Energy-exergy performance assessment with optimization guidance for the components of the 396-MW combined-cycle power plant

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Abstract
In this study, energy-exergy analysis is performed for the gas-fired combined-cycle power plant which will be constructed in Kuantan and Kapar in the Malay Peninsula in 2020. The main objectives of this current study are first and second law evaluation of the powerplant's main components consisting of the gas turbine unit, condenser, heat recovery steam generator, and triple-pressure steam turbines. By energy-exergy analysis, we can not only evaluate the exact magnitude of exergy destruction and efficiency in each component separately but analyze either major or trivial effects of environment condition variations on plant components. Performance effects are also pinpointed when the pressure ratio, reference temperature, HPT and condenser pressure, reheating and stack temperature change. In the current investigation, more than 808 MW of exergy destruction to the environment mainly occurs in the gas turbine cycle with around 83.79% of total exergy destruction, followed by HRSG (11.3%), steam turbines (roughly 2.58%), and condenser (2.54%). Additionally, exhaust gas fractions from the gas turbine across the combustor are calculated as well wherein the average molar fraction of nitrogen, oxygen, carbon dioxide, and water vapor is calculated to be 76.42%, 17.57%, 1.37%, and 4.64%, respectively. In conclusion, several constructive possibilities for CCPP's performance development based on the obtained results are introduced. With regard to the construction of this project in 2020, energy-exergy assessment, as well as optimization guidance, can be worthwhile for better modifying some operating conditions and taking advantage of the trivial destruction rate simultaneously.

Keywords
combined-cycle power plant, combustion chamber, energy, exergy, gas turbine, heat recovery steam generator, thermal efficiency
1 | INTRODUCTION

As the first half of the 21st century continues to unfold, several controversial debates regarding improving combined-cycle powerplants and reducing their environmental impacts have arisen, most of which used to be out of the human being’s comprehension in days gone by. When the steam (bottoming) cycle is combined with the gas unit (upping) cycle in the combined-cycle, the steam cycle improves overall net output and first-law efficiency on account of its greater efficiency and operational flexibility. The importance of this increased energy becomes significant when data gathered in the United States tell more than half of energy generation is utilized in the residential and commercial localities, and over two-fifths of them are utilized in US industries. Since then, a wide range of researchers have proposed several certain designs and performance optimizations to reduce the catastrophic effects of fuel consumption and increase plant efficiency simultaneously. By implementing energy-exergy analysis, we can evaluate such potential climate impacts that are appeared due to the exergy destruction of the powerplant. In thermodynamics, exergy is defined as the maximum amount of work that is achievable for a system as it is brought into equilibrium with a reference environment. Exergy analysis is a thermodynamic analysis technique based on the second law providing an alternative and illuminating approach for investigating energy processes rationally and meaningfully. The clear difference between energy and exergy analysis is that by exergy analysis, we can gather some proper information regarding the exact locations of inefficiencies and the magnitude of them which cannot be given by means energy analysis. Therefore, second law assessment can be expressed as a powerful tool which can help us to optimize the system design and improve its energy conversion. Simply put, it is a kind of helpful approach for clarifying the distinction between energy wasted to the environment and internal irreversibility in the system. In fact, any sort of irreversibility detrimentally affects the energy conversion of the combustion-based systems including internal-combustion engines, industrial furnaces for materials processing or even residential and commercial space heating furnaces.

