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Exergy Analysis of 144 Mw Combined Cycle Power Plant Kotri Pakistan

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Abstract: The conventional energy analysis evaluates the performance of a thermodynamic system generally on its quantity only. It gives no information about the effect of irreversibilities on performance that occurs inherently during any thermodynamic process. On the other hand, exergy analysis, based on the second law of thermodynamics recognizes magnitudes and locations of the losses due to these irreversibilities. Therefore, the application of exergy analysis for thermodynamic evaluation of conventional power plants is steadily growing. This paper deals with the exergy analysis performed on a 144 MW combined cycle power plant situated at Kotri, Pakistan. The exergy destruction models are used to assess the losses occurred in the key components of the power plant. The results indicate that the total exergy destruction of the power plant is around 288.5 MW; combustion chambers contribute a major share of 168 MW (58.2%) followed by heat recovery steam generators (HRSGs) with 43 MW (14.8%). The energy efficiency and exergy efficiency of the power plant are calculated as 34.41 and 33.40% respectively. Finally, some suggestions regarding the efficiency improvement of the power plant components are given.

Keywords: combined cycle power plant, exergy, exergy analysis, exergy efficiency.

1. INTRODUCTION

Energy is the prime mover of socio-economic development of any country. The world energy needs rely heavily on fossil fuels for electricity generation. Currently, 80% of electricity in the world is approximately produced from the fossil fuels-fired power plants (Erdem *et al* 2009). In Pakistan, the total installed capacity of electricity generation is approximately 20,000 MW. In 2009-10, the total electricity generation is approximately 95,608 GWh, nearly 69% of which has been generated by fossil fuel resources, HDIP, Pakistan Energy Yearbook (2010). Most of the thermal power plants in Pakistan are generating electricity while consuming natural gas and furnace oil. Due to spiraling oil prices, shortage of gas supply and very low conversion efficiency due to aging and poor management, not only the unit price of electricity is skyrocketing but the gap between demand and supply is also widening. Consequently, an environmental impact associated with the inefficient burning of fossil fuels is also a great concern. Therefore, there is a continuing need of modifications in thermal power plants for effective and efficient utilization of the scarce resources of fossil fuels. The combined cycle power plant is a popular modification in this regard which involves a gas power cycle topping a vapor power cycle. Such a cycle has a higher overall efficiency than either of the cycles executed individually. The performance of thermal power plants is generally evaluated through analysis based on the first law of thermodynamics. It has some inherent limitations which are overcome by exergy analysis which is based on the second law of thermodynamics that characterizes the work potential of a system and recognizes the extents and locations of energy.

Therefore, exergy analysis has been widely used by many researchers in evaluation, optimization and improvement of thermal power plants. (Erdem *et al.*, 2009) have provided a comparative study of nine coal-fired power plants situated in Turkey from energetic and exergetic viewpoints. Via an exergy analysis, (Woudstra *et al.*, 2010) evaluated the combined cycle power plants with alternative designs. According to their results, most of the exergy is lost during the combustion process. (Cihan *et al.*, 2006) performed the energy and exergy analyses of a combined cycle power plant in Turkey and found that the combustion chamber, gas turbine and HRSG are the main sources of irreversibilities, representing over 85% of the overall exergy losses. (Balli *et al.*, 2007) carried out the exergetic performance analysis of a gas turbine cogeneration power plant located in Turkey and showed that nearly 68% of the overall exergy destruction is occurring in the combustion chamber. Many investigators have linked exergy with economics, called as thermoeconomics or exergoeconomics for thermodynamic cycle's evaluation. (Ghaebi *et al.*, 2011) presented thermoeconomic analysis of a trigeneration system with gas turbine as prime mover. They have showed that the energy and exergy efficiencies can be improved with increase in the air compressor pressure

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ratio, turbine inlet temperature, and pinch temperature in the low-pressure evaporator of HRSG. (Colpan *et al.,* 2006) dealt with thermoeconomic analysis of Bilkent combined cycle cogeneration power plant located in Turkey and found that the combustion chamber is the most irreversible part of the system with a 74 % exergy destruction. The researchers have also emphasized the role of exergy in environmental analysis and referred it as exergoenvironmental analysis. In this regard, (Ahmadi *et al*., 2011) have reported a comprehensive exergy, exergoeconomics and exergoenvironmental analyses of combined cycle power plants. (Kanoglu *et al.*, 2007) emphasized the need of understanding the definitions of energy and exergy efficiencies for improved energy management in thermal power plants. The exergy concept has also played an important role in making an energy policy as elaborated by Dincer(2002) and Rosen *et al.*,(2008). According to them exergy does not only address the impact of energy resource use on the environment but prove to be a suitable technique for promoting the goal of improved energy conversion efficiency.

