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RESEARCH PAPER

Component-Level Energy and Exergy Assessment of a 378 MW Gas-Fired Combined Cycle Power Plant Using a Training Simulator and Engineering Equation Solver

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Abstract

Combined cycle power plants (CCPPs) generate a large share of Kuwait's electricity, but the location and size of their internal losses are seldom reported component by component. This study applies energy and exergy analysis to a nominal 378 MW natural-gas-fired CCPP (252 MW gas turbine and 126 MW steam turbine) reproduced by the SPM-5601 training simulator at the Higher Institute of Energy, Kuwait. Design-point data from the simulator were used as inputs to mass, energy and exergy balances solved in Engineering Equation Solver for the air compressor, combustion chamber, gas turbine, heat recovery steam generator (HRSG), high-, intermediate- and low-pressure steam turbines, and condenser. The total exergy destruction rate was 965.8 MW. The combustion chamber accounted for 724 MW (75.0%), followed by the HRSG (109.2 MW; 11.3%) and the gas turbine (77.1 MW; 8.0%), so that the gas turbine block was responsible for 83.6% of the destroyed exergy, while the three steam turbines and the condenser contributed only 2.6% and 2.5%. The combustion chamber also showed the lowest exergy efficiency (79.2%). Combustion irreversibility and the gas-to-steam temperature mismatch in the HRSG are therefore the main targets for improvement. The simulator-based procedure offers a repeatable tool for operator training and for screening operating changes before they are tried on a real unit. *2026 ijrei.com. All rights reserved*

1. Introduction

Kuwait generates its electricity almost entirely in fuel-fired thermal power stations, and its demand is dominated by air conditioning, which produces a sharp peak during the long, hot summer. Under these conditions every percentage point of plant efficiency translates directly into fuel savings and lower emissions. For this reason, new capacity in Kuwait, such as the Sabiya plant, has increasingly been built as combined cycle units, which convert a larger fraction of the fuel energy into electricity than simple-cycle gas turbines or conventional steam plants [1, 2]. A combined cycle couples a gas turbine (Brayton) topping cycle with a steam (Rankine) bottoming cycle. Advances in blade cooling and thermal-barrier coatings now allow gas turbine inlet temperatures above about 1400 °C, whereas live-steam temperatures in conventional steam plants remain close to 600 °C [3]. Because heat is added to the gas turbine at a much higher mean temperature, the topping cycle is thermodynamically attractive, but its exhaust still leaves the expander at roughly 500–600 °C. Passing this exhaust through a heat recovery steam generator (HRSG) to raise steam for a Rankine cycle recovers much of that energy, which is why combined cycle plants reach higher overall efficiencies than either cycle alone [3, 4]. Energy (first-law) analysis tracks how much energy enters and leaves each component, but it treats all forms of energy as equivalent and therefore cannot show where the useful work potential of the fuel is actually lost. Exergy is the maximum useful work that a system or stream can deliver while it comes into equilibrium with a defined reference environment (the dead state) [5, 6]. Unlike energy, exergy is

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not conserved: it is destroyed by irreversibilities such as chemical reaction, heat transfer across a finite temperature difference, mixing and friction, and it is lost with streams discharged to the surroundings. An exergy balance written for each component therefore separates internal destruction from external losses and ranks components by the work potential they consume [5–7]. This is the information designers and operators need to decide which equipment to improve first.

Plant simulators used for operator training reproduce the behaviour of an actual unit with a consistent, fully specified set of operating data. Such data are well suited to exergy analysis, because every stream needed to close the balances is available at a known operating point. The present study uses the SPM-5601 combined cycle simulator installed at the Higher Institute of Energy in Kuwait as the data source. Its objectives are (i) to quantify the exergy destruction rate and the exergy efficiency of each main component at the design point, (ii) to rank the components by their contribution to total exergy destruction, and (iii) to identify which components offer the largest scope for performance improvement.