Several researchers also have conducted research on CCPPs as well as different integrated systems with the intention of finding the proper rage of better system performance. That is, the integrated solar farm which was deduced to operate fossil fuel powerplant cycles was assessed by Ahmadi et al. in which he concluded that the current case can be the best choice because net energy and exergy efficiencies show the 18.29% increase when compared to the simple cycle. Soo In et al. worked on a straightforward optimization approach of the HRSG models focusing on removing destroyed exergy to find optimum evaporation temperature in HRSG. Naserabad et al. also proposed that using a double-pressure level of HRSG has a better efficiency generation than the single one. Ahmadi et al. calculated that the generation power when GT Model ABB GT8C together with single-pressure HRSG are utilized is 7.6 times greater than its simple CHP cycle. Conducting the objective assessments of the different power plants based upon energy-exergy analysis can be found in works of in which the effects of various design cycle values on plant performance have been analyzed. Furthermore, they believe that managing the amount of combustion excess air at the optimal level, suitable isolating of the boiler body, and making use of outlet gas energy can be useful methods for reducing exergy degradation. A study on coal-based thermal power plants was performed by Kaushik pointing out that the major exergy loss in the gas-fired and coal-based power plant is the combustion chamber and boiler, respectively, and also efficiency at different boiler pressures and temperatures was assessed. Ahmadi et al. also prepared graphs to illustrate the upward trend of CO₂ production against arising steam to heat sink and P/TGS. Toghraie et al. investigated the effects of geometric characterizations on the solar chimney power plant. When the inlet air flow rate varies, different types of effectiveness and second law efficiency were studied by Ma et al. in a custom liquid desiccant air conditioning (LDAC) system. Ersayin and Ozgener re-evaluated the first and second law efficiencies after installation of new heat recovery steam generator and gas turbine units in 2015, and improve its efficiencies compared with 2009 to a great extent. Ahmadi et al. pointed out that one-level HRSG with one reheat phase can increase energy and exergy efficiencies by 76.8% and 73% in the integrated cycle of solar and Montazeri steam powerplant, and to take the precaution of reproducing CO₂, feed water to the solar field should not be exceeded 31.3 kg/s. Abuelnuor et al. carried out an exergy analysis for the 180 MW CCPP called Gharri “2” consisting of two gas turbines. According to, combustor per se had the biggest exergy destruction, compared with the gas turbine and the HRSG. The vast majority of renewable energy sources (RES) depending considerably on climate patterns cannot be employed for 24-hour power generation owing to unusual climate patterns in some localities. Therefore, contrasting with the self-utilization of such systems, combinations of more RES production called HRES is more likely to be trustworthy (eg, Hybrid Renewable Energy System), and be able to resolve emissions, efficiency, technical and economic issues, especially in untouchable areas. Works of show the effect of ambient temperature on changing the performances of either gas or steam cycle is much more sensitive than relative humidity and pressure so that owing to 1 K increment in the compressor air inlet temperature and environment temperature, power output reduction was investigated to be nearly 0.6% and 1%, respectively. The comprehensive assessment of the steam power plant has performed in the analysis of Ameri et al. in which major exergy is destructed in the condenser by 307 MW. Gupta
and Kaushik recommended that exergy destruction can decrease if solar thermal energy would be utilized as an aided source for feed water preheating. Farhad et al. introduced an innovative efficient design of the feedwater heaters network in steam power plants aiming 0.6%-1.7% reduction in fuel consumption as well as a 1%-2.7% reduction in condenser load with a target pinch temperature of 3°C. Cihan, Vidal, and Koch also carried out an exergy analysis with an eye toward achieving optimization algorithm considering varied conditions and devices (e.g., two and three GT cycles or adding refrigeration cycle), and Sanjay et al. specified the sources of the degradation in a reheat gas-steam CCPP utilizing air as a coolant and performed an exergy analysis with another different means of cooling (R3PR) in a selected configuration to boost efficiency. Adibhatla et al. by comparing the consequences of exergetic analysis when operation works in the constant pressure and pure sliding pressures found out that during sliding pressure, noticeable reduction in exergy losses and yet exergy efficiency increment for the gas cycle components can take place when throttle pressure is maintained. Ghaebi et al. also investigated the effects of system performance in CCHP focusing on pinch temperatures and pressures in the LP and HP evaporator of the HRSG. In the work of Ahmadi et al., Shahid Montazeri power plant with an eye toward repowering suggestion was studied by replacing high- and low-pressure steam flash heaters with two heat recovery heat exchangers as well as changing different gas turbine configurations. Akbari et al. also assessed the supply boiler repowering of an existing steam power plant by using energy-exergy analysis. In addition, Adibhatla et al. implemented energy, exergy, and economic (3E) analysis for evaluating an integrated solar direct steam generation combined-cycle power plant. Boyaghi et al. employed sensitivity analysis to investigate pressure ratio and TIT effects on advanced exergy assessment and found out both first and second law efficiencies increase when TIT and compressor pressure ratio goes up.

2 | THE POWER PLANT CYCLE SPECIFICATION

The present combined-cycle power plant which will be located Kuantan and Kapar in the Malay Peninsula contains three blocks consisting of the gas unit (frame 9F.05), one steam cycle which is driven by electric generators, accompanied by triple-pressure equipment in the heat recovery steam generator. A schematic diagram of this CCPP can be found in Figure 1. Air from the environment enters the air compressor at $P = 1.013$ bar and $T = 32°C$ and after compression with $PR = 18.3$ at the next state, it supplies to the combustion chamber in which by $17.48$ kg/s of fuel (natural gas) is injected to it. After expansion, the hot gases are utilized to provide the HRSG energy. Pressure drop and heat loss during the combustion process are assumed to be 3% and 2%, respectively. Noteworthy, the HRSG built of several heat exchangers is an essential connecting link between the steam and gas unit to generate superheated steam before exhaust gases reach the stack. The stack minimum temperature and pressure are $84.4°C$ and $1.013$ bar, respectively. Steam
turbines are operated at three levels on every evaporator in the steam cycle, at 4.8 bar (LP), 29.4 bar (IP), and 31.5 bar (HP). Owing to the reheating effect, the steam inlet temperatures for either HPT or IPT are identical which are equal to 560°C. There is an attemperator before the reheated steam entering IPT, which helps to adjust the steam temperature. Outlet expanded steam coming from LPT before reaching condenser is at 0.081 bar and 41.8°C. The temperature drop during the condensing process is approximately 2°C and back to the cycle (LP economizer) at 60°C and 6.8 bar. All of the performances and key data of the present CCPP have been gathered from the final report, prepared in February 2016 by Tokyo Electric Power Services Co., Ltd. for The Ministry of Economy, Trade, and Industry.39

The calculation is applied based on the following principles:

- At the dead state points, $P_0 = 1.01$ bar and $T_0 = 32°C$
- All the processes are assumed steady states.
- Heat losses and pressure drop from CC are deemed to be 2% of the LHV and 3%, respectively.
- Type of the fuel injected to the CC is natural gas
- The ideal-gas mixture basic assumptions are applied for the air together with flue gases.

2.1 Exergy analysis

The first law and the main exergy balance equations are defined below, respectively40:

$$\sum Q_k + \sum \dot{m} \left( h_i + \frac{C_i^2}{2} + gZ_i \right) = \sum \dot{m} \left( h_o + \frac{C_o^2}{2} + gZ_o \right)$$

(1)

$$\sum \left( 1 - \frac{T_0}{T} \right) Q_k + \sum (\dot{m}_i \Psi_i) = \sum \Psi_m + \sum (\dot{m}_o \Psi_o) + I_{\text{destroyed}}$$

(2)

In thermodynamics, the exergy of a system is the maximum amount of useful work possible which could be produced by a system in a reversible operation and this is the main property of all energy flows and streams depending on the characteristics of the system such as pressure, temperature, chemical features, environment, and even electric potential. Exergy destruction is an effective measure for the calculation of irreversibility which is the source of performance destruction. Hence, the exergy analysis is for calculating the proportion of exergy destruction which is useful for recognizing the proportion, element, and the exact source of thermodynamic inefficiencies in various systems. Therefore, exergy analysis is able to help optimizing and improving systems and designs.

