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SECOND LAW AND SENSITIVITY ANALYSIS OF A COMBINED CYCLE POWER PLANT IN TURKEY

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Abstract: In this study, exergy and sensitivity analysis of a combined cycle gas turbine (CCGT) power plant in Ankara, Turkey, is performed with real plant data sets. Exergy efficiency of each component and overall plant is determined by calculating exergy destructions. In addition, a sensitivity analysis is carried out by changing some critical parameters of the cycle. The effects of parameters on exergy efficiency, CO_2 emissions of the plant and heat transfer area is presented. The total exergy destruction and net exergy efficiency of the plant is calculated 228.05 MW and 50.11 % respectively. Exergy analysis shows that the major exergy destructions take place in combustion chamber with the value of 124.07 MW and 54.41% of the total exergy destruction of the overall plant. The effect of ambient temperature, duct burner exhaust temperature, steam injection mass flow rate, steam turbine inlet pressure, high pressure and low pressure evaporator pinch point temperature difference, high pressure and low pressure economizer sub cooling temperature are examined by sensitivity analysis. The variation of exergy efficiency, CO_2 emissions and heat transfer areas are given.
Keywords: Combined cycle power plant, exergy analysis, sensitivity analysis, exergy.

TÜRKİYE'DEKİ BİR KOMBİNE GÜÇ ÇEVRİMİ SANTRALİNİN İKİNCİ YASA VE DUYARLILIK ANALİZLERİ

Özet: Bu çalışmada Ankara'da bulunan bir gaz türbinli kombine güç çevrimi santralinin ekserji ve duyarlılık analizleri gerçek verilere göre yapılmıştır. Herbir elemanın ve tüm santralin ekserji verimi, ekserji kayıplarının hesaplanması ile bulunmuştur. Bununla birlikte, çevrimde bazı kritik parametrelerin değiştirilmesi ile duyarlılık analizi yapılmıştır. Bu parametrelerin ekserji verimi, CO_2 emisyonları ve ısı transferi alanına olan etkileri sunulmuştur. Santralin toplam ekserji kaybı ve net ekserji verimleri sırasıyla 228.05 MW ve %50.11 olarak hesaplanmıştır. Ekserji analizi, en büyük ekserji kaybının 124.07 MW değeriyle ve santralin toplam ekserji kaybının %54.41 değeriyle yanma odasında olduğunu göstermiştir. Ortam sıcaklığı, son yakıcı çıkış sıcalığı, buhar enjeksiyonu kütlesel debisi, buhar türbini giriş giriş basıncı, yüksek basınç ve düşük basınç buharlaştırıcısı yaklaşım sıcaklığı farkı, yüksek basınç ve düşük basınç ekonomizeri aşırı soğutma sıcaklığı etkileri duyarlılık analizi ile incelenmiştir. Ekserji verimi, CO_2 emisyonları ve ısı transferi alanlarına olan etkiler verilmiştir.
Anahtar kelimeler: Kombine çevrim santrali, ekserji analizi, duyarlılık analizi, ekserji.

NOMENCLATURE

e	Specific exergy [kJ kg^{-1}]
E	Time rate of exergy [MW]
ε	Exergy efficiency [%]
h	Specific enthalpy [kJ kg^{-1}]
m	Time rate of mass[kg s^{-1}]
P	Pressure [kPa]
Q	Time rate of heat loss [MW]
s	Specific entropy [kJ kg^{-1}K^{-1}]
T	Temperature [ºC]
W	Time rate of work [MW]
Subscripts	
CV	Control volume
D	Destruction
i	Inlet
o	Outlet
CH	Chemical
F	Fuel
KN	Kinetic
P	Product
PH	Physical
PT	Potential
ST	Steam turbine
GT	Gas turbine
com	Compressor
0	Dead state condition

INTRODUCTION

Sharp increasing energy prices in the recent years have encouraged the researchers and industries to develop systems, having maximum efficiency and minimum energy losses. Generally, two methods are utilized in the analysis of energy conversion processes. First one is the conventional energy analysis, which deals with the first law of thermodynamics and the second one is exergy analysis, which deals with the second law of thermodynamics. The idea of conversion efficiency based solely on first law of thermodynamic consideration is misleading, because the scale of energy quality can be quantified only by an entropy analysis (Belli, 2001). In contrast to the energy analysis, exergy analysis method considers the quality of energy as well as its quantity (Wu and Nikulshina, 2001). Additionally, in a complicated thermal system, exergy analysis shows the main location of exergy losses, their origins and also, identifies the components or processes, where the highest inefficiencies appear, in order to improve their performances, or to develop new components or processes.

Srinivas et. al. (2008) analyzed a dual pressure combined cycle and investigated the effect of steam injection on its performance. Fiaschi et. al. (1998) studied a semi-closed gas turbine combined power plant in terms of exergy balances and efficiency, for the purpose of identifying the critical plant devices, considering several operating conditions. Aljundi (2009) used exergy analysis to identify and quantify the components having largest energy and exergy losses in a steam power plant. Besides, the effects of ambient air temperature variation on the power plant are also presented. Ganapathy et. al. (2009) applied energy and exergy analysis on lignite fired power plant and compared the energy and exergy losses of the individual components of the plant. They indicated that the maximum energy losses of 39% occurred in the condenser, whereas the maximum exergy losses of 42.37% occurred in the combustor. Cihan et. al. (2006), performed energy-exergy analysis of a combined cycle power plant located in Lüleburgaz and the results show that combustion chambers, gas turbines and heat recovery steam generators (HRSG) are the main sources of irreversibilities involving more than 85% of the overall exergy losses. Constructive and thermal suggestions for these devices were suggested to improve the efficiency of the system. Ballı et. al. (2007) performed exergy analysis of a power plant located in Eskişehir. Some modifications were also suggested and exergy efficiency of the modified power plant was increased.

