RESEARCH ARTICLE

Energy, exergy, and economic analysis of an integrated solar combined cycle power plant

Hamza Moussa Benabdellah1 | Adel Ghenaiet2

1Laboratory of Energy Mechanics, Faculty of Engineering Science, M’Hamed Bougara University, Avenue of Independence, Boumerdes, Algeria
2Laboratory of Energy Conversion Systems, Faculty of Mechanical Engineering and Process Engineering, University of Science and Technology Houari Boumediene, Algiers, Algeria

Abstract

Integrated Solar Combined Cycle (ISCC) power plants based on Parabolic Trough Concentrators (PTCs) are the most efficient way for solar into electrical energy conversion. However, due to operation in several climate conditions, they need more efforts in their adaptation. This paper presents a techno-economic assessment of an ISCC - PTC system operating at Hassi R’mel site (Algerian Sahara) for which a new thermal storage system is incorporated. The obtained results reveal noticeable enhancements in the solar energy conversion and the overall power plant performance, hence offering better stability to the grid. As shown, the net solar thermal energy conversion ratio may reach up to 14%. Energy and exergy efficiencies are respectively equal to 56.06% and 53.29%, and about 46% of exergy received by the system is destructed. Moreover, the obtained Levelized Cost of Energy (LCOE) is around 9.75 ¢/kWh which is very promising, but still higher compared with a simple Combined Cycle (CC). At this geographical site, the modified ISCC - PTC power plant will allow saving about 30 million $ in natural gas consumption and 13 million $ in emission taxes.

KEYWORDS

energy and exergy analysis, integrated solar combined cycle, parabolic trough concentrators, thermal energy storage, levelized cost of energy

1 INTRODUCTION

Solar thermal energy is the most abundant clean renewable energy source which could be used to generate electricity by hybridization with Simple Rankine Cycle (SRC) or CC. The integration of solar thermal energy via a combination of Concentrated Solar Power (CSP) technology with CC has resulted in a continuous power supply to the grid, in addition to the improvement of thermal efficiency and reduction in CO₂ emission. Indeed, such hybridization does not suffer from inefficiencies related to the start-up and shut down of steam turbine, and the solar thermal energy is converted at high efficiency. There are four types of CSP technologies that could be used: Parabolic Trough Concentrator (PTC), Linear Fresnel Reflector (LFR), Solar Tower (ST), and Stirling Solar Dish. PTCs are used to collect the solar thermal energy to generate steam directly into absorber tubes or indirectly using another fluid. The first concept is named Direct Steam Generation (DSG), whereas the second one is based on Heat Transfer Fluid (HTF) as will be presented later. Over the last 20 years, several countries oriented their interest to such technologies and many Integrated Solar Combined...
Cycle (ISCC) power plants were put in service: Spain, USA, India, Brazil, Morocco, Egypt, Kuwait, and Algeria. Thermodynamic approaches are essential in preliminary thermal cycle selections. However, performing a comprehensive exergy analysis will provide a complete picture of the system behavior and provides reasonable assessments of performance and makes steps towards its improvement. Many researchers tackled such complex power plants under different aspects. Kane and Favrat were the first to investigate in details the sub-components heat exchangers considering the pinch method to integrate solar thermal energy into CCs. Baghernejad et al. made energy and exergy analysis of the ISCC system in Yazd (Iran) and calculated the values of energy and exergy efficiencies, and found them equal to 46.17% and 45.6%, respectively. Another study of solar energy integration using DSG was done by Gupta and Kaushik, thereby concluding that the condenser and solar components are the primary sources of exergy destruction. Reddy et al. carried out exergy analysis for a solar power plant based on LFR and evaluated exergy destruction. Rovira et al. presented a comparison between four Heat Recovery Steam Generation (HRSG) configurations using HTF and others with DSG, to decrease exergy destruction in HRSG and select an optimum layout. They concluded that HRSG optimization could help to improve the performance but it is not a key issue to define the best configuration. Franchini et al. compared four layouts to integrate PTC and ST into HRSG of both ISCC and SRC under the southern Spain climate. They concluded that ISCC coupled with ST assures the highest annual solar to electric efficiency of 21.8%. Peng et al. studied exergy destruction of a typical solar hybrid coal-fired power plant using the energy utilization diagram methodology and showed that exergy destruction in ISCC is lower than in solar thermal plant. Babaelahi et al. tried to optimize energy and exergy efficiencies in several SRC configurations to propose the optimum among them. According to the literature, most of the ISCC power plants are those using PTC technology. Indeed, many techno-economic studies have found that the integration of PTC with CC provides an interesting way for solar electricity generation, in addition to environmental and economic benefits. Algeria has embarked on a programme announced in 2011 aimed at increasing electricity production, especially through renewable clean energies. By the year 2030, Algeria will be relying on 37% of energy from renewable sources which could reduce the cost of production and protect the environment. The first ISCC - PTC power plant installed at Hassi R’Mel has been considered as a pilot model, producing 160 MW where 22 MW are through solar energy. It consists of classical CC and solar field through which the concentration of sunlight is reflected on the absorber and transferred via HTF to the solar steam generator. Researchers from Algeria investigated the thermal cycle of this power plant. Behar et al. showed that during sunny periods the net electricity ratio from solar energy can reach up to 15%, and quite similar value of 14.4% is obtained by Achour et al. whereas Abdelhafidi et al. found a value of 18% in June, 15% in March and 8% in December. Recently, Amani and Ghenaiet proposed a novel hybridization of the solar central receiver system via an open volumetric air receiver with the CC power plant. Their results reveal noticeable enhancements in solar energy conversion and plant performance, where the net solar energy ratio reached up to 23.5%. Derbal-Mokrane et al. used TRNSYS to show that this power plant can produce 150 MW with an efficiency of 52% vs. 49% for CC. The present study consists of in-depth analyses involving energy, exergy modelling added to an economic evaluation for ISCC - PTC power plant integrating a new thermal energy storage system to compensate for the power output during nights and cloudy days.

2 | POWER PLANT CONFIGURATION

The first ISCC - PTC power plant is located at Hassi R’Mel about 750 m above the sea level and 500 km from Algiers. The daily ambient temperature varies in-between 21 - 50°C in the summer and from −10°C to 20°C in the winter. This region is blessed daily by 9.5h of sunshine, offering an annual averaged direct normal irradiation (DNI) estimated at 7138 Wh/m²/day. As shown in Figure 1, this power plant consists of a solar field, a power block of two Gas Turbine (GT) units, one steam turbine unit, two HRSG with a simple pressure level, and one Solar Steam Generator (SSG) added to the air cooler system. The supplement of solar thermal energy provides an increase in steam mass flow of the Rankine cycle. During nightly operation, it operates as ISCC - PTC, while it operates as a conventional CC power plant during nights or cloudy days when solar radiation is insufficient and two Duct Burners (DBs) or supplementary firings are used. SSG operates as a boiler in parallel to the HRSGs for enhancing the quantity of steam generated during the sunny periods, thus only an evaporator section is used to avoid using preheating and superheating exchangers. This power plant works in three modes; ISCC system in days or at solar hours with full turbines or even with one GT; conventional CC at non-solar hours and GT when the steam turbine is not functioning. The heat exchangers are the important subsystem, hence the method of pinch point and approach point are used in the thermodynamic modelling.
2.1  |  **Power block layout**

The power unit consists of two identical GTs Siemens SGT-800 of the nominal capacity of 40 MW each, and one steam turbine Siemens SST-900 of the nominal capacity of 80 MW fed by two identical HRSGs of single pressure with supplementary firing. Each HRSG is equipped with two superheaters, evaporator, economizer, low-pressure evaporator, and low-pressure economizer. Table 1 provides some data for the power systems. The two GTs are powered by the natural gas from the Hassi R’Mel deposit, which is rich in CH₄ about 85% and has a lower heating value of 45,778 kJ/kg. Each GT is equipped with a cooling system (chiller) to lower the ambient air temperature to 15°C before entering the compressor to increase the thermal efficiency. An air-cooled condenser is equipped with 15 fans with a rated thermal capacity of 147,600 kW. This technology is a substitute for the conventional water condenser due to water unavailability in this arid region. As a result, water consumption is reduced by 90% but the thermal efficiency is lesser because of increased condensation pressure. Indeed, the air temperature of days that rarely drops below 30°C serves to condensate the water at a temperature of 52°C and pressure 0.14 bar.

