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Thermodynamic analysis of a combined cycle power plant located in Jordan: A case study

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Abstract Efficiency and electrical power output of combined cycle power plants vary according to the ambient conditions. The amount of these variations greatly affects electricity production, fuel consumption, and plant incomes. Obviously, many world countries have a wide range of climatic conditions, which impact the performance of power plants. In this paper, a thermodynamic analysis of an operating power plant located in Jordan is performed with actual operating data acquired from the power plant control unit. The analysis is performed by using first and second laws of thermodynamics. Energy and exergy efficiencies of each component of the power plant system are calculated and the effect of ambient temperature on the components performance is studied. The effects of gas turbine pressure ratio, gas turbine inlet temperature, load and ambient conditions on the combined cycle efficiency, power outputs and exergy destruction are investigated. Energy and exergy efficiencies of the combined cycle power plant are found as 45.29%, and 42.73% respectively when the ambient temperature is 34 °C. Furthermore, it is found that the combustion chamber has the largest exergy destruction rate among the system components. The results showed that 73% of the total exergy destruction occurs in the combustion chamber when the ambient temperature is 34 °C. Moreover, the results show that the second major exergy loss is in HRSC. The results show that the energy and exergy efficiency of the combined cycle power plant decreases as the ambient temperature increases. According to the calculation results, improvement and modification suggestions are presented.

Keywords: Combined cycle power plant; Energy efficiency; Exergy efficiency; Exergy destruction; Exergy losses

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Nomenclature

C_p	–	specific heat at constant pressure, kJ/kgK
C_V	–	specific heat at constant volume, kJ/kgK
$\dot{E}$	–	energy rate, kW
h	–	enthalpy, kJ/kg
LHV	–	lower heating value, MJ/kg
$\dot{m}$	–	mass flow rate, kg/s
P	–	pressure, bar
$\dot{Q}$	–	heat flow, kW
$\dot{q}$	–	specific heat flow, kJ/kg
R	–	gas constant, J/g K
r	–	pressure ratio
s	–	entropy, kJ/kgK
T	–	temperature, °C
W	–	work, kJ
$\dot{W}$	–	power, kW
w	–	specific work, kJ/kg
$\dot{X}$	–	exergy rate, kW

Greek symbols

η	–	efficiency
ζ	–	fuel factor
ψ	–	specific exergy, kJ/kg

Subscripts

a	–	air
CC	–	combustion chamber
$CCPP$	–	combined cycle power plant
des	–	destruction
f	–	fuel
GT	–	gas turbine
g	–	exhaust gas
in	–	inlet
out	–	exit
o	–	dead state
SC	–	simple cycle
ST	–	steam turbine

Abbreviations

ACC	–	air cooled condenser
CC	–	combustion chamber

CCPP	–	combined cycle power plant
Ec	–	economizer
Evap	–	evaporator
ex	–	exergy
FW	–	feed water
GT	–	gas turbine
HP	–	high pressure
HRSG	–	heat recovery steam generator
IP	–	intermediate pressure
LP	–	low pressure
SC	–	simple cycle
SH	–	super heater
ST	–	steam turbine
SEPCO	–	Samara Electric Power Company
th	–	thermal

1 Introduction

Gas turbines (based on the Brayton cycle) are increasingly used in combination with steam turbines (based on the Rankine cycle), either to generate electricity alone in combined cycles, or to produce in cogeneration both electrical power and heat for industrial processes [1]. A combined cycle featuring one or several gas turbines and a steam cycle is a power plant option commonly used for power production that offers a high efficiency. Obviously, the combined power cycle plants play an important role in the present energy sector. This cycle has a higher thermal efficiency than either of the cycles executed individually so it is of greatest interest of all [2].

Generally, the performance of thermal power plants is evaluated through energy analysis based on the first law of thermodynamics, including thermal efficiency of the simple cycle and combined cycle. Energy efficiencies can be non-intuitive or even deceptive [3]. Energy losses during the operation process can be large, and the amount of lost energy is thermodynamically useless due to its low quality. Recently, the exergy analysis based on the second law of thermodynamics is used as a useful method in evaluation and improvement of the power plants. The exergy efficiencies and destruction provide measures of approach to ideality. Such results allow unlimited opportunities for improvements. Exergy analysis can determine the magnitude and location of losses in the power plants and within individual components. Exergy analysis deals with the quality of energy but energy analysis deals with quantity of energy. Therefore, complete system characteristics can be calculated using both energy and exergy analysis.

Several studies are found in the literature that used the energy and exergy analysis of thermal power plants [1–25]. Moreover, most of these studies have focused on the performance analysis and performance enhancement of the combined cycle power plant (CCPP) using various technologies. Ibrahim *et al.* presented a review of the latest thermodynamics analysis of each system components of a CCPP independently and determine the exergy destruction of the plant [4]. They found that most of energy losses occurred in the condenser, while the highest exergy destruction occurred in the combustion chamber (CC). Moreover, based on the review, it was found that the increasing ambient temperature causes an evident decrease in the power production by the gas turbine. Finally, they proved that both energy and exergy analyses should be used to enhance the performance of CCPP. Kakaras reported that the gas turbine output and efficiency is a strong function of the ambient air temperature [5]. He found that depending on the gas turbine type, the power output is reduced between 5% and 10% of the ISO-rated power output (15 °C) for every 10 °C increase in ambient air temperature. Ersayin *et al.* studied the combined cycle power plant located in Turkey using the first and second laws of thermodynamics [6]. Energy and exergy efficiencies of each component of the power plant system were investigated. Most of exergy destruction was found in the combustion chamber. Energy and exergy efficiencies of the combined cycle power plant were found as 56% and 50.04% respectively. Petrakopoulou *et al.* analyzed a combined cycle power plant using both conventional and advanced exergetic analyses [7]. It was found that 87% of the total exergy destruction takes place within the combustion chamber component. Furthermore, about 68% of the total exergy destruction in the combustion chamber cannot be avoided (unavoidable exergy destruction). Almutairi *et al.* studied an actual combined cycle power plant (CCPP) in Kuwait using energy and exergy analysis [8]. The plant has an advanced triple pressure reheat heat recovery steam generator (HRSG). They used sensitivity analysis of HRSG to differentiate between the sources of irreversibility. Aljundi calculated energy and exergy losses of a steam power plant located in Jordan using energy and exergy analysis [9]. They found that the major exergy destruction takes place in the boiler system where 77% of the fuel exergy input to the cycle was destroyed. Srinivas *et al.* presented optimization analysis of the performance of CCPP with different types of HRSG having a single pressure, dual pressure, and triple pressure reheat [10]. They found that the optimal HRSG configuration led to improvement in steam generation and therefore, in the output of the steam turbine.

The previous studies on combined cycle power plant have divided the exergy into four main groups as: physical, chemical, kinetic and potential exergy. The kinetic and potential exergy are found to be insignificant components through the analysis and were ignored. On the other hand, the physical and chemical exergy were found to be significant components of the exergy through the analysis [11–14]. The chemical exergy of a substance is the highest work available that can be obtained at standard conditions at constant pressure and temperature (1 atm and 25 °C respectively) [15]. Hence, the chemical exergy is considered the most important quantity in the combustion process. Any deviations of the pressure and temperature from the standard conditions lead to the generation of the physical exergy [16].

Dhar Garg *et al.* evaluated the performance of a combined cycle cogeneration configuration in India based on energy and exergy analysis approaches [16]. They investigated the effect of operating conditions on combined cycle efficiency, power outputs, and exergy destruction. Woudstra *et al.* studied three different heat recovery steam generators (HRSG) [17]. These HRSG systems are designed and modeled using the computer program called Cycle-Tempo. They found out that the highest losses in the gas turbine are caused by (thermal) combustion of fuel. Furthermore, the triple pressure system has a higher efficiency when compared with a single pressure system. This is due to the reduction of the exergy loss of heat transfer in HRSG. Bagdanavicius *et al.* found out that the combined cycle power plant (CCPP) is the most exergy efficient system with the lowest exergy cost of electricity and heat produced [18].