In general, exergy for each system is defined as follows:

$$\dot{E_x} = \dot{E}_{ph} + \dot{E}_{ke} + \dot{E}_{pe} + \dot{E}_{ch}$$

(3)

The exergy of potential ($\dot{E}_{pe}$) and kinetic ($\dot{E}_{ke}$) are related to the arranged movements, and elevation of particles, respectively. There are no exergy losses or entropy differences in these two terms of exergy. Therefore, in this study, the potential and kinetic types of exergy are removed because of negligible effects on different components.

Equations of mass, energy, and exergy balance for a standard volume in steady-state disregarding potential and kinetic energy terms are respectively expressed as follows:

$$\sum \dot{m}_i = \sum \dot{m}_o$$

(4)

$$Q - W = \sum \dot{m}_o h_o - \sum \dot{m}_i h_i$$

(5)

$$W + \sum (\dot{m}_i \Psi_i) = Q + \sum (\dot{m}_o \Psi_o)$$

(6)

where the net exergy transferred by heat $\Psi_q$ at a specific temperature equals to:

$$\Psi_q = Q \left( 1 - \frac{T_0}{T} \right)$$

(7)

Also, the specific flow exergy for each streamline can be calculated by the following Equation40:

$$\Psi_{ph} = h - h_0 - T_0 (S - S_0)$$

(8)

where subscript 0 indicates the dead state, and T and S are stream temperature and specific entropy, respectively.

Then, the total exergy rate associated with a fluid stream at temperature $T_s$ can be approximately assessed in accordance with the following equation:

$$\dot{E}_{ph} = \dot{m} \Psi_m = \dot{m} \left[ (h - h_0) - T_0 (s - s_0) \right] = C_p \left[ (T - T_0) - T_0 \ln \left( \frac{T}{T_0} \right) \right]$$

(9)

where $C_p$ (specific heat capacity) can be calculated as follows41:

$$C_{p,\text{air}} = C_0 + C_1 T + C_2 T^2 + C_3 T^3$$

(10)

$$C_{p,\text{gases}} = 0.991615 + 6.997 \frac{T}{10^3} + 2.713 \frac{T^2}{10^6} - 1.224 \frac{T^3}{10^{10}}$$

(11)
TABLE 1 The exergy destruction rate and exergy efficiency equations for components of the power plant

| Component          | Exergy destruction rate | Exergy efficiency |
|--------------------|-------------------------|-------------------|
| Air compressor     | \( I = T_0 (S_{gen} - m_0 T_0 (S_{out} - S_a) \) | \( \eta_{AC} = 1 - \frac{I_{AC}}{W_{AC}} \) |
| Gas turbine        | \( I_{GT} = \Psi_{in} - \Psi_{out} - W_{GT} \) | \( \eta_{GT} = \frac{W_{GT}}{\Psi_{in} - \Psi_{out}} \) |
| Combustion chamber | \( I_{CC} = \Psi_{in} - \Psi_{out} + \Psi_{Fuel} \) | \( \eta_{CC} = \frac{\Psi_{out} - \Psi_{in}}{\Psi_{in} - \Psi_{out}} \) |
| Steam turbine      | \( I_{ST} = \sum \Psi_{in} - \sum \Psi_{out} - W_{ST} \) | \( \eta_{ST} = 1 - \frac{\Psi_{out} - \Psi_{in}}{\Psi_{in} - \Psi_{out}} \) |
| Condenser          | \( I_{HRSG} = \sum \Psi_{in} - \sum \Psi_{out} \) | \( \eta_{HRSG} = 1 - \frac{\Psi_{out} - \Psi_{in}}{\Psi_{in} - \Psi_{out}} \) |

For choosing each component as a control volume in the CCPP in the steady-state, exergy destruction and exergy efficiency can be calculated as it is shown in Table 1. Moreover, temperature after compressor due to irreversibility is defined as follows:

\[ T_2 = T_1 \left[ 1 + \frac{1}{\eta_{AC}} \left( \frac{S_{out} - S_a}{S_{gen} - S_a} \right) \right] \] (12)

Where \( \eta_{AC} \), \( PR_{AC} \), and \( \delta_a \) are pressure ratio, the efficiency of air compressor and, respectively.