1.1 Previous studies

Exergy-based assessment of combined cycle and steam plants has been reviewed by Kaushik et al. [4] and Ibrahim et al. [3]. Both reviews identify the combustion chamber as the dominant site of exergy destruction in gas-fired plants, with the HRSG and the condenser as the next most important locations of loss. Several studies have examined how the gas turbine cycle configuration affects the performance of the complete plant. Polyzakis et al. [8] compared simple, intercooled, reheat and intercooled–reheat gas turbine cycles as topping cycles and found that a reheat gas turbine gave the highest overall combined cycle efficiency. Sanjay et al. [9] carried out a component-wise energy and exergy analysis of a steam-cooled reheat gas–steam combined cycle and showed how the choice of blade coolant and the operating mode of the steam cycle change the distribution of losses among components.

Two studies have addressed the Sabiya combined cycle plant in Kuwait. Almutairi et al. [1] modelled the plant, which has a triple-pressure reheat HRSG, using operating data and found that 60.9% of the total exergy destruction occurs in the combustion chamber; they also examined the effects of climatic conditions, pressure ratio, pinch-point temperature and condenser pressure. In a follow-up study, Almutairi et al. [2] compared three gas turbine types as primary movers for the same plant and reported that the GT26 reheat engine gave the lowest waste exergy (43.93%, against 47.27% for the 9FA and 45.17% for the LM6000).

Case studies of plants elsewhere show the same pattern. Pinninti et al. [10] analysed the Gautami combined cycle plant in India using operating data and reported that the combustion chamber was responsible for the largest single exergy loss. Pinto et al. [11] simulated an 820 MW natural gas combined cycle plant in Brazil at design conditions and attributed about 70% of the irreversibilities to the combustion chamber. Elwardany et al. [12] reported that the combustion chambers of a 750 MW plant in Egypt accounted for 53.3% of the total exergy loss, and Topal and Tanbay [13] obtained thermal and exergy efficiencies of 59.70% and 58.52% for the renovated 1220 MW Hamitabat plant in Turkey. Omara et al. [14] extended conventional exergy analysis of the Garri 1 plant in Sudan with advanced exergy analysis, and Babaei Jamnani and Kardgar [15] evaluated the main components of a 396 MW plant in Malaysia and proposed operating guidelines for performance improvement.

Ambient temperature is particularly relevant for Gulf conditions. Wan and Chen [16] showed with operating data that the power output of a natural gas combined cycle plant fell by 22.6% and its energy efficiency from 57.28% to 56.3% as the ambient temperature rose from 5 to 35 °C, and Egware and Ighodaro [17] evaluated the effect of ambient air temperature on the exergy-based sustainability of a 153 MW gas turbine plant. Other work has focused on the HRSG and the steam side. In and Lee [18] optimised HRSG design by minimising the unavailable exergy, i.e., the sum of the exergy destroyed and the exergy lost through heat transfer in the HRSG. Naserabad et al. [19] applied multi-objective optimisation to HRSG configurations used for repowering steam power plants, and Ahmadi et al. [20] evaluated solar-assisted parallel feedwater heating as a repowering option for an Iranian steam plant. Farhad et al. [21] combined pinch technology with exergy analysis to redesign the feedwater heater network of steam power plants, and Ahmadi and Dincer [22] coupled exergy analysis with environmental impact assessment and genetic-algorithm optimisation of a cogeneration plant.

These studies confirm that component-level exergy analysis is an established route to locating performance limits. Most of them, however, rely on operating data from specific commercial plants that are not openly available for teaching or for repeating the analysis. The present work applies the same method to a fully documented simulator configuration, so that the results can be reproduced and extended in training and research.

2. Materials and Methods

2.1 Combined cycle power plant simulator

The plant analysed is the SIMTRONICS SPM-5601 combined cycle power plant simulator (DSS-100, version 7.0) operated by the Department of Mechanical Power at the Higher Institute of Energy, Kuwait [23]. The simulator is used to train combined cycle operators and comprises the following main systems (Figure 1):

- a natural-gas-fired gas turbine (air compressor, combustion chamber and expander) driving its own generator;
- a heat recovery steam generator with steam drum, reheater and duct burner;
- a three-casing steam turbine (high-, intermediate- and low-pressure sections) driving a second generator;
- a surface condenser with condensate pumps and hotwell;
- a boiler feedwater system with deaerator and feedwater pumps; and
- a cooling tower with circulating water pumps.