There have been inadequate studies on the exergetic assessment of thermal systems in Pakistan. In this regard, current work deals with an exergy analysis of a 144 MW combined cycle power plant Kotri, located in Pakistan. The main objectives of this work are to present thermodynamic model for the power plant, and evaluate its performance by analyzing the exergetic parameters using design data.

2. MATERIAL AND METHODS

2.1 Process description and assumptions made

The power plant investigated is equipped with four similar gas turbine (GT) units (A, B, C and D) each with capacity of 25 MW, topping a steam turbine (ST) unit with additional capacity of 44 MW, as shown in **(Fig.1)**. Each GT-unit consists of (i) compressor, (ii) combustion chamber, (iii) gas turbine and combined with ST-unit by a heat recovery steam generator (HRSG). The ST-unit consists of (i) condenser, (ii) deaerator, (iii) high-pressure drum, (iv) low-pressure economizer, (v) high-pressure economizer, (vi) high-pressure evaporator,and (vii) superheater. The power plant also includes number of pumps and auxiliary equipments. The decision variables and parameters of the power plant are presented in **(Table 1).**

Table 1. Decision variables and parameters of the power plant

Decision variables	Value
Air compressor pressure ratio	9.5
Gas turbine inlet temperature	1198 K
Gas turbine outlet temperature	787 K
Isentropic efficiency of air compressor	85 %
Combustion efficiency	98 %
Electromechanical efficiency	97 %
Parameters	
Net power generated by gas turbine units	4x25 MW
Net power generated by steam turbine unit	44 MW
Air compressor inlet pressure	101.325kPa
Air compressor inlet temperature	288 K
Exhaust gas temperature	443 K
Steam turbine inlet pressure	3950kPa
Steam turbine inlet temperature	746 K
Steam mass flow rate	47.44 kg/s
Condenser pressure	10kPa
Steam drum pressure	5400kPa
Deaerator pressure	300kPa
Cooling water temperature difference	7 K

This study is based on the design conditions with following assumptions:

i. The system operates in a steady-state condition.
ii. Air and constituents of combustions gas are treated as ideal gas.
iii. The combustion reaction of natural gas (mainly CH_4) is complete. Only chemical exergy of the fuel is considered.
iv. No pressure loss in fluid streams while flowing through combustion chamber, HRSG and pipes except the steam pipe to steam turbine.
v. Kinetic energy (and exergy), potential energy (and exergy) of fluid streams and pump work are neglected.

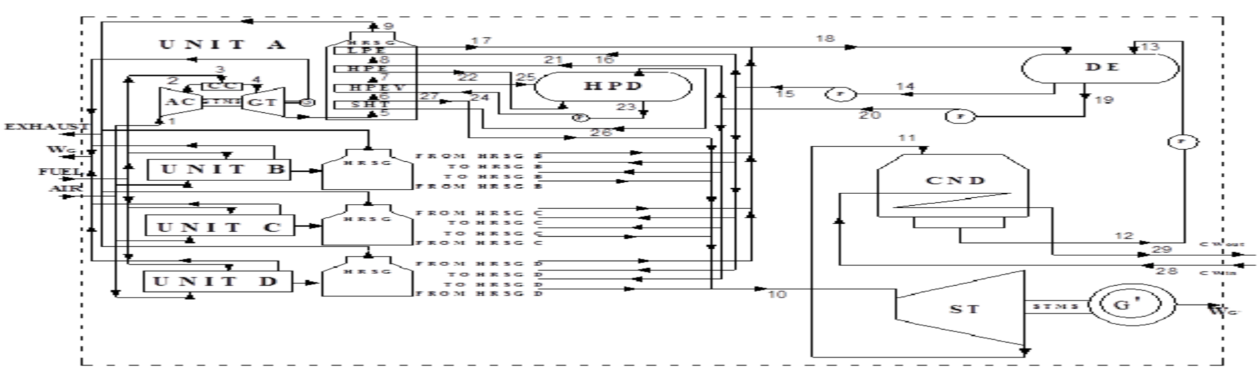

Figure 1 Schematic view of combined cycle power plant investigated

Thermodynamic model equations

The thermodynamic equations of exergy analysis related to power plant components are given below:

