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Energy, exergy, economic and environmental (4E) analyses of a geothermal power plant with NCGs reinjection

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Abstract

In this study, a conceptual geothermal power plant with Non-Condensable Gases (NCGs) reinjection is analyzed from energy, exergy, exergo-economic, and environmental perspectives. The cycle is evaluated in considering both subcritical and supercritical working conditions. A sensitivity analysis comprising the variation of condensing temperature, turbine inlet pressure, and amount of NCGs within the geofluid is performed to assess the system performance from the thermodynamic, economic, and environmental points of view. The exergy efficiency results for the supercritical and the subcritical cycles are 52% and 50.5%, respectively. Results showed increasing the condensing temperature decreases exergy efficiency, net power, and exergetic performance parameters and increases total exergy destruction, exergo-destruction impact, and specific power cost in both working cases. According to the obtained results, an increase in the inlet pressure of the turbine in the supercritical case study reduces exergy efficiency and exergetic performance and increases the exergo-destruction impact factor. However, it increases the cycle exergy efficiency and exergetic performance criteria first and then decreases in a subcritical case. The LCOE for the supercritical cycle is 6.96 (*c€/kWh*) and for the subcritical is 5.52 (*c€/kWh*). The highest exergy destruction rates are related to the main heat exchanger, condenser, and turbine, respectively.

Keywords: Geothermal, Exergy, Exergo-economic, NCG, Reinjection

1. Introduction

In recent decades, the increasing world population and the higher energy demand caused an increase in fossil fuel consumption, leading to global warming, ozone layer depletion, and environmental pollution. Therefore, renewable energy utilization and the employment of more efficient technologies that provide lower environmental impacts are flourishing. Several alternative energies can be exploited, such as solar,

geothermal, wind, etc. Geothermal energy is influenced by the location, depth, temperature of the reservoir, geological features of the region, and site-specific technology. In addition, a geothermal power plant (GPP) is not affected by weather conditions. Geothermal fluid includes different quantities of Non-Condensable Gases (NCGs) that are mainly composed of CO_2 and other contaminants. These gases originate from degassing magma, seldom from organic sediment decomposition and metamorphic decarbonization [1]. The existence of NCGs in geothermal fluids mainly causes corrosion, calcite deposition, lower efficiency, and health, safety, and environmental risks [2].

Many research types have been carried out on geothermal from energy, economic and environmental aspects. Lazzaretto and Tsatsaronis [3] introduced a methodology for determining and measuring exergetic efficiencies and exergy related costs in thermal systems. Their methodology was according to the Specific Exergy Costing (SPECO) method. In this method, the component's fuel and product are defined by using a systematic record of all exergy additions to and removals from all the exergy streams of the system. They applied basic principles from business administration to calculate the costs. Coskun et al. [4] proposed a modified exergoeconomic model using exergy and cost accounting analyses for geothermal power plants. They evaluated the relations between the capital costs and the exergetic loss/destruction of the system components. Kecebas [5] investigated geothermal district heating systems from energy, exergy, economic and environmental viewpoints. He applied actual operation data to evaluate the system performance, first and second laws efficiencies, specific exergy index, exergetic improvement potential, and exergy losses. Tempesti and Fiaschi [6] carried out a thermo-economic analysis of a micro combined heat and power Organic Rankine Cycle (ORC) system that includes geothermal and solar energies as the heat source. They compared three different working fluids from the thermo-economic viewpoint. They also evaluated the proposed system's cost and determined the most proper working fluid in a thermo-economic assessment. Tan and Kecebas [7] conducted a thermodynamic and economic evaluation of geothermal district heating systems (GDHS) using exergy-based methods.

Zare [8] studied and compared three different ORC configurations for binary geothermal power plants from thermodynamic and economic aspects. Their study showed that ORC with an internal heat exchanger has the best thermodynamic performance than other configurations. Yari et al. [9] carried out the thermodynamics and exergo-economic evaluation of the trilateral power cycle and compared the ORC results with the Kalina cycle. Their results showed that trilateral Rankine cycle net power is higher than the other considered cycles. Zhao and Wang [10] conducted exergy and exergoeconomic analysis of a flash-binary geothermal power plant. They did a parametric evaluation to find the effects of critical thermodynamic parameters such as flash pressure, ORC turbine inlet pressure, and ORC turbine inlet temperature on the system performance. Fiaschi et al. [11] did exergy and exergoeconomic analysis of two case studies of geothermal power plants in Italy. They compared these cycles from an exergoeconomic

aspect. Their results showed R1233zd(E) as working fluid has the best exergoeconomic performance compared with others. Leveni et al. [12] conducted an energy and exergy analysis of ORC integrated with a LiBr/water absorption chiller from the geothermal reservoir of Torre Alfina in central Italy. They developed the thermodynamic model in Engineering Equation Solver (EES) software. They did a sensitivity analysis to study the plant operating range and the possibility of design adapting to the customers' requirements.

Manente et al. [13] analyzed and compared integrated flash-binary and two-phase binary layouts. Their two considered plant layouts integrate the water absorption and reinjection of H_2S and CO_2 . Talluri et al. [14] performed energy, exergy, and exergoeconomic analysis of thermoelectric energy storage with geothermal heat integration. They did a sensitivity analysis on the main system parameters' impacts on the produced power cost per electric energy unit. Ehyaei et al. [15] carried out exergy and exergoeconomic analysis of a combined geothermal cooling and power system. Their considered system was an ORC as an upstream cycle and the absorption chiller as the downstream cycle. Alirahmi et al. [16] carried out exergy and exergoeconomic analysis of a multi-generation system that gets its heat from geothermal and solar energies. They did multi-objective optimization of the system to improve the system performance. They also considered different working fluids to find the best one based on the system's working parameters. Samadi and Kazemi [17] studied a new combined ORC and geothermal cycle with a zeotropic working fluid from an exergoeconomic aspect. They did optimization of their proposed cycle in order to optimize the operating parameters. Their obtained results showed that rising Isobutane mole fraction in Isobutane/Isopentane mixture causes a reduction in return on investment as an economic factor.

By considering these conducted research on geothermal power plants, it is found that few studies have been performed on a geothermal power plant with NCG's reinjection. Most researchers have analyzed geothermal power plants with an assumption of zero NCGs amount in geofluid; however, the NCGs presence should be considered for a more accurate evaluation of the geothermal cycle. The presence of NCGs in geofluid is inevitable and can affect the geothermal power cycle performance. Reinjection of NCGs is crucial as their existence needs a turbine and condenser with more corrosion-resistant material and design.

In this research, a geothermal power plant with NCGs gas reinjection is evaluated in subcritical and supercritical working conditions from energy, exergy, and exergo-economic perspectives. This cycle's modeling is developed in MATLAB, and in order to calculate thermodynamic properties, REFPROP 9.1 [18] is used. The exergo-economic analysis is done according to the SPECO method [3]. As generally more than 90% of NCGs in a geothermal fluid are carbon dioxide; just carbon dioxide is considered in this study's thermodynamic evaluation. One of the main benefits of this power plant is reducing the environmental impact of the geothermal power cycle by reinjection of NCGs. The literature review showed a research gap on analyzing geothermal plants with the reinjection of NCGs that needs much more research and this study

considers a novel technology for NCGs reinjection. The main aim of this paper is not the total reinjection technology but the power plant configuration of a total reinjection. In addition, the performance of the plant is assessed, which requires two intercoolers and one pre-cooler and two post-coolers decline the needed power for compression of NCGs [19, 20]. Furthermore, the supercritical and subcritical working states are evaluated to find their effects on system performance from energy, exergy, economic, and environmental aspects. To the best of our knowledge, these working conditions have not been investigated before for a geothermal power cycle with total reinjection of NCGs.

2. System description

The schematic diagram of the considered cycle is illustrated in Figure 1. The selected working fluid for the ORC cycle is R1233zd(E), and the percentage of CO₂ in geofluid is considered 8% which is a typical value of Italian basins [19]. R1233zd(E) is an HFO (HFOs are organic compounds comprised of hydrogen, fluorine, and carbon) that is a new fluid with low environmental impact and zero Ozone Depletion Potential (ODP) and is a substitute for the R245fa. In this cycle, geothermal fluid comes from the production well (point 30) and gets its heat to the organic working fluid of the ORC cycle. This high enthalpy vapor drives the turbine and generates electricity. The ORC fluid flows into a regenerator to preheat the fluid before entering the main heat exchanger. Then, it enters the condenser, and it is cooled down to saturated liquid. After that, a pump increases the pressure, after passing through the regenerator, enters into the main heat exchanger to start a new cycle.

The mixture of NCGs and water from point 40 flows into a pre-cooler and exchanges heat with water, and its temperature reduces. Afterward, it enters into the first-stage compressor, and its pressure increases. This trend continues with two other compressors, intercoolers and post-cooler 1. Obtained heat from these post-coolers and intercoolers are used in the district heating system. The produced water for district heating usage is about 75 °C. Finally, compressed CO₂ reinjects into the reinjection well. Another exit stream from the evaporator contains steam condensate (point 31), which enters in a post-cooler 2 before entering the reinjection well. The stream is cooled until the maximum possible temperature before silica precipitation will take place. Lastly, two reinjected streams combines in the depth of 600 meters where both fluids have the same pressure value. The separated CO₂ could not be applied for other applications because it contains some contaminants. The temperature-entropy diagram of the system is shown in Figure 2.