In this study, exergy analysis of a combined cycle gas turbine (CCGT) power plant is performed, in Ankara, in an exergy aspect. The exergy efficiency of each component and overall plant is studied by calculating exergy destructions. Furthermore, a sensitivity analysis is carried out by changing some critical parameters of the system. The real data sets are simulated in THERMOFLEX software. The obtained data from sensitivity analysis is to be considered for the further researches, for the thermoeconomic analysis of the power plant.

The Combined Cycle Power Plant

A schematic diagram of the power plant is given in Fig.1. General Electric GE7251FB model gas turbine-generator with 373 MW electrical power (in ISO conditions), CMI model heat recovery steam generator and ALSTOM steam turbine-generator with 92.26 MW electrical power is used. Net power output and net electric efficiency of the plant is calculated 272 MW and 53.13% respectively. Ambient air with 15ºC enters the cycle through a filter and then compressed with a pressure ratio of 18.5 in the compressor. Compressed air enters to the combustion chamber where combustion takes place and natural gas is used as fuel source with a 46,280.2 kJ/kg LHV. At this point a bled steam from HP turbine is injected into the combustion chamber to improve the net power and minimize the NO_x emissions of the power plant. The combustion gasses leave the gas turbine at 633°C and enter duct burner. After the duct burner, combustion gasses pass through the heat exchangers inside of the heat recovery steam generator (HRSG) to produce steam at three different pressure levels. Steam at three different pressure levels from the HRSG are led to high-intermediate- and low-pressure steam turbines that are connected to each other via a common shaft. Produced steam rotates the steam turbines while passing through the narrow openings between the casing and the blades. This rotational mechanical energy is converted to electrical energy on the steam turbine generator, connected on the same shaft. The expanded steam, leaving the low-pressure steam turbine, is condensed in a condenser and then fed back to integral deaerator for utilizing the waste heat capacity of flue gases. A forced draft cooling tower ejects heat to the environment.

METHODOLOGY

Exergy is the highest available work, which in a certain circumstance could be acquired from a certain thermal system, as it proceeds to a specified final state in equilibrium with its surroundings. Exergy is not conserved as energy and destructed in the system due to the internal or external irreversibilities. For a real process, the exergy input always exceeds the exergy outputs; this unbalance is due to irreversibilities, which is known as exergy destruction (2001). The higher the value of exergy means more work which can be obtained from a system.

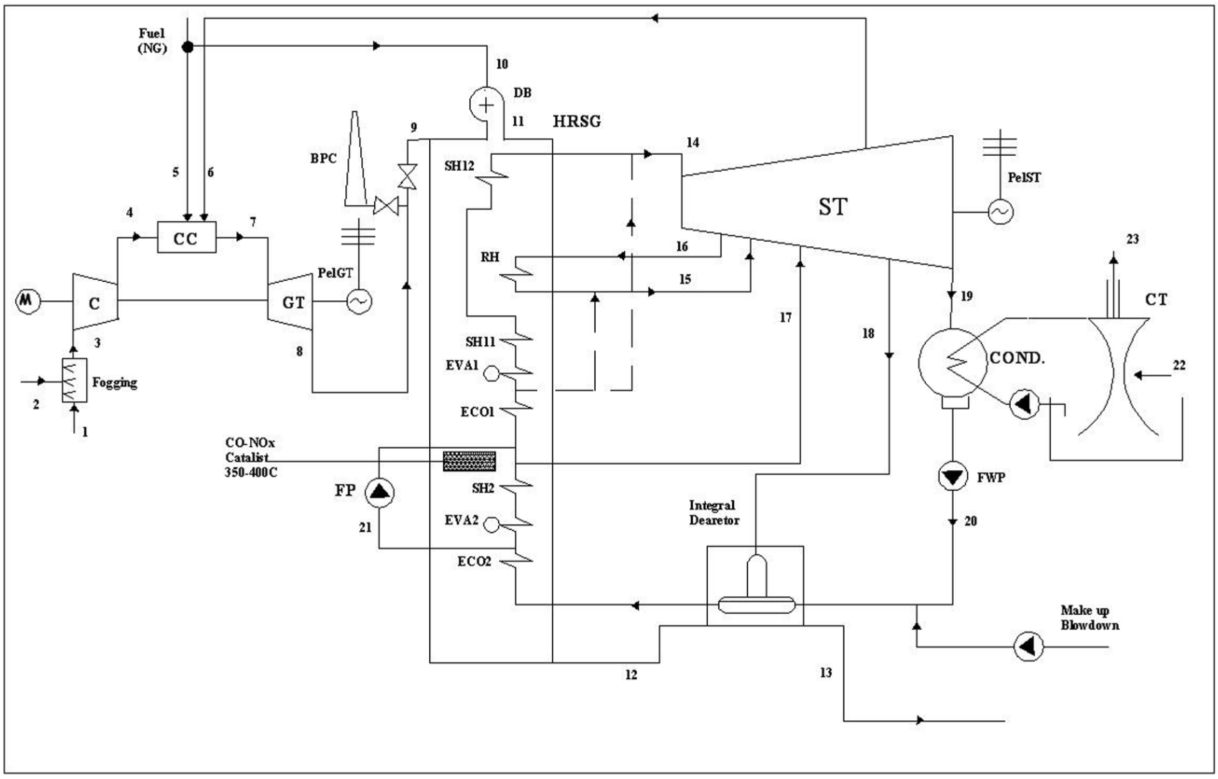

Figure 1. Schematic diagram of the combined cycle power plant

Exergy Destruction

General form of exergy equation for an open system control volume can be given as follows [Bejan and Tsatsaronis, 1996 –Bejan, 1988].