Continuity: $\sum \dot{m}_{in} - \sum \dot{m}_{out} = 0$ (1)

Energy: $\dot{Q} - \dot{W} + \sum \dot{m}_{in} h_{in} - \sum \dot{m}_{out} h_{out} = 0$ (2)

For combustion process, Eq. (2) is written as

$$\dot{Q}_c = \sum \dot{N}_r (h_f^o + \overline{h} - \overline{h^o})_r - \sum \dot{N}_p (h_f^o + \overline{h} - \overline{h^o})_p \quad (3)$$

Exergy: $\dot{X}_Q - \dot{X}_W + \sum \dot{X}_{in} - \sum \dot{X}_{out} - \dot{X}_D = 0$ (4)

Where $\dot{X}_Q = \sum (1 - \frac{T}{T_o}) \dot{Q}$ (5)

$\dot{X}_W = \dot{W}$ (6)

and $\dot{X} = \dot{m}\, x$ (7)

where $x = (h - h_o) - T_o(s - s_o)$ (8)

For an ideal gas, the entropy change in Eq.(8) is given as

$$s - s_o = C_{P,avg} ln(\frac{T}{T_o}) - R ln(\frac{P}{P_o}) \quad (9)$$

The chemical exergy of the fuel is given as

$$\dot{X}_3 = \dot{m}_3 x_3 \quad (10)$$

An approximate value for the specific chemical exergy of gaseous hydrocarbon fuels C_aH_b is given in (Balliet et al 2007):

$$x_{ch} = x_3 = (1.033 + 0.0169 \frac{b}{a} - \frac{0.0698}{a})(LHV)_{fuel} \quad (11)$$

The lower heating value of CH_4 is taken as 50.05 MJ/kg.

Thermodynamic parameters

The energy efficiency (η_E) of the power plant components defined as

$$\eta_{E,i} = \left[\frac{(desired\ output)_i}{(required\ input)_i} \right] x100\% \quad (12)$$

The overall energy efficiency (η_E) of the power plant is defined as

$$\eta_{E,\ power\ plant} = \left[\frac{\dot{W}_G + \dot{W}_{G'}}{\dot{Q}_{in,\ powerplant}} \right] x100\% \quad (13)$$

where $\dot{Q}_{in,\ power\ plant} = 4\, \dot{m}_3 (LHV)_{fuel}$ (14)

The exergy efficiency (η_X) of the power plant components is defined as:

$$\eta_{X,i} = \left[1 - \frac{\dot{X}_{D,i}}{\dot{X}_{in,i}} \right] x100\% \quad (15)$$

The exergy efficiency (η_X) of the overall power plant is given as

$$\eta_{X,\ power\ plant} = \left[\frac{\dot{W}_G + \dot{W}_{G'}}{\dot{X}_{in,\ powerplant}} \right] x100\% \quad (16)$$

where $\dot{X}_{in,\ power\ plant} = 4\, \dot{X}_{fuel}$ (17)

The relative exergy destruction ratio (RXDR) is defined as ratio of the exergy destruction of power plant component to the exergy destruction of overall power plant

$$(RXDR)_i = \left[\frac{\dot{X}_{D,i}}{\dot{X}_{D,poewerplant}} \right] x100\% \quad (18)$$

The exergy improvement potential rate ($\dot{IP}$) of the power plant components is defined as follows

$$\dot{IP}_i = [1 - \eta_{X,i} / 100]\ x\ \dot{X}_{D,i} \quad (19)$$

3. RESULTS AND DISCUSSION

In this section energy and exergy analyses on the power plant are performed with assumptions made above and parameters given in **(Table 1).**

The molar compositions of air consist of 77.48% N_2, 20.59% O_2, 0.03% CO_2 and 1.90% H_2O, (Balli *et al* 2007). The complete combustion equation on molar basis is then given as

$CH_4 + 28.76\ N_2 + 7.64\ O_2 + 0.01CO_2 + 0.70\ H_2O \longrightarrow$
$28.76\ N_2 + 5.64\ O_2 + 1.01\ CO_2 + 2.70\ H_2O$ (20)

Using Eq.(3) with 2 % heat loss from the combustion chamber, the fuel-to-air ratio (FAR) is determined as 0.0269 kmol fuel/kmol air or 0.0150 kg fuel/kg air. The mass composition of the combustion gas is obtained as 74.63% N_2, 16.73% O_2, 4.12% CO_2 and 4.51% H_2O. The molecular mass (M_g), average specific heat at constant pressure ($C_{Pg,avg}$) and gas constant (R_g) of the combustion gas are 28.31 kg/kmol, 1.135 kJ/kgK and 0.29 kJ/kgK respectively. The specific heat at constant pressure for air (C_{Pa}) is taken as 1.005 kJ/kgK.