Integrating vapor compression and vapor absorption cooling systems to a combined cycle plant for inlet air cooling is investigated by [19]. They found that inlet air cooling can lead to an increase of the plant's specific power output by 9.02%. However, to operate the vapor compression system, power is extracted from the gas turbine output. Boonnasa *et al.* integrated a steam operated absorption chiller to a CCPP for the compressor inlet air cooling [20]. The gas turbine power output increased by about 10.6%, though the power output of the steam turbine was decreased by about 2.43% due to the steam extracted to operate the absorption chiller. Integration of a chilled-water thermal energy storage system to precool the inlet air of CCPP is proposed by [21,22]. The temperature of the thermal energy storage system was maintained by an absorption refrigeration system powered by waste heat from the flue gases of CCPP. Although there is an improvement, a large volume of the storage tank limits the feasibly of this integration. Oko and Njoku investigated the performance of an existing combined cycle power plant augmented with a waste heat fired organic Rankine cycle power plant for extra power generation [23]. They found out that exergy and energy efficiencies of the thermal power plant improved by 1.95% and 1.93%, respectively.

In this work, energy and exergy analysis of actual power plant is performed with actual operating data acquired from the power plant control unit. Moreover, the effect of seasonal operating conditions is studied. The analysis is performed by using the first and second laws of thermodynamics. Energy and exergy efficiencies of each component of the power plant system are calculated.

2 Power plant studied

The basic components of phase 3 of the Samra Electric power plant considered in this study are shown in Fig. 1. The actual operating data of a combined cycle power plant located in Jordan are considered in this study. The power plant has a total power capacity of 1100 MW, covering almost 40% of Jordan load. It has four phases: phase one, phase two, phase three and phase four. Phase three of the Samra Electric Power Company is studied in this paper. Phase three of the plant has two Alstom gas turbine units, two heat recovery steam generators (HRSG) and one Alstom steam turbine with total power about 430 MW. Fresh air at ambient conditions enters into the compressor, where its pressure and temperature are raised. The compressed air enters into the combustion chamber and is burned with the fuel. The high energy and enthalpy combustion gases expand in the gas turbine to produce power. The exhaust gases from the gas turbine can be used as the energy source in the bottoming cycle. The energy from the exhaust gases is recovered by heat recovery steam generators to produce steam. The produced steam in HRSG enters into the steam turbine to produce work, and to be finally cooled by an air cooled condenser.

The Samra Electric Power Company has two 142×2 MW gas turbines and one 140 MW steam turbine. The combustion gases enter the gas turbines at 1095 °C and steam enters the steam turbine at three pressure levels, high pressure (490 °C, 105.2 bar), intermediate pressure + reheat (480 °C, 21.5 bar) and low pressure (262 °C, 3.3 bar). The specification of the gas turbine is listed in Tab. 1 at full load running on natural gas as a fuel [24], whereas the specification of the steam turbine is listed in Tab. 2. The molar percentage composition of natural gas that is used in the Samra Electric Power Company is listed in Tab. 3.

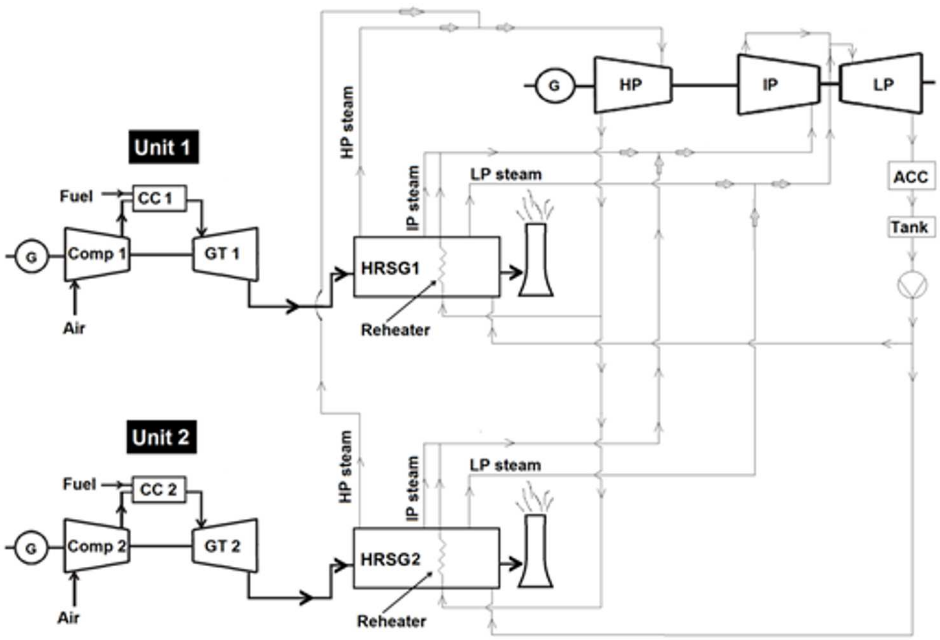

Figure 1: The schematic diagram of basic components of phase 3 of Samra Electric Power Company.

Table 1: Specification of gas turbine [24].

Model	Alstom GT13E2 (2005)
Net heat rate	9524 kJ/kWh, LHV
Number of compressor stages	21 stages
Compressor pressure ratio (r)	16.9
Number of turbine stages	5 stages
Turbine inlet temperature (TIT)	1095 °C
Temperature after turbine (TAT)	505 °C
Frequency	50 Hz
Shaft speed	single shaft 3000 rpm
Main fuel	Natural gas
Secondary fuel	Fuel oil (diesel)

Table 2: Specification of steam turbine [24].

Model	Alstom
Nominal power	140 MW
Frequency	50 Hz
shaft speed	single shaft 3000 rpm
HP steam Temperature & pressure	490 °C, 105.2 bar
IP + reheat steam Temperature & pressure	480 °C, 21.5 bar
LP steam Temperature & pressure	262 °C, 3.3 bar
air cooled condenser Temperature & pressure (ACC)	57.3 °C, 0.176 bar

Table 3: Natural gas composition.

Components	Molar percentage
Methane	99.85%
Propane	0.07%
Nitrogen	0.08%

3 Thermodynamic analysis

Thermodynamic analyses of the thermal power plant are performed using both the first and second law of thermodynamics. The exergy analysis is based on the second law of thermodynamics. There are two forms of energy that are transferred to or from a system: work $\dot{W}$, and heat $\dot{Q}$.

3.1 Energy analysis

Energy analysis for the combined cycle power plant was applied to two cycles; the Brayton cycle (top cycle) and the Rankine cycle (bottom cycle). The mass and energy conservation equations applied to a control volume after ignoring kinetic and potential energy are:

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

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

The thermal efficiency of the Brayton cycle is given as

$$\eta_{th,GT} = \frac{\dot{W}_{net,GT}}{\dot{Q}_{cc}} \quad \text{or} \quad \eta_{th,GT} = \frac{\dot{w}_{net,GT}}{\dot{q}_{cc}} \, . \tag{3}$$

Thermal efficiencies of the combined cycle are given as

$$\eta_{th,CCPP} = \frac{\dot{W}_{net,GT} + \dot{W}_{ST,net}}{\dot{Q}_{cc}} \, . \tag{4}$$

The power consumed by the compressor in the Brayton cycle is

$$\dot{W}_{comp,in} = \dot{m}_a(h_2 - h_1) = \dot{m}_a C_{pa}(T_2 - T_1) \, , \tag{5}$$

where T_1 and T_2 are air temperatures at the inlet and outlet of the compressor, respectively. C_{pa} denotes the average specific heat of the air between T_1 and T_2, given as

$$\begin{aligned} C_{pa}(T) &= 1.04841 - \frac{3.83719}{10^4}T + \frac{9.45378}{10^7}T^2 \\ &- \frac{9.45378}{10^{10}}T^3 + \frac{7.92981}{10^{14}}T^4 \, . \end{aligned} \tag{6}$$