$$\frac{dE_{cv}}{dt} = \sum_j \left(1 - \frac{T_0}{T_i}\right)\dot{Q}_j - \left(\dot{W}_{cv} - p_0 \frac{dV_{cv}}{dt}\right)$$
$$+ \sum_i \dot{m}_i e_i - \sum_e \dot{m}_e e_e - \dot{E}_D \tag{1}$$

The exergy equation for the system at steady state conditions is given in Eq. 2, where time rate variations, specified in Eq. 1, are neglected. Rearranging Eq.2 gives the exergy destruction of a steady state open system for a control volume.

$$0 = \sum_j \left(1 - \frac{T_0}{T_i}\right)\dot{Q}_j - \left(\dot{W}_{cv}\right) + \sum_i \dot{m}_i e_i - \sum_e \dot{m}_e e_e - \dot{E}_D \tag{2}$$

$$\dot{E}_D = \sum_j \dot{E}_{qj}\dot{Q}_j - \left(\dot{W}_{cv}\right) + \sum_i \dot{E}_i - \sum_e \dot{E}_e \tag{3}$$

In the absence of nuclear, magnetic, electrical, and surface tension effects, the total exergy of a system $\dot{E}$ can be divided into four components. Neglecting potential and kinetic energy, Eq. (5) is obtained.

$$\dot{E} = \dot{E}_{PH} + \dot{E}_{CH} + \dot{E}_{PT} + \dot{E}_{KN} \tag{4}$$

$$\dot{E} = \dot{E}_{PH} + \dot{E}_{CH} \tag{5}$$

The specific physical exergy can be expressed as follows where subscript "0" indicates reference conditions.

$$e_{PH} = h - h_0 - T_0\left(s - s_0\right) \tag{6}$$

Total exergy rate can be written as;

$$\dot{E} = \dot{m}\left[h - h_0 - T_0\left(s - s_0\right) + \bar{e}_{CH}\right] \tag{7}$$

Here, chemical exergy of a substance can be obtained from standard chemical exergy tables [Bejan and Tsatsaronis, 1996 – Kotas, 1985] relative to specification

of the environment. For mixtures containing gases other than those present in the reference tables, chemical exergy can be evaluated with following equation.

$$\bar{e}_{CH} = \sum x_n (\bar{e}_{CH})_n + \bar{R}T_0 \sum x_n \ln x_n \tag{8}$$

Here, x_n is the mole fraction of the k_{th} gas in the mixture and $\bar{R}$ is the universal gas constant. In the exergy analyses, another significant matter which must be noticed is the reference conditions. In this study, the atmospheric temperature and pressure are taken as 25ºC and 101.32 kPa respectively.

Exergy Efficiency

Exergy efficiency shows the percentage of the fuel exergy provided to a system that is found in the product exergy. Moreover, the difference between 100% and the actual value of the exergy efficiency, expressed as a percent, is the percentage of the fuel exergy wasted in this system as exergy destruction and exergy loss [Bejan and Tsatsaronis, 1996].

$$\varepsilon = \frac{\dot{E}_P}{\dot{E}_F} \quad (9)$$

The exergy of the product shows the desired outcome of the system, while the exergy of the fuel represents the total given resources into the system. The desired outcome and the given resource could be different for the different components of the power plant. Therefore, output and input streams of each component should be separately examined.

Exergy Analysis of Combined Cycle Power Plant Components

To evaluate the exergy destruction of the overall power plant the exergy destruction in the individual components must be calculated. By taking each component as a control volume, the exergy equations for each one can be derived from the general exergy equation given in Eq. 3. It should be noted that, in this study, the exergy destructions caused by the heat losses from the components are neglected, since it is assumed that the boundary temperature of the each component (T_j) is equal to the dead state temperature (T_0). Exergy expressions and efficiencies of the each component are given in Appendix 1.

The thermodynamic properties of each stream of the combined cycle power plant are presented in Table 1 and also the physical, chemical and total exergy rate of air, combustion gases, fuel, steam and water at these points were computed and summarized in Table 2. The chemical exergy of the air at streams 1, 3, 4, 22 and 23 is assumed as zero since the composition of the air in these states resembles almost the air composition of reference environment as given in Table 3. The exergy destruction of the pumps is also neglected due to almost zero enthalpy difference between the inlet and outlet water.

The physical and chemical exergy of each stream are calculated according to the Eq. 6 and 8. Stream 5 indicates the fuel (natural gas) and as given in Table 2, the highest chemical exergy value belongs to the fuel which the composition of the natural gas by volume; 87% methane, 8.46% ethane, 0.36% H_2, 0.07% O_2, 3.63% H_2O, 0.09% CO, 0.34 CO_2.

