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USING TWO TYPES OF HEAT RECOVERY STEAM GENERATOR FOR FULL REPOWERING A STEAM POWER PLANT AND ITS ANALYSIS BY EXERGY METHOD

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ABSTRACT

Repowering is defined as adding gas turbine unit(s) to a steam cycle and using exhaust gases to increase cycle efficiency. There are two main methods of repowering: full repowering and partial repowering. Full repowering method is used as a common and tested method to reconstruct old steam power plants and enhance their efficiency. In this research Be'sat steam power plant in Tehran has been considered as a reference steam power plant. This old power plant has been designed by General Electric Corporation with 31.46% efficiency, while its current efficiency is 26.81%. Two separate combined cycles with two different heat recovery steam generators have been designed by full repowering (dual pressure, single pressure). Thermal and exergy efficiencies of the repowered cycles have been computed by exergy analysis; their weaknesses and strengths in using gas turbine exhaust gas exergy