The mass conservation in the combustion chamber is

$$\dot{m}_g = \dot{m}_a + \dot{m}_f \quad \text{or} \quad \dot{m}_3 = \dot{m}_2 + \dot{m}_5 \, , \tag{7}$$

where $\dot{m}_g, \dot{m}_a,$ and $\dot{m}_f$ denote the combustion gases flow rate ($\dot{m}_3$), air flow rate ($\dot{m}_2$) and fuel flow rate ($\dot{m}_5$), respectively. The heat supplied by the combustion chamber is

$$\dot{Q}_{cc} = \dot{m}_f \mathrm{LHV} \, , \tag{8}$$

where LHV is the average lower heat value for the natural gas. The generated power by the gas turbine is

$$\dot{W}_{GT,out} = \dot{m}_4(h_3 - h_4) = \dot{m}_g C_{pg}(T_3 - T_4) \, , \tag{9}$$

where T_3 and T_4 are air temperatures at the inlet and outlet of the turbine, respectively. C_{pg} denotes the average specific heat of the combustion gases between T_3 and T_4 given as [25]

$$C_{pg}(T) = 0.991615 - \frac{6.99703}{10^5}T + \frac{2.7129}{10^7}T^2 - \frac{1.22442}{10^{10}}T^3 \, . \tag{10}$$

The net generated power by the simple cycle is

$$\dot{W}_{net,GT} = \dot{W}_{GT,out} - \dot{W}_{comp,in} \, . \tag{11}$$

The heat recovered by HRSG (water cycle side) is given by

$$\begin{aligned} \dot{Q}_{HRSG} &= \dot{m}_{19}(h_{19} - h_{16}) + \dot{m}_{10}(h_{10} - h_{16}) + \dot{m}_{10}(h_{15} - h_{10}) \\ &+ \dot{m}_{6}(h_{6} - h_{16}) + \dot{m}_{9}(h_{9} - h_{6}) + (\dot{m}_{14} - \dot{m}_{13})(h_{15} - h_{21}). \end{aligned} \tag{12}$$

The specific heat recovered by HRSG (exhaust gas from the gas turbine) is given as

$$\dot{q}_{HRSG} = C_p(T_{HRSH,in} - T_{HRSH,out}) = C_p(T_4 - T_{HRSH,out}) \, , \tag{13}$$

where C_p is the specific heat averaged between the temperature at the HRSG inlet and outlet. The mass flow rate of gases in HRSG and gas turbine is

$$\dot{m}_g = \frac{\dot{Q}_{HRSG}}{\dot{q}_{HRSG}} \, . \tag{14}$$

The generated power by the steam turbine is

$$\dot{W}_{ST,out} = \dot{m}_{20}h_{20} - \dot{m}_{21}h_{21} + \dot{m}_{22}h_{22} + \dot{m}_{23}h_{23} - \dot{m}_{24}h_{24} \, . \tag{15}$$

The net generated power by the Rankine cycle can be found as

$$\dot{W}_{ST,net} = \dot{W}_{ST,out} - \dot{W}_{cons} \, , \tag{16}$$

where the consumed power is

$$\dot{W}_{cons} = \dot{W}_{FWpump,in} + \dot{W}_{condensate.pump,in} + \dot{W}_{ACC} + \dot{W}_{auxiliary.system} \, . \tag{17}$$

4 Exergy analysis

Exergy represents the upper limit of the amount of work a device can deliver without violating any thermodynamic laws [1]. In other words, exergy is the maximum useful work that can be obtained from the system at a given state in a specified environment. Exergy destruction is the wasted work during a process between two specified states. Reversible work of the turbine is the maximum work output at the minimum work of the pump and compressor. This can be achieved when exergy destruction of the components equals zero. In this study, potential and kinetic exergy are neglected, physical and chemical exergy are included for each component in the combined cycle power plant. In the combined cycle power plant, exergy is transferred by work, heat and mass flow. Exergy transfer by work is

$$\dot{X}_{work} = \dot{W} \; . \tag{18}$$

Exergy transfer by heat is

$$\dot{X}_{heat} = \left(1 - \frac{T_0}{T_i}\right)\dot{Q}_i \, , \tag{19}$$

where subscript i denotes hot source.

Exergy transfer by mass flow is

$$\dot{X}_{mass} = \dot{m}\psi \; , \tag{20}$$

where ψ is the specific exergy for water and steam given as

$$\psi = (h - h_0) - T_0(s - s_0) \; . \tag{21}$$

The specific exergy for air and combustion gases (as ideal gas) is

$$\psi = C_p\left[T - T_0 - T_0 \ln\left(\frac{T}{T_0}\right)\right] + RT_0 \ln\left(\frac{P}{P_0}\right) \; , \tag{22}$$

where R is the gas constant given as

$$R = C_P - C_V = C_p\left(1 - \frac{1}{K}\right) \; , \tag{23}$$

where K is the specific heat ratio.

Chemical specific exergy for fuel is

$$\psi = \zeta \mathrm{LHV}_f \; . \tag{24}$$

In general fuel factor $\zeta = 1.06$ for natural gas [6]. The exergy efficiency of the Brayton cycle is

$$\eta_{ex,SC} = \frac{\dot{W}_{net,GT}}{\dot{X}_f} = \frac{\dot{w}_{net,GT}}{\dot{x}_f} \; , \tag{25}$$

where $\dot{x}_f$ is the specific exergy for fuel.

The exergy efficiency of the combined cycle is

$$\eta_{ex,CCPP} = \frac{\dot{W}_{net,GT} + \dot{W}_{ST,net}}{\dot{X}_f} \,. \tag{26}$$

A general form for the steady state exergy balance equation is

$$\dot{X}_{in} = \dot{X}_{out} + \dot{X}_{destroyed} \,. \tag{27}$$

Exergy analysis is applied to both cycles in the combined cycle power plant (Brayton and Rankine cycle). The destroyed exergy of the compressor ($\dot{X}_{destroyed,comp}$) can be calculated by applying the exergy balance equation as

$$\dot{X}_1 + \dot{W}_{comp,in} = \dot{X}_2 + \dot{X}_{destroyed,comp} \,. \tag{28}$$

Then, the exergy efficiency of the compressor is

$$\eta_{ex,comp} = \frac{\dot{W}_{rev,comp,in}}{\dot{W}_{comp,in}} \,, \tag{29}$$

where the reversible work consumed by the compressor is $\dot{W}_{rev,comp,in} = \dot{X}_2 - \dot{X}_1$. The destroyed exergy of the combustion chamber ($\dot{X}_{destroyed,CC}$) is obtained by applying the exergy balance equation to the combustion chamber as

$$\dot{X}_2 + \dot{X}_5 = \dot{X}_3 + \dot{X}_{destroyed,CC} \,. \tag{30}$$

Then, the exergy efficiency of the combustion chamber is

$$\eta_{ex,CC} = \frac{\dot{X}_3 - \dot{X}_2}{\dot{X}_f} \,. \tag{31}$$

The destroyed exergy of the gas turbine ($\dot{X}_{destroyed,GT}$) can be obtained by

$$\dot{X}_3 = \dot{X}_4 + \dot{W}_{GT,out} + \dot{X}_{destroyed,GT} \,. \tag{32}$$