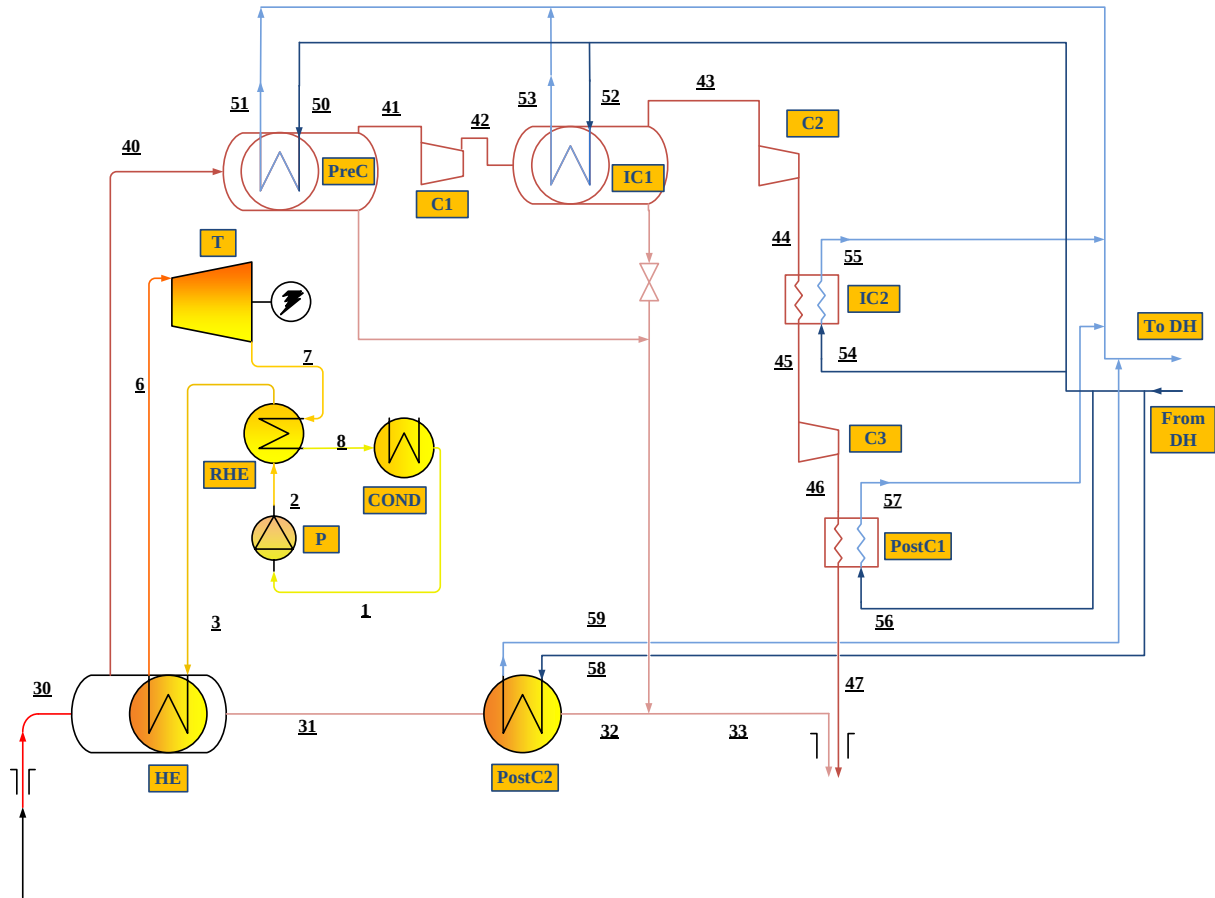


Fig. 1. Schematic diagram of considered geothermal power cycle

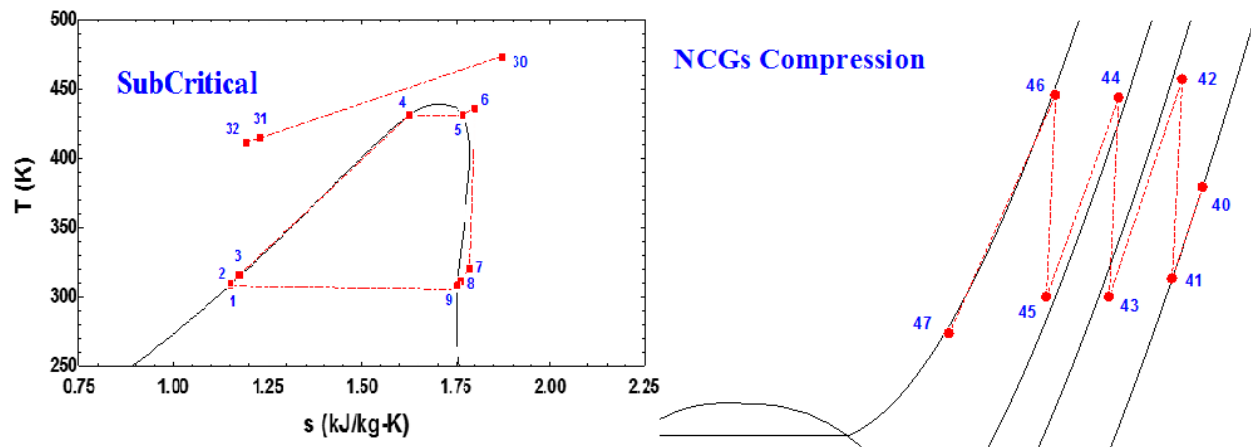


Fig. 2. Temperature-entropy diagram

3. Energy and exergy analysis

The following assumptions are considered in system modeling:

- The pump, turbine, and compressors have been assumed as adiabatic devices [8].
- The kinetic and potential energy exchanges are negligible [21].
- Steady-state condition is considered for each process in the cycles [21].
- The pressure losses in cycle components and piping are assumed negligible [21].
- The outlet fluid of the condenser is considered saturated liquid [22].
- The ambient temperature and pressure are supposed 15 °C and 1.013 kPa [19].
- The inlet pressure of the turbine in the supercritical condition is considered $1.06 \times P_{crit}$ [23].

In this study, both energy and exergy modeling has been done according to the thermodynamics' first and second laws. The operational parameters of the cycle are reported in Table 1. The properties of the ORC working fluid are presented in Table 2. The mass, and energy balance equations and exergy accounting are written in modeling this cycle as follows [24]:

$$\sum \dot{m}_i = \sum \dot{m}_e \quad (1)$$

$$\sum(\dot{m}h)_i + \dot{Q}_i + \dot{W}_i = \sum(\dot{m}h)_e + \dot{Q}_e + \dot{W}_e \quad (2)$$

$$\sum(\dot{m}.ex)_i + \dot{E}_{Q_i} + \dot{E}_{W_i} = \sum(\dot{m}.ex)_e + \dot{E}_{Q_e} + \dot{E}_{W_e} + \dot{E}_D \quad (3)$$

The specific exergy can be written as follows [24, 25]:

$$ex_{ph,k} = (h - h_0)_k - T_0(s - s_0)_k \quad (4)$$

In addition, exergy can be obtained from [24]:

$$\dot{E} = \dot{m}.ex \quad (5)$$

Thermal and exergy efficiency of the cycle could be written [21]:

$$\eta_{th} = \dot{W}_{net} / \dot{Q}_{in} \quad (6)$$

$$\eta_{ex} = \dot{W}_{net} / \dot{E}_{in} \quad (7)$$

Table 1

Input parameters in thermodynamic modeling

Parameter	Unit	Value
P_{30}	<i>kPa</i>	1000
T_{30}	°C	180
$\dot{m}_{30}$	<i>kg/s</i>	11.1
$P_{6,Subcritical}$	<i>kPa</i>	3130
$P_{6,Supercritical}$	<i>kPa</i>	3686.9
T_{32}	°C	89
η_p	%	85
η_T	%	88
η_C	%	82
x_1	-	0
P_0	<i>kPa</i>	101.3
T_0	°C	15

Table 2

Properties of the R1233zd(E)

Parameter	Unit	Value
Critical pressure	<i>kPa</i>	3572.6
Critical temperature	°C	165.5
GWP	-	1
ODP	-	Non-ozone depleting

Exergetic performance criteria (EPC) is useful for technical decisions and can be calculated as shown below [26]:

$$EPC = \dot{W}_{net} / \dot{E}_{D,Tot} \quad (8)$$

Relative exergy destruction defines as the ratio of exergy destruction of the kth component to total exergy destruction of the system and could be written as [27, 28]:

$$Y_{D,k} = \dot{E}x_{D,k} / \dot{E}x_{D,Tot} \quad (9)$$

3.1. Heat exchanger surface calculation

The heat exchanger area has a direct impact on its cost calculation. In this study, to the more accurate area and heat transfer coefficient calculation, the discretization method is selected for the main heat exchanger area calculation. In this method, the heat exchanger area divides into smaller sections, and the parameter is obtained for each section according to the energy balance equation. The area of each section could be written as follows:

$$A(j) = \dot{Q}(j) / [U(j) \times \Delta T_{LMTD}(j)] \quad (10)$$

In the above equation,

$$\Delta T_{LMTD} = \frac{(T_{hot,out} - T_{cold,in}) - (T_{hot,in} - T_{cold,out})}{\ln((T_{hot,out} - T_{cold,in}) / (T_{hot,in} - T_{cold,out}))} \quad (11)$$

Finally, the area can be calculated as follow:

$$A = \sum A(j) \quad (12)$$

At supercritical working conditions, to evaluate the heat exchangers and heat transfer coefficients, simple heat transfer correlations are not suitable. As the fluids do not have a stable state, then Pseudo-critical point properties should be applied. Karellas et al. [29] indicated that as the variation of thermodynamic properties such as thermal conductivity, dynamic viscosity, specific heat capacity at the Pseudo-critical state is rapid, Jackson correlation should be applied. Jackson and Hall introduced a correlation to counteract the effect of variations around Pseudo-critical point as below [30]:

$$Nu(j)_b = 0.0183 \times Re(j)_b^{0.82} Pr(j)^{0.5} \left(\frac{\rho_{wa}(j)}{\rho_b(j)} \right)^{0.3} \left(\frac{\bar{c}_p(j)}{c_{pb}(j)} \right)^n \quad (13)$$