Table1. Thermodynamic properties of each stream of the power plant

Point	State	T[˚C]	P [kPa]	$\dot{m}$ [kg/s]	*h* [kJ/kg]	*s* [kJ/kg K]
1	Air	25	101.32	427.8	-161.82	6.8671
2	Water	15.25	1724	0.92	65.62	0.2250
3	Air	19.74	101.26	428.7	-195.64	6.8609
4	Air	425.2	1869	428.7	226.40	6.9243
4'	Air to combustion	425.2	1869	383.9	226.40	6.9243
4''	Air to turbine	425.2	1869	44.76	226.40	6.9243
5	Fuel	59.91	2999	10.85	-4183.10	8.8536
6	Steam	388.48	4482	10.74	3180.5	6.6629
7	Combustion gases	1353.4	1776	450.3	-89.42	8.3116
8	Combustion gases	632.6	104.83	450.3	-1020	8.3965
9	Combustion gases	631.49	104.58	450.3	-1021.35	8.3957
10	Fuel	25	2068	0.218	-4247.67	8.8083
11	Combustion gases	648.89	104.52	450.3	-1022.63	8.4202
12	Combustion gases	157.8	101.71	450.5	-1595.41	7.5472
13	Combustion gases	114.44	101.45	450.5	-1642.87	7.4319
14	Steam	534.64	12066	69.26	3441.09	6.6001
15	Steam	537.78	3599	56.67	3535.86	7.2486
16	Steam	362.71	3624	56.37	3132.8	6.6858
17	Steam	256.82	15513	9.62	2938.09	6.7198
18	Steam	285.29	600	0.68	3031.45	7.3163
19	Water/steam	38.72	6.89	79.97	2366.11	7.6235
20	Water(condenser)	38.73	42.78	67.76	162.28	0.5553
20	Water(deaerator)	35.31	118.6	79.38	147.98	0.5092
21	Water	105.05	1715	79.97	441.59	1.3639
22	Air	25	101.32	2540.1	-161.82	6.8671
23	Air	33.1	101.32	2592.1	-153.67	6.8888

Table 2. Physical, chemical and total exergy of each stream

Point	State	E^{PH}[MW]	E^{CH}[MW]	E_{total}[MW]
1	Air	0.00	0.00	0.00
2	Water	0.003	0.0023	0.0053
3	Air	0.0021	0.00	0.002
4	Air	172.82	0.00	172.82
4'	Air to combustion	154.76	0.00	154.76
4''	Air to turbine	18.04	0.00	18.04
5	Fuel	8.90	523.40	532.3
6	Steam	12.84	0.03	12.87
7	Combustion gases	576.5	17.43	593.92
8	Combustion gases	146.06	17.43	163.5
9	Combustion gases	145.56	17.43	163
10	Fuel	0.17	10.51	10.7
11	Combustion gases	152.81	17.77	170.58
12	Combustion gases	12.10	17.77	29.88
13	Combustion gases	6.21	17.77	23.98
14	Steam	102.36	0.17	102.53
15	Steam	78.17	0.14	78.31
16	Steam	64.5	0.14	64.63
17	Steam	9.04	0.02	9.06
18	Steam	0.58	0.002	0.58
19	Water/steam	7.82	0.2	8.02
20	Water (condenser)	0.09	0.17	0.26
20	Water (deaerator)	0.07	0.2	0.26
21	Water	3.17	0.2	3.37
22	Air	0.00	0.00	0.00
23	Air	4.3	0.00	4.3

RESULTS

General Results of the Exergy Analysis

Table 3 provides a list of the exergy destruction and exergy efficiency data within the components of the power plant. It is clear that the maximum exergy destruction occurs in combustion chamber with 72.41% of total exergy destruction due to chemical reactions, heat transfer and frictions. According to the data, given in Table 3, the irreversibility associated with chemical reactions is the significant source of exergy destruction. Exergy destruction in HRSG and gas turbine with the percentage of 8.24 and 8.05% are the next most prominent components to improvement. In contrast with the first law analysis, indicating that the greatest energy loss occurs in the condenser, the exergy analysis of this plant shows that only 0.67% of the total exergy is lost in the condenser. The total plant exergy destruction is computed to be 228.04 MW. It is observed from Table 3 that, compressor, gas turbine and steam turbine have the maximum exergy efficiencies of 95.85, 95.3, 88.89% and HRSG, condenser, deaerator and fogger have the minimum exergy efficiencies of 67.20, 45.76, 42.70 and 39.57%. The overall plant exergy efficiency is computed 50.1%. Compared to the net first law efficiency of the plant, 53.13%, the exergy efficiency is less than energy efficiency. According to the results of exergy analysis, combustion chamber, gas turbine and HRSG should be designed to decrease exergy destruction percentages.

Sensitivity Analysis

A comprehensive sensitivity analysis is carried out to examine the influences of the variation of the combined cycle parameters on exergy efficiency, CO_2 emissions and heat transfer areas. The selected parameters are ambient temperature, duct burner exhaust temperature, steam injection mass flow rate, steam turbine inlet pressure, high and low pressure evaporator pinch point temperature difference, high and low pressure economizer sub-cooling temperature.

Table3. Exergy destruction data and exergy efficiency in different points of the power plant

Component	Exergy destruction [MW]	Percent exergy Destruction [%]	Exergy efficiency [%]
Fogger	0.0032	0.0014	39.57
Compressor	7.48	3.28	95.85
C. Chamber	165.1	72.41	77.39
Gas turbine	18.36	8.05	95.3
Bypass stack	0.5	0.22	99.69
Duct burner	3.09	1.35	98.22
HRSG	18.79	8.24	67.20
Steam turbine	11.53	5.05	89
Condenser	1.52	0.67	45.74
Deaerator	3.37	1.48	42.70
Overall plant	**228.04**	**100**	**50.1**

The effect of the variation of ambient air temperature on exergy efficiency of the power plant is given in Fig. 2. The exergy efficiency of the power plant improves by increasing ambient temperature. At 25ºC exergy efficiencyhas a peak point and the reason of this is the selection of reference temperature. Fig. 3 shows the effect of ambient temperature on CO_2 emission of the combined cycle. CO_2 emissions decrease with increasing ambient temperature, due to decreasing fuel consumption. However, the net power production of power plant decreases with increasing ambient air temperature. The net power production of a gas turbine is in direct proportion to ambient air temperature. As a result, Fig. 3 can be related to reduction of net power production of gas turbine.

