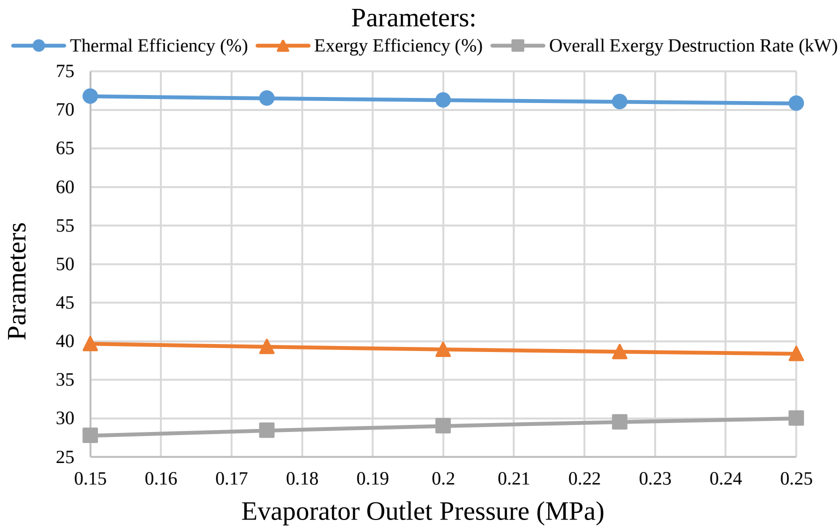

Fig. 3. Effect of the evaporator outlet pressure on the thermal efficiency, exergy efficiency, and overall exergy destruction ratio

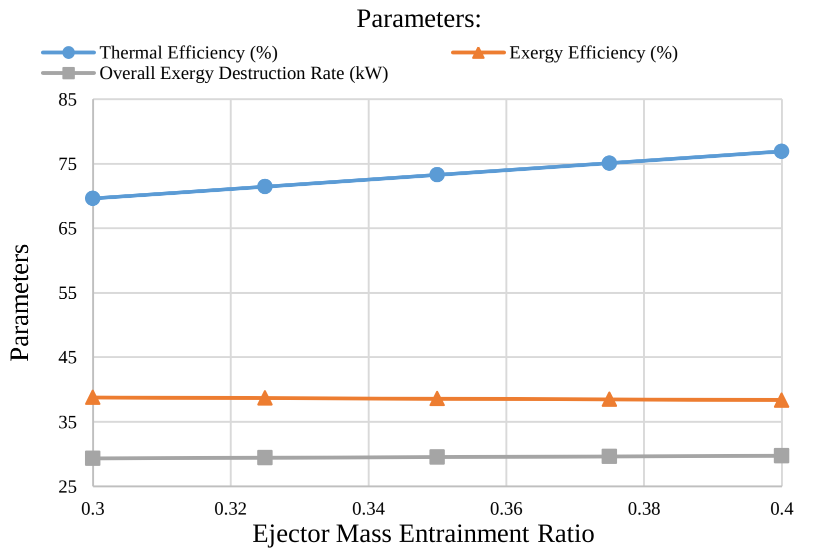

Fig. 4. Effect of the ejector mass entrainment ratio on the thermal efficiency , exergy efficiency, and overall exergy destruction ratio

The accountability of each operated component in the proposed novel CCHP system has been shown in Fig. 6. As the main purpose of this paper, it is necessary to determine the main source of irreversibility in the whole combined system. Upon doing this, one can take all the necessary steps to reduce the losses by different methods. In the proposed system, the generator represents the main source of losses, and is followed by the condenser. So, all attempts need to be taken in order to decrease the irreversibility of these components, which will result in the overall irreversibility reduction of the CCHP system.

6. Conclusion

A novel CCHP system has been presented and analyzed based on the first and second laws of thermodynamics. The proposed system was integrated from an ORC, an ERC, and an HPC to produce power output, cooling output, and power output simultaneously. The energetic and exergetic analyses of the proposed cycle were conducted to demonstrate its efficiency. The accountability of the irreversibility of each component was specified where it was shown that the generator and the condenser have the largest

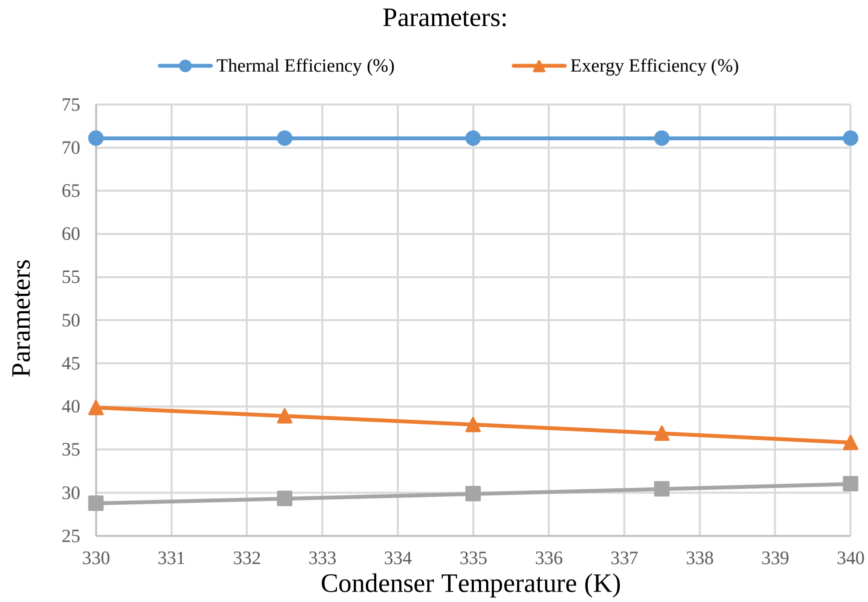

Fig. 5. Effect of the condenser temperature on thermal efficiency, exergy efficiency, and overall exergy destruction ratio

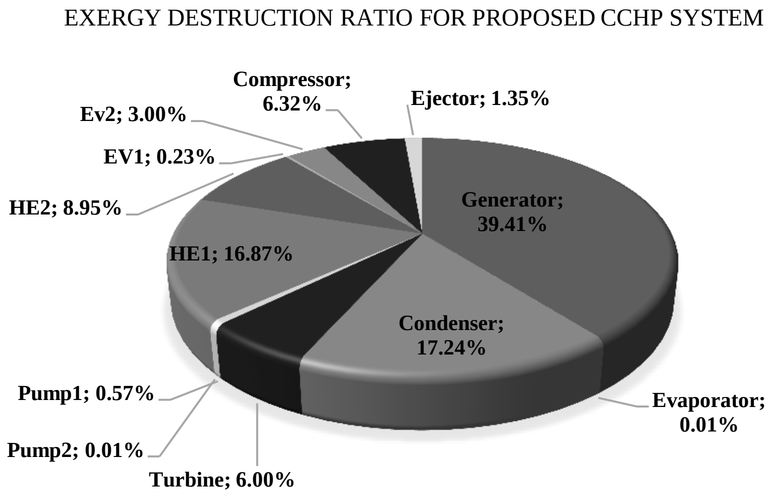

Fig. 6. Contribution of each component on the irreversibility of the proposed CCHP system

irreversibility of 39.41% and 17.24%, respectively. At the end, a parametric study of the different key elements on the performance of the proposed cycle was investigated. It was shown that the proposed cycle performs well in high generator pressure and low evaporator outlet pressure, based on the first and second laws of thermodynamics.

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