The design operating conditions supplied with the simulator

are listed in Table 1. At these conditions, the gas turbine generator delivers 252 MW and the steam turbine generator 126 MW, giving a nominal gross output of 378 MW. The fuel gas flow is 69.1 × 10^3 Sm^3/h, the expander exhaust leaves at 590 °C and the flue gas leaves the HRSG stack at 195.6 °C. The low-pressure turbine exhausts to the condenser at 0.081 bar and 41.8 °C.

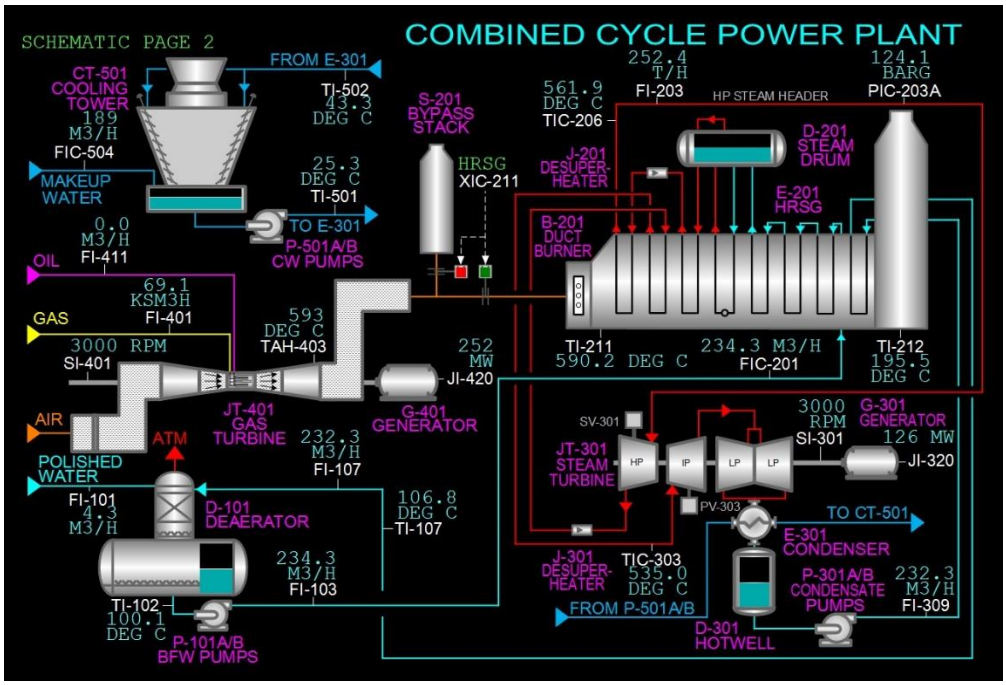


Figure 1: Process schematic of the SPM-5601 combined cycle power plant as displayed by the simulator

Table 1: Design operating conditions of the SPM-5601 combined cycle power plant simulator

No	Parameter	Value	Unit
1	Gas turbine power	252	MW
2	Steam turbine power	126	MW
3	Turbine shaft speed	3000	rpm
4	Generator shaft speed	1800	rpm
5	Frequency	50	Hz
6	HP steam produced	252.3	t/h
7	Condensate to deaerator	232.1	m^3/h
8	Fuel gas pressure	27.6	barg
9	Blowdown	1.96	t/h
10	MP steam to ejector	0.68	t/h
11	Expander exhaust temperature	590	°C
12	Fuel gas to combustion	69.1	10^3 Sm^3/h
13	BFW to users	234.1	m^3/h
14	Supply flow	7504	m^3/h
15	Filter flow	57.5	m^3/h
16	Make-up water	4.3	m^3/h
17	Deaerator vent	2.49	t/h
18	Air compressor discharge pressure	21.9	barg
19	Condensate return	232.1	m^3/h
20	IP steam reheater temperature	535	°C
21	Ambient temperature	20	°C
22	HRSG stack outlet temperature	195.6	°C

barg: gauge pressure; Sm^3: standard cubic metre; HP/IP/MP: high-, intermediate- and medium-pressure steam; BFW: boiler feedwater.