Fig. 4, depicts the exergy efficiency of the power plant with respect to the increasing duct burner exhaust temperature. Duct burner exhaust temperature is shifted from 635 to 660ºC to simulate the effect of additional fuel input on exergy efficiency. Exergy efficiency decreases due to the increasing additional fuel consumption to reach desired exhaust temperatures. However, the effect of duct burner exhaust temperature to the exergy efficiency is low due to the fuel consumption when compared to the total fuel consumption of the combined cycle.

As a result of additional fuel consumption, CO_2 emissions of the plant increases with increasing exhaust temperatures as given in Fig. 5. HRSG heat transfer areas should be designed properly by the designer for starting-up additional burner at minimum level.

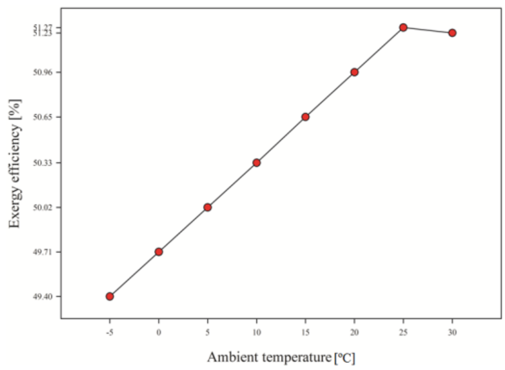

Figure 2. Effect of ambient temperature on exergy efficiency

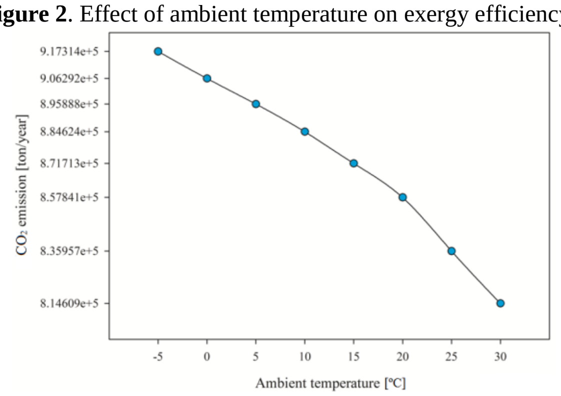

Figure 3. Effect of ambient temperature on CO2 emissions

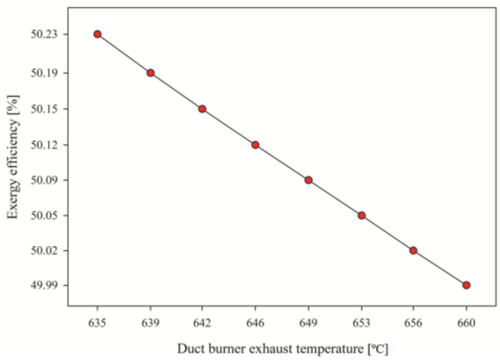

Figure 4. Effect of duct burner exhaust temperature on exergy efficiency

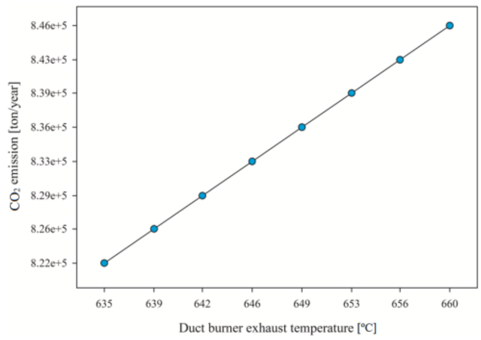

Figure 5. Effect of duct burner exhaust temperature on CO2 emissions

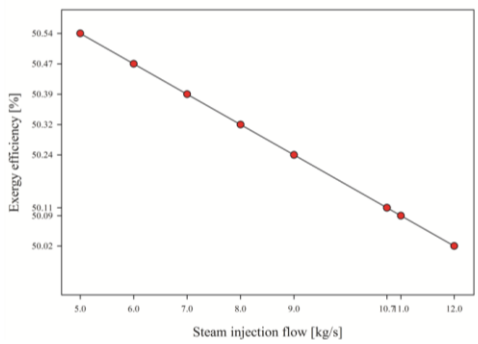

Figure 6. Effect of steam injection flow on exergy efficiency

The effect of steam injection mass flow rate on the exergy efficiency of the power plant is given in Fig. 6. Steam injection mass flow rate is varied from 5 to 12 kg/s in the simulations. The exergy efficiency decreases from 50.52 to 50.02%,while steam injection increases the net power output. On the other hand, as given in Fig. 7, steam injection mass flow rate has a direct relation with CO_2 emissions. According to the results net power and net fuel consumption increases with steam injection and it affects CO_2 emissions of the plant.

The effect of steam turbine inlet pressure on exergy efficiency is given in Fig. 8. Exergy efficiency increases with increasing inlet pressure of steam. In the analysis, steam pressure varies from 80 to 150 bar and as a result, exergy efficiency increases from 49.73 to 50.02%. For the simulated cycle, after 120 bar of inlet pressure exergy efficiency increases slightly.