In the above correlation, for we n have:

$$n = \begin{cases} 0.4 & \text{if } T_b < T_{wa} < T_{pc} \text{ and } T_{pc} < T_b < T_{wa} \\ 0.4 + 0.2(T_{wa}/T_{pc} - 1) & \text{if } T_b < T_{pc} < T_{wa} \\ 0.4 + 0.2(T_{wa}/T_{pc} - 1) \left(1 - 5(T_b/T_{pc} - 1) \right) & \text{if } T_{pc} < T_b < 1.2T_{pc} \end{cases} \quad (14)$$

In the above equations, T_b , T_{wa} , and T_{pc} are bulk, wall, and Pseudo-critical temperatures, respectively. In order to the calculation of condenser area below equation is used to obtain the Nusselt number [31]:

$$\begin{cases} Nu = 1.47 Re^{-1/3} & , \text{ if } Re \leq 30 \\ Nu = \frac{Re}{1.08 Re^{1.22} - 5.2} & , \text{ if } 30 \leq Re \leq 1800 \\ Nu = \frac{Re}{8750 + 58 Pr^{-0.5} (Re^{0.75} - 253)} & , \text{ if } Re \geq 1800, Pr \geq 1 \end{cases} \quad (15)$$

Also, for other heat exchangers Dittus-Boelter equation could be written [31]:

$$Nu = 0.23 \times Re^{0.8} Pr^n \quad (16)$$

In this equation, n for cooling is 0.3 and for heating is 0.4 [31]. The convective heat transfer coefficient (α) can be written [29]:

$$\alpha = Nu \lambda / d \quad (17)$$

The mean overall heat transfer coefficient is introduced as [29]:

$$\frac{1}{U} = \frac{1}{\alpha} + \frac{1}{\alpha_{hot}} + \frac{\delta}{\lambda} + R_f \quad (18)$$

4. Exergo-economic analysis

Several methodologies can be used for the exergoeconomic analysis of the cycle. In this research, a specific exergy costing method (SPECO) is chosen to carry out the exergoeconomic analysis [3]. Then for this

target, the cost balance equation based on input and output costs of each component can be written as follows [32-34]:

$$\sum \dot{C}_{e,k} + \dot{C}_{w,k} = \sum \dot{C}_{i,k} + \dot{C}_{q,k} + \dot{Z}_k \quad (19)$$

Here [35],

$$\dot{C} = c \dot{E}x \quad (20)$$

Where, $\dot{C}_{w,k}$ and $\dot{C}_{q,k}$ are the cost rates associated with the work and heat of the component, respectively. c is the cost per unit of exergy, $\dot{C}$ is the cost rate associated with the exergy streams, and $\dot{Z}_k$ denotes the total cost rate associated with the capital investment ($\dot{Z}_k^{CI}$) and operation and maintenance ($\dot{Z}_k^{OM}$) costs. Then, $\dot{Z}$ for the k th component can be expressed as [36]:

$$\dot{Z}_k = \dot{Z}_k^{CI} + \dot{Z}_k^{OM} = \frac{Z_k \times CRF \times \varphi}{N \times 3600} \quad (21)$$

Where N is annual plant operation hours, which is considered 7446 hours [37]. The capital recovery factor can be defined as below [37]:

$$CRF = \frac{i(1+i)^n}{(1+i)^n - 1} \quad (22)$$

Here, i is the interest rate considered 10%, and n is the useful life of the cycle that is supposed to be 30 years in this study. The purchasing equipment cost correlations are brought in Table 3. To exergoeconomic modeling, cost balance and auxiliary equations are written, which are presented in Table 4.

Table 3

Equations of equipment purchasing cost

Component	Cost correlation	Coefficients	Ref
Pump	$Z = \log(\dot{W}_p) + a(\dot{W}_p)^2 + b(\dot{W}_p) + c$	$a = -0.03195$ $b = 467.2$ $c = 2.048 \times 10^4$	[38]
Turbine	$\log C_0 = [K_1 + K_2(\log \dot{W}_T) + K_3(\log \dot{W}_T)^2]$ $Z = C_0 \times F_M$	$K_1 = 2.6259$ $K_2 = 1.4398$ $K_3 = -0.1776$	[39]
Heat exchangers	$Z = \log(A) + a(A)^2 + b(A) + c$	$a = -0.06395$ $b = 947.2$ $c = 227.9$	[38]
Post-C 1,2 & PC & IC 1,2	$Z = \log(A) + a(A)^2 + b(A) + c$	$a = 0.2581$ $b = 891.7$ $c = 2.605 \times 10^4$	[38]
Condenser	$Z = \log(A) + a(A)^2 + b(A) + c$	$a = 0.01764$ $b = 617.4$ $c = 3.31 \times 10^4$	[38]
Compressors	$Z = \log(\dot{W}_{Comp}) + a(\dot{W}_{Comp})^2 + b(\dot{W}_{Comp}) + c$	$a = 0.03867$ $b = 446.7$ $c = 1.378 \times 10^5$	[38]

In order to consider inflation, CEPCI (Chemical Engineering Plant Cost Index) is applied. This index is updated to the year 2020 [40]. Besides, the exchange rate of converting the US Dollar to the Euro is considered 0.9.

$$PEC_k = Z_k \times \left(\frac{CEPCI_{2020}}{CEPCI_{2001}} \right) \quad (23)$$

Well drilling cost (WDC) estimation is done according to the below correlation [41]:

$$WDC = 5.329 \times 10^5 n \cdot \log(z) + 0.2156 n \cdot z^2 + 9.655 \times 10^5 \quad (24)$$

Which z is the depth of well (m) and n is the number of production and injection wells. The specific power cost can be obtained as follows [42]:

$$c_{Tot} = PEC_{Tot}/\dot{W}_{net} \quad (25)$$

Table 4

Cost balance and auxiliary equations of each component

Component	Main equations	Auxiliary equations
Pump	$c_2\dot{E}x_2 = c_1\dot{E}x_1 + c_{W_p}\dot{W}_p + \dot{Z}_p$	$c_{W_p} = c_{W_t}$
RHE	$c_3\dot{E}x_3 + c_8\dot{E}x_8 = c_2\dot{E}x_2 + c_7\dot{E}x_7 + \dot{Z}_{HE}$	$c_7 = c_8$
HE	$c_6\dot{E}x_6 + c_{31}\dot{E}x_{31} + c_{40}\dot{E}x_{40} = c_3\dot{E}x_3 + c_{30}\dot{E}x_{30} + \dot{Z}_{HRSG}$	$c_{30} = c_{31}$ $c_{40} = c_{30}$
Turbine	$c_7\dot{E}x_7 + c_{W_t}\dot{W}_t = c_6\dot{E}x_6 + \dot{Z}_t$	$c_6 = c_7$
Condenser	$c_1\dot{E}x_1 + c_{21}\dot{E}x_{21} = c_8\dot{E}x_8 + c_{20}\dot{E}x_{20} + \dot{Z}_{cond}$	$c_{20} = 0$ $c_{21} = c_{20}$
Pre-cooler	$c_{41}\dot{E}x_{41} + c_{51}\dot{E}x_{51} = c_{40}\dot{E}x_{40} + c_{50}\dot{E}x_{50} + \dot{Z}_{PC1}$	$c_{50} = 0$ $c_{40} = c_{41}$
Compressor 1	$c_{42}\dot{E}x_{42} = c_{41}\dot{E}x_{41} + c_{W_{c1}}\dot{W}_{c1} + \dot{Z}_{c1}$	$c_{W_{c1}} = c_{W_t}$
Intercooler 1	$c_{43}\dot{E}x_{43} + c_{53}\dot{E}x_{53} = c_{42}\dot{E}x_{42} + c_{52}\dot{E}x_{52} + \dot{Z}_{IC1}$	$c_{52} = c_{50}$ $c_{43} = c_{42}$
Compressor 2	$c_{44}\dot{E}x_{44} = c_{43}\dot{E}x_{43} + c_{W_{c2}}\dot{W}_{c2} + \dot{Z}_{c2}$	$c_{W_{c2}} = c_{W_t}$
Intercooler 2	$c_{45}\dot{E}x_{45} + c_{55}\dot{E}x_{55} = c_{44}\dot{E}x_{44} + c_{54}\dot{E}x_{54} + \dot{Z}_{IC2}$	$c_{54} = c_{50}$ $c_{45} = c_{44}$
Compressor 3	$c_{46}\dot{E}x_{46} = c_{45}\dot{E}x_{45} + c_{W_{c3}}\dot{W}_{c3} + \dot{Z}_{c3}$	$c_{W_{c3}} = c_{W_t}$
Post-cooler 1	$c_{47}\dot{E}x_{47} + c_{57}\dot{E}x_{57} = c_{46}\dot{E}x_{46} + c_{56}\dot{E}x_{56} + \dot{Z}_{PC2}$	$c_{56} = 0$ $c_{47} = c_{46}$
Post-cooler 2	$c_{32}\dot{E}x_{32} + c_{59}\dot{E}x_{59} = c_{31}\dot{E}x_{31} + c_{58}\dot{E}x_{58} + \dot{Z}_{PC3}$	$c_{58} = 0$ $c_{32} = c_{31}$

Specific Investment Cost (SIC) is an economic indicator that presents the ratio of the total capital to net power of the cycle and can be defined as follows [43]:

$$SIC = F_s \times \frac{C_{TCL}}{\dot{W}_{net}} \quad (26)$$

Here, F_s is the correction factor of overhead expenses [43].