2.2  |  **Solar field layout**

The solar field shown in Figure 2 has a total area of 183,120 m² and consists of 224 PTCs. The field is composed of 56 loops spread over two surfaces, north and south fields and each contains 28 loops of four modules divided into two rows. The module consists of 12 segments, each with several mirrors. Table 2 illustrates the characteristics of the solar field and Table 3 describes ET150 collector and HTF specifications.

2.3  |  **Energy storage system**

An energy storage system is added to restore the solar thermal energy during nights and when energy to heat HFT is insufficient over the low nominal temperature, hence offering better stability to the grid. This solution as shown in Figure 3 consists of two storage tanks, hot and cold. During days when the solar energy is available, HTF is pumped from the cold tank towards the ET150 collector and thereafter, after reaching its nominal temperature, transfers its energy to the hot tank to be stored. During nights, HTF stored in the hot storage tank is pumped to SSG to transform the pre-heated water into saturated steam, and afterwards HTF is stored in the cold tank.
| Gas turbine (SGT-800) | Ambient pressure | 0.928 bar |
|----------------------|------------------|-----------|
| Ambient temperature  | 35°C             |           |
| Intake compressor air temperature | 15°C |           |
| Compressor pressure ratio | 20.2 |           |

| | Compressor isentropic efficiency | 88% |
| Inlet turbine temperature | 1200°C |
| Turbine isentropic efficiency | 88% |
| Exhaust mass flow rate | 120.2 kg/s |

| | Exhaust temperature | 550°C |
| LHV of natural gas | 45,778 kJ/kg |
| Net output power | 40 MW |
| Thermal efficiency | 35% |

| Steam turbine (SST-900) | Inlet steam temperature | 560°C |
| Inlet steam pressure | 83 bar |
| Steam mass flow rate | 70 kg/s |
| Condensate temperature | 52°C |
| Condensate pressure | 0.14 bar |
| Isentropic efficiency | 90% |
| Full output capacity | 80 MW |

| HRSG (single pressure without reheat) | Fuel mass flow rate in the DBs (night) | 0.66 kg/s |
| Approach temperature | 25°C |

| | Pinch temperature | 25°C |
| Pressure losses in flue gas side | 0.025 bar |
| Pressure losses in water/steam side | 16 bar |
| Inlet water temperature | 60°C |
| Exit stack temperature | 100°C |
| Thermal efficiency | 98.5% |

| SSG | Inlet water temperature | 195°C |
| Inlet water pressure | 93 bar |
| Exit steam temperature | 372°C |
| Water/steam mass flow rate | 22.6 kg/s |
| Inlet HTF temperature | 393°C |
| Exit HTF temperature | 290°C |
| HTF mass flow rate | 205 kg/s |
| Pressure losses in water/steam side | 5.8 bar |
| Pressure losses in HTF side | 2 bar |
| Thermal efficiency | 98% |
3 | ENERGY ANALYSIS

The energy analysis is performed assuming an average day temperature of 35°C (Hassi R’Mel,\textsuperscript{29}) selected as the reference temperature. Assumptions are enumerated as follows:

- HTF cycle in ET150 collector consists of synthetic oil (VP-1) which presents a proven maturity in all solar thermal power plants and has an operational temperature range of 13–395°C. To avoid decomposition its temperature is limited to 393°C. During sunny days HTF circulation pump operates and HTF recovers the solar heat energy and transfer it to SSG. The heat flux associated with HTF conveys a thermal power of 99.8 MWth at low exergy since its maximum temperature is 393°C.
The water/steam valves between SSG and two HRSGs are kept closed until HTF has reached its normal working temperature. As the HTF system reaches the start-up temperature of SSG the feedwater control valve opens, and at this time some of the feed water is withdrawn from two HRSGs and converted to saturated steam in SSG and thereafter returned to HRSGs where it is superheated.

- From the two open-cycle GTs the residual heat energy is recovered in two HRSGs.
- The Rankine steam cycle consists of one steam turbine of a single pressure level.
- When solar energy is unavailable the plant works as pure CC represented by the ‘night mode’, while during the sunny periods, the power plant operates in the hybrid mode (ISCC-PTC) represented by the ‘day mode’.
- At the inlet and outlet of each part of the power plant, the real thermodynamic properties of air/gas, water/vapour and thermal oil are evaluated locally based on the pressure and temperature information and by considering the standard thermodynamic tables via five dedicated subroutines.

### 3.1 Solar field

The solar field consists of 224 PTC elements ‘ET150’ shared in two zones. Each zone has 28 loops, each loop consists of four collectors and each collector has an aperture area of 817.5 m² and contains a series of 12 modules each has seven mirrors along a horizontal axis between pylons and four mirrors in the vertical cross-section. Each mirror is supported on the structure at four points on its backside. The collectors of each module have one axis and are aligned at the north–south line to follow the sun from east to west. Some of the solar field parameters are mentioned in Table 4.

The DNI could be evaluated by a simple model such as Hottel, Liu and Jordan (HLJ). However, for more accuracy the satellite data corresponding to Hassi R’Mel site of the latitude of 33.8° are used to plot the annual variations of DNI in Figure 4. The maximum may reach 800 W/m² while the annual average is about 660 W/m².

The solar field parameters and specifications are given in Tables 2–5. To evaluate the thermal performance of the solar field the energy balance between solar radiation and heat absorption by HTF and the heat losses are considered:

$$Q_{\text{PTC}} = A_c \text{DNI} \eta_0 \cos \theta K(\theta) - A_{\text{abs}} U_{\text{abs}} (T_{\text{abs}} - T_a)$$  \hspace{1cm} (1)
FIGURE 4  Estimated DNI daily and monthly

TABLE 5  Site parameters

| Parameter          | Symbol | Value | Unit |
|--------------------|--------|-------|------|
| Latitude           | φ      | 33.8  | °    |
| Ambient temperature| T_a   | 20    | °C   |
| Solar constant     | I_s   | 1367  | W/m² |
| Relative humidity  | RH    | 58    | %    |

The incidence angle modifier is given as follow:

\[ K(\theta) = 1 - 2.23073 \times 10^{-4} \theta - 1.1 \times 10^{-4} \theta^2 - 3.185 \times 10^{-6} \theta^3 - 4.855 \times 10^{-8} \theta^4 \]  

(2)

The absorber temperature is evaluated from:

\[ T_{abs} = \sqrt{\alpha \text{ DNI} C_t \frac{\eta_0}{\sigma E}} + T_a \]^4  

(3)

The variation of thermal loss coefficient versus receiver pipe temperature is usually expressed with a polynomial equation, where coefficient \( a, b, \) and \( c \) were obtained experimentally.\(^{31}\)

\[ U_{abs} = a + b (T_{abs} - T_a) + c(T_{abs} - T_a)^2 \]  

(4)

The useful solar energy gained by HTF is given by:

\[ Q_{SF} = N_L \cdot C_L \cdot Q_{PTC} \]  

(5)

\( C_L, N_L \) are respectively the number of collectors in each row and the number of lines.
The mass flow rate of HTF is estimated from energy gained and enthalpy difference of HTF:

\[ m_{HTF} = \frac{Q_{SF}}{\Delta h_{HTF}} \]  

(6)

The solar field efficiency is the ratio between the net energy gained by HTF and the total quantity of solar beam reaching the mirrors:

\[ \eta_{SF} = \frac{Q_{SF}}{DNI \cdot A_P \cdot N_L \cdot C_L} \]  