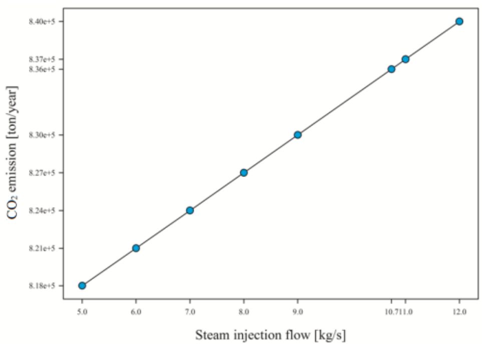

Figure 7. Effect of steam injection flow on CO_2 emission (Steam injection flow rate 10.7 kg/s is taken as main case)

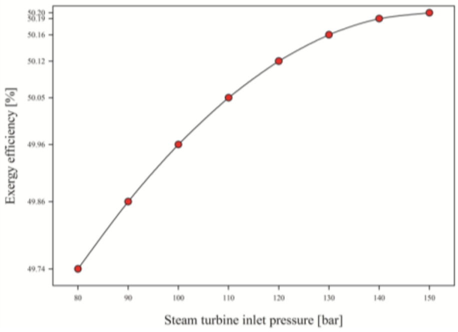

Figure 8. Effect of inlet pressure of steam turbine on exergy efficiency

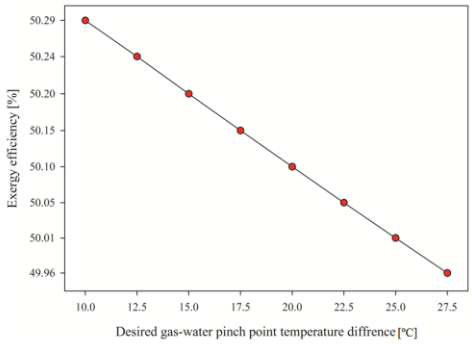

Figure 9. Effect of HP evaporator pinch point temperature difference on exergy efficiency

The temperature difference of the pinch point of the HP evaporator in HRSG is illustrated in Fig. 9. Exergy efficiency decreases when temperature difference varied from 10 to 28°C, while the heat transfer area decreases from 23,357 to 15,150 m^2 as indicated in Figure 10.

The effect of the temperature difference of the LP evaporator gas-water pinch point on exergy efficiency of the combined cycle power plant is given in Figure 11. Exergy efficiency decreases with an increase in this parameter. Also, as illustrated in Figure 12, total heat transfer area reduces with increasing pinch point temperature difference. Exergy efficiency of the power plant decreases sharply after 18ºC pinch temperature difference (Fig.11). According to Kehlhofer et. al (1999), the pinch point temperature difference should be in the range of 8-15 K.

Figure 13 and 14; show the effect of exit sub-cooling temperature of high pressure economizer on exergy efficiency of plant and heat transfer area of HP economizer. Both parameters decrease with increase in temperature of exiting water of economizer. When considering the first investment cost of a power plant sub-cooling temperature (approach temperature) has a great effect. In the literature, it is suggested that sub-cooling temperature should be taken 5 to 12 K.

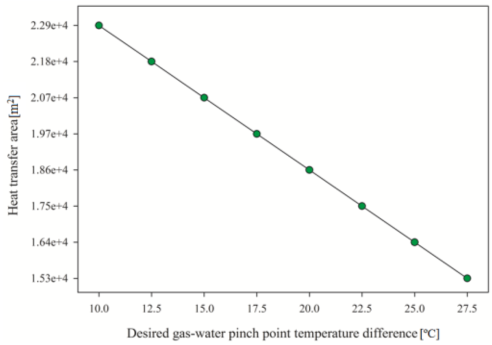

Figure 10. Effect of HP evaporator pinch point temperature difference on heat transfer area

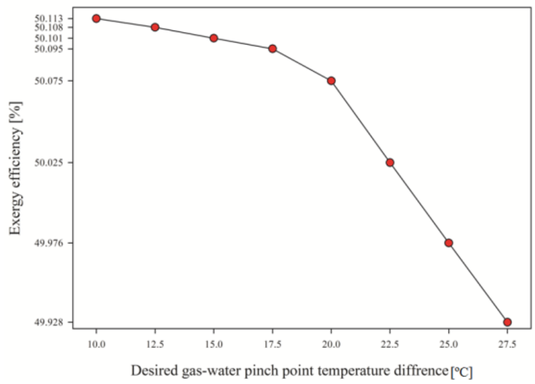

Figure 11. Effect of LP evaporator pinch point temperature difference on exergy efficiency

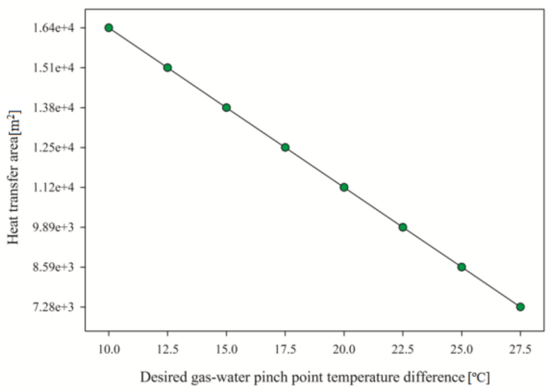

Figure 12. Effect of LP evaporator pinch point temperature difference on heat transfer area

Figure 15 and 16; show the effect of the outlet water temperature of LP economizer is same as HP economizer. Exergy efficiency and heat transfer area decreases with increasing exit temperature. Water exit sub-cooling temperature on exergy efficiency variation is smaller however, heat transfer area decreases as twice when compared to minimum and maximum temperature. As shown from Figure 15, after 8ºC sub-cooling exergy efficiency decreases sharply and 8ºC water exit sub-cooling temperature can be accepted as optimum design temperature difference of the LP economizer.