Generally, the exergy destruction rate can be evaluated as follows:

\[ I = T_0 S_{gen} = m_0 T_0 \left( S_2 - S_1 \right) \] (13)

where entropy generation is calculated by the following equation

\[ (S_2 - S_1) = C_p \ln \frac{T_2}{T_1} - R \ln \frac{P_2}{P_1} \] (14)

where \( R_a \) is calculated as follows:

\[ R_a = C_{p,a} \left( \frac{\gamma_a - 1}{\gamma_a} \right) \] (15)

2.2 Combustion process

Combustion is a chemical reaction during which an intended fuel is oxidized and a high quantity of thermal energy is produced. During conducting thermodynamic analysis, it is essential to consider chemical internal energy while combustion is happening associated with the formation and destruction of chemical bonds between atoms. As a matter of fact, hydrocarbons denoted by the general formula of \( C_nH_m \) (eg, methane, ethane, etc) are the familiar and popular types of fuels, whereas they release significant greenhouse gases. Nitrogen which has the major percentage in the oxidizer does not take part in combustion, but at high temperatures, it may be converted to nitrogen oxides (NOx). In combustion analysis, the air is treated as a combination of \( O_2 \) and \( N_2 \). Thus, other gases (water vapor and \( CO_2 \)) are neglected and dry air is approximated as 21% of \( O_2 \) and 79% of \( N_2 \).

\[ 1 \text{ kmol } O_2 + 3.76 \text{ kmol } N_2 = 4.76 \text{ kmol air} \] (16)

A global combustion reaction utilizing \( CH_4 \) as the main fuel can be written as below:

\[ CH_4 + \left( 2O_2 + 7.52N_2 \right) \rightarrow CO_2 + 2H_2O + 7.52N_2 + \text{ heat} \] (17)

At higher temperatures, some nitrogen will be converted to nitrogen oxides (NOx)\(^2\) which is crucial in environmental analysis.

\[ CH_4 + 2O_2 + N_2 \rightarrow CO_2 + 2H_2O + N_2 + CO + NOX + \text{ heat} \] (18)

Combustion process analyses become much more rewarding when we understand although energy losses to the environment for every CCPP can be variable depending upon each operating condition, roughly over one-third of fuel useful energy which is destroyed during the combustion process cannot be utilized in electrical power production. Entropy generation has a direct relation with an increase of exergy destruction can be separated into three sub-processes including heat transfer which can be expressed as an internal thermal energy exchange, combined diffusion, and the product constituent mixing process.\(^3\) Combustion happens within the scope of elevated temperatures over which all the products of combustion (including the water vapor) behave as ideal gases.

2.3 Chemical exergy

Chemical exergy shows the maximum amount of useful energy possible which can be extracted when the streamflow changing from an environment state to a dead state. An environment state is defined when a particular system is thermally and mechanically at the equilibrium position, but not chemically. We can find chemical exergy of mixture gases at \( P_0 \) and \( T_0 \) from the following Equation\(^4\):

\[ ... \]
\[ \Psi_{ch} = \sum X_k \frac{\Psi_{K}^{ch}}{RT_0} \sum X_k \ln X_k, \quad (19) \]

where \( X_k, \frac{\Psi_{K}^{ch}}{RT_0} \), and \( \bar{R} \) are the molar fraction of gases in fuel which can be extracted from Table 2, specific chemical exergy which is evident in Table 3, and specific gas constant.

Specific chemical exergy can be calculated by the following table:

Also, the chemical exergy for fuel injected into the combustion chamber is defined as follows\(^{(44)}\):

\[ \Psi_{fuel} = \varphi LHV. \quad (20) \]

For most hydrocarbons, \( \varphi \) is near unity (\( \varphi_{CH_4} = 1.06 \)) and \( LHV_{CH_4} = 802361 \text{ kj/kmol} \).

To calculate the molar fraction of exhaust gases from the gas turbine including nitrogen, oxygen, carbon dioxide, and water vapor, the following equations are taking into consideration.

\[ X_{O_2} = \left( \frac{0.2059 - 2\lambda}{\lambda + 1} \right), \quad X_{CO_2} = \left( \frac{0.0003 + \lambda}{\lambda + 1} \right), \quad X_{H_2O} = \frac{0.019 + 2\lambda}{\lambda + 1} \quad \text{(21)} \]

The molar breakdown of the combustion gases which is known as \( \bar{\lambda} \) and is determined as follows:

\[ \bar{\lambda} = \frac{\bar{N}_F}{\bar{N}_a} \quad \text{and} \quad \bar{\lambda} + 1 = \frac{\bar{N}_F + \bar{N}_a}{\bar{N}_a} \quad \text{(22)} \]

### RESULTS AND DISCUSSION

In this part, the evaluation results based on the first and second laws are graphically prepared for the intended CCPP at varied reference temperatures, pressure ratios, condenser, HPT, and reheat pressures. The proper diagnosis by means of energy-exergy assessment can identify and evaluate the exact amount of deficiencies that help the power plant simulators to improve the powerplant processes. As is specified in Figure 1 the schematic figure, two different processes work with air and exhaust gases from GT. According to Table 4, for all the processes, the CCPP operation reference states remained at 32°C and 1.013 bar. Noted that the molar fraction of gases in the air has been specified in Table 5. By deducing from what has been discussed about second law analysis in Table 1, the results of exergy destruction rate, exergy proportion, and efficiency can be extracted from Table 6, Figure 2, and Figure 3 in which the combustion chamber per se has the major impact in exergy loss increment in the CCPP, by around 723 MW and 75% of total exergy destruction which can be related to the bigger quantity of entropy generation coming from a sharper temperature difference, chemical reactions, mixing, and large temperature differences. While air compressor with roughly 5.8 MW covers merely 0.62% of total wasted exergy, the HRSG also shows approximately

### TABLE 2 The proportion of exhaust gases from the combustion chamber

| Type of gas    | Molar fraction (%) |
|----------------|-------------------|
| Nitrogen       | 75.55             |
| Oxygen         | 13.46             |
| Carbon dioxide | 3.13              |
| Water vapor    | 7.86              |