(7)

PTC optical efficiency is given by the authors in Reference 33 according to Figure 5.

\[ \eta_o = \rho \tau \alpha \gamma (1 - A_f \tan \theta) \cos \theta \]  

(8)

where \( \rho \) is the reflectance of mirror, \( \tau \) transmittance of glass cover, \( \alpha \) absorptivity of mirror, \( \gamma \) intercept factor, \( \theta \) incidence angle, \( A_f \) geometric factor, \( A_P \) collector aperture area, and \( A_L \) area of shadow surface on the reflector:

\[ A_f = \frac{A_L}{A_P} \]  

(9)

Energy reflected by the collector is:

\[ Q_s = Q_i \eta_o \]  

(10)

First law efficiency for the collector is given by:

\[ \eta_S = \frac{Q_s}{Q_i} \]  

(11)

Useful energy delivered to the fluid in the receiver is given by:

\[ Q_u = N \cdot m_{HTF} (h_{HTF_o} - h_{HTF_i}) \]  

(12)

Efficiency for receiver is:

\[ \eta_F = \frac{Q_u}{Q_s} \]  

(13)

Overall efficiency of collector-receiver subsystem:

\[ \eta_{SF} = \frac{Q_u}{Q_i} \]  

(14)
3.2 | HRSGs and SSG

Gas–steam temperature profiles serve to evaluate the steam generation since the principal inputs to HRSGs are the outlet temperature and the mass flow of gases. The enthalpy of air/gases and water/vapour are evaluated locally based on pressure and temperature information. In low gas temperature heat recovery systems the steam pressure and pinch point are critical in determining the gas–steam temperature profiles and the exit gas temperature from the economizer cannot be arbitrarily assumed. The temperature profiles are determined based on Table 6 and the definitions of pinch point \( \Delta T_{pp} = T_{g7} - T_{sat} \) and approach point \( \Delta T_{ap} = T_{sat} - T_{w18} \). The temperature profiles are required to evaluate the steam generation, considering the exhaust gases parameters with the principle input parameters of HRSG are gas temperature outlet \( (T_{g22}) \) and gases mass flow \( (m_g) \).

With gases inlet temperature to economizer given by:

\[
T_{g7} = T_{sat} + T_{pp}
\]  

(15)

The energy balance of superheater and evaporator yields:

\[
m_g (h_{g22} - h_{g7}) = m_s (h_{s33} - h_{sat})
\]  

(16)

From above Equation (16) the steam generated by each HRSG during nights or cloudy periods is easily determined. Energy balance of the economizer

\[
m_g (h_{g35} - h_{g7}) = m_s (h_{w18} - h_{w33})
\]  

(17)

The performance of HRSG and SSG could be estimated by knowing the value of global heat exchange coefficient multiplied by area when operating in ‘night mode’. The update in ‘day mode’ is obtained according to Reference 35.

\[
(UA) = (UA)_n \left( \frac{m_g}{m_{g,n}} \right)^{0.65} \left( \frac{m_s}{m_{s,n}} \right)^{0.15}
\]  

(18)

3.3 | Power block

The work output of GT is that produced by turbine minus consumed by compressor including cooling fraction, transmission, and generator losses. The air mass flow is estimated from the power \( P_{GT} \) and specific work \( w_{GT} \) as follows:

\[
m_a = \frac{P_{GT}}{\eta_e \eta_m w_{GT}}
\]  

(19)

The thermal efficiency of GT unit is given by:

\[
\eta_{GT} = \frac{P_{GT}}{m_f LHV}
\]  

(20)

The specific fuel consumption of natural gas is the ratio of fuel flow rate and output power.

\[
sfc = \frac{m_f}{P_{GT}}
\]  

(21)

### Table 6: Pinch and approach points\[^{34}\]

| Evaporator type | Plain tubes | Finned tubes | For both |
|-----------------|-------------|--------------|---------|
| Gas inlet temp (°C) | Pinch point (°C) | Pinch point (°C) | Approach point (°C) |
| 650–900         | 60–85       | 20–35        | 20–40   |
| 375–650         | 40–60       | 5–20         | 5–20    |
In GT performance, typical values for combustion chamber efficiency are about 0.98,\textsuperscript{36} for pressure drop in the combustion chamber and HRSGs between 2\% and 6\% and for filters about 0.005–0.015 bar.\textsuperscript{37} The required data for the models of GT Siemens SGT 800 and steam turbine Siemens SST 900 are summarized in Table 1.

The work output from steam turbine converted into electrical energy is that from steam turbine minus consumed by pumps multiplied by transmission and generator efficiencies. Therefore the total output from the power plant converted into electrical power is given by:

\[ P_{\text{ISCC}} = 2P_{\text{GT}} + P_{\text{ST}} \quad (22) \]

The input fuel energy consumed by the power plant is given by:

\[ Q_{\text{ISCC}} = \dot{m}_f \text{LHV} \quad (23) \]

The overall energy efficiency of the power plant is defined as the ratio between the electrical power produced and the fuel energy consumed.

\[ \eta_{\text{ISCC}} = \frac{P_{\text{ISCC}}}{Q_{\text{ISCC}}} \quad (24) \]

4 | EXERGY ANALYSIS

This analysis based on the second principle of thermodynamics is a powerful tool to evaluate and improve a thermal system. The total exergy can be divided into four components: physical exergy \( E_{ph} \), chemical exergy \( E_{ch} \), kinetic exergy \( E_{kn} \), and potential exergy \( E_{pt} \).\textsuperscript{38}

\[ E = E_{ph} + E_{ch} + E_{kn} + E_{pt} \quad (25) \]

Physical specific exergy for a simple compressible pure substance is given as:

\[ e_{ph} = (h - h_0) - T_a (s - s_0) \quad (26) \]

\( T_0 \) ambient temperature and \( s_0 \) initial entropy. The local specific entropy used interpolated values from the standard thermodynamic properties of materials via dedicated subroutines.

The chemical exergy of a system is equal to the amount of theoretical work when the system is carried by reversible chemical reactions from its initial state to the dead state with its environment. The chemical exergy is given by the following relation\textsuperscript{39}:

\[ e_{ch} = RT_a \sum_{k=1}^{n} x_k \ln \frac{x_k}{x_k^0} \quad (27) \]

where \( R \) specific gases constant and \( x_k \) and \( x_k^0 \) are components fractions of the exhaust gases and the gases at dead state, respectively. The composition of natural gas from Hassi R’Mel production site is given in Table 7.

The chemical exergy of fuel is given as below, where \( M \) is the gas molecular weight.

\[ e_{ch} = \frac{1}{M} \text{LHV} \quad (28) \]

The kinetic exergy and potential exergy are neglected.

In all power systems, the product exergy is less than the fuel exergy due to losses and exergy destruction and thus this relation\textsuperscript{41} is derived.

\[ E_P = E_P + E_D + E_L \quad (29) \]
Table 7 Molar fraction of natural gas components

| Component | Molar fraction (%) |
|-----------|--------------------|
| CH₄       | 83.7               |
| C₂H₆      | 7.7                |
| C₃H₈      | 1.9                |
| C₄H₁₀     | 1.1                |
| C₅H₁₂     | 0.8                |
| N₂        | 4.8                |

Exergy efficiency (rational efficiency) is a parameter that provides a true measure of the performance of a system from a thermodynamic viewpoint. This efficiency is generally more meaningful and useful than any other efficiency based on the first or second laws of thermodynamics, including thermal efficiency. Bejan et al. identified exergy rates for different components related to fuel exergy and product exergy at steady-state:

\[ \varepsilon = \frac{E_p}{E_F} \]  (30)

Using Equation (29), Equation (30) of exergy efficiency becomes:

\[ \varepsilon = 1 - \frac{E_D + E_L}{E_F} \]  (31)

The exergy efficiency evaluated for different subsystems of the power plant:

Compressor:

\[ \varepsilon_{AC} = 1 - \frac{E_{DC} + E_{LC}}{W_C} \]  (32)

Combustion chamber:

\[ \varepsilon_{CC} = 1 - \frac{E_{DCC} + E_{LCC}}{E_{ch}} \]  (33)

Gas power turbine:

\[ \varepsilon_{GT} = 1 - \frac{E_{DGPT} + E_{LGPT}}{W_{GPT}} \]  (34)

Steam power turbine:

\[ \varepsilon_{ST} = 1 - \frac{E_{DSPST} + E_{LSPT}}{W_{SPT}} \]  (35)

For an ideal process the exergy input to PTC comes from the exergy of solar radiation. The maximum exergy can be obtained from radiation by the collector as follows:

\[ E_i = Q_s \left[ 1 - \frac{4T_s}{3T_a} (1 - 0.28 \ln f) \right] \]  (36)

where \( T_a \) is ambient temperature, \( T_s \) sun temperature about 5777 K, and \( f \) dilution factor \((1.3 \times 10^{-5})\) a mixing ratio of solar radiation from the sun and radiation from the surrounding.