The mass flow rates of air, fuel and combustion gas are calculated as 139.16, 2.09 and 141.25 kg/s, respectively. The enthalpy of combustion gas at different temperature values have been calculated from its constituents' enthalpy with respect to their mass percentage as given below:

$$h_g = 0.7463 h_{N_2} + 0.1673 h_{O_2} + 0.0412 h_{CO_2} + 0.0451 h_{H_2O} \quad (21)$$

Since the inlet and exit conditions are the same for all the components of gas turbine units;

thermodynamic properties are determined for gas turbine unit-A only. Firstly, different thermodynamic properties at various state points in the power plant are calculated and tabulated in **(Table 2)**. Moreover, the results of energy analysis are obtained and tabulated in (**Table 3)**.

Table 2. The energy rate, exergy rate and other thermodynamic properties at various power plant locations labelled in Fig.1.

State	P (MPa)	T (K)	$\dot{m}$ (kg/s)	$\dot{E}$ (MW)	$\dot{X}$ (MW)
1	0.1013	288	139.16	40.8	-0.01
2	0.9625	594	139.16	85.1	40.9
3	-	298	2.09	104.6	107.8
4	0.9625	1198	141.25	189.6	106.7
5	0.1013	787	141.25	119.5	29.3
6	0.1013	727	141.25	109.7	23.3
7	0.1013	557	141.25	83.0	9.2
8	0.1013	466	141.25	69.0	3.8
9	0.1013	443	141.25	65.5	2.7
10	3.95	746	47.44	160.6	61.6
11	0.01	318.8	47.44	115.2	7.1
12	0.01	318.8	47.44	9.1	0.14
13	0.30	318.8	47.44	9.1	0.15
14	0.30	353	47.44	15.9	0.92
15	0.60	353.0	47.44	15.9	0.92
16	0.60	353.0	11.86	3.9	0.23
17	0.60	403.0	11.86	6.5	0.76
18	0.60	403.0	47.44	25.9	3.0
19	0.30	353.0	47.44	15.9	0.92
20	6.00	353.0	47.44	15.9	0.92
21	6.00	353.0	11.86	3.9	0.23
22	5.40	541.8	11.86	13.9	3.6
23	5.40	541.8	11.86	13.9	3.6
24	5.40	541.8	11.86	13.9	3.6
25	5.40	541.8	11.86	33.1	12.1
26	5.40	541.8	11.86	33.1	12.1
27	4.33	748.0	11.86	40.1	15.5
28	0.20	305.0	3626.6	486.3	1.3
29	0.20	312.0	3626.6	592.4	5.0

It can be noted from this table, the back work ratio in gas turbine units is very high (0.63), that is 63 % of the energy produced by gas turbine is consumed in air compression. Similarly, the energy efficiency of the combined power plant (34.41%) is considerably higher than the efficiency of the gas turbine units (23.90%) or the steam turbine unit (28.44%) operating alone.

The values in Table 2 are also used to calculate the inlet exergy rate, the exit exergy rate, the exergy destruction, the exergy efficiency, the relative exergy destruction ratio and the exergy improvement potential for each component and of the combined cycle power plant. These values are tabulated in **(Table 4).** According to Table 4, the following is resulted:

- The exergy efficiency of AC, CC, GT, HRSG, ST, CND, and DE are calculated as 92.4, 71.8, 93.1, 59.7, 85.2, 52.5 and 57.4 %, respectively. The exergy efficiency of each component/stream is shown in (**Fig.2)**.
- The highest exergy destruction among the power plant components occurs in the combustion chamber with 167.94 MW (58.2%), which is in agreement with the results of (Woudstra *et al* 2010), Cihan *et al.*, 2006, Balli *et al.*, 2007 and Ahmadi *et al* 2011), followed by HRSG with 42.87 MW (14.8%). Therefore the total exergy improvement potential of these components is the highest (64.71 MW). The exergy destruction rate of the power plant components are exhibited in **(Fig. 3).**
- The exergy efficiency of the gas turbine units, the steam turbine unit and the combined power plant are 23.18, 69.43 and 33.40%, respectively.