4.1. Return on investment

Return on investment (ROI) defines a ratio of net profit to the total investment cost which can be expressed as a percentage or ratio. ROI presents the amount of investment return comparative to the cost of investment [43].

$$ROI = \frac{(1-t_{corp})(S_{annual}-C_{TPC})}{C_{TCI}} \quad (27)$$

In the above correlation, t_{corp} is the corporate tax rate in Italy which is considered according to [43, 44]. C_{TPC} and C_{TCI} are the total production cost and the cost of total capital investment, respectively. Also S_{annual} is annual sales revenue can be written as below [43]:

$$S_{annual} = N \times \dot{W}_{net} \times C_{el} \quad (28)$$

C_{el} is average electricity prices for the industry in Italy (\$/kWh) [43, 45]. The equations of cost calculation of the power plant are presented in Table 5 [43].

Table 5

Total capital investment and production cost [43]

Cost type	Correlation
Total direct permanent investment	$C_{DPI} = TC_B + C_{site} + C_{serv} + C_{alloc}$
Total depreciable capital	$C_{TDC} = C_{DPI} + C_{cont}$
Cost of contingencies and contractor's fee	$C_{cont} = 0.18C_{DPI}$
Total permanent investment	$C_{TPI} = C_{TDC} + C_{land} + C_{royal} + C_{startup}$
Total capital investment	$C_{TCI} = C_{TPI} + C_{WC}$
Cost of wages and benefits	$C_{WB} = 0.035C_{TDC}$
Cost of salaries and benefits	$C_{SB} = 0.035C_{WB}$
Cost of materials and services	$C_{MS} = C_{WB}$
Cost of maintenance overhead	$C_{MO} = 0.05C_{WB}$
Direct manufacturing costs	$C_{DMC} = C_{WB} + C_{SB} + C_{MS} + C_{MO}$
Cost of property taxes and liability insurance	$C_{PI} = 0.02C_{TDC}$

Fixed manufacturing costs
 Total annual cost of manufacture
 General expenses
 Total production cost

$$\begin{aligned} C_{FIX} &= C_{PI} \\ C_{COM} &= C_{DMC} + C_{FIX} \\ C_{GE} &= 0 \\ C_{TPC} &= C_{COM} + C_{GE} \end{aligned}$$

To estimate the electricity generated cost in an energy system, the Levelized Cost of Energy (LCOE) can be used as a beneficial economic indicator. Low-cost electricity generation can lead to investors' profitability in a specific period; this is a proper explanation of a relatively low LCOE value [46]. The LCOE can be calculated as below [43]:

$$LCOE = \frac{C_{TCI} + \sum_{t=1}^n \left(\frac{C_{TPC}}{(1+i)^t} \right)}{\sum_{t=1}^n \left(\frac{G_t}{(1+i)^t} \right)} \quad (29)$$

In the above equation, G_t is generated electrical power in year t (kWh).

4.2. Pay-Back Period

The Pay-Back Period (PBP) is the amount of time required to recoup an investment. Generally, a shorter pay-back period means the investment's risk level based on the initial investment cost is for a shorter period. PBP formula can be written [43]:

$$PBP = \frac{C_{TDC}}{Cash\ flow} \quad (30)$$

The pay-back period is calculated for both with and without considering well drilling costs.

5. Environmental analysis

This cycle is compared with conventional fossil fuel power plants, which consume fuels and then cause greenhouse gas emissions. As geothermal power plant does not need fuel for generating electricity and gets its required energy from geothermal fluid, their advantages compared with conventional fossil fuel power plant could be explained by a reduction in fuel consumption and carbon dioxide emission in a year as follows [47]:

$$M_{CO_2} = \alpha_{CO_2} \times N \times \left(\frac{W_{net}}{1000} \right) \quad (31)$$

$$M_{Petro} = \alpha_{Petro} \times N \times \left(\frac{W_{net}}{1000} \right) \quad (32)$$

In the above equation, α_{Petro} is the average needed petroleum for generating 1 kWh of electricity which is considered 0.266 (*lit/kWh*) [10], and α_{CO_2} is the amount of released carbon dioxide from fossil fuels for producing 1 kWh that is considered 0.849 (*kg/kWh*) [10]. It has been assumed that with total reinjection of NCGs, the CO₂ emission of plant operation is close to zero.

There are controversial arguments among researchers about renewable energies from an environmental point of view. As analyzing a geothermal cycle from energetic, exergetic, and economic aspects is essential, evaluating it from the exergo-destruction side is essential. Then exergo-destruction indicators are formulated in order to evaluate the systems from exergy and environmental viewpoints. The exergo-destruction impact factor represents the environmental impacts of the irreversibilities of the system. The exergo-destruction impact factor can be calculated as below [48, 49]:

$$f_{ei} = \dot{E}x_{D_{Tot}} / \dot{E}x_{in} \quad (33)$$

The exergo-environmental impact index is an indicator that shows how much is a system environmentally damaging. It can be obtained from [48, 49]:

$$\theta_{ei} = 100 \times f_{ei} / \eta_{ex} \quad (34)$$

Another indicator is exergo-environmental impact improvement which represents the relation between the energy system and the environment. This indicator should be higher as much as possible, unlike the exergo-environmental impact index, and could be calculated from [48, 49]:

$$\theta_{eii} = 1 / \theta_{ei} \quad (35)$$

Also, the exergetic stability factor is defined as follows [48, 49]:

$$f_{es} = \frac{\dot{E}x_{out}}{\dot{E}x_{in} + \dot{E}x_{D_{Tot}} + \dot{E}x_{uu}} \quad (36)$$

Furthermore, the exergetic sustainability index could be written [48, 49]:

$$\theta_{est} = f_{es} \times \theta_{eii} \quad (37)$$

Which this parameter should be as high as possible for an effective system.

6. Results

In this research, a geothermal power plant with total reinjection of NCG is considered from energy, exergy, economic and environmental points of view. Modeling and analyzing this cycle has been done in both subcritical and supercritical working configurations and the obtained results have been compared. The conducted mathematical modeling of the present study has been compared with the presented value of ref. [50] to examine our modeling. Based on Table 6, results showed good compatibility between current work and reference values. The thermodynamic and exergy results are brought in Table 7. The main thermodynamic and economic parameters, yearly avoided fossil fuel consumption and CO₂ emission of both the subcritical and supercritical cases are presented in Table 8. The calculated LCOE for the supercritical cycle is 6.96 (*c€/kWh*) and for subcritical is 5.52 (*c€/kWh*), which is attractive for geothermal power plant compared with other renewable energies. The exergo-economic values of the cycle of both working conditions are presented in Table 9. The pay-back period is obtained 7.1 and 7.7 years for subcritical and supercritical cases respectively. The make-up wells have not been considered as they can differ from plant to plant and region to region.

Table 6

The validation of our model with ref. [50]

Parameter	Unit	Ref value [50]	Our model	Difference (%)
T_7	°C	72.5	72.4	-0.14
T_3	°C	57.62	57.59	-0.05
h_6	<i>kJ/kg</i>	536.3	532.8	-0.65
h_8	<i>kJ/kg</i>	383.8	380.2	-0.93
$\dot{m}_1$	<i>kJ/s</i>	44.06	44.02	-0.09
A_{Cond}	<i>m</i> ²	1699	1685	-0.82
η_{th}	%	20.57	20.12	-2.18
η_{ex}	%	63.72	62.81	-1.42
$\dot{W}_{net}$	<i>kW</i>	2481	2422	-2.37
$\dot{C}_{Tot}$	<i>M\$/year</i>	1.78	1.76	-1.12

Table 7

Thermodynamic results of each component

Component	Heat (kW)		Work (kW)		Exergy efficiency (%)	
	SubC	SuperC	SubC	SuperC	SubC	SuperC
Pump	-	-	321.2	377.8	86.0	86.2
RHE	902.8	1005.4	-	-	90.1	89.8
HE	26476	26476	-	-	73.6	75.4
Turbine	-	-	5074.4	5272	88.9	89.5
Condenser	21723	21582	-	-	26.1	26.3
Pre-cooler	20.8	20.8	-	-	79.3	79.3
Compressor 1	-	-	41.8	41.8	86.6	86.6
Intercooler 1	51.5	51.6	-	-	69.1	69.2
Compressor 2	-	-	39.9	39.9	86.5	86.6
Intercooler 2	50.7	50.7	-	-	71.1	71.1
Compressor 3	-	-	37.5	37.6	86.5	86.5
Post-cooler 1	73.2	73.2	-	-	76.6	76.7
Post-cooler 2	42.9	42.9	-	-	68.7	68.7

Table 8

Main thermodynamic parameters of the cycle in both cases

Case	η_{th} (%)	η_{ex} (%)	EPC	SIC	M_{CO_2} (kt)	M_{Petro} (m^3)	$\dot{Z}_{k,Tot}$ ($\text{€}/h$)	$\dot{C}_{Des,Tot}$ ($\text{€}/h$)
SubC	17.42	50.50	1.1017	1.8686	29.2	9178.3	132.9	93.4
SuperC	17.95	52.02	1.1736	1.8465	30.1	9456.5	172.8	96.5

Table 9

Exergoeconomic values of the cycle

Component	$\dot{Z}_k$ ($\text{€}/h$)		$\dot{C}_{D,k}$ ($\text{€}/h$)		$\dot{Z}_k + \dot{C}_{D,k}$ ($\text{€}/h$)		f_k (%)	
	SubC	SuperC	SubC	SuperC	SubC	SuperC	SubC	SuperC
Pump	5.83	8.89	2.06	2.70	7.89	11.59	73.8	76.7
RHE	3.71	4.86	0.22	0.28	3.93	5.14	94.4	94.5
HE	19.40	25.92	29.5	27.51	48.9	53.15	39.7	48.5
Turbine	72.97	93.67	17.71	19.47	90.68	113.1	80.5	82.8