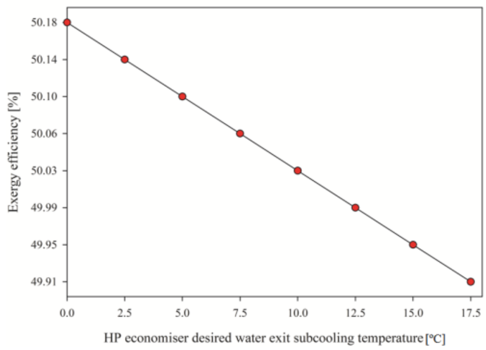

Figure 13. Effect of HP economizer sub-cooling temperature on exergy efficiency

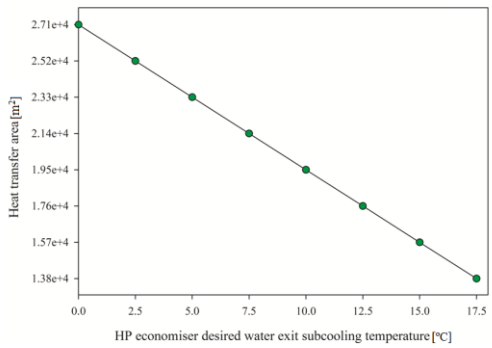

Figure 14. Effect of HP economizer sub-cooling temperature on heat transfer area

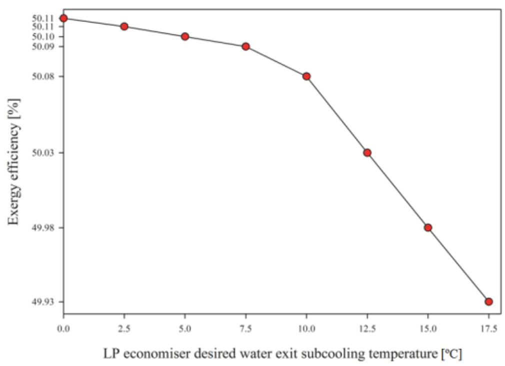

Figure 15. Effect of LP economizer sub-cooling temperature on exergy efficiency

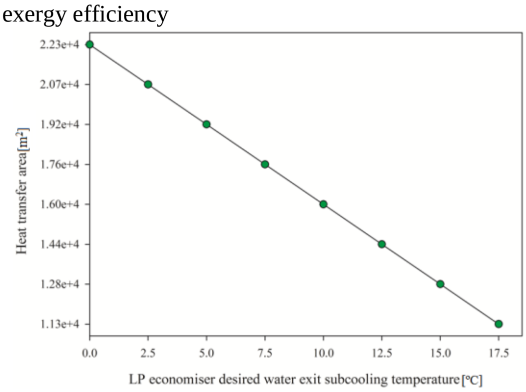

Figure 16. Effect of HP economizer sub-cooling temperature on heat transfer area

CONCLUSION

In this study, exergy and sensitivity analysis were performed in a combined cycle power plant with real data sets and the exergy destruction rates of each component and exergy efficiencies were evaluated.

1. Total exergy destruction of the plant is 228.05 MW and the net exergy efficiency of the power plant is 50.11%.

2. Exergy analysis shows that the major exergy destructions take place in combustion chamber by 165.1 MW and 77.39% of the total exergy destruction of the overall plant.

3. The exergy efficiency of the plant increases with increasing environment temperature however, net power decreases simultaneously. Duct burner exit temperature and steam injection mass flow rate to the combustion chamber decreases the exergy efficiency due to additional fuel consumption. Inlet pressure of steam to the turbine increases the exergy efficiency.

4. High pressure evaporator pinch point temperature difference, low pressure evaporator pinch point temperature difference, high pressure economizer sub-cooling temperature and low pressure economizer sub-cooling temperature, causes a reduction of exergy efficiency of the power plant with increasing parameters. However, in case of determining the heat transfer area with acceptable exergy efficiency has a vital importance of first investment cost of HRSG.

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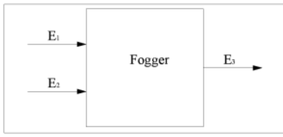
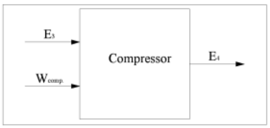
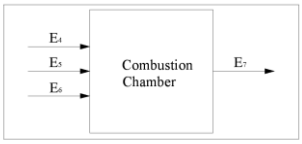
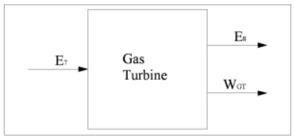
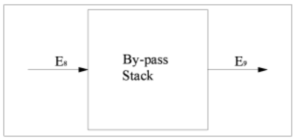
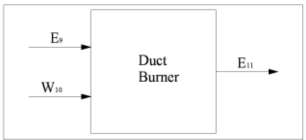
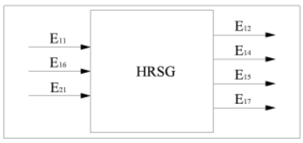
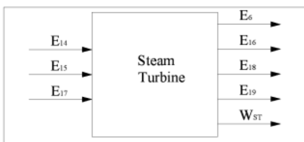
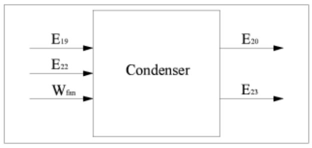
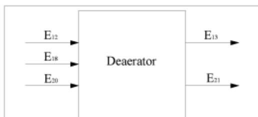
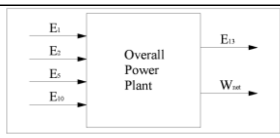
Appendix 1. Exergy formulas and efficiencies of each component of the power plant