### TABLE 3 Standard chemical exergy values of some combustion gases at \( P_0 \) and \( T_0 \)

| Constituent | \( \Psi_{K}^{ch} \) (kj/mol) | Constituent | \( \Psi_{K}^{ch} \) (kj/mol) |
|-------------|------------------------------|-------------|------------------------------|
| N\(_2\)     | 0.72                         | He          | 30.37                        |
| O\(_2\)     | 3.97                         | Ne          | 27.19                        |
| CO\(_2\)    | 19.87                        | Ar          | 11.69                        |
| H\(_2\)O    | 9.49                         | Kr          | 34.36                        |

### TABLE 4 Operating Conditions of the Power Plant

| Operating Condition        | Value            |
|----------------------------|------------------|
| Mass flow rate of fuel     | 17.48 kg/s       |
| Mass flow rate of air       | 605 kg/s         |
| Stack gas temperature      | 84.4°C           |
| Exhaust gas temperature    | 657.3°C          |
| Type of fuel               | Natural gas      |
| Mass flow rate of the make-up water | 121 kg/s               |
| Total outlet steam from HRSG | 68.36 kg/s       |
| Ambient temperature        | 32°C             |
| Ambient pressure           | 1.013 bar        |
| Thermal efficiency of CCPP | 55.5%            |
| Power plant power output   | 396 MW           |
| Efficiency of gas turbine  | 38.7%            |

### TABLE 5 Molar fraction of gases in the air

| Type of gas    | Molar fraction (%) |
|----------------|--------------------|
| Nitrogen       | 77.48              |
| Oxygen         | 20.59              |
| Carbon dioxide | 0.03               |
| Water vapor    | 1.9                |
110 MW of exergy destruction which is related to the flame high temperature, temperature differences among the streams, friction and the heat dissipated to the surroundings. It is followed by the gas turbine covering around 8% of total irreversibilities. The proportion of condenser is 2.54% attributed to the temperature difference between condenser water and the environment. Noteworthy, the steam turbines include 2.58% of total inefficiencies which can be due to the throttling in governing valves and wasted heat from its body. As expected, according to Figure 3 the combustion chamber, as well as HRSG, shows the least second law efficiency at 41% and 44%, respectively. As such, the greatest second law

| Component | Exergy Destruction (kW) | Percentage of exergy destruction (%) | Exergy Efficiency (%) |
|-----------|------------------------|-------------------------------------|-----------------------|
| Air compressor | 5826.7                  | 0.62                                | 96.31                 |
| Gas turbine  | 76926                   | 7.97                                | 84                    |
| Combustor   | 723880                  | 75.2                                | 41.5                  |
| HPT         | 2045.25                 | 0.21                                | 92.7                  |
| IPT         | 4073.47                 | 0.42                                | 92.1                  |
| LPT         | 18830                   | 1.95                                | 67.5                  |
| Condenser   | 24527.87                | 2.54                                | 90.83                 |
| HRSG        | 109149                  | 11.3                                | 44.15                 |
| Total       | 965258.29               | 100                                 | 100                   |

**FIGURE 2** Exergy destruction percent for all of the components in the CCPP

**FIGURE 3** Exergy efficiency for all of the components in the CCPP
efficiency is observed in the air compressor at 96% which may be pertinent to the widening gap between air compressor output and exergy destruction rate.

Figure 4 is comprised of the effect of the environment temperature variation (between 25°C, and 40°C) on the component exergy destruction rate when base pressure keeps at 1.013 bar. As is readily apparent, as the ambient temperature increases the exergy loss from all components goes up consequently apart from condenser reaching around 16.5MW at 40°C which may be attributed to the less energy loss to the coolant during the condensation process when the base temperature increases. In general, the more the temperature difference with the environment, the more adverse effect on overall system performance will be created. Nonetheless, the variation has a negligible effect on the exergy losses of triple-pressure steam turbines where these components see the slight upward trends as a result of the reheating system and the type of working fluid. The most immense increment can be found in the gas turbine in which the exergy loss increases by approximately more than 22MW which can be owing to the higher exergy input from the combustion chamber and entropy generation.

Variation of exergy destruction rate in the gas cycle components against pressure ratios from 10 to 20 is shown in Figure 5, wherein the ambient temperature is kept constant at 32°C. Firstly, the greater the pressure ratio, the higher the second law destruction is affected by it as the more entropy generation is produced as it has a direct relation with exergy loss. The biggest increment occurs in the gas turbine by nearly 80 MW which may be due to the bigger difference between the inlet and outlet temperature of the gas turbine. Whereas it slightly impacts combustion irreversibility(by 30MW) in the light of less temperature alteration, It must be underlined that the air compressor witnesses the least upward trend by 1.5 MW.

Figure 6 illustrates the molar fraction change against increasing ambient temperature when the heat loss, combustion pressure, and fuel mass flow rate keep at its reference point. The salient point is that although the temperature of the surrounding varies from 25°C to 40°C, produced hot gases from the combustor do not change significantly. Generally, the molar breakdown of the combustion gases ($\lambda$) which is defined as the molar fraction of fuel to air can play a pivotal role in changing the combustion gases molar fraction. Hence, it slightly drops when the temperature goes up. The negligible effect here consists of going up molar fractions of N₂ and decreasing this proportion for other gases. That is, the proportion of N₂ changes from 76.38 to somewhere more than 76.45 when the temperature reaches 40°C. One of the root causes of combustor considerable exergy degradation rate is the chemical exergy occurring when a substance is brought from the reference environment state to the dead state by a process including heat transfer and exchange of substances only with the reference environment. Thus, if we can properly control and adjust the fraction of the outlet gases from the combustor, we will be capable of identifying the proper proportions with lesser rate of inefficiencies in the CCPP. Therefore, by finding the proper interval of ambient temperature and investigating average temperature differences, the most suitable reference temperature interval accompanied by the lowest amount of chemical exergy can be extracted. When the base temperature changes from 25°C to 40°C, its average molar fraction can be found in Table 7.