Condenser	5.00	6.37	42.58	44.99	47.58	51.36	10.6	12.4
Pre-C	3.34	4.28	0.007	0.007	3.35	4.29	99.7	99.8
C1	3.49	4.46	0.25	0.28	3.74	4.74	93.2	93.9
IC1	3.02	3.88	0.14	0.17	3.16	4.05	95.3	95.7
C2	3.35	4.25	0.25	0.27	3.60	4.52	93.1	93.9
IC2	3.20	4.06	0.19	0.23	3.39	4.29	94.3	94.6
C3	3.13	3.99	0.23	0.26	3.36	4.25	93.1	93.9
Post-C 1	3.42	4.36	0.25	0.31	3.67	4.67	93.1	93.4
Post-C 2	2.98	3.85	0.03	0.03	3.01	3.88	98.9	99.1

Figure 3 shows the exergy efficiency and destruction variations with condensation temperature for subcritical and supercritical conditions. According to the results, increasing the condensation temperature causes a decrease in exergy efficiency and an increase in total exergy destruction in both subcritical and supercritical working states. Rising condensation temperature causes an increase in mean heat transfer temperature difference in the condenser, resulting in an increase in the irreversibilities and a reduction in efficiency. Figure 4 illustrates the change in net power and exergo-destruction impact factor with condensation temperature for subcritical and supercritical conditions. It is clear from this graph that rising the condensation temperature decreases the net power of the cycle, which is because of more exergy destruction and increases the exergo-destruction impact factor in both working situations, which is compatible with results presented in [8]. The higher value of the exergo-destruction impact factor demonstrates the system has more environmental damages and directly relates to the system's exergy destruction. As a cycle with supercritical working conditions has lower exergy destruction, then the exergo-destruction impact factor in the supercritical working state is lower than subcritical. Figure 5 represents the change in EPC and specific power cost against condensation temperature for subcritical and supercritical conditions. With an increase of condensation temperature, it can be observed in lessening the EPC in both working conditions, which is because of an increase in irreversibility and exergy destruction with condensation temperature enhancement and a decrease in net power. Besides, specific power cost rises with rising the condensation temperature due to a decline in net power.

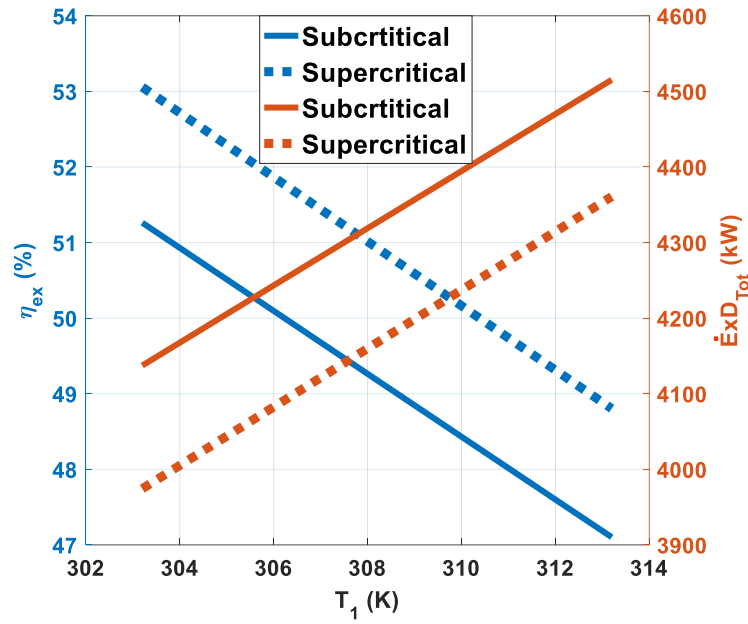


Fig. 3. Variations of exergy efficiency and total exergy destruction with condensation temperature

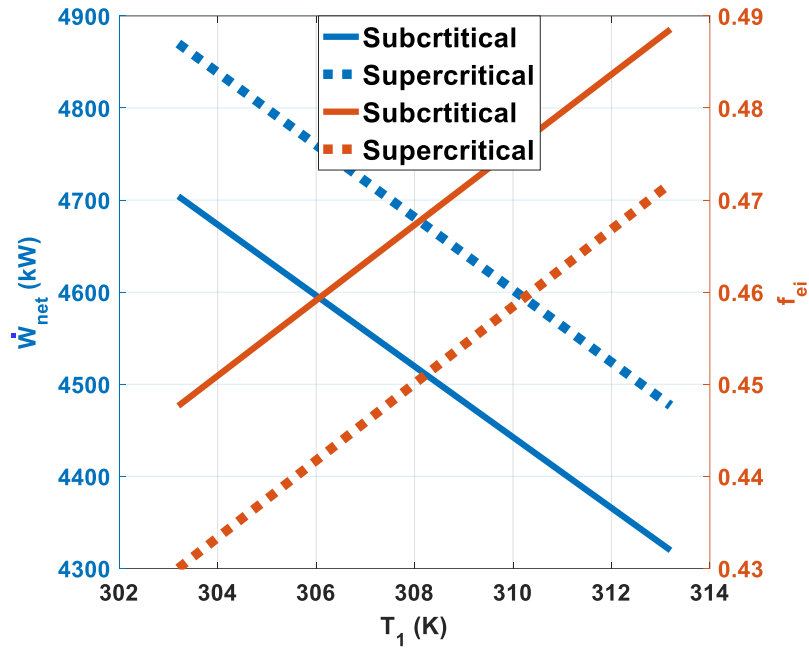


Fig. 4. Variations of net power and exergo-destruction impact factor with condensation temperature

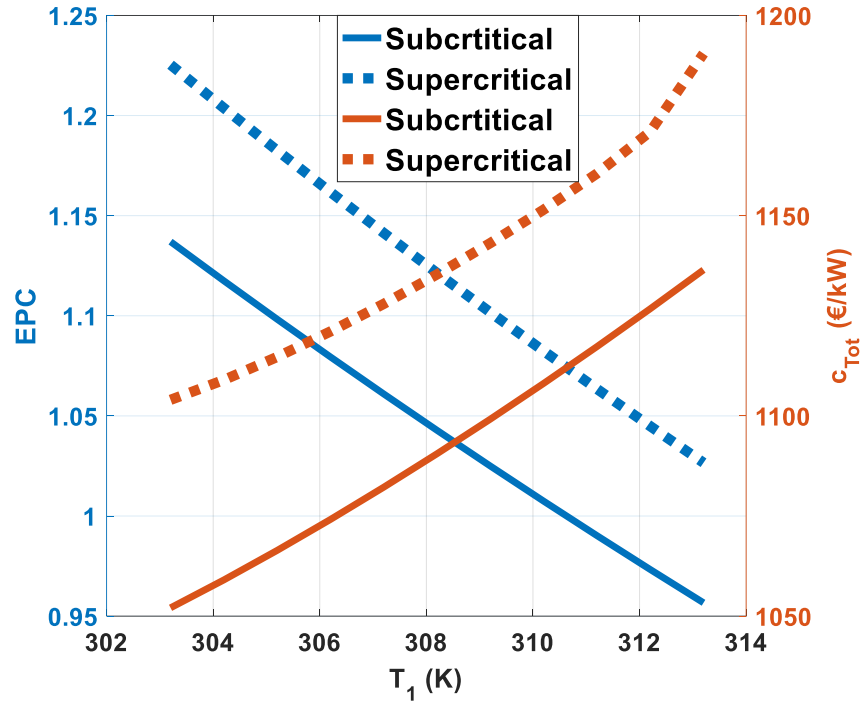


Fig. 5. Variations of EPC and specific power cost with condensation temperature

Variations of exergy efficiency and total exergy destruction of the cycle with an inlet pressure of the turbine for the subcritical state are shown in Fig. 6. With increasing the turbine's inlet pressure, cycle exergy efficiency increases first, and then at a pressure higher than 3030 kPa decreases. The exergy efficiency reaches its highest value at 3030 kPa, which is about 50.5%. This trend is converse for exergy destruction; it increases up to pressure of 3030 kPa and then reduces. Variations of the pay-back period and exergo-destruction impact factor with an inlet pressure of the turbine for the subcritical state are presented in Fig. 7. It can be seen that with increasing the inlet pressure of the turbine, the pay-back period reduces first (up to a pressure of 2700 kPa) and then increases at higher pressures. The pay-back period pattern is similar to the specific power cost trend as it was expected. The exergo-destruction impact factor variation trend decreases firstly, as it is expected according to the system's exergy destruction, and after reaching the lowest value at 3030 kPa, it rises. This is because of exergy destruction variations, as the irreversibilities have the minimum value, then exergo-destruction impact factor minimized in this point, and with destruction increment, it rises too. This pattern of irreversibilities variations is mainly because the condenser and main heat exchanger exergy destructions change with turbine pressure increase as they have the most prominent destruction of the system. In both equipment, the destruction firstly decreases and then rises with pressure enhancement.