Then, the exergy efficiency of the gas turbine is

$$\eta_{ex,turb} = \frac{\dot{W}_{turb,out}}{\dot{W}_{rev,turb,out}} \,, \tag{33}$$

where the reversible work generated by the gas turbine is $\dot{W}_{rev,turb,out} = \dot{X}_3 - \dot{X}_4$. The destroyed exergy of the heat recovery steam generator ($\dot{X}_{destroyed,HRSG}$) is calculated as

$$\begin{aligned} \dot{X}_4 + \dot{X}_6 + \dot{X}_8 + \dot{X}_{10} + \dot{X}_{12} + \dot{X}_{14} + \dot{X}_{16} + \dot{X}_{18} = \dot{X}_{HRSG,out} \\ + \dot{X}_7 + \dot{X}_9 + \dot{X}_{11} + \dot{X}_{13} + \dot{X}_{15} + \dot{X}_{17} + \dot{X}_{19} + \dot{X}_{destroyed,HRSG} \,. \end{aligned} \tag{34}$$

The exergy efficiency of the heat recovery steam generator is

$$\eta_{ex,HRSG} = \frac{1}{\dot{X}_4 - \dot{X}_{HRSG,out}} \Big[\dot{X}_7 + \dot{X}_9 + \dot{X}_{11} + \dot{X}_{13} + \dot{X}_{15} + \dot{X}_{17} + \dot{X}_{19} - (\dot{X}_6 + \dot{X}_8 + \dot{X}_{10} + \dot{X}_{12} + \dot{X}_{14} + \dot{X}_{16} + \dot{X}_{18})\Big] . \tag{35}$$

Applying the exergy balance equation to the steam turbine, the destroyed exergy of the steam turbine is

$$\dot{X}_{20} + \dot{X}_{22} + \dot{X}_{23} = \dot{W}_{ST,out} + \dot{X}_{21} + \dot{X}_{24} + \dot{X}_{destroyed,ST} . \tag{36}$$

Then, the exergy efficiency of the steam turbine is

$$\eta_{ex,ST,net} = \frac{\dot{W}_{ST,out}}{\dot{W}_{rev,ST,out}} , \tag{37}$$

where the reversible work generated by the steam turbine is

$$\dot{W}_{rev,ST,out} = \dot{X}_{20} + \dot{X}_{22} + \dot{X}_{23} - (\dot{X}_{21} + \dot{X}_{24}) . \tag{38}$$

The total exergy destruction of a cycle is the sum of the exergy destruction of each component in that cycle, and is given as

$$\dot{X}_{destroyed,CCPP} = \left(\dot{X}_{destroyed,GT} + \dot{X}_{destroyed,comp} + \dot{X}_{destroyed,CC}\right)_{SC,1} + \left(\dot{X}_{destroyed,GT} + \dot{X}_{destroyed,comp} + \dot{X}_{destroyed,CC}\right)_{SC,2} + \dot{X}_{destroyed,HRSG,1} + \dot{X}_{destroyed,HRSG,2} + \dot{X}_{destroyed,ST} . \tag{39}$$

All the readings of temperature, pressure, and flow are acquired from the power plant control unit. The enthalpy and entropy are calculated using the Engineering Equation Solver (EES) software. Details of Unit 1, HRSG1, and steam turbine are shown in Fig. 2. The analysis is performed for the whole year to study the seasonal effect. Five cases of operating conditions are considered. In order to simplify the calculations, several assumptions are made and can be summarized as follows:

a) steady state flow in each stream in the combined cycle is assumed,

b) air and combustion gases are considered ideal gases,

c) heat losses to the environment are neglected,

d) mechanical efficiencies are assumed to be 100%,

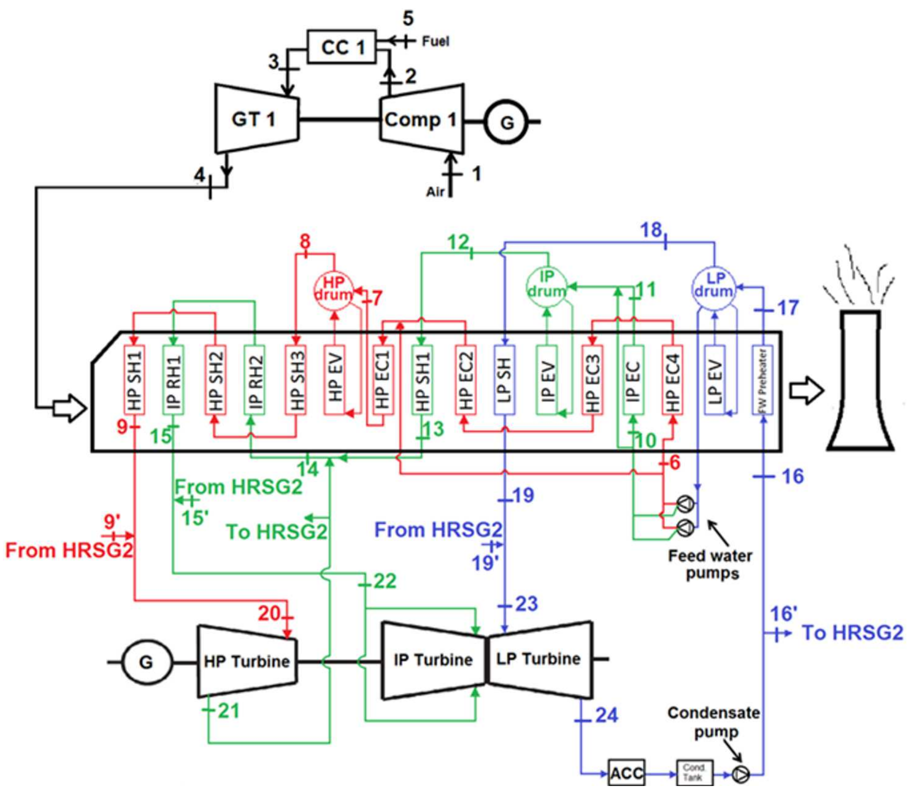

Figure 2: Unit 1, HRSG1 in details and steam turbine.

e) potential and kinetic energy are neglected, and

f) heat loss through HRSG is assumed to be 1.24% [24].

The thermodynamic parameters for each stream of phase 3 of SEPCO at full load with two running gas turbines, for case 1 when temperature ambient $T_{amb} = 34\,^{\circ}\mathrm{C}$, are listed in Tab. 4.

5 Calculations sequence

Figure 3 shows a flowchart of the calculation sequences.

Table 4: Thermodynamic parameters of each stream of phase 3 of SEPCO at full load with two running gas turbines, when $T_{amb} = 34\,^{\circ}\mathrm{C}$ (Case 1).

	Point	Type of stream	Temperature, °C	Pressure, bar	Mass flow rate, kg/s
GT 1	1	Compressor inlet	34	0.934	483.10 (calculated)
	2	Compressor outlet	429.5	13.4	483.10 (calculated)
	3	Turbine inlet	1095	13.4	491.2 (calculated)
	4	Turbine outlet	502	–	491.2 (calculated)
	5	Natural gas	–	–	8.1
GT 2	1’	compressor inlet	34	0.934	482.39 (calculated)
	2’	Compressor outlet	428	13.2	482.39 (calculated)
	3’	Turbine inlet	1095	13.2	490.29 (calculated)
	4’	Turbine outlet	509	–	490.29 (calculated)
	5’	Natural gas	–	–	7.9
HRSG 1	6	HP ECin	157.3	173.2	39.1
	7	HP ECout	306	173.2	39.1
	8	HP Shin	318.4	109.2	44.8
	9	HP Shout	497.9	107.5	44.8
	10	IP ECin	154	58.7	4.5
	11	IP ECout	220.1	58.7	4.5
	12	IP Shin	226	25.4	10.9
	13	IP Shout	292.5	24.8	10.9
	14	IP RHin	291.5	21.6	47.2
	15	IP RHout	489.2	21.6	47.2
	16	LP FWin	56.8	20.5	62.4
	17	LP ECout	134.3	20.5	62.4
	18	LP Shin	154	4.6	10
	19	LP Shout	257	3.7	10
Steam turbine	20	HP Steam (Turbine inlet)	496.9	106.3	85
	21	HP Steam (Turbine outlet)	290.5	22.5	85
	22	IP Steam (Turbine inlet)	488	21.2	95.8
	23	LP Steam (Turbine inlet)	236	3.4	20
	24	LP Steam (Turbine outlet)	54.8	0.156	115.8
HRSG 2	6’	HP ECin	158.4	172.7	41.6
	7’	HP ECout	309.8	172.7	41.6
	8’	HP Shin	318.3	109	40.2
	9’	HP Shout	448.9	107.8	40.2
	10’	IP ECin	155	59	8
	11’	IP ECout	219.7	59	8
	12’	IP Shin	225	24.7	9.3
	13’	IP Shout	299.2	24.2	9.3
	14’	IP RHin	294	21.7	48.6
	15’	IP RHout	492	21.7	48.6
	16’	LP FWin	56.8	20.6	63.9
	17’	LP ECout	142.6	20.6	63.9
	18’	LP Shin	155.2	4.8	10
	19’	LP Shout	223	3.7	10