As is illustrated in Figure 1, the steam cycle includes triple-pressure steam turbines which states of each of its inlet
stream plays an important role in changing component performance especially electricity production. Thus, we investigated the effects of HPT pressure variation on power output and the steam cycle overall efficiency. As is illustrated in Figure 7, the more main steam pressure, the less power output will be extracted from the bottoming cycle. As a matter of fact, the pressure increment can result in dropping the energy required to change the phase of water together with the latent heat of vaporization. Furthermore, less power will be generated owing to the expansion range increment across the high-pressure turbine, and consequently, the cycle efficiency decreases. It is also notable that seldom could you increase this pressure out of its certain range because the cycle efficiency may adversely change. Noteworthy, power output and cycle efficiency drop are approximately 1 MW and 0.05%, at every 10-bar inlet pressure increment.

Moreover, Figure 8 is comprised of condenser pressure variation impact on its cycle efficiency and power output. The performance of the steam cycle almost highly depends on the condenser pressure. As is clearly visible, both values reduce when it comes to condenser pressure increment. For every 0.01 bar condenser pressure increment, around 1 MW power generation and 0.07% of steam cycle efficiency is decreased. The underlying cause of this phenomenon is the increase of working fluid temperature in the heat dissipation process. Plus, when the condenser pressure increases, owing...
to the greater cooling water temperature, the steam will condense at a higher pressure leading to less expansion through the steam turbine, and hence reduced output, as well as efficiency, is extracted. In addition to the impact of pressure, the effective parameters which are affected by changing condenser pressure are condenser temperature, the cooling mass flow rate, and air leakage.

Initially, reheater typically is implemented to decrease the moisture existed in the steam and prevent the erosion of turbine blades during the expansion. Additionally, it is noted, the axiomatic upshot of the reheating process inside of the HRSG is the enhancing performance of the steam turbine. Simply put, the reheating process can change power production as well as the average temperature at which heat is increasing in the steam generator which highly depends on final feed temperature. As it is presented in Figure 9, the steam cycle power output is investigated when the temperature varies from 520°C to 640°C and its pressure remains stable. As the temperature increases for each 40°C, power production is increased by approximately 5.5 MW. It should be pointed out that the temperature of reheating steam must be kept within specified limits. Because higher reheat temperature leads to power generation increment, it can result in damage to reheater metal.
The stack temperature is the temperature of the combustion gases which are appeared as dry and water vapor which exit from HRSG and enters the chimney which is a ventilation structure. The underlying cause of wasting energy from the chimney in that energy is not transferred from the fuel to the steam or hot water. Stack energy due to its essential importance in changing plant performance, amount of wasted heat from flue gases, moisture loss, and particularly HRSG efficiency can play an important role in second law evaluation. In Figure 10 as the

### TABLE 7  Average molar fraction at $P_0 = 1.013$bar and $25°C < T_P<40°C$

| Type of gas      | Average molar fraction (%) |
|------------------|----------------------------|
| Nitrogen         | 76.42                      |
| Oxygen           | 17.57                      |
| Carbon dioxide   | 1.37                       |
| Water vapor      | 4.64                       |
stack temperature raises, the exergy destruction is increased consequently so that by 0.15 MW destruction drop for each around 4°C increment. By contrast, the second law efficiency experiences an upward trend to 39.5% at 226°C. On the whole, the self-evident parameters in stack temperature distinction can be boiler room temperature, operating pressure, and firing rate. Putting less heat into the boiler results in lower stack temperature leads to higher fuel-to-steam efficiency and a more effective heat exchanger. Therefore, a proper way to manipulate an efficiency value and less destruction rate can be used at the cooler time of the day.

4  |  CONCLUSION AND OPTIMIZATION SUGGESTIONS

In this article, the energy-exergy analysis of a gas-fired combined-cycle power plant components has been comprehensively investigated in which also the effects of different values on gas and steam cycles equipment have been analyzed. To sum up, the following points can be extracted from the current assessment:

- The major source of irreversibility is covered by the combustion chamber by around 724 MW (75.2%) which can be on account of the internal heat transfer, the mixing process, fuel oxidation, and sharper temperature difference in which by modifying molar fraction of the combustion gases or adding an air preheater, we can reduce chemical exergy, and consequently exergy destructed by combustion chamber. HRSG and combustion chamber constitute the least second law efficiency at around 41% and 46%, respectively.
- The reheating system has an essential impact on keeping exergy losses of steam turbines at their lower rates so that triple-pressure turbines merely cover nearly 2.58% of the overall rate exergy loss.
- Although base temperature increment leads to decreasing condenser exergy destruction, other components experience an increase of inefficiency. Thus, it is proposed to operate CCPP at the cooler time. Although there is a particular range to this change.
- The higher the pressure ratio with its design ratio, the more irreversibilities are affected by the gas cycle. By keeping low pressures, although not less than a specific range, we can reduce the destruction rate of exergy.
- The effect of base temperature on molar fractions of CO₂ and H₂O has been found to be higher than N₂ and O₂. Hence, based on Figure 6, by modifying reference temperature which affects the combustion gas fractions, we can control chemical exergy, and consequently irreversibility in the combustor.
- High-pressure streamline (HPT to RHT) in the steam cycle cannot be desirable as it has an adverse effect on the cycle efficiency enhancement, although it should not be exceeded a particular range.
- Keeping condenser pressure at a low level is significantly recommended to be taken into consideration during the power plant simulation because condenser pressure is directly linked to the amount of work done by the turbine. So it affects output and hence the efficiency of the cycle.
- The reheating process plays a pivotal role in augmenting the turbine performance as a great deal of irreversibility will be addressed due to the existing reheater.
- Figure 10 re-emphasizes that the lower the stack temperature, the more effective the heat exchanger design.