Components	$\sum \dot{E}_{in}$	$\sum \dot{E}_{out}$	$\dot{W}$	$\dot{E}_D$	**Schematic**	$\varepsilon = \frac{\dot{E}_P}{\dot{E}_F}$
Fogger	$\dot{E}_1 + \dot{E}_2$	$\dot{E}_3$	-	$\dot{E}_1 + \dot{E}_2 - \dot{E}_3$		$\frac{\dot{E}_3}{\dot{E}_1 + \dot{E}_2}$
Compressor	$\dot{E}_3$	$\dot{E}_4$	$-\dot{W}_{com}$	$-\left(-\dot{W}_{com}\right) + \dot{E}_3 - \dot{E}_4$		$\frac{\dot{E}_4 - \dot{E}_3}{\dot{W}_{com}}$
Combustion Chamber	$\dot{E}_4 + \dot{E}_5 + \dot{E}_6$	$\dot{E}_7$	-	$\dot{E}_4 + \dot{E}_5 + \dot{E}_6 - \dot{E}_7$		$\frac{\dot{E}_7}{\dot{E}_4 + \dot{E}_5 + \dot{E}_6}$
Gas Turbine	$\dot{E}_7$	$\dot{E}_8$	$\dot{W}_{GT}$	$-\left(\dot{W}_{GT}\right) + \dot{E}_7 - \dot{E}_8$		$\frac{\dot{W}_{GT}}{\dot{E}_7 - \dot{E}_8}$
By-pass stack	$\dot{E}_8$	$\dot{E}_9$	-	$\dot{E}_8 - \dot{E}_9$		$\frac{\dot{E}_9}{\dot{E}_8}$
Duct burner	$\dot{E}_9 + \dot{E}_{10}$	$\dot{E}_{11}$	-	$\dot{E}_9 + \dot{E}_{10} - \dot{E}_{11}$		$\frac{\dot{E}_{11}}{\dot{E}_9 + \dot{E}_{10}}$
HRSG	$\dot{E}_{11} + \dot{E}_{16} + \dot{E}_{21}$	$\dot{E}_{12} + \dot{E}_{14} + \dot{E}_{15} + \dot{E}_{17}$	-	$(\dot{E}_{11} + \dot{E}_{16} + \dot{E}_{21}) - (\dot{E}_{12} + \dot{E}_{14} + \dot{E}_{15} + \dot{E}_{17})$		$\frac{(\dot{E}_{14} + \dot{E}_{17} - \dot{E}_{21}) + (\dot{E}_{15} - \dot{E}_{16})}{\dot{E}_{11} - \dot{E}_{12}}$
Steam Turbine	$\dot{E}_{14} + \dot{E}_{15} + \dot{E}_{17}$	$\dot{E}_6 + \dot{E}_{16} + \dot{E}_{18} + \dot{E}_{19}$	$\dot{W}_{ST}$	$-\left(\dot{W}_{ST}\right) + (\dot{E}_{14} + \dot{E}_{15} + \dot{E}_{17}) - (\dot{E}_6 + \dot{E}_{16} + \dot{E}_{18} + \dot{E}_{19})$		$\frac{\dot{W}_{ST}}{(\dot{E}_{14} + \dot{E}_{15} + \dot{E}_{17}) - (\dot{E}_6 + \dot{E}_{16} + \dot{E}}$
Condenser	$\dot{E}_{19} + \dot{E}_{22}$	$\dot{E}_{20} + \dot{E}_{23}$	$-\dot{W}_{fan}$	$-\left(-\dot{W}_{fan}\right) + (\dot{E}_{19} + \dot{E}_{22}) - (\dot{E}_{20} + \dot{E}_{23})$		$\frac{\dot{E}_{19} + \dot{E}_{22}}{\dot{W}_{fan} + (\dot{E}_{20} + \dot{E}_{23})}$
Deaerator	$\dot{E}_{12} + \dot{E}_{18} + \dot{E}_{20}$	$\dot{E}_{13} + \dot{E}_{21}$	-	$(\dot{E}_{12} + \dot{E}_{18} + \dot{E}_{20}) - (\dot{E}_{13} + \dot{E}_{21})$		$\frac{\dot{E}_{21} - \dot{E}_{20} - \dot{E}_{18}}{\dot{E}_{12} - \dot{E}_{13}}$
Overall Power Plant	$\dot{E}_1 + \dot{E}_2 + \dot{E}_5 + \dot{E}_{10}$	$\dot{E}_{13}$	$\dot{W}_{net}$	$(\dot{E}_1 + \dot{E}_2 + \dot{E}_5 + \dot{E}_{10}) - \dot{E}_{13} - \left(\dot{W}_{net}\right)$		$\frac{\dot{W}_{net}}{\dot{E}_1 + \dot{E}_2 + \dot{E}_5 + \dot{E}_{10}}$