Exergy of receiver, with \( T_r \) is absorber temperature:

\[ E_r = Q_s \left( 1 - \frac{T_a}{T_r} \right) \]  (37)
Exergy loss is:

\[ E_L = E_i - E_r \]  (38)

Second law efficiency:

\[ \varepsilon_R = \frac{E_R}{E_i} \]  (39)

Exergy of collector:

\[ E_c = Q_F \left( 1 - \frac{T_a}{T_C} \right) \]  (40)

Useful exergy delivered:

\[ E_u = N \dot{m}_{HTF} \left[ (h_{HTF_0} - h_{HTF_1}) - T_a (s_{HTF_0} - s_{HTF_1}) \right] \]  (41)

Second law efficiency

\[ \varepsilon_C = \frac{E_u}{E_c} \]  (42)

The overall exergy efficiency of the collector-receiver is given by:

\[ \varepsilon_{SF} = \frac{E_u}{E_i} \]  (43)

5 | ECONOMIC ANALYSIS

It is essential to estimate the average net cost of electricity generation over the power plant lifetime using LCOE given in unit of currency per kilowatt-hour (¢/kWh). The evaluation of LCOE compares between the power plant configurations while CC plant is considered as the reference plant. The economic lifetime of the steam power block is expected to reach 30 years while the life expectancy of GT is about 15 years, after which a replacement cost has to be considered. LCOE formulated as below represents a suitable method for evaluating the cost-efficiencies of diverse technologies.\(^{43}\)

\[ \text{LCOE} = \frac{\sum_{j=1}^{n} C_j (1+d)^j}{\sum_{j=1}^{n} E_j (1+d)^j} \]  (44)

where \( C_j \) is the total annual cost in the \( j \)th year, \( d \) discount rate, \( E_j \) total output energy, and \( n \) number of years. The total annual cost is the sum of capital cost (INV), operation and maintenance cost (O&M), and fuel cost (PVF). The capital cost involves investment cost in GT units, steam turbine unit, and solar unit. It is remarked from Equation (44) that the LCOE of a system varies inversely with the output energy. For a given total cost it decreases at more efficient systems. The economic assumptions data for calculating LCOE are presented in Table 8.

6 | RESULTS AND DISCUSSION

Energy and exergy models are implemented in computer code considering design parameters at nominal conditions for the ambient temperature of 35°C and pressure of 0.928 bars. The site DNI design value for the power plant corresponds to 750 W/m². At nominal the conditions this plant can provide 160 MW of electricity. One half of the output is obtained by the two GTs and the other half by the steam turbine. The solar field contributes to generating superheated steam by providing 50 MWth and contributes to more than 22 MW or almost 14% of electricity produced.
The results of energy analysis are firstly discussed, including thermal efficiencies, solar shares and net power. Second, the exergy efficiencies and exergy destruction rates are presented to underline the useful exergy. The incident solar energy onto the collector system is equal to 137.52 MW and the total energy absorbed by absorbers is equal to 114.49 MW, and so the exergy incident is about 121.04 MW and that absorbed is 99.81 MW. The difference is due to the optical efficiency of field mirrors. The sharing of this exergy by different systems is clearly shown in Figure 6.

Based on these implemented models, the results of energy and exergy analysis are presented in Tables 9 and 10, and the comparison is provided in Table 11.

Exergy provided by each component of the power plant is the result of fuel exergy minus the destruction and losses. The exergy flow quantified at different points of the power plant is summarized in Table 12. As seen, the combustors and

---

**Table 8: Assumptions and data**

| Assumptions and data                        | Value  |
|--------------------------------------------|--------|
| Life expectancy of solar field (year)      | 30     |
| Life expectancy of steam unit (year)       | 30     |
| Life expectancy of gas unit (year)         | 15     |
| Annual discount rate R (year)              | 10     |
| Capacity factor                            | 0.8    |
| **Direct costs**                           |        |
| Specific cost of solar field $C_{\text{sol}}$ ($/kW) | 1400   |
| Specific cost of steam unit $C_{\text{st}}$ ($/kW) | 635    |
| Specific cost of gas unit $C_{\text{gt}}$ ($/kW) | 235    |
| Specific cost of storage unit $C_{s}$ ($/kW) | 524    |
| Contingency (% of direct costs)            | 10     |
| **Indirect costs**                         |        |
| Engineering, procurement and construction (% of direct costs) | 13     |
| O&M cost factor of solar field $k_{\text{sol}}$ (%) | 1.5    |
| O&M cost factor of steam unit of CC $k_{\text{st}}$ (%) | 2      |
| O&M cost factor of gas unit $k_{\text{gt}}$ (%) | 5      |
| O&M cost factor of storage unit $k_{s}$ (%) | 2      |
| Fuel price ($/kft^3) based on May 2020     | 2.25   |
| Emission ($$/ton)                          | 9.9    |
TABLE 9   Energy analysis of solar field

| System      | Energy received (MW) | Energy supplied (MW) | Energy loss (MW) | Energy loss (%) | Energy efficiency (%) |
|-------------|---------------------|----------------------|------------------|-----------------|-----------------------|
| Collector   | 137.52              | 114.49               | 23.03            | 16.75           | 83.25                 |
| Receiver    | 114.49              | 51.25                | 63.24            | 55.24           | 44.76                 |
| Solar field | 137.52              | 51.25                | 86.27            | 62.73           | 37.27                 |

TABLE 10   Exergy analysis of solar field

| System      | Exergy received (MW) | Exergy delivered (MW) | Exergy loss (MW) | Exergy loss (%) | Exergy efficiency (%) |
|-------------|----------------------|-----------------------|------------------|-----------------|-----------------------|
| Collector   | 121.04               | 99.81                 | 21.23            | 17.54           | 82.46                 |
| Receiver    | 99.81                | 29.89                 | 69.92            | 70.05           | 29.95                 |
| Solar field | 121.04               | 29.89                 | 91.15            | 75.31           | 24.69                 |

TABLE 11   Energy and exergy comparison analysis of solar field

| System      | Irreversibility (MW) | Energy loss (%) | Exergy loss (%) | Energy efficiency (%) | Exergy efficiency (%) | Difference (%) |
|-------------|----------------------|-----------------|----------------|-----------------------|-----------------------|----------------|
| Collector   | 21.23                | 16.75           | 17.54           | 83.25                 | 82.46                 | 0.79           |
| Receiver    | 69.92                | 55.24           | 70.05           | 44.76                 | 29.95                 | 14.82          |
| Solar field | 91.15                | 62.73           | 75.31           | 37.27                 | 24.69                 | 12.57          |