To sum up, from what has carried out above, the logical conclusion on average is that with energy-exergy evaluation, numerous considerations for the CCPPs can be implemented to identify irreversibilities and use several methods to improve plant performance. Nevertheless, such improvements are accompanied by either economic or environmental implications which will be investigated as this paper’s second part in the future.

NOMENCLATURE

| Symbol | Description |
|--------|-------------|
| eₓ     | Specific exergy [kJ/kg] |
| Ex, Ψ | Exergy[kJ] |
| Ex     | Exergy flow rate [kW] |
| ṁ      | Mass flow rate [kg/s] |
| T      | Temperature [°C, K] |
| P      | Pressure [Bar, kPa] |
| Cₚ     | Specific heat at constant pressure [kJ/kg K] |
| h      | Specific enthalpy [kJ/kg] |
| Qₓ     | Heat transfer rate [kW] |
| Xₑ     | Molar fraction [%] |
| E      | Energy [kJ] |
| r      | Pressure ratio |
| R      | Gas constant [J/g K] |
| h      | Specific enthalpy [J/kg] |
| s      | Specific entropy [kJ/kg K] |
| ηth    | Thermal efficiency [%] |
| Ψₑ     | Chemical exergy [kJ] |
| e      | Exergy efficiency [%] |
| I      | Exergy destruction rate [kW] |
| W      | Work [KJ] |
| δ      | Efficiency [%] |
| W      | Power [kW] |
| Eₑ     | Physical exergy [kJ] |
| Eₚₑ    | Physical exergy |
| Eₑₚₑ   | Potential exergy |
| Eₑₖₑ   | kinetic exergy |
References

1. Dincer I, Rosen MA. Exergy: energy, environment, and sustainable development. 2nd ed. Elsevier; 2013:1-547. https://www.sciencedirect.com/book/9780080970899/exergy

2. Ahmadi P, Dincer I, Rosen MA. Exergy, exergoeconomic and environmental analyses and evolutionary algorithm based multi-objective optimization of combined cycle power plants. Energy. 2011;36(10):5886-5898.

3. Dunbar WR, Lior N. Sources of combustion irreversibility. Combust. Sci. Technol. 1994;103(1–6):41-61.

4. Ahmadi G, Toghraie D, Akbari OA. Solar parallel feed water heating repowering of a steam power plant: A case study in Iran. Renew. Sustain. Energy Rev. 2017;77:474-485.

5. Jong Soo SYL. Optimization of heat recovery steam generator through exergy analysis for combined cycle gas turbine power plants. Int. J. energy Res. 2009;31(2007):135-147.

6. Nasrabad SN, Mehrpanahi A, Ahmadi G. Multi-objective optimization of HRSG configurations on the steam power plant repowering specifications. Energy. 2018;159:277-293.

7. Ahmadi G, Toghraie D, Akbari OA. Technical and environmental analysis of repowering the existing CHP system in a petrochemical plant: A case study. Energy. 2018;159:937-949.

8. Ahmadi GR, Toghraie D. Energy and exergy analysis of Montazeri Steam Power Plant in Iran. Renew Sustain Energy Rev. 2016;56:445-463.

9. Tiwari AK, Hasan MM, Islam M. Exergy analysis of combined cycle power plant: NTPC Dadri, India. Int. J. Thermodyn. 2013;16(1):36-42.

10. Regulagadda P, Dincer I, Naterer GF. Exergy analysis of a thermal power plant with measured boiler and turbine losses. Appl Therm Eng. 2010;30(8-9):970-976.

11. Ibrahim TK, Basrawi F, Awad OI, et al. Thermal performance of gas turbine power plant based on exergy analysis. Appl Therm Eng. 2017;115:977-985.

12. Kaushik SC, Reddy VS, Tyagi SK. Energy and exergy analyses of thermal power plants: A review. Renew Sustain Energy Rev. 2011;15(4):1857-1872.

13. Ahmadi G, Toghraie D, Akbari O. Energy, exergy and environmental (3E) analysis of the existing CHP system in a petrochemical plant. Renew Sustain Energy Rev. 2019;99(2018):234-242.

14. Toghraie D, Karami A, Afrand M, Karimipour A. Effects of geometric parameters on the performance of solar chimney power plants. Energy. 2018;162:1052-1061.

15. Ma Y, Fazlizadeh MA, Toghraie D, Talebizadehsardari P. Natural convection energy recovery loop analysis, part I: energy and exergy studies by varying inlet air flow rate. Heat Mass Transf. 2020;56:1685-1695. https://doi.org/10.1007/s00231-019-02766-x

16. Ersayin E, Ozgener L. Performance analysis of combined cycle power plants: A case study. Renew Sustain Energy Rev. 2015;43:832-842.

17. Ahmadi G, Toghraie D, Azimian A, Akbari OA. Evaluation of synchronous execution of full repowering and solar assisting in a 200 MW steam power plant, a case study. Appl Therm Eng. 2017;112:111-123.