2.2 Computational procedure

Mass, energy and exergy balances for every component were coded in Engineering Equation Solver (EES) [24]. EES solves the full set of non-linear algebraic equations simultaneously and provides built-in thermodynamic property functions for water/steam, air and the combustion gas species, so stream enthalpies and entropies were evaluated directly from the simulator pressures, temperatures and flow rates. The calculated stream properties were then used to obtain the exergy of each stream and the exergy destruction and efficiency of each component. The reference (dead) state was taken as T_0 = 20 °C, the design ambient temperature of the simulator, and P_0 = 1.013 bar.

2.3 Assumptions

The analysis rests on the following assumptions:

- all components operate at steady state at the design point;
- air and combustion products behave as ideal-gas mixtures;
- the fuel is pure methane, treated as an ideal gas;
- combustion is complete, and nitrogen does not take part in the reaction;
- pressure drops in heat exchangers and connecting pipework are neglected;
- the HRSG is modelled as a dual-pressure unit;
- heat losses from component casings to the surroundings are neglected; and
- changes in kinetic and potential energy are negligible.

2.4 Governing equations

For each component treated as a control volume at steady state, the mass and energy balances are

$$\sum \dot{m}_{\text{in}} = \sum \dot{m}_{\text{out}} \tag{1}$$

$$\dot{Q}_{\text{cv}} - \dot{W}_{\text{cv}} = \sum \dot{m}_{\text{out}} h_{\text{out}} - \sum \dot{m}_{\text{in}} h_{\text{in}} \tag{2}$$

where $\dot{Q}_{cv}$ and $\dot{W}_{cv}$ are the heat transfer and power crossing the control-volume boundary and h is the specific enthalpy. The corresponding exergy balance is

$$\sum \left(1 - \frac{T_0}{T_k}\right) \dot{Q}_k + \sum \dot{m}_{\text{in}} \psi_{\text{in}} = \dot{W}_{\text{cv}} + \sum \dot{m}_{\text{out}} \psi_{\text{out}} + \dot{E}_{\text{D}} \tag{3}$$

in which the first term is the exergy transferred with heat $\dot{Q}_k$ at boundary temperature T_k, and $\dot{E}_D$ is the rate of exergy destruction caused by irreversibilities inside the component. The exergy associated with a heat transfer at temperature T is

$$\dot{E}^{Q} = \left(1 - \frac{T_0}{T}\right) \dot{Q} \tag{4}$$

Neglecting kinetic, potential and chemical contributions, the specific physical flow exergy of a stream and the corresponding exergy rate are

$$\psi = (h - h_0) - T_0 (s - s_0) \tag{5}$$

$$\dot{E} = \dot{m} \psi \tag{6}$$

where the subscript 0 denotes properties at the dead state and s is the specific entropy. For an ideal gas with constant specific heat c_p at a pressure close to the reference pressure, Eq. (5) reduces to the thermal exergy

$$\psi_T = c_p \left[(T - T_0) - T_0 \ln \frac{T}{T_0}\right] \tag{7}$$

The exergy destruction rate and the exergy efficiency of each component were obtained from the fuel–product definitions in Table 2 [5, 6], where $\dot{E}_{fuel}$ is the chemical exergy of the fuel supplied to the combustion chamber. The share of component k in the total exergy destruction of the plant is

$$y^*_{\text{D,k}} = \frac{\dot{E}_{\text{D,k}}}{\dot{E}_{\text{D,tot}}} \times 100 \tag{8}$$