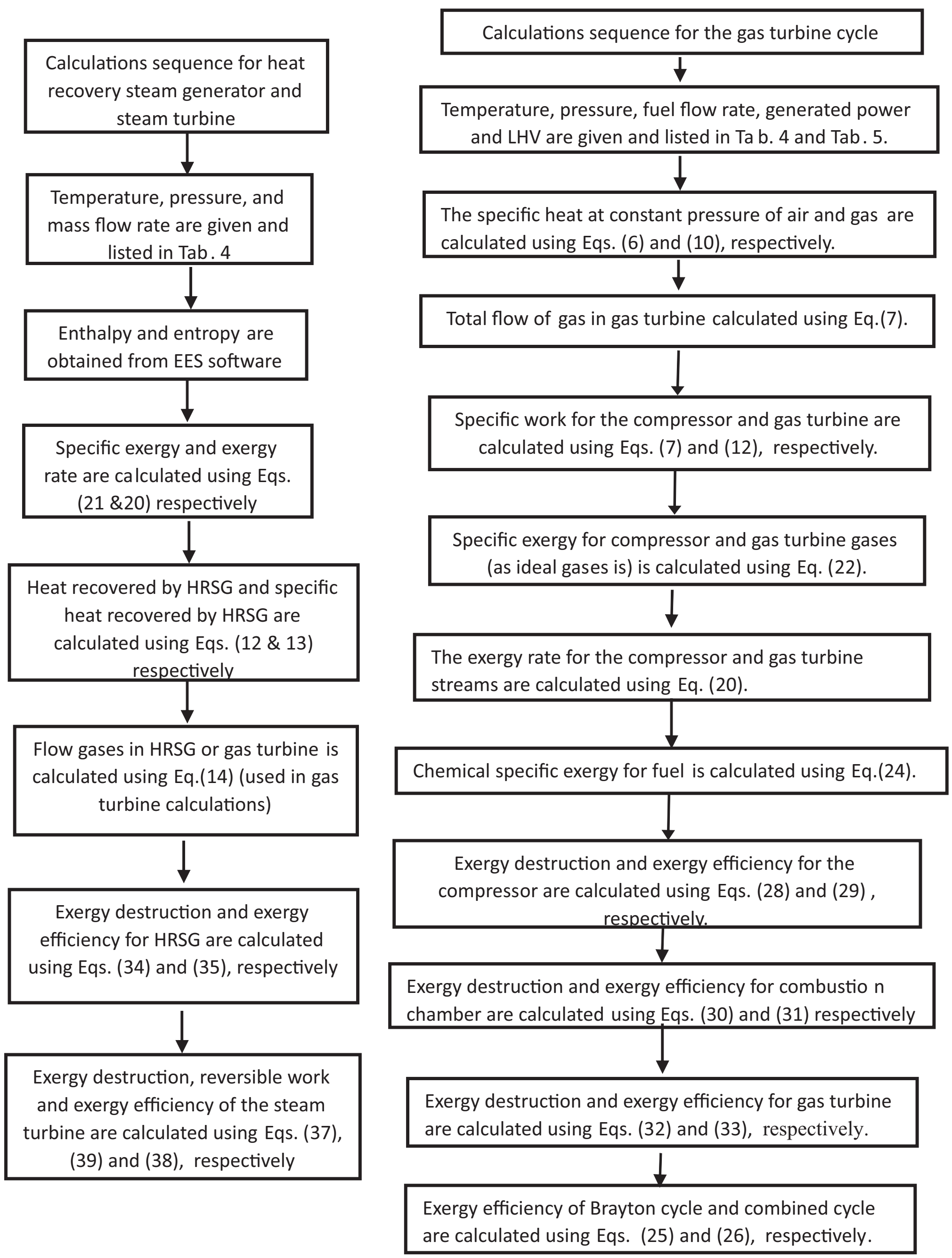

Figure 3: Flowchart of the calculation sequences.

6 Results and discussions

The energy and exergy analyses are presented for Samra Electric Power Company CCPP. Five different cases are investigated depending on the ambient temperatures (see Tab. 5). The generated power and natural gas flow rate measurements are recorded and listed in Tab. 5.

Table 5: Studied cases description.

Case No.	Ambient temperature	Generated energy (2*GT)	Steam turbine power	Total generated power	Fuel gas flow rate (2*GT)
Unit	^{0}C	MW	MW	MW	kg/s
Case 1	34.00	139.65	130.00	409.29	15.40
Case 2	26.00	146.40	132.00	424.79	16.00
Case 3	17.00	155.00	133.00	443.00	16.20
Case 4	12.00	158.00	134.00	450.00	16.90
Case 5	5.00	166.00	134.00	466.00	17.00

Table 6: Comparison between the present model and results of Yilmazoglu *et al.* [29].

		Steam turbine	HRSG	Compressor	Gas turbine	Combustion chamber
Exergy destruction (MW)	Present model (MW)	11.64	20.29	7.02	20.20	170.01
	Yilmazoglu *et al.* [29]	11.53	18.79	7.48	18.36	165.10
	Percentage error	0.95	7.98	6.15	10.02	2.97
Exergy efficiencies (%)	Present model	88.80	69.22	96.22	94.95	76.05
	Yilmazoglu *et al.* [29]	89.00	67.20	95.85	95.30	77.39
	Percentage error	0.22	3.01	0.39	0.37	1.73

In order to validate the developed model, plant specifications and data presented in [21] are input into the present model and its results are compared against published results presented in [21]. Table 6 shows that our present model matches very well previously published data. Using the procedure discussed previously, specific exergies are calculated for the Brayton cycle for all cases and are presented in Tab. 7. Enthalpy, entropy, specific exergy and exergy rate for HRSG1, HRSG2 and steam turbine are presented in Tab. 8 for case No. 1, while details for other cases are not included in the article. Exergy destruction in each component of the combined cycle is shown in Tab. 9, whereas the percentage of exergy destruction in each component is shown in Tab. 10 for all investigated cases of operation of CCPP. The total plant exergy destruction for Case 1 is calculated to be 633.34 MW.

Table 7: Specific exergies in the Brayton cycle (kJ/kg).