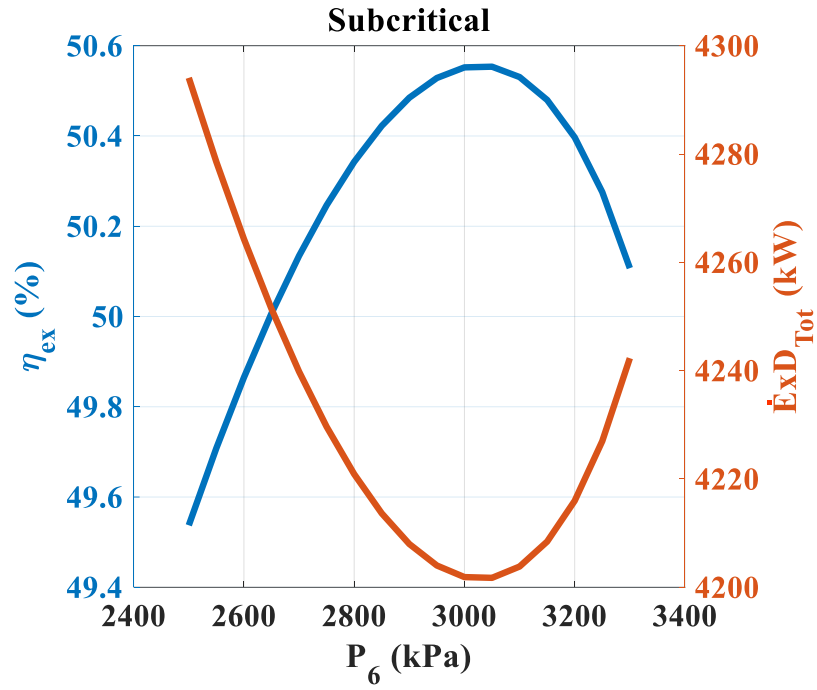


Fig. 6. Variations of exergy efficiency and total exergy destruction with inlet pressure of turbine for subcritical condition

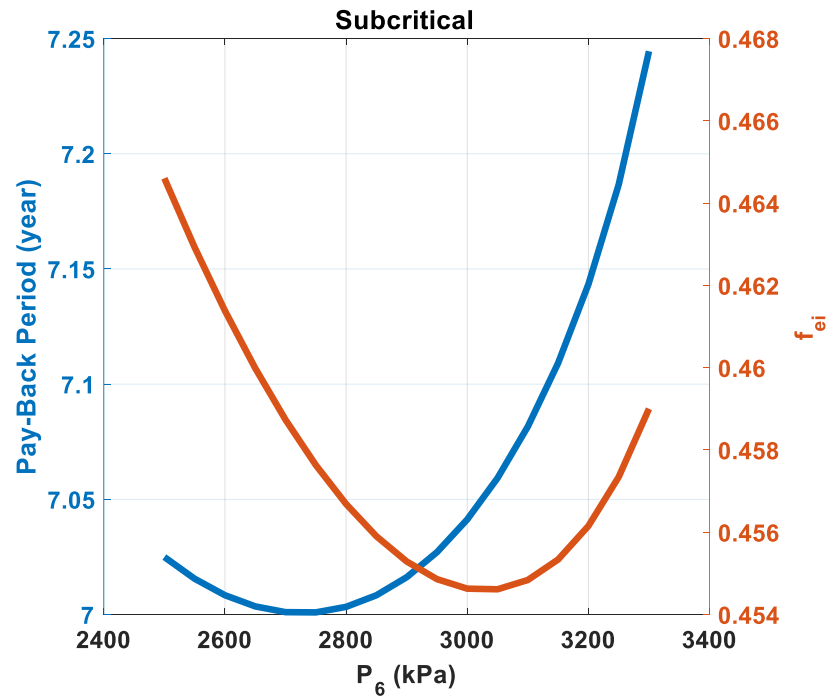


Fig. 7. Variations of pay-back period and exergo-destruction impact factor with inlet pressure of turbine for subcritical condition

Figure 8 implies variations of EPC and specific power cost with an inlet pressure of the turbine's subcritical working state. It can be found that rising the inlet pressure of the turbine causes an increase in exergetic performance criteria. After reaching the highest value at the pressure of 3030 kPa, it decreases. It is reasonable according to the exergy destruction trend against turbine inlet pressure, as they have the opposite trend. This trend for specific power cost is reversed so that it reduces firstly, and after reaching a minimum value at the pressure of 2750 kPa, it increases with more slope at higher pressures. It can be found from the results that increasing the specific power cost results in growth in pay-back time.

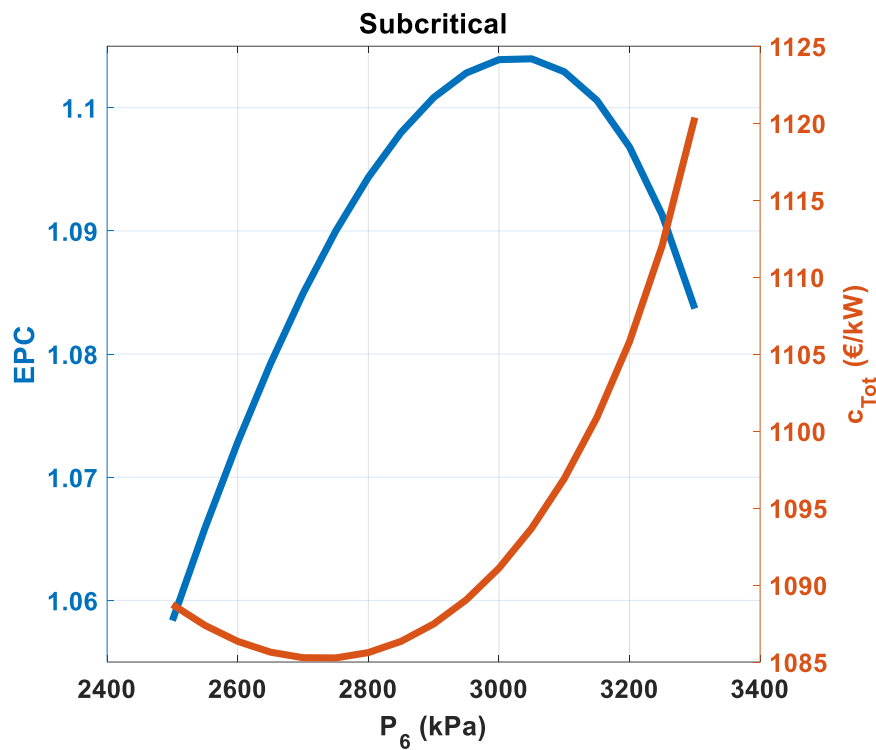


Fig. 8. Variations of EPC and specific power cost with inlet pressure of turbine for subcritical condition

Figure 9 illustrates the change in exergy efficiency and total exergy destruction of the cycle against the turbine's inlet pressure for supercritical working conditions. Results show that rising the inlet pressure of the turbine reduces the exergy efficiency and increases exergy destruction. Pressure rising leads to increases the heat absorption of organic working substances and decreases the proportion of constant temperature evaporating process to the whole heating process. As shown in figure 10, growth in inlet pressure of turbine in supercritical state causes an increase in both pay-back period and exergo-destruction impact factor. From figure 11, it is evident that growth in turbine inlet pressure results in a reduction in the system's exergetic performance. As the operating system in supercritical condition needs equipment, components with higher

capacity, and different working conditions, the specific power cost increases with turbine inlet pressure enhancement.

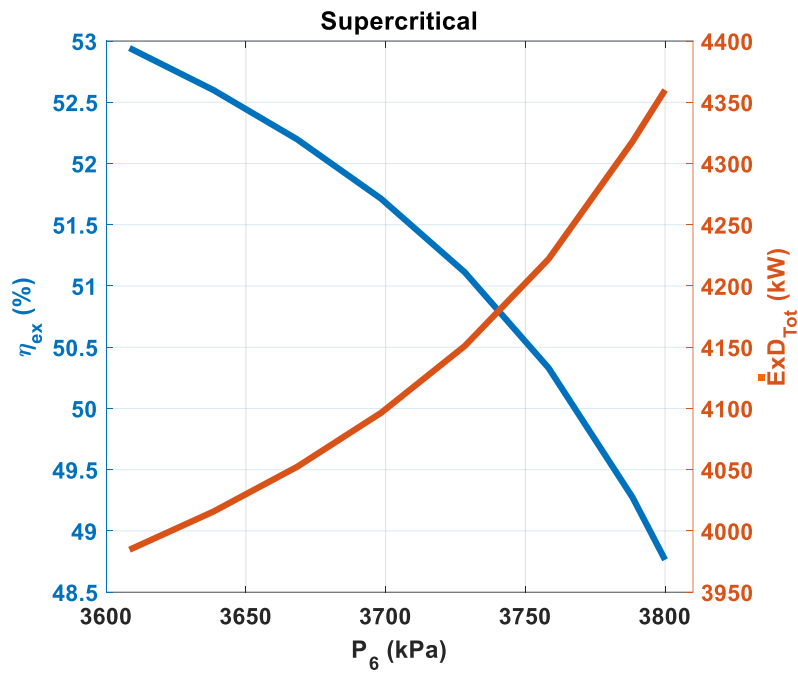


Fig. 9. Variations of exergy efficiency and total exergy destruction with inlet pressure of turbine for supercritical condition