Table 2: Exergy destruction rate and exergy efficiency of the plant components

Component	Exergy destruction rate and exergy efficiency
Air compressor	$\dot{E}_{\text{D,AC}} = \dot{W}_{\text{AC}} - (\dot{E}_{\text{out}} - \dot{E}_{\text{in}})$ $\varepsilon_{\text{AC}} = \frac{\dot{E}_{\text{out}} - \dot{E}_{\text{in}}}{\dot{W}_{\text{AC}}}$
Combustion chamber	$\dot{E}_{\text{D,CC}} = \dot{E}_{\text{air}} + \dot{E}_{\text{fuel}} - \dot{E}_{\text{gas}}$ $\varepsilon_{\text{CC}} = \frac{\dot{E}_{\text{gas}}}{\dot{E}_{\text{air}} + \dot{E}_{\text{fuel}}}$
Gas turbine	$\dot{E}_{\text{D,GT}} = (\dot{E}_{\text{in}} - \dot{E}_{\text{out}}) - \dot{W}_{\text{GT}}$ $\varepsilon_{\text{GT}} = \frac{\dot{W}_{\text{GT}}}{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}$
HRSG	$\dot{E}_{\text{D,HRSG}} = (\dot{E}_{\text{g,in}} - \dot{E}_{\text{g,out}}) - (\dot{E}_{\text{s,out}} - \dot{E}_{\text{w,in}})$ $\varepsilon_{\text{HRSG}} = \frac{\dot{E}_{\text{s,out}} - \dot{E}_{\text{w,in}}}{\dot{E}_{\text{g,in}} - \dot{E}_{\text{g,out}}}$
Steam turbines (HP, IP, LP)	$\dot{E}_{\text{D,ST}} = (\dot{E}_{\text{in}} - \dot{E}_{\text{out}}) - \dot{W}_{\text{ST}}$ $\varepsilon_{\text{ST}} = \frac{\dot{W}_{\text{ST}}}{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}$
Condenser and feedwater heaters	$\dot{E}_{\text{D,HX}} = (\dot{E}_{\text{h,in}} - \dot{E}_{\text{h,out}}) - (\dot{E}_{\text{c,out}} - \dot{E}_{\text{c,in}})$ $\varepsilon_{\text{HX}} = \frac{\dot{E}_{\text{c,out}} - \dot{E}_{\text{c,in}}}{\dot{E}_{\text{h,in}} - \dot{E}_{\text{h,out}}}$
Pumps	$\dot{E}_{\text{D,P}} = \dot{W}_{\text{P}} - (\dot{E}_{\text{out}} - \dot{E}_{\text{in}})$ $\varepsilon_{\text{P}} = \frac{\dot{E}_{\text{out}} - \dot{E}_{\text{in}}}{\dot{W}_{\text{P}}}$

Subscripts: in/out, inlet/outlet; g, exhaust gas; s, steam; w, feedwater; h and c, hot and cold streams of a heat exchanger (for the condenser, condensing steam and cooling water).

3. Results and Discussion

3.1 Distribution of exergy destruction among components

Table 3 and Figure 2 give the exergy destruction rate of each component at the design point. The plant destroys 965.8 MW of exergy in total. The combustion chamber accounts for 724 MW, more than six times the next largest value, which is found in the HRSG (109.15 MW). The gas turbine expander follows with 77.1 MW, while the condenser (24.6 MW), the low-pressure turbine (18.9 MW), the air compressor (5.83 MW) and the intermediate- and high-pressure turbines (4.1 and 2.1 MW) destroy comparatively little.

Table 3: Exergy destruction rate and share of total exergy destruction in the main components

Component	Symbol	$\dot{E}_D$ (MW)	y^*_D (%)
Air compressor	AC	5.83	0.60
Gas turbine	GT	77.10	7.98
Combustion chamber	CC	724.00	74.97
HP turbine	HPT	2.10	0.22
IP turbine	IPT	4.10	0.42
LP turbine	LPT	18.90	1.96
Condenser	CO	24.60	2.55
HRSG	HRSG	109.15	11.30
Total		965.78	100.00

Expressed as shares of the total (Eq. (8); Table 3 and the bar labels in Figure 2), the combustion chamber is responsible for 75.0% of the destroyed exergy, the HRSG for 11.3% and the gas turbine for 8.0%. The three components of the gas turbine block together account for 83.6%, while the whole steam turbine train (2.6%) and the condenser (2.5%) are minor contributors. This ranking agrees with the pattern reported for

other gas-fired combined cycle plants, where the combustor is consistently the largest source of irreversibility, with reported shares between about 53% and 70% [1, 3, 11, 12].