Table 3. The energy transfer and energy efficiency values of the power plant components

Quantity	Value
Total power consumed in air compression, $\dot{W}_{AC, total}$	177.03 MW
Total power produced by gas turbines, $\dot{W}_{GT, total}$	280.12 MW
Net mechanical power produced by gas turbine units, $\dot{W}_{net, mechanical}$	103.09 MW
Net electric power produced by gas turbine units, $\dot{W}_{G}$	100.00 MW
Back work ratio of gas turbine units, BWR	0.63
Total heat rate supplied to the power plant, $\dot{Q}_{in, power plant}$	418.42 MW
Energy efficiency of HRSG	71.60 %
Total electric power produced by steam turbine unit, $\dot{W}_{G'}$	44.00 MW
Heat rate rejected from the condenser, $\dot{Q}_{condenser}$	106.11 MW
Total electric power produced by combined cycle power plant, $\dot{W}_{G}+\dot{W}_{G'}$	144.00 MW
Energy efficiency of gas turbine units	23.90 %
Energy efficiency of steam turbine unit	28.44 %
Overall energy efficiency of power plant, $\eta_{E, power plant}$	34.41 %

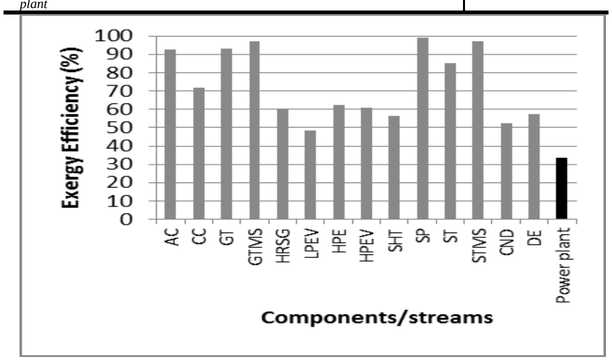

Fig.2. Exergy efficiency of the power plant components/streams

Table 4. The exergy rate, exergy efficiency and exergy improvement potential of the power plant components (adapted from Table 2)

Component /streams	$\dot{X}_{in}$ (kW)	$\dot{X}_{out}$ (kW)	$\dot{X}_{D}$ (kW)	η_x (%)	RXDR (%)	$\dot{IP}$ (kW)
AC	177000	163493	13507	92.4	4.7	1031
CC	594793	426850	167943	71.8	58.2	47420
GT	426849	397224	29625	93.1	10.3	2056
GTMS	103076	100000	3076	97.0	1.1	92
HRSGs	106238	63372	42866	59.7	14.8	17296
LPEVs	4407	2128	2279	48.3	0.8	1179
HPEs	21447	13315	8132	62.1	2.8	3083
HPEVs	56450	34415	22035	61.0	7.6	8601
SHTs	23934	13514	10420	56.5	3.6	4536
SP (27-10)	62161	61593	568	99.1	0.2	5
ST	61591	52473	9118	85.2	3.1	1350
STMS	45359	44000	1359	97.0	0.5	41
CND	6971	3662	3309	52.5	1.1	1571
DE	3198	1836	1362	57.4	0.5	580
Stacks			10857		3.8	
CW_{out} (29)			4983		1.7	
Power plant	432573	144000	288573	33.4	100	192226

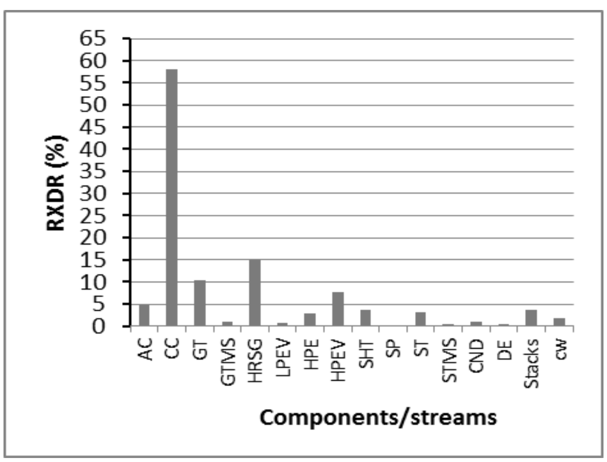

Fig.3. Relative exergy destruction ratio of the power plant components/streams

The exergy analysis, presented in Table 4, is much more revealing than the energy analysis. For example, energy analysis diverts our attention towards the condenser for the performance improvement of the power plant since it rejects a considerable amount of energy (106 MW). Since the condenser temperature of 318.8 K and cooling water inlet temperature of 305.0 K are used in the analysis, the maximum possible power production from the energy rejected from the condenser would be only 4.6 MW. Therefore the exergy destruction ratio of condenser is only 1.1%. In order to obtain this meagre amount of power, various technical and economic constraints could be encountered. The combustion chamber, on the other hand, is having highest exergy destruction ratio of 58.2%, whereas the same is working with an energy loss of only 2.0%. Thus exergy analysis diverts our attention to the combustion process rather than the condensation process for performance improvement. Hence, the exergy analysis serves to identify and locate the actual inefficiencies within the power plant.