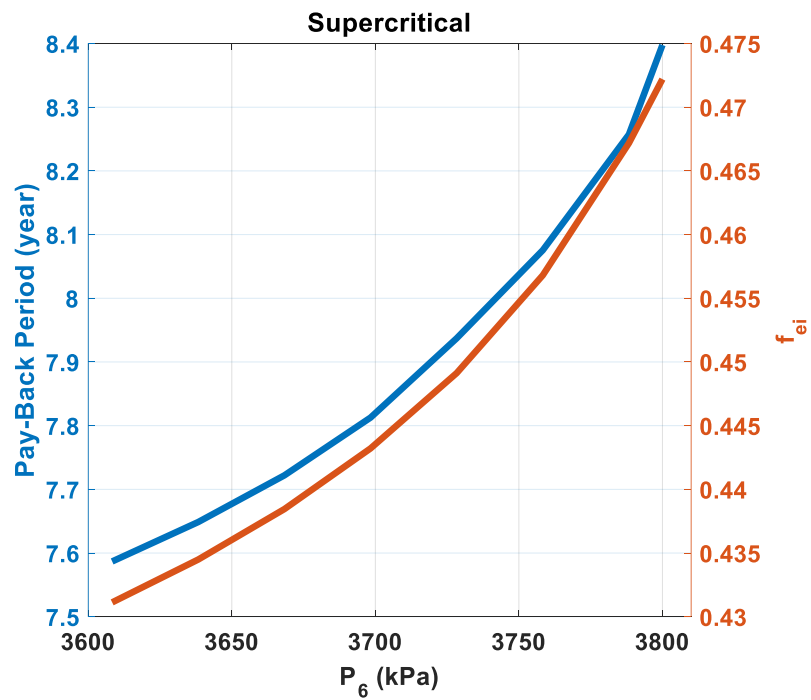


Fig. 10. Variations of pay-back period and exergo-destruction impact factor with inlet pressure of turbine for supercritical condition

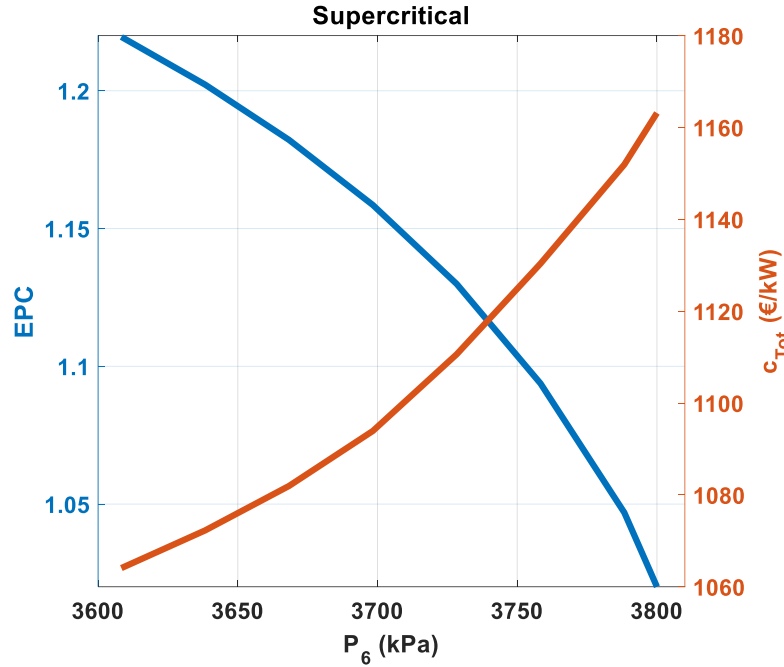


Fig. 11. Variations of EPC and specific power cost with inlet pressure of turbine for supercritical condition

Variations of exergy efficiency and specific power cost with NCGs percentage for both working conditions are presented in figure 12. Based on the results, increasing the NCGs' amount reduces the system's exergy efficiency as it can raise the system irreversibilities and raises the power cost in both subcritical and supercritical conditions. Power cost growth in the subcritical state is with a higher slope compared with supercritical. As observed from figure 13, by raising the NCGs' amount, generated net power of the cycle decreases, which is compatible with the analysis done by Gokcen and Yildirim [51]. The presence of NCGs results in growth in compressors' work and then harms net power. NCGs comprise lower recoverable specific energy than steam. The gases dilute the geothermal steam and lessen the gross turbine generation in the geothermal power plant. However, the exergo-destruction impact factor increases in both working situations. The effect of NCGs' amount variation on specific investment costs and the pay-back period has been drawn in Fig. 14. The increase in specific investment costs is due to decreased net power and increased total capital investment. As is expected, the investment cost and investment return increase by raising the percentage of NCGs in geofluid. NCGs should be separated from the brine as they degrade the condenser's heat transfer efficiency. The main effect of this is to enhance the condenser operating pressure, which

diminishes turbine potential. Consequently, overcoming this effect needs bigger condensers with higher heat transfer area and consequently with higher costs. Results show that the existence of NCGs in geofluid harms system performance, reduces the system's productivity, and increases system expenses. It causes a reduction in generated turbine work and an increase in exergy destruction of the system. The exergy destruction ratio of each component is illustrated in figure 15. This bar graph shows that maximum exergy destruction is related to the main heat exchanger (about 59%), condenser, and turbine, respectively, in both subcritical and supercritical cases. Results show that the destruction of the compression part is too much lower than the ORC part. The heat exchanged in the main heat exchanger is shown in figure 16.

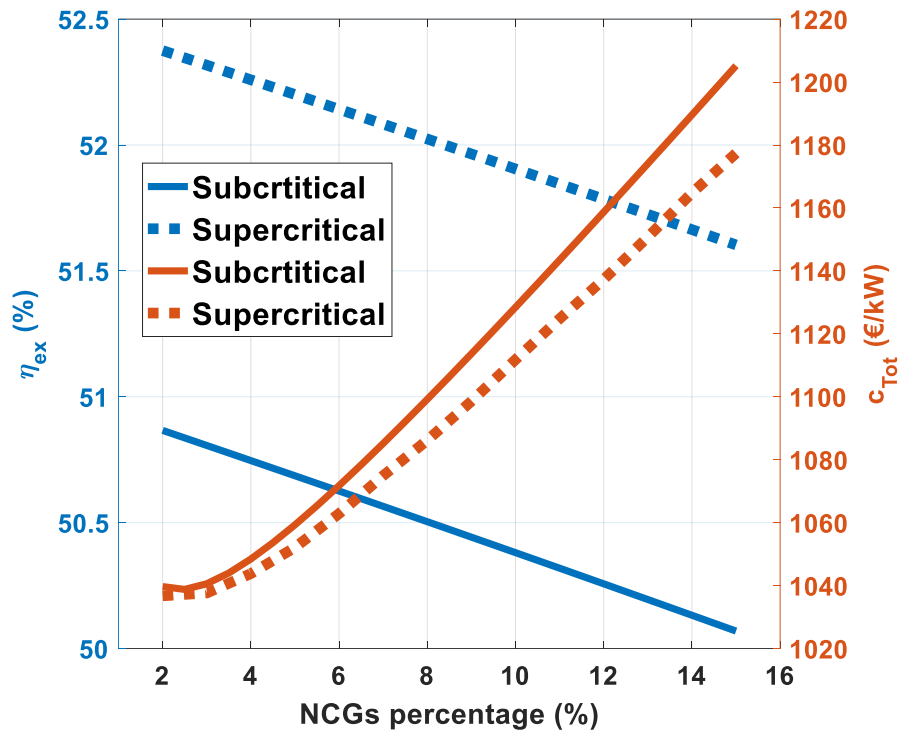


Fig. 12. Variations of exergy efficiency and specific power cost with NCGs percentage

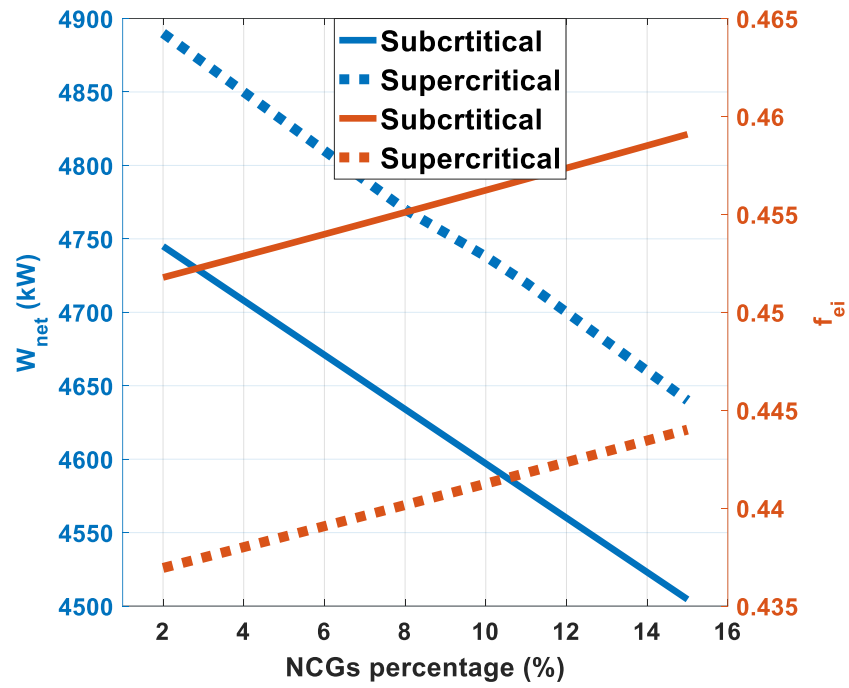


Fig. 13. Variations of net power and exergo-destruction impact factor with NCGs percentage