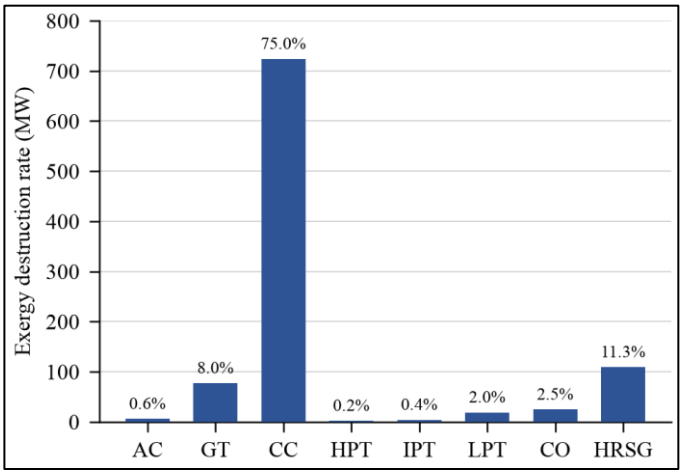


Figure 2: Exergy destruction rate (MW) of the main components; bar labels give the share of total exergy destruction (AC: air compressor; GT: gas turbine; CC: combustion chamber; HPT/IPT/LPT: high-, intermediate- and low-pressure turbines; CO: condenser)

The dominance of the combustion chamber is a consequence of the process itself rather than of a fault in the equipment. Converting the chemical exergy of methane into thermal energy of the combustion gas involves an unrestrained chemical reaction, internal heat transfer across very large temperature differences, and mixing of fuel, air and products, each of which generates entropy [5, 6]. Part of this destruction can be avoided only by raising the temperature at which the products leave the combustor, i.e., the turbine inlet temperature, or by preheating the reactants.

The HRSG is the second largest source of destruction. Exhaust gas enters at 590 °C and gives up its heat to water and steam that evaporate at a fixed saturation temperature, so a large temperature difference remains over much of the heat-transfer surface. In addition, the gas leaving the stack at 195.6 °C still carries exergy that is rejected to the surroundings. Both effects explain why researchers have focused on exergy-based optimisation of HRSG design parameters, such as the number of pressure levels and the pinch and approach temperatures [1, 18, 19]. The condenser behaves differently. In energy terms it rejects a large amount of heat, but this heat is released at 41.8 °C, only slightly above the 20 °C dead state, so its work potential is small. The condenser therefore appears as a minor exergy destroyer (2.5%), illustrating why energy analysis alone would overstate the cold end's improvement potential and understate the combustors.

3.2 Exergy efficiency of the components

Table 4 and Figure 3 report the percentage exergy destruction rate and the exergy efficiency obtained for each component, including the feedwater pumps and heaters. The combustion chamber again stands out, with the highest destruction rate (41.6%) and the lowest exergy efficiency (79.2%). The low-pressure steam turbine (86.7%) and the HRSG (87.3%) have the next lowest efficiencies, whereas the gas turbine (97.5%), the feedwater heaters (97.6%) and the condenser (98.5%) show the highest values. The feedwater heaters have a destruction rate of 9.7%, the second highest in Table 4.

Table 4: Exergy destruction rate and exergy efficiency of the plant components

Component	Exergy destruction rate (%)	Exergy efficiency (%)
Air compressor	2.70	90.40
Gas turbine	1.60	97.50
Combustion chamber	41.60	79.20
HP steam turbine	0.97	90.00
LP steam turbine	3.10	86.70
Feedwater pumps	0.04	94.51
Feedwater heaters	9.70	97.60
Condenser	2.15	98.50
HRSG	7.40	87.30