	GT 1					GT 2				
Type of stream	Compressor inlet	Compressor outlet	Turbine inlet	Turbine outlet	Natural gas	Compressor inlet	Compressor outlet	Turbine inlet	Turbine outlet	Natural gas
Point	1	2	3	4	5	1′	2′	3′	4′	5′
Case 1 (12 °C)	0	389.79	921.91	198.07	62203.53	0	387.47	920.59	202.82	62203.53
Case 2 (17 °C)	0	388.49	931.95	202.02	60663.63	0	385.94	930.71	205.45	60663.63
Case 3 (26 °C)	0	383.64	943.66	205.05	60047.81	0	381.17	942.51	209.20	60047.81
Case 4 (34 °C)	0	381.40	949.17	209.56	57580.15	0	379.27	948.05	213.76	57580.15
Case 5 (5 °C)	0	375.68	957.64	214.89	57242.82	0	373.92	956.56	218.43	57242.82

The performance of CCPP, thermal and exergy efficiency of the simple cycle are presented graphically in Fig. 4. It shows that the total generated power is decreasing as the ambient temperature increases. It is clear from Fig. 4 that the generated power is greatly affected by the air temperature entering the compressor (ambient temperature). Furthermore, Fig. 4 shows that the total exergy destruction increases as the ambient temperature increases. The primary cause for growth in total exergy destruction is high exergy destruction in the combustion chamber. Some exergy destructions can be avoided, others cannot be avoided, in many cases a small part of the exergy destruction in the combustion chamber can be avoided [19].

Table 8: Enthalpy, entropy, specific exergy and exergy rate for HRSG1, HRSG2, and steam turbine, when $T_{amb} = 34$ °C (Case 1).

	Point	Type of stream	Enthalpy	Entropy	Specific exergy	Exergy rate
	–	–	kJ/kg	kJ/kg.K	kJ/kg	kW
HRSG 1	6	HP ECin	674	1.897	99.950	3908.049
	7	HP ECout	1368	3.274	371.211	14514.354
	8	HP Shin	2713	5.568	1011.953	45335.499
	9	HP Shout	3359	6.547	1357.400	60811.524
	10	IP ECin	652.9	1.877	84.990	382.455
	11	IP ECout	954	2.512	191.145	860.153
	12	IP Shin	2806	6.257	893.430	9738.388
	13	IP Shout	2990	6.615	967.524	10546.013
	14	IP RHin	2998	6.688	953.113	44986.938
	15	IP RHout	3442	7.363	1189.888	56162.718
	16	LP FWin	239.5	0.790	5.361	334.495
	17	LP ECout	566	1.678	59.183	3693.025
	18	LP Shin	2757	6.877	654.090	6540.901
	19	LP Shout	2979	7.443	702.328	7023.281
Steam turbine	20	HP Steam (Turbine inlet)	3358	6.550	1355.479	115215.724
	21	HP Steam (Turbine outlet)	2993	6.661	956.402	81294.179
	22	IP Steam (Turbine inlet)	3440	7.369	1186.046	113623.216
	23	LP Steam (Turbine inlet)	2937	7.401	673.222	13464.442
	24	LP Steam (Turbine outlet)	2400	7.385	141.134	16343.329
HRSG 2	6'	HP ECin	678.7	1.908	101.273	4212.961
	7'	HP ECout	1390	3.311	381.852	15885.047
	8'	HP Shin	2714	5.570	1012.339	40696.032
	9'	HP Shout	3361	6.549	1358.786	54623.201
	10'	IP ECin	657.2	1.887	86.220	689.761
	11'	IP ECout	943.1	2.508	181.473	1451.785
	12'	IP Shin	2807	6.271	890.132	8278.229
	13'	IP Shout	3008	6.657	972.630	9045.460
	14'	IP RHin	3004	6.696	956.657	46493.535
	15'	IP RHout	3448	7.369	1194.046	58030.640
	16'	LP FWin	239.5	0.790	5.361	342.536
	17'	LP ECout	601.5	1.764	68.281	4363.162
	18'	LP Shin	2758	6.861	660.002	6600.021
	19'	LP Shout	2909	7.307	674.080	6740.801

Table 9: Exergy destruction for all components of CCPP.

Component	Case 1 (34 °C)	Case 2 (26 °C)	Case 3 (17 °C)	Case 4 (12 °C)	Case 5 (5 °C)
X_{des} compressor 1	13.45	13.78	14.64	15.25	15.28
X_{des} gas turbine 1	13.97	13.47	13.53	11.78	11.06
X_{des} combustion chamber 1	239.32	217.61	217.12	208.61	193.83
X_{des} compressor.2	13.35	14.28	14.37	14.91	15.66
X_{des} gas turbine 2	13.21	13.64	12.37	10.03	11.11
X_{des} combustion chamber 2	226.96	194.69	179.93	179.28	167.52
X_{des} HRSG1	48.35	50.99	51.92	52.87	54.59
X_{des} HRSG2	50.07	54.84	52.14	52.60	54.62
X_{des} steam turbine	14.67	9.47	9.52	5.47	5.01
X_{des} total (MW)	633.34	582.77	565.54	550.80	528.68

Table 10: Percentage of exergy destruction for all components of CCPP turbine (%).

Component	Case 1 (34 °C)	Case 2 (26 °C)	Case 3 (17 °C)	Case 4 (12 °C)	Case 5 (5 °C)
X_{des} compressor 1	2.12	2.37	2.59	2.77	2.89
X_{des} gas turbine 1	2.21	2.31	2.39	2.14	2.09
X_{des} combustion chamber 1	37.79	37.34	38.39	37.87	36.66
X_{des} compressor.2	2.11	2.45	2.54	2.71	2.96
X_{des} gas turbine 2	2.09	2.34	2.19	1.82	2.10
X_{des} combustion chamber 2	35.84	33.41	31.82	32.55	31.69
X_{des} HRSG1	7.63	8.75	9.18	9.60	10.33
X_{des} HRSG2	7.91	9.41	9.22	9.55	10.33
X_{des} steam turbine	2.32	1.63	1.68	0.99	0.95
X_{des} total (MW)	100.00	100.00	100.00	100.00	100.00

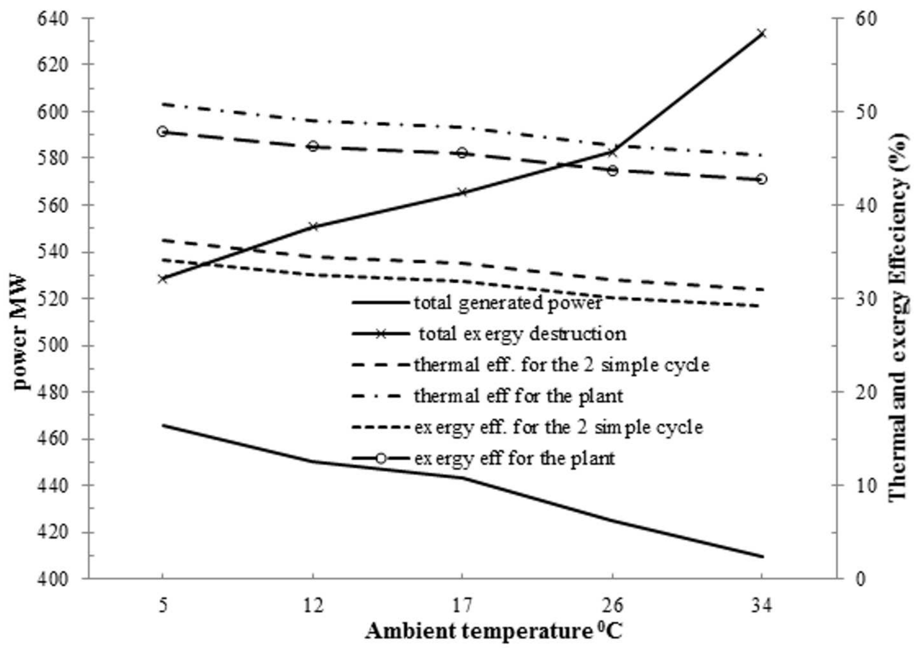

Figure 4: Power plant performance.