TABLE 12   Exergy flow at different components

| Component      | Exergy fuel ExF (MW) | Exergy product ExP (MW) | Exergy destruction ExD (MW) | Exergy loss ExL (MW) | Relative irreversibility χ (%) | Exergy efficiency ζ (%) |
|----------------|----------------------|------------------------|---------------------------|---------------------|-------------------------------|------------------------|
| Compressors    | 100.1                | 93.9                   | 6.19                      | 0                   | 1.24                          | 93.81                  |
| Combustion chambers | 237.3              | 162.3                  | 75.0                      | 0                   | 13.57                         | 68.40                  |
| Gas turbines   | 263.9                | 250.6                  | 13.33                     | 0                   | 1.90                          | 94.95                  |
| Steam turbine  | 99.2                 | 89.7                   | 9.45                      | 0                   | 1.61                          | 90.47                  |
| Condenser      | 7.8                  | 1.2                    | 6.6                       | 0                   | 1.21                          | 15.35                  |
| Duct burners   | 63.6                 | 40.9                   | 22.6                      | 0                   | 4.09                          | 64.42                  |
| HRSGs          | 102.14               | 79.9                   | 22.2                      | 0                   | 0.52                          | 78.22                  |
| SSG            | 29.9                 | 25.9                   | 3.1                       | 0.87                | 1.34                          | 86.61                  |
| SF             | 99.8                 | 29.9                   | 57.6                      | 12.3                | 5.55                          | 29.94                  |
| Stacks         | 0                    | 0.0                    | 0.0                       | 6.7                 | 0.00                          | 0.00                   |
| System         | 300.8                | 160.3                  | 120.6                     | 19.9                | -                             | 53.29                  |

the solar field are the most destructors of exergy, in the proportions of 24.36% and 17.45%, respectively. Table 13 presents details of the physical parameters, such as temperature, mass flow rate, enthalpy and exergy of air, gases, water and steam for each fluid state.

To identify the exergy destruction and to represent impacts on principal exergy fuel, an exergy Grassmann diagram is specified in Figure 7. Similarly, an energy Grassmann diagram shown in Figure 8 provides a comparison between the first and second law of thermodynamics. The exergy Grassmann diagram reveals that 46.71% of exergy input to the power plant is lost and destroyed while the remaining exergy is useful. The exergy efficiency of the power plant equal to 53.29% considering ISCC without thermal storage. By adding thermal storage, the exergy efficiency becomes 67.57% which is higher compared to other layouts shown in Figure 9.
| Point | Description               | Fluid   | Phase      | Mass flow rate (kg/s) | Temperature (°C) | Pressure (bars) | Enthalpy (kJ/kg) | Exergy (MW) |
|-------|---------------------------|---------|------------|------------------------|------------------|----------------|------------------|-------------|
| 0     | Dead state                | Air     | -          | -                      | 35               | -              | 0.928            | 292.43      |
| 0'    | Water                     | -       | -          | -                      | 35               | -              | 1.013            | 79.75       |
| 0''   | Oil                       | -       | -          | -                      | 19               | -              | 1.013            | 12          |
| 25    | Air compressor inlet      | Air     | -          | 235.472                | 453.44           | 17.74          | 320.65           | 100.05      |
| 3     | Compressed air            | Air     | -          | 235.472                | 453.44           | 17.74          | 320.65           | 100.05      |
| 4     | Fuel gas GT inlet         | Gas     | -          | 240.40                 | 1200             | 17.54          | 235.02           | 263.90      |
| 1     | Natural gas providing CC  | Natural gas| -        | 4.928                  | 25               | 33             | 3856.85          | 237.28      |
| 5     | Fuel gas to HRSG          | Gas     | -          | 240.40                 | 550              | 0.953          | 565.61           | 61.19       |
| 21    | Superheated steam ST inlet| Water   | Superheated steam| 69.386            | 560              | 83             | 3543.16          | 99.16       |
| 14    | superheated steam providing deaerator | Water | Superheated steam| 0.43             | 201.85          | 4.5            | 2862.38          | 0.29        |
| 11    | Saturated steam from ST to condenser | Water | Saturated x = 0.87 | 68.956         | 52              | 0.1363         | 2352.83          | 7.82        |
| 44    | Air cooled condenser      | Air     | -          | 14,532.828            | 35               | 0.928          | 116.56           | 0.26        |
| 12    | Saturated water condenser outlet | Water | Compressed water| 68.956            | 52              | 0.1363         | 217.7            | 0.12        |
| 42    | Condensed air to environment | Air | -          | 14,532.828            | 40               | 0.928          | 111.49           | 1.44        |
| 13    | Pumped water to HRSG      | Water   | Compressed water| 68.956             | 52              | 7             | 218.47           | 0.17        |
| 23    | Natural gas providing DB  | Natural gas| -        | 1.32                  | 25               | 33             | 3856.85          | 63.56       |
| 22    | Fuel gas to SHE2          | Gas     | -          | 241.72                 | 750              | 0.953          | 583.59           | 102.14      |
| 35    | Fuel gas to environment   | Gas     | -          | 241.72                 | 100              | 0.928          | 1326.93          | 6.73        |
| 10    | ECO1 outlet water         | Water   | Compressed water| 69.386            | 195             | 94.1           | 833.47           | 9.89        |
| 16    | ECO2 inlet water         | Water   | Compressed water| 46.766            | 195             | 89.2           | 833.47           | 6.65        |
| 24    | DECO inlet water         | Water   | Compressed water| 78.346            | 60              | 7             | 251.73           | 0.36        |
| 31    | Superheated steam from SHE1 | Water | Compressed water| 70.036            | 147.91          | 4.5           | 623.22           | 4.98        |
| 33    | Total superheated steam to SHE2 | Water | Saturated vapor| 46.766            | 372             | 87.2           | 3039.75          | 53.51       |
| 46    | Synthetic oil from SF to SSG | Oil | -          | 235.65                 | 392              | 14             | 3240.73          | 118.62      |
| 51    | Synthetic oil from SSHE to SEVA | Oil | -          | 235.65                 | 378.2            | 13.5           | 3211.89          | 232.43      |
| 53    | Synthetic oil from SSG to SF | Oil | -          | 235.65                 | 292              | 12             | 3028.97          | 207.36      |
| 54    | Water from HRSG to SSG    | Water   | Compressed water| 22.62             | 195              | 94.1           | 833.47           | 3.22        |
| 55    | Superheated steam from SSG to HRSG | Water | Saturated vapor| 22.62             | 372             | 87.2           | 3039.75          | 25.88       |
More detail is depicted in Figure 10, where the highest exergy loss fraction in the power plant is 19.12% from combustors, heat transfer, and frictions which are the main sources of irreversibility. Indeed, chemical reactions are the most significant source of exergy destruction. The second-largest exergy destruction occurs in the solar field with 13.70% of total system exergy, due to the importance of temperature transformation between solar radiation and HTF. The third-largest destructors of exergy are compressor, gas and steam power turbines in amounts of 29 MW or 5.67% of total exergy destroyed in the power plant. As shown in Figure 11 steam turbine represents about 32%, and good management of aerodynamic design can minimize these losses. Also, the compressor has the least exergy loss about 11% each. After the main components are the heat exchangers including HRSGs, SSG, condenser, and stacks with 7.76% corresponding to 36.6 MW of input exergy related to heat transfer and frictions. According to Figure 12, the two HRSGs have the most destroyed exergy about 56%, and 10% for SSG, while the condenser destroys about 17%.