One way to reduce the exergy destruction in the combustion chamber is to reduce the temperature difference, which causes higher turbine inlet temperature. An increase in the energy and exergy efficiencies with rise in the inlet temperature has been reported in (Ghaebi *et al*., 2011). However, maximum inlet temperature is restricted by the turbine material strength. With the use of some special superalloys in gas turbine parts, the inlet temperature can be increased. The HRSG is among the components of greatest exergy destruction (15%), where HPEV contributes a major share (50%), as shown in Table 4. The pinch analysis to optimize the pinch point and approach point, the optimization of pressure levels and the mass flow ratio can reduce exergy destruction in the HRSG. The stack exergy destruction is nearly 11 MW. Since the temperature of the gas is high, some exergy can be recovered to enhance the plant performance, either by operating an organic Rankine or Kalina cycle to produce some additional power or producing process heat (cogeneration). The efficiencies of gas turbine and steam turbine units may be improved through multi-stage compression/expansion of fluids with intercooling/reheating, and regeneration processes.

5. CONCLUSIONS

This study reveals that the exergy analysis is more useful thermodynamic tool than energy analysis for the performance assessment of the power plant. It is concluded that the combustion chambers cause the maximum exergy destruction (58%), followed by the HRSGs (15%), gas turbines (10%), air compressors (5%), and stack gas (4%). The energy and exergy efficiencies of the power plant are calculated as 34.41 and 33.40%, respectively. Arrangements proposed to reduce the exergy destruction rates in combustion chambers and HRSGs are higher gas turbine inlet temperature and optimization of the HRSGs operating parameters, respectively. Similarly multi-stage compression with intercooling for air compressors and multi-stage expansion with reheating for gas turbines and steam turbine are proposed.

Nomenclature

BWR	**back work ratio (%)**
C_p	**specific heat at constant pressure (kJ/kgK)**
$\dot{E}$	**energy flow rate (kW)**
h	**specific enthalpy (kJ/kg)**
$\bar{h}$	**molar specific enthalpy (kJ/kmol)**
$\dot{IP}$	**improvement potential rate (kW)**
LHV	**lower heating value (kJ/kg)**
M	**molar mass (kg/kmol)**
$\dot{m}$	**mass flow rate (kg/s)**
$\dot{N}$	**molar flow rate (kmol/s)**
P	**pressure (MPa)**
$\dot{Q}$	**heat transfer rate (kW)**
R	**gas constant (kJ/kgK)**
RXDR	**relative exergy destruction ratio (%)**
s	**specific entropy (kJ/kgK)**
T	**temperature, (K)**
$\dot{W}$	**power (kW)**
x	**specific exergy flow (kJ/kg)**
$\dot{X}$	**exergy flow rate (kW)**
η	**efficiency, %**

Abbreviations

AC	**air compressor**
CC	**combustion chamber**
CND	**condenser**
CW	**cooling water**
DE	**deaerator**
GT	**gas turbine**
GTMS	**gas turbine mechanical shaft**
HPE	**high-pressure economizer**
HPEV	**high-pressure evaporator**
HRSG	**heat recovery steam generator**
LPE	**low-pressure economizer**
SHT	**superheater**
SP	**steam pipe**
ST	**steam-turbine**
STMS	**steam-turbine mechanical shaft**

Subscripts

avg	**average**
a	**number of carbon**
b	**number of hydrogen**
c	**combustion**
ch	**chemical**
D	**destruction**
E	**energy**
f	**formation**
G	**generator with gas-turbine unit**
G'	**generator with steam-turbine unit**
g	**combustion gas**
i	**i^{th} component of the power plant**
Q	**heat transfer**
r	**reactants**
p	**products**
s	**isentropic**
W	**work transfer**
X	**exergy**
o	**dead (environment or reference) state**

Superscript

o	**standard reference state of 25°C and 1 atm.**

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