The gas turbines have a constant volume engine, increasing the ambient temperature will decrease the mass flow rate of the air. In other words, the effect of ambient temperature is extended to the compressor power consumption, which increases with high ambient temperatures, due to the decrease in air density. The thermal efficiency and exergy efficiency decrease as the ambient temperature increases. Exergy efficiencies of CCPP components are presented graphically in Fig. 5. It is noted from Tabs. 9 and 10 that the gas turbine, compressor and steam turbine have the maximum exergy efficiencies of 96.1%, 94.7, and 89.9% (Case 1), respectively, and HRSG and combustion chamber have the minimum exergy efficiencies of 46.4% and 53% (Case 1). The overall combined cycle power plant energy efficiency is computed as 45.29% compared to the exergy efficiency of the plant, 42.73% (Case 1). Thus, the energy efficiency is larger than the exergy efficiency. The chemical reactions lead to raise the exergy destruction [27], so the combustion chamber has the minimum exergy efficiency in the system. The chemical reactions, heat transfer and friction are the major causes of exergy destruction in the combustion chamber [21]. With temperature changes the pressure ratio of the compressor (and gas turbine) as well as the air flow (and combustion gas flow) will change. The change of these two parameters maintains the exergy efficiency of the compressor and gas turbine constant. The exergy efficiency of the steam turbine decreases as the ambient temperature increases. Hence, there is a slight decrease in power as the temperature increases. HRSG exergy efficiencies are slightly changed by ambient temperature differences. The temperature difference between the streams represents the major source of irreversibilities in HRSG [8].

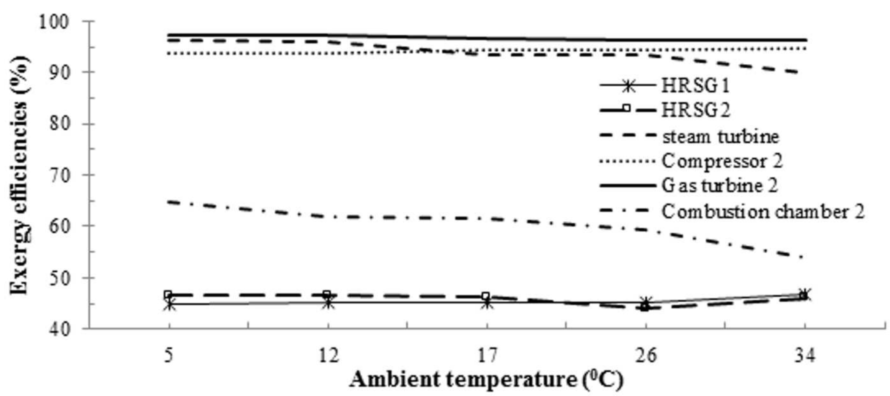

Figure 5: Exergy efficiency of CCPP components.

For all ambient temperature values, the combustion chamber has the largest exergy destruction percentage as shown in Tab. 10. It is readable that the maximum exergy destruction occurs in combustion chambers with 74% (Case 1) of the total exergy destruction due to heat transfer, chemical reactions, and friction. According to the data given in Tab. 10, the irreversibility associated with chemical reactions is the important source of exergy destruction. When the ambient temperature equals 35 °C, the percentage of exergy destruction of compressors, gas turbines, steam turbine and HRSG are 4%, 4%, 2%, and 16%, respectively. According to the results of exergy analysis, the combustion chamber and HRSG should be planned to decrease exergy destruction percentages.

Figure 6 shows the variation of pressure ratio for compressor 1 and compressor 2 with respect to the ambient temperature. It can be seen from

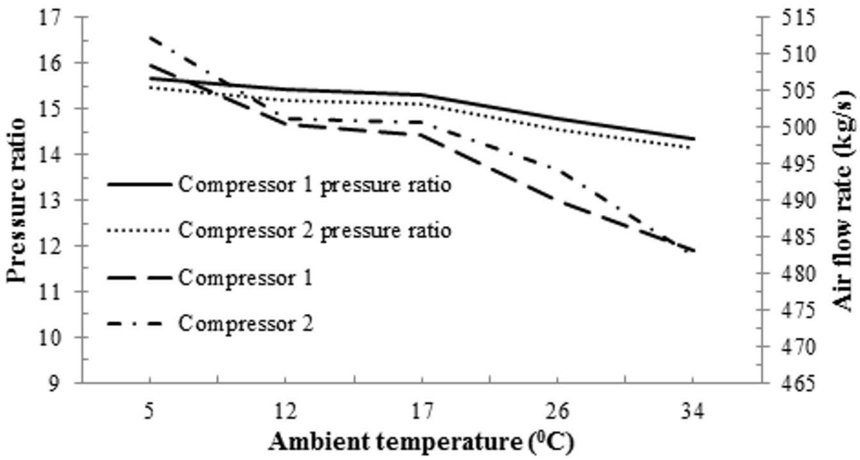

Figure 6: The variation of pressure ratio and air flow rate for compressor 1 and compressor 2 versus ambient temperature.

the results presented in Fig. 6 that the compressor pressure ratio and the compressor air flow decreases as the ambient temperature increases.

The thermal and exergy efficiency of the simple cycle and CCPP are presented graphically in Fig. 7. It shows that the thermal efficiency and exergy efficiency are maximum at full load and minimum at minimum load (72% of full load). If 100% of load is taken as a reference case, the thermal efficiency and exergy efficiency for the two gas turbines (simple cycle) is decreased by 9.78% when the load is decreased to 83% and decreased by 13.07% when the load is decreased to 72%. The thermal efficiency and exergy efficiency of the plant are decreased by 4.46% when the load is decreased to 83% and decreased by 5.27% when the load is decreased to 72%.

Exergy efficiencies of CCPP components are presented graphically in Fig. 8. The exergy efficiency of the combustion chamber is decreased slightly when the load is increased, since the fuel (natural gas) flow rate is increased. Furthermore, the exergy efficiency of the compressor and gas turbine are increased slightly when the load is increased. If 100% of load is taken as a reference, the exergy efficiency for the combustion chamber 1 is increased by 5.19% when the load is decreased to 83% and increased by 7.1% when the load is decreased to 72%. The exergy efficiency for the combustion chamber 2 is increased by 5.32% when the load is decreased to 83% and increased by 8.56% when the load is decreased to 72%. Moreover, the exergy efficiency of the steam turbine is increased slightly when the

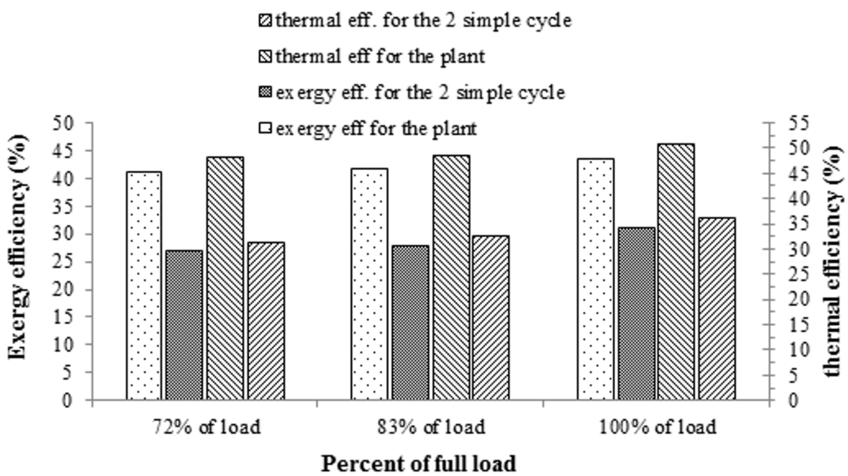

Figure 7: Thermal efficiency of the simple cycle and CCPP versus percentage of full load.