Exergy efficiencies for each component are depicted in Figure 13, successively gas power turbine, compressor, and steam power turbine and SSG and HRSGs which their respective efficiencies are: 95% for gas power turbine, 93% for...
**FIGURE 10** Exergy destruction

![Pie chart showing exergy destruction categories: Useful Exergy 53%, Solar Field Loss 14%, Combustors Losses 19%, Stack & Condenser Losses 3%, Compressors & Turbines Losses 6%, and HRSGs Losses 4%]

**FIGURE 11** Exergy destruction of rotating components

![Pie chart showing exergy destruction for rotating components: Compressor 1 11%, Compressor 2 11%, Gas Turbine 1 23%, Gas Turbine 2 23%, and Steam Turbine 32%]

**FIGURE 12** Exergy destruction of heat exchangers

![Pie chart showing exergy destruction for heat exchangers: Stack 17%, Condenser 17%, HRSG 1 28%, HRSG 2 28%, and SSG 10%]

**FIGURE 13** Exergy efficiencies of ISCC’s components

![Bar chart showing exergy efficiencies for ISCC's components: Compressors, Combustion Chambers, Gas Turbines, Steam Turbine, Condenser, Duct Burners, HRSGs, SSG, and SF]

compressor, 90% for steam turbine and 86% for SSG, and 78% for HRSGs. These values indicate the highest performance of turbomachinery components compared with heat exchangers systems. The exergy efficiencies of the combustion chamber and DBs are 68% and 64%, respectively, due to low air ambient temperature compared to combustion temperature. The relatively low efficiency of the combustor indicates such deficiencies could be reduced by preheating air and reducing the air-gas ratio, and subsequently, the performance of the combustor could be improved.

Far behind, the exergy efficiency of the solar field is around 30% due to irreversibility created by heat transfer involving too large temperature difference owing to the limited temperature of HTF. On the other hand, efforts are needed to reduce exergy loss and increase exergy efficiency by researching new HTF able of absorbing more exergy.

The energy Grassmann diagram shows that 56.06% of energy input to the system is converted into electricity. The main losses take place first in the condenser at about 15.35%, followed by the solar field for 12.36% and the stack where 6.25% is lost which is smaller compared with that of the condenser. The solar field and condenser behave differently from exergy analysis; the solar field has an important exergy loss of 13.70% while the loss in the condenser is about 1.26% of input exergy. This behaviour could be explained by the fact that the solar field has higher quality energy loss, whereas in condenser energy loss is of low quality (has less potential or ability to be utilized for work).

In critical weather conditions such as in winter, the solar radiation reaching the power plant's site becomes weak and according to Figure 4 the DNI is only equal to 540 W/m². The computation revealed a drop in energy and exergy efficiencies for about 51.06% and 48.75%, respectively. This is because of using the supplementary firing in HRSGs where more natural gas is burned to maintain the same electrical power production.

This subsection discusses the economic aspects. Figures 14 and 15 present the contribution of different investments without considering environmental costs and with environmental costs in Figures 16 and 17. LCOE is strongly affected by the initial investment especially in the solar part and the newly added storage system added to fuel consumption. The investment in solar integration is recovered in fuel consumption but at higher LCOE. In the case of ISCC-PTC power
The estimated LCOE is about 9.75 €/kWh which is higher than that of CC found equal to 6.38 €/kWh. By adopting the new storage system LCOE may reach about 11.88 €/kWh which is even higher.

Environmental issues have an important impact on exploitation cost. Figure 18 gives a comparison based on added taxes due to CO₂ emissions as calculated from the specific fuel consumption via combustion gases products. Accordingly, ISCC-PTC with a thermal storage system is the cleanest system since it preserves more than 26 million $ per year compared to CC alone and thus avoids 0.3 million ton of CO₂ emission per year and subsequently cutting about 13 million $ per year if the solar plant does not use the storage system such as the case of Hassi R’Mel power plant.

In term of natural gas consumption, Figure 19 provides PVF in million dollars per year while comparing ISCC-PTC system and CC. The former could save 30.76 million $ and if equipped with storage, it could save about 60 million $ per year, which is a powerful indicator in such layout selection.
Figure 20 makes a comparison between the global costs and depicts that the solar installations need an important cost in the first investment, but with the time of operation ISCC will require lesser fuel consumption and subsequently lower emission and emission taxes. In a conclusion, ISCC with storage has an important initial investment, but thereafter it will economize in fuel and taxes generated by CO₂ emissions, and extra costs are amortized in about 3 years which show the feasibility of this solution.

Once again when the solar radiation reaching the power plant is weak and for the same rated power it is required to activate the supplementary firing. As consequence, this will add more costs due to increased fuel consumption, and subsequently, LCOE will increase by 0.07 to become equal to 9.82 ¢/kWh.

7 | CONCLUSION

Based on energy, exergy and economic analyses carried out for an ISCC-PTC power plant operating under the Saharan climate, this technology is still competitive in the medium and short term. The obtained results show that the rotating components have the highest exergy efficiency about 95%, 93% and 90% respectively for the power GT, compressor and power steam turbine. However, lesser efficiencies are obtained for SSG and HRSGs which are equal to 86% and 78%, respectively. The solar field and the condenser have the worst efficiencies of 30% and 15%, respectively, due to the large temperature differences between HTF and solar field and between water/steam and ambient air. The main burner and supplementary firing are the main destructors of exergy about 14.69% and 4.43%, respectively. Subsequently, the preheating of air could reduce this deficiency. The solar field is another important exergy destructor at around 14%. Moreover, the Improvement of collectors' efficiency could improve the exergy efficiency of the solar field and the whole ISCC-PTC system. The thermal storage system; despite its added cost, in the course of operation will reduce the natural gas consumption and emission taxes. The extra cost could be amortized in about 3 years which consolidates the feasibility of this solution.
**PEER REVIEW INFORMATION**

*Engineering Reports* thanks the anonymous reviewers for their contribution to the peer review of this work.

**PEER REVIEW**

The peer review history for this article is available at https://publons.com/publon/10.1002/eng2.12404.

**DATA AVAILABILITY STATEMENT**

The data that support the findings of this study are available from the corresponding author upon reasonable request.

**CONFLICT OF INTEREST**

The authors declare no potential conflicts of interest with respect to the research, authorship and/or, publication of this article.