18. Abuelnuor AAA, Saqr KM, Mohiedein SAA, Dafallah KA, Abdullah MM, Nogoud YAM. Exergy analysis of Garri ‘2’ 180 MW combined cycle power plant. Renew Sustain Energy Rev. 2017;79:960-969.

19. Izadyar N, Ong HC, Chong WT, Leong KY. Resource assessment of the renewable energy potential for a remote area: A review. Renew Sustain Energy Rev. 2016;62:908-923.

20. Bhandari B, Lee KT, Lee GY, Cho YM, Ahn SH. Optimization of hybrid renewable energy power systems: A review. Int J Precis Eng Manuf Green Technol. 2015;2(1):99-112.

21. Erdine O, Uzunoglu M. Optimum design of hybrid renewable energy systems: Overview of different approaches. Renew Sustain Energy Rev. 2012;16(3):1412-1425.

22. El Hadik AA. The impact of atmospheric conditions on gas turbine performance. J EngGas Turbines Power. 1990;112(4):590-596.

23. Ibrahim TK, Rahman MM, Abdalla AN. Optimum gas turbine configuration for improving the performance of combined cycle power plant. Procedia Eng. 2011;15:4216-4223.

24. Shi X, Agnew B, Che D, Gao J. Performance enhancement of conventional combined cycle power plant by inlet air cooling, inter-cooling and LNG cold energy utilization. Appl Therm Eng. 2010;30(14-15):2003-2010.

25. Ameri M, Ahmadi P, Hamidi A. Energy, exergy and exergoeconomic analysis of a steam power plant: A case study. Int J Energy Res. 2009;33(5):499-512.

26. Gupta MK, Kaushik SC. Exergetic utilization of solar energy for feed water preheating in a conventional thermal power plant. Int J Energy Res. 2009;33(6):593-604.

27. Farhad S, Saffar-Avmal M, Younessi-Simkani M. Efficient design of feedwater heaters network in steam power plants using pinch technology and exergy analysis. Int J Energy Res. 2009;32(1):1-11.

28. Cihan A, Hacihafozoglu O, Kahveci K. Energy-exergy analysis and modernization suggestions for a combined-cycle power plant. Int J Energy Res. 2006;30(2):115-126.

29. Vidal A, Best R, Rivero R, Cervantes J. Analysis of a combined power and refrigeration cycle by the exergy method. Energy. 2006;31(15):3401-3414.

30. Koch C, Czesla F, Tsatsaronis G. Optimization of combined cycle power plants using evolutionary algorithms. Chem Eng Process Process Intensif. 2007;46(11):1151-1159.

31. Sanjay Y, Singh O, Prasad BN. Energy and exergy analysis of steam cooled reheat gas-steam combined cycle. Appl Therm Eng. 2007;27(17–18):2779-2790.
32. Adibhatla S, Kaushik SC. Energy and exergy analysis of a super critical thermal power plant at various load conditions under constant and pure sliding pressure operation. Appl Therm Eng. 2014;73(1):51-65.
33. Ghaebi H, Amidpour M, Karimkashi S, Rezayan O. Energy, exergy and thermoeconomic analysis of a combined cooling, heating and power (CCHP) system with gas turbine prime mover. Int J Energy Res. 2011;35(8):697-709.
34. Ahmadi G, Toghraie D. Parallel feed water heating repowering of a 200 MW steam power plant. J Power Technol. 2015;95(4):288-301.
35. Akbari O, Marzban A, Ahmadi G. Evaluation of supply boiler repowering of an existing natural gas-fired steam power plant. Appl Therm Eng. 2017;124:897-910.
36. Adibhatla S, Kaushik SC. Energy, exergy and economic (3E) analysis of integrated solar direct steam generation combined cycle power plant. Sustain Energy Technol Assessments. 2017;20:88-97.
37. Boyaghchi FA, Molaie H. Sensitivity analysis of exergy destruction in a real combined cycle power plant based on advanced exergy method. Energy Convers Manag. 2015;99:374-386.
38. Boyaghchi FA, Heidarnejad P. Thermoeconomic assessment and multi objective optimization of a solar micro CCHP based on Organic Rankine Cycle for domestic application. Energy Convers Manag. 2015;97:224-234.
39. Tokyo L. Electric Power Services Co., “Study on Economic Partnership Projects in Developing Countries in FY2015”, Tokyo, 2016.
40. Calli O, Colpan CO, Gunerhan H. Chapter 2.3 - Performance Assessment of a Biomass-Fired Regenerative ORC System Through Energy and Exergy Analyses. Exergetic, Energetic and Environmental Dimensions. London, UK: Elsevier; 2018:253-257. https://www.sciencedirect.com/science/article/pii/B978012813734500159
41. Melek Y, Aytun OU. An energy benchmarking model based on artificial neural network method utilizing US Commercial Buildings Energy Consumption Survey (CBECS) database. Int J Energy Res. 2007;31(2007):135-147.
42. Speight JG. Handbook of Industrial Hydrocarbon Processes. Gulf Professional Publishing: Elsevier. 2019:1-806. https://www.elsevier.com/books/handbook-of-industrial-hydrocarbon-processes/speight/978-0-12-809923-0
43. Abam FI, Ugot IU, Igbong DI. Thermodynamic assessment of grid-based gas turbine power plants in Nigeria. J Emerg Trends Eng Appl Sci. 2011;2(6):1026-1033.
44. Dincer I, Rosen MA, Ahmadi P. Optimization of Energy Systems, 53, 9. Chichester, UK: John Wiley & Sons Ltd; 2017:1-453. https://onlinelibrary.wiley.com/doi/book/10.1002/9781118894484

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