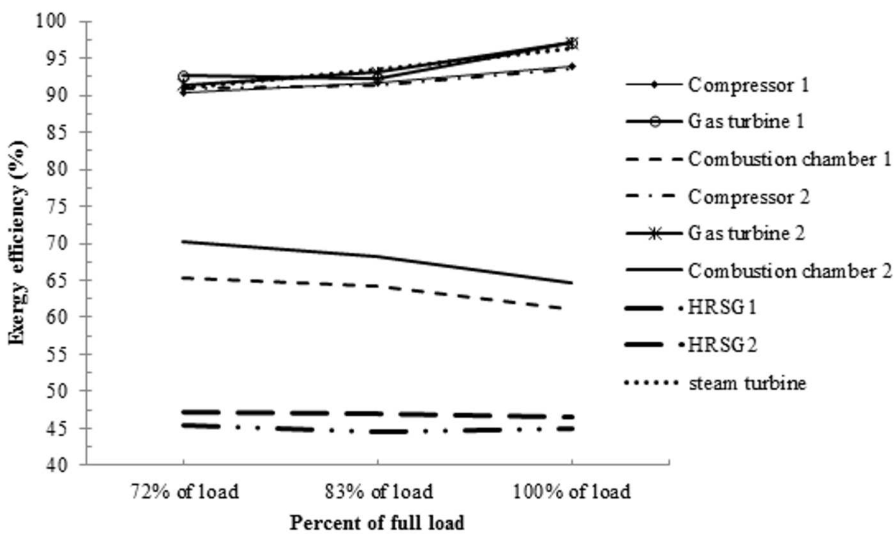

Figure 8: Exergy efficiency of CCPP components versus percentage of full load.

load is increased. HRSG exergy efficiencies have been slightly changed by load differences. The total exergy destruction increases when the load is increased. The exergy efficiency of the steam turbine is decreased by 3.12% when the load is decreased to 83% and decreased by 5.46% when the load is decreased to 72%. The total exergy destruction is decreased by 7.65% when the load is decreased to 83% and decreased by 18.09% when the load is decreased to 72%.

The variations of thermal and exergy efficiencies of CCPP when HRSG pressure levels are increased or decreased are presented graphically in Fig. 9. It shows that the thermal efficiency and exergy efficiency are increasing by 0.443% and 0.495% when the pressure levels (HP, IP, and LP) are increased by 10% of actual pressure and 20% of actual pressure respectively and decreasing by 0.815% and 1.15 % when the pressure levels (HP, IP, and LP) are decreased by 10% of actual pressure and 20% of actual pressure respectively.

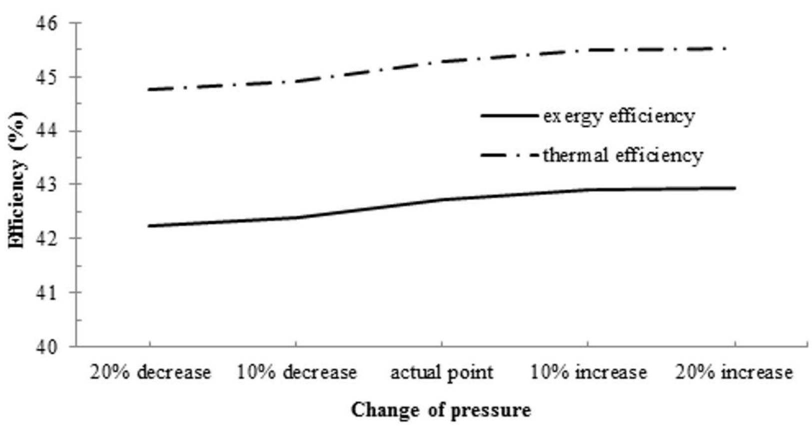

Figure 9: Thermal efficiency (%) and exergy efficiency (%) of CCPP versus pressure.

Variations of steam turbine exergy efficiency with HRSG pressure levels are presented graphically in Fig. 10. It shows that the steam turbine exergy efficiency is increased by 0.752% and 0.291% when the pressure levels (HP, IP, and LP) are increased by an amount of 10% and 20% of actual pressure respectively. The entropy generation associated with a 20% actual pressure increase is larger than that generated with a 10% pressure increase. This can explain why exergy efficiency of the steam turbine is lowered for a 20% increase in actual pressure. The steam turbine exergy efficiency is decreased by 1.562% and 1.61% when the pressure levels (HP, IP, and LP) are decreased by 10% and 20% of actual pressure, respectively. Furthermore, results presented in Fig. 10 show that the steam turbine exergy destruction decreases by 6.068% and 1.346% when the pressure levels (HP, IP, and LP) are increased by 10% and 20% of actual pressure respectively. Steam turbine exergy destruction is increased by 12.68% and 13.43% when the pressure levels (HP, IP, and LP) are decreased by 10% and 20% of actual pressure respectively.

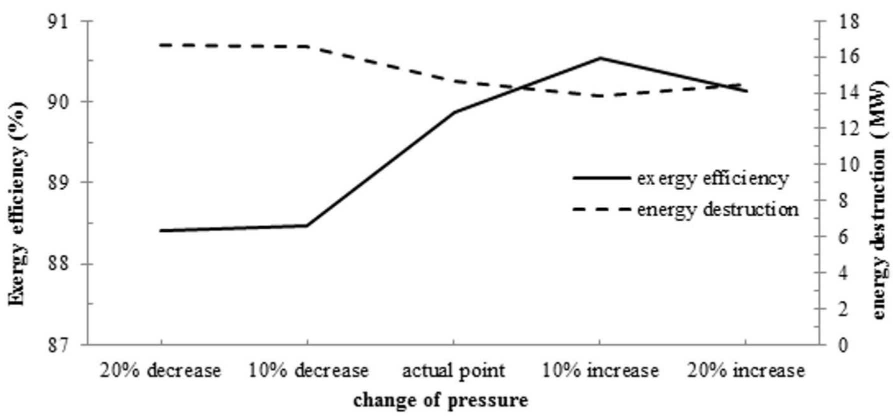

Figure 10: Steam turbine exergy efficiency (%) versus pressure.

Table 11 exhibits the comparison between the findings of the present study and other published studies as further validation of our study.

Table 11: Comparison with other studies.

Reference No.	Location	Year	Total power (MW)	Thermal efficiency of CCPP (%)	Exergy efficiency of CCPP (%)	Exergy destruction for steam turbine
[6]	Turkey	2015	119.20	56.00	50.04	6
[29]	Ankara, Turkey	2010	272.00	53.13	50.10	11.5
[30]	Pakistan	2013	144.00	34.41	33.40	5.7
[8]	Kuwait	2015	650.00	54.50	–	20.5
[31]	Dadri, India	2013	277.00	55.00	50.00	21
Present study (Case 4)	Jordan	2016	450.00	49.02	46.24	5.4
Present study (Case 1)	Jordan	2016	409.29	45.29	42.73	2.3%

7 Conclusion

The irreversibility of each part of the combined cycle power plant located in Jordan using the exergy and energy analyses are evaluated. The effect of ambient temperature, gas turbine pressure ratio, gas turbine inlet temperature, and load conditions on the combined cycle efficiency, power outputs and exergy destruction are investigated. The results show that

- Energy and exergy efficiencies of the combined cycle power plant are found as 45.29% and 42.73% (Case 1), respectively.
- The gas turbine, compressor, and steam turbine have the maximum exergy efficiencies of 96.1%, 94.7%, and 89.9% (Case 1), respectively.
- HRSG and combustion chamber have the minimum exergy efficiencies of 46.4% and 53% (Case 1).
- The maximum exergy destruction occurs in combustion chambers with 74% (Case 1) of the total exergy destruction due to heat transfer, chemical reactions, and friction.
- The irreversibility associated with chemical reactions is the important source of exergy destruction.
- The energy and exergy efficiencies of the combined cycle power plant are decreasing as the ambient temperature increases.
- The combined cycle power plant has the maximum energy and exergy efficiencies at full load.
- The thermal efficiency and exergy efficiency are increased when pressure levels (HP, IP, and LP) are increased.
- The steam turbine exergy efficiency is increased with an increase of pressure.

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