**NOTATIONS**

- \( A_{\text{abs}} \) absorber area \([\text{m}^2]\)
- \( A_f \) geometric factor
- \( A_L \) area of shadow surface on the reflector \([\text{m}^2]\)
- \( A_p \) collector aperture area \([\text{m}^2]\)
- \( C_c \) concentration ratio
- \( C_L \) collector number in each row
- \( C_{p_s} \) specific heat \([\text{J/kg K}]\)
- \( C_j \) total annual cost \([\text{$}]\)
- \( d \) discount rate
- \( \text{DNI} \) direct normal irradiation \([\text{W/m}^2]\)
- \( E \) system exergy \([\text{J}]\)
- \( E_i \) input exergy \([\text{J}]\)
- \( E_R \) absorbed exergy in receiver \([\text{J}]\)
- \( E_C \) absorbed exergy in receiver \([\text{J}]\)
- \( E_j \) total output energy \([\text{J}]\)
- \( E_{\text{ph}} \) physical exergy \([\text{J}]\)
- \( E_{\text{ch}} \) chemical exergy \([\text{J}]\)
- \( E_{\text{kn}} \) kinetic exergy \([\text{J}]\)
- \( E_{\text{pt}} \) potential exergy \([\text{J}]\)
- \( E_F \) fuel exergy \([\text{J}]\)
- \( E_P \) product exergy \([\text{J}]\)
- \( E_D \) destruct exergy \([\text{J}]\)
- \( E_L \) loss exergy \([\text{J}]\)
- \( E_{\text{DC}} \) compressor destructed exergy \([\text{J}]\)
- \( E_{\text{LC}} \) compressor loss exergy \([\text{J}]\)
- \( E_{\text{DCC}} \) combustion chamber destructed exergy \([\text{J}]\)
- \( E_{\text{LCC}} \) combustion chamber loss exergy \([\text{J}]\)
- \( E_{\text{DGT}} \) gas turbine destructed exergy \([\text{J}]\)
- \( E_{\text{LGT}} \) gas turbine loss exergy \([\text{J}]\)
- \( E_{\text{DST}} \) steam turbine destructed exergy \([\text{J}]\)
- \( E_{\text{LST}} \) steam turbine loss exergy \([\text{J}]\)
- \( e \) mass system exergy \([\text{J/kg}]\)
- \( e_{\text{ph}} \) mass physical exergy \([\text{J/kg}]\)
- \( e_{\text{ch}} \) mass chemical exergy \([\text{J/kg}]\)
- \( e_{\text{kn}} \) mass kinetic exergy \([\text{J/kg}]\)
- \( e_{\text{pt}} \) mass potential exergy \([\text{J/kg}]\)
- \( f \) dilution factor
- \( h \) enthalpy \([\text{J/kg}]\)
- \( h_{\text{HTF}} \) enthalpy HTF \([\text{J/kg}]\)
INV  capital cost [$]
LCOE  levelized cost of energy [¢/kWh]
LHV  low heating value [kJ/kg]
\( M \)  gas molecular weight [g/mol]
\( \dot{m}_{HTF} \)  heat thermal fluid mass flow [kg/s]
\( \dot{m}_g \)  gases mass flow [kg/s]
\( N_L \)  number of lines in solar field
O&M  operation and maintenance [$]
PVF  annual fuel cost [$]
P_{GT}  gas turbine output [MW]
P_{ST}  steam turbine output [MW]
P_{ISCC}  plant output [MW]
\( R \)  specific gas constant [J/kg.K]
\( Q_{PTC} \)  useful energy gained by absorber [kW]
\( Q_{SF} \)  useful energy gained by HTF [kW]
\( Q_i \)  energy reflected by the collector [kW]
\( Q_l \)  energy incident on collector [kW]
\( Q_a \)  energy absorbed by receiver [kW]
\( S_{HTF} \)  entropy of HTF [J/K]
\( s_0 \)  initial entropy [kJ/kg K]
\( s \)  entropy [kJ/kg K]
\( T_a \)  ambient temperature [°C]
\( T_{abs} \)  absorber temperature [°C]
\( T_s \)  sun temperature [°C]
\( T_{sat} \)  saturated temperature [°C]
\( T_r \)  receiver temperature [°C]
\( \Delta T_{ap} \)  approach point temperature [°C]
\( \Delta T_{pp} \)  pinch point temperature [°C]
\( U_{abs} \)  absorber thermal loss coefficient [kW/m2 K]
\( UA \)  global heat exchange coefficient by area[W/K]
\( x \)  component fraction of exhaust gases
\( x^0 \)  component fraction of gases at dead

**GREEK SYMBOLS**
\( \alpha \)  absorptivity
\( \eta \)  efficiency
\( \eta_o \)  optical efficiency of parabolic trough collectors
\( \eta_{SF} \)  solar field energetic efficiency
\( \varepsilon \)  absorber emissivity
\( \varepsilon_{AC} \)  compressor exergetic efficiency
\( \varepsilon_{CC} \)  combustion chamber exergetic efficiency
\( \varepsilon_{GT} \)  gas turbine exergetic efficiency
\( \varepsilon_{ST} \)  steam turbine exergetic efficiency
\( \varepsilon_{SF} \)  solar field exergetic efficiency
\( \varepsilon \)  ISCC exergetic efficiency
\( \theta \)  angle of incidence [rad]
\( \tau \)  atmospheric transmissivity
\( \sigma \)  Steflane-Boltzmann constant [W/m² K⁴]
\( \rho \)  reflectivity
\( \gamma \)  intercept factor
\( \phi \)  Latitude [°]
SUBSCRIPTS
a ambient
abs absorber
ap approach point
c collector
ch chemical
D destruct
e electrical
f fuel
GT gas turbine
g gases
HTF heat thermal fluid
ISCC integrated solar combined cycle
i input
j year
kn kinetic
L loss
m mechanical
n night mode
o output
P product
ph physical
pp pinch point
pt potential
PTC parabolic trough concentrator
r receiver
s sun
SPT steam power turbine
SF solar field

AUTHOR CONTRIBUTIONS
Hamza Moussa Benabdellah: Data curation; formal analysis; investigation; methodology; software; validation; visualization; writing-original draft. ADEL GHENAIET: Conceptualization; formal analysis; investigation; methodology; supervision; writing-original draft; writing-review & editing.

ORCID
Adel Ghenaiet https://orcid.org/0000-0001-5112-3551

REFERENCES
1. Tasbirul IM, Nazmul H, Ahmad Baharuddin A, Rahman S. A comprehensive review of state-of-the-art concentrating solar power (CSP) technologies: current status and research trends. Renew Sustain Energy Rev. 2018;91:987-1018. https://doi.org/10.1016/j.rser.2018.04.097.
2. Hafez AZ, Attia AM, Eltwab HS, et al. Design analysis of solar parabolic trough thermal collectors. Renew Sustain Energy Rev. 2018;82:1215-1260. https://doi.org/10.1016/j.rser.2017.09.010.
3. San Miguel G, Corona B. Economic viability of concentrated solar power under different regulatory frameworks in Spain. Renew Sustain Energy Rev. 2018;91:205-218. https://doi.org/10.1016/j.rser.2018.03.017.
4. Xu X, Vignarooban K, Xu B, Hsu K, Kannan AM. Prospects and problems of concentrating solar power technologies for power generation in the desert regions. Renew Sustain Energy Rev. 2016;53:1106-1131. https://doi.org/10.1016/j.rser.2015.09.015.
5. Purohit I, Purohit P. Technical and economic potential of concentrating solar thermal power generation in India. Renew Sustain Energy Rev. 2017;78:648-667. https://doi.org/10.1016/j.rser.2017.04.059.
6. Vieira de Souza LE, Gilmanova Cavalcante AM. Concentrated solar power deployment in emerging economies: the cases of China and Brazil. Renew Sustain Energy Rev. 2017;72:1094-1103. https://doi.org/10.1016/j.rser.2016.10.027.
7. Tsikalakis A, Tomsi T, Hatzigyrgiou ND, et al. Review of best practices of solar electricity resources applications in selected Middle East and North Africa (MENA) countries. Renew Sustain Energy Rev. 2011;15:2838-2849. https://doi.org/10.1016/j.rser.2011.03.005.
8. Servert JF, Cerrajero E. Assessment on Egypt’s CSP components manufacturing potential. Energy Proca. 2015;69:1498-1507. https://doi.org/10.1016/j.egypro.2015.03.100.

9. Binamer AO. Al-Abdaliya integrated solar combined cycle power plant: case study of Kuwait, Part I. Renew Energy. 2019;131:923-937. https://doi.org/10.1016/j.renene.2018.07.076.

10. Stambouli AB, Khiat K, Flazi S, Kitamura Y. A review on the renewable energy development in Algeria: current perspective, energy scenario and sustainability issues. Renew Sust Energ Rev. 2012;16:4445-4460. https://doi.org/10.1016/j.rser.2012.04.031.

11. . Stambouli AB. Promotion of renewable energies in Algeria: strategies and perspectives. Renew Sustain Energ Rev. 2011;15:1169-1181. https://doi.org/10.1016/j.rser.2010.11.017.

12. Kane M, Favrat D. Approche de conception et d’optimisation de centrale solaire intégrée à cycle combiné inspirée de la méthode du pincement (Partie I: paliers de récupération)synthesis and optimization approach for integrated solar combined cycle systems based on pinch tec. Int J Therm Sci. 1999;38:501-511. https://doi.org/10.1016/S1290-0729(99)80023-4.

13. Kane M, Favrat D. Approche de conception et d’optimisation de centrale solaire intégrée à cycle combiné inspirée de la méthode du pincement (Partie II: réseau d’échangeurs de chaleur)synthesis and optimization approach for integrated solar combined cycle systems based on p. Int J Therm Sci. 1999;38:512-524. https://doi.org/10.1016/S1290-0729(99)80024-6.

14. Baghernajad A, Yaghoubi M. Exergy analysis of an integrated solar combined cycle system. Renew Energy. 2010;35:2157-2164. https://doi.org/10.1016/j.renene.2010.02.021.