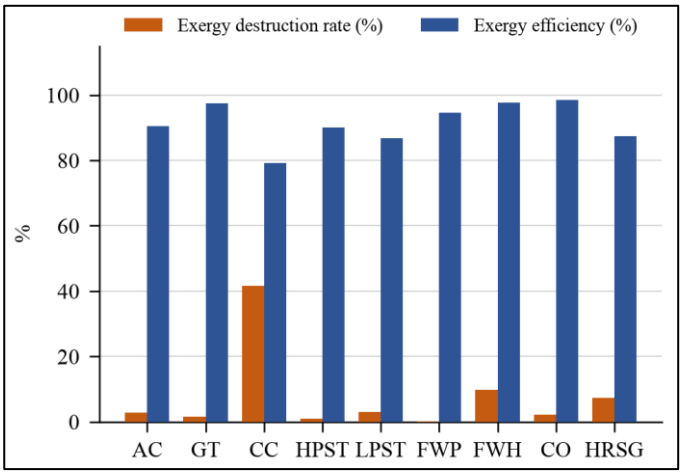


Figure 3: Exergy destruction rate and exergy efficiency of the plant components (HPST/LPST: high- and low-pressure steam turbines; FWP: feedwater pumps; FWH: feedwater heaters)

3.3 Implications for performance improvement

Because more than four-fifths of the destroyed exergy is concentrated in the gas turbine block, measures acting on the topping cycle have the largest potential effect on overall plant performance. The following options follow from the present ranking together with findings reported in the literature; their quantitative benefit for the SPM-5601 configuration was not evaluated in this study.

3.3.1 Higher turbine inlet temperature

Raising the firing temperature reduces the temperature gap across which combustion takes place and increases the exhaust temperature available to the HRSG. This requires improved blade cooling, for example steam cooling as analysed by Sanjay et al. [9].

3.3.2 Reheat gas turbine cycles

Polyzakis et al. [8] found that reheat in the gas turbine gives the highest combined cycle efficiency among the configurations they compared, and the reheat GT26 engine gave the lowest waste exergy for the Sabiya plant [2].

3.3.3 Compressor inlet-air cooling

Summer air temperatures in Kuwait frequently exceed 45 °C. Since combined cycle output and efficiency fall as ambient temperature rises [16, 17], cooling the intake air lowers compressor work and restores gas turbine output.

3.3.4 Fuel preheating

Using low-grade heat to preheat the fuel gas reduces the temperature difference between reactants and products in the combustor.

3.3.5 HRSG optimization

Adding a pressure level, reducing pinch-point temperature differences or lowering the stack temperature where acid-dew-point limits allow would reduce both the internal destruction and the stack loss identified in Section 3.1 [18, 19].

4. Conclusions

A component-level energy and exergy analysis of a nominal 378 MW gas-fired combined cycle plant was carried out using design-point data from the SPM-5601 training simulator and balances solved in EES. The main conclusions are as follows.

- The plant destroys 965.8 MW of exergy at the design point.
- The combustion chamber is the dominant source of irreversibility, with 724 MW (75.0% of the total), and it also has the lowest exergy efficiency (79.2%).
- The HRSG (11.3%) and the gas turbine expander (8.0%) are the next largest contributors; the gas turbine block as a whole account for 83.6% of the total.
- The steam turbines (2.6% combined) and the condenser (2.5%) destroy little exergy, although the condenser rejects a large quantity of low-temperature energy.
- Improvement efforts should therefore target combustion irreversibility (turbine inlet temperature, reheat, fuel and air preconditioning) and the gas-to-steam temperature match in the HRSG.

The main advantage of the approach is that a training simulator supplies a complete and consistent data set, so the analysis can be reproduced and extended without access to proprietary plant records. Its limitations are that the results refer to a single design-point condition reproduced by a simulator rather than to measured data from an operating unit, that no uncertainty analysis was performed, and that off-design operation, ambient temperature variation and the costs of the improvement options were not quantified. The model can be applied in operator and engineering training, and for screening operating changes before they are tried on a real unit. Future work will extend it to parametric studies of ambient temperature, compressor pressure ratio, turbine inlet temperature and condenser pressure, and to an exergoeconomic assessment.

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