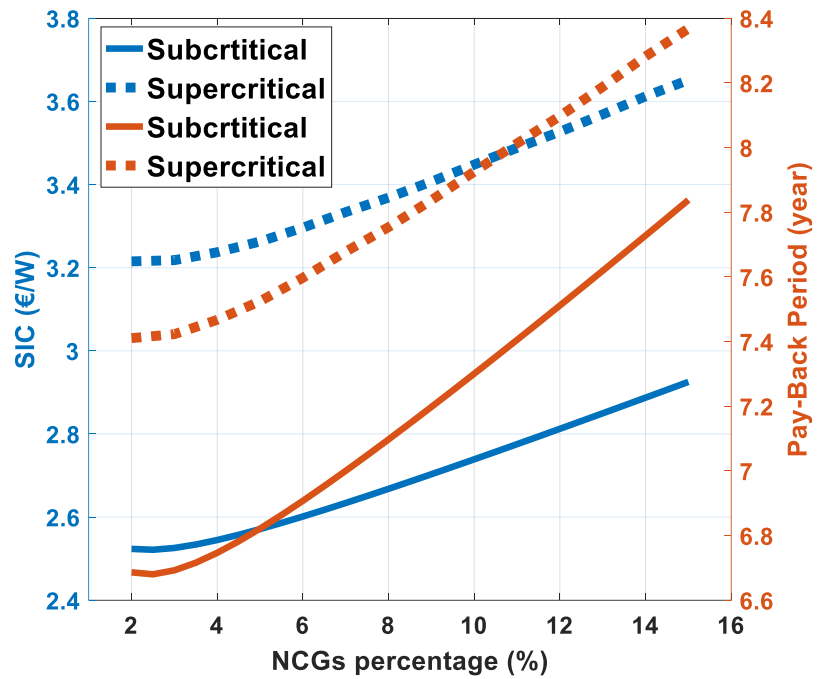


Fig. 14. Variations of SIC and pay-back period with NCGs percentage

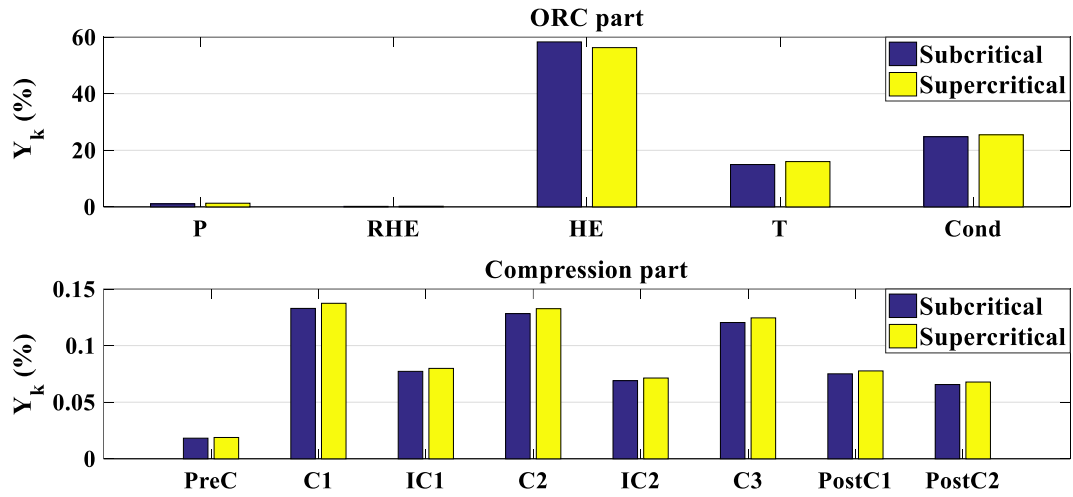


Fig. 15. Exergy destruction ratio of component in both subcritical and supercritical

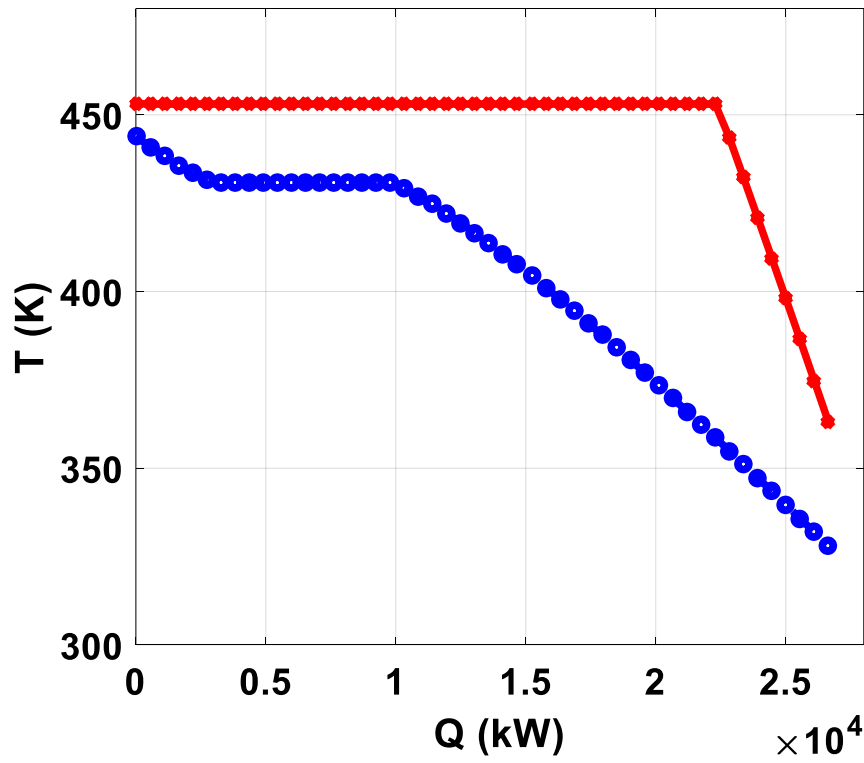


Fig. 16. Heat transfer profile of main heat exchanger

7. Conclusion

In this paper, energy, exergy, exergo-economic, and environmental assessment of a geothermal power plant with NCG reinjection are performed. The subcritical and supercritical working conditions are evaluated and compared. The thermodynamic modeling is expanded in MATLAB software and using REFPROP 9.1. A sensitivity analysis of condensing temperature, turbine inlet pressure, and amount of CO₂ in geofluid is done on system performance from thermodynamic, economic, and environmental sides. The following results are obtained:

- Increasing the condensing temperature results in decreased exergy efficiency, net power of the cycle and exergetic performance criteria, increased total exergy destruction, exergo-destruction impact factor, and specific power cost in subcritical and supercritical cases.
- Rising the inlet pressure of the turbine in the subcritical working state increases the cycle exergy efficiency and exergetic performance criteria at first and after reaching the maximum value, decreases. This pattern for total cycle exergy destruction is converse. Moreover, the pay-back period reduces first (up to a pressure of 2730 *kPa*) and then increases at higher pressures. The exergo-destruction impact factor variation trend is decreasing firstly, and after reaching to lowest value at 3030 *kPa*, it rises. The trend of specific power cost change is reducing firstly, and after reaching a minimum value at a pressure of 2730 *kPa*, it increases with more slope at higher pressures.
- Growth in the inlet pressure of the turbine in a supercritical working situation reduces the exergy efficiency, exergetic performance of the system, and the specific power cost and increases exergy destruction, pay-back period exergo-destruction impact factor.
- Increasing the amount of CO₂ in geofluid reduces the exergy efficiency, net power of the system and raises the power cost, exergo-destruction impact factor, the investment cost, and investment return in both subcritical and supercritical conditions.
- The pay-back period is obtained 7.1 and 7.7 years for subcritical and supercritical cases respectively.
- The calculated LCOE for the supercritical cycle is 6.96 (*c€/kWh*) and for subcritical is 5.52 (*c€/kWh*), These values are attractive for geothermal power plants compared with other renewable energies.
- The highest exergy destruction rate is related to the main heat exchanger, condenser, and turbine, respectively, in both subcritical and supercritical working conditions.

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Nomenclature

c	Specific exergy cost, ($\text{€}/kJ$)
$\dot{C}$	Cost rate associated with exergy transfer, ($\text{€}/s$)
$CEPCI$	Chemical Engineering Plant Cost Index
CRF	Capital Recovery Factor
ex	Specific exergy, (kJ/kg)
G	Generated electrical power, (kWh)
$\dot{E}_x$	Exergy rate, (kW)
EPC	Exergetic performance criteria
F	Correction factor
f	Exergo-economic factor
f_{ei}	Exergo-destruction impact factor
f_{es}	Exergetic stability factor
h	Specific enthalpy, (kJ/kg)
i	Rate of interest
$LCOE$	Levelized Cost of Energy, ($\text{€}/kWh$)
$LMTD$	Logarithmic mean temperature difference
$\dot{m}$	Mass flow rate, (kg/s)
N	Annual plant operation hours
Nu	Nusselt
P	Pressure, (kPa)
PBP	Pay-back period, ($yaer$)
PEC	Purchased Equipment Costs (€)
Pr	Prandtl number
$\dot{Q}$	Heat transfer rate, (kW)
Re	Reynolds number
s	Specific entropy, (kJ/kgK)
SIC	Specific investment cost, ($\text{€}/W$)
T	Temperature, ($^{\circ}C$)
U	Heat transfer coefficient, (W/Km^2)
$\dot{W}$	Power, (kW)
x	Quality
y	Exergy destruction ratio

z	Well depth
Z	Capital cost of components, (€)
$\dot{Z}$	Capital cost rate of components, (€/s)

Greek Symbols

η	Efficiency, (%)
φ	Maintenance factor
θ_{ei}	Exergo-environmental impact index
θ_{eii}	Exergo-environmental impact improvement
θ_{est}	Exergetic sustainability index
α	Convective heat transfer coefficient
λ	Thermal conductivity (W/mK)

Subscripts

b	Bulk
$Cond$	Condenser
C	Compressor
D	Destruction
e	Exit
ex	Exergy
HE	Heat exchanger
i	Inlet
IC	Inter-cooler
T	Turbine
P	Pump
pc	Pseudo-critical
$PostC$	Post-cooler
$PreC$	Pre-cooler
q	Specific heat rate
RHE	Regenerative heat exchanger
th	Thermal
TCI	Total capital investment
TDC	Total depreciable capital
Tot	Total
w	Specific work rate
wa	Wall
0	Ambient

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