15. Gupta MK, Kaushik C. Exergyanalysis and investigation for various feed water heaters of direct steam generation solar–thermal power plant. Renew Energy. 2010;35:1228-1235. https://doi.org/10.1016/j.renene.2009.09.007.

16. Siva Reddy V, Kaushik SC, Tyagi SK. Exergetic analysis of solar concentrator aided natural gas fired combined cycle power plant. Renew Energy. 2012;39:114-125. https://doi.org/10.1016/j.renene.2011.07.031.

17. Rovira A, Montes MJ, Varela F, Gil M. Comparison of heat transfer fluid and direct steam generation technologies for integrated solar combined cycles. Appl Therm Eng. 2015;25:264-274. https://doi.org/10.1016/j.applthermaleng.2012.12.008.

18. Franchini G, Perdic Ichizzi A, Ravelli S, Barigozzi G. A comparative study between parabolic trough and solar tower technologies in solar rankine cycle and integrated solar combined cycle plants. Sol Energy. 2013;98:302-314. https://doi.org/10.1016/j.solener.2013.09.033.

19. Peng S, Wang Z, Hong H, Xu D, Jin H. Exergy evaluation of a typical 330 MW solar-hybrid coal-fired power plant in China. Energy Conver Manage. 2014;85:848-855. https://doi.org/10.1016/j.enconman.2013.12.073.

20. Babaelahi M, Mofidipour E, Rafat E. Combined energy-exergy-control (CEEC) analysis and multi-objective optimization of parabolic trough solar collector powered steam power plant. Energy. 2020;201:117641. https://doi.org/10.1016/j.energy.2020.117641.

21. Ministry of Energy and Mines New energies, renewable energies and mastery of energy. http://www.energy.gov.dz. Accessed September 2018.

22. Behar O, Khellaf A, Mohamedi K, Belha M. Instantaneous performance of the first integrated solar combined cycle system in Algeria. Energy Proca. 2011;6:185-193. https://doi.org/10.1016/j.egypro.2011.05.022.

23. Amani M, Ghenait E. Novel hybridization of solar central receiver system with combined cycle power plant. Energy. 2020;201:117627:17. https://doi.org/10.1016/j.energy.2020.117627.

24. Abdelhafidi N, Bachari N, Abdelhafidi Z, Cheknane A, Mohkache A, Castro L. Modeling of integrated solar combined cycle power plant (ISCC) of Hassi R’ mel, Algeria. Int J Energy Sect Manag. 2020;16(3):505-526.

25. Amani M, Ghenaiet A. Novel hybridization of solar central receiver system with combined cycle power plant. Energy. 2020;201:117627. https://doi.org/10.1016/j.energy.2020.117627.

26. Derbal-Mokrane H, Bouaichoua S, El Gharbi N, Belhamel M, Benzaoui A. Modeling and numerical simulation of an integrated solar combined cycle system in Proc Eng. 2012;33:199-208. https://doi.org/10.1016/j.proeng.2012.01.1194.

27. Khalidi F. Energy and exergy analysis of the first hybrid solar-gas power plant in Algeria. Paper presented at: Proceedings of the 25th International Conference on Efficiency, Cost, Optimization, Simulation and Environmental Impact of Energy Systems(ECOS 2012). Perugia, Italy; June 26–29, 2012.

28. Geyer M, Lüpfert F, Osuna R, et al. EUROTRough-parabolic trough collector developed for cost efficient solar power generation. Paper presented at: Proceedings of the 11th International Symposium on Concentrating Solar Power and Chemical Energy Technologies; 2002.

29. National Meteorological Office (ONM) Report 2014. Algeria, 2014.

30. Romero-Alvarez M, Zarzba E. Chap 21, Plataforma Solar de Almeria-CIEMAT. Concentrating Solar Thermal Power. Boca Raton, FL: Taylor & Francis Group; 2007.

31. Cohen GE, Kearney DW, Kolb GJ. Final report on the operation and maintenance improvement program for concentrating solar power plants. SAND99-1290, United States; 1999. https://doi.org/10.2172/8378.

32. Cohen GE, Kearney DW, Kolb GJ. Final report on the operation and maintenance improvement program for concentrating solar power plants. SAND99-1290, United States; 1999. https://doi.org/10.2172/8378.

33. Duffie JA, Beckman WA. Solar Engineering of Thermal Processes. 2nd ed. New York, NY: Wiley; 1991.

34. Romero-Alvarez M, Zarzba E. Chap 21, Plataforma Solar de Almeria-CIEMAT. Concentrating Solar Thermal Power. Boca Raton, FL: Taylor & Francis Group; 2007.

35. Cohen GE, Kearney DW, Kolb GJ. Final report on the operation and maintenance improvement program for concentrating solar power plants. SAND99-1290, United States; 1999. https://doi.org/10.2172/8378.

36. J.A. Duffie and W.A. Beckman,”Solar Engineering of Thermal Processes”, 4th, Hoboken, NJ: John Wiley & Sons, Inc, 2013. https://doi.org/10.1002/9781118671603.

37. Ganapathy V. Industrial Boilers and Heat Recovery Steam Generators. New York: Marcel Dekker; 2003.

38. Ganapathy V. Steam Generators and Waste Heat Boilers for Process and Plant Engineers. Boca Raton, FL: Taylor & Francis Group; 2015.

39. Razak AMY. Industrial Gas Turbines: Performance and Operability. Boca Raton, FL: Taylor & Francis Group; 2007.

40. Kim T, Ro S. Comparative evaluation of the effect of turbine configuration on the performance of heavy-duty gas turbines. ASME 1995; 95-GT-334:V004T1A019. https://doi.org/10.1115/95-GT-334.

41. Bejan A, Tsatsaronis G, Moran M. Thermal Design and Optimization. New York, NY: Wiley; 1996.
39. Colpan CO. Exergy Analysis of Combined Cycle Cogeneration Systems. Ankara, Turkey: Graduate School of Natural and Applied Sciences, Middle East Technical University; 2005.

40. J. P. Mota and S. Lyubchik, Recent Advances in Adsorption Processes for Environmental Protection and Security, Amsterdam, Netherlands: Springer, 2008. https://doi.org/10.1007/978-1-4020-6805-8.

41. Almutairi A, Pericles P, Nawaf A-M. Energetic and Exergetic analysis of combined cycle power plant: Part-1 operation and performance. Energies. 2015;8(12):14118-14135. https://doi.org/10.3390/en8121418.

42. Winter C-J, Sizmann RL, Vant-Hull LL. Solar Power Plants: Fundamentals, Technology, Systems, Economics. Berlin, Germany: Springer-Verlag; 1991.

43. Wang X, Kurdgelashvili L, Byrne J, Barnett A. The value of module efficiency in lowering the levelized cost of energy of photovoltaic systems. Renew Sustain Energy Rev. 2011;15(9):4248-4254. https://doi.org/10.1016/j.rser.2011.07.125.

44. Horn M, Führing H, Rheinländer J. Economic analysis of integrated solar combined cycle power plants: a sample case: the economic feasibility of an ISCCS power plant in Egypt. Energy. 2004;29:935-945. https://doi.org/10.1016/S0360-5442(03)00198-1.

45. Hosseini R, Soltani M, Valizadeh G. Technical and economic assessment of the integrated solar combined cycle power plants in Iran. Renew Energy. 2005;30(10):1541-1555. https://doi.org/10.1016/j.renene.2004.11.005.

46. Nezammahalleh H. Exergy analysis of DSG parabolic trough collectors for the optimal integration with a combined cycle. Int J Exergy. 2015;16(1):72-96. https://doi.org/10.1504/IJEX.2015.067300.

How to cite this article: Benabdellah HM, Ghenaiet A. Energy, exergy, and economic analysis of an integrated solar combined cycle power plant. Engineering Reports. 2021;e12404. https://doi.org/10.1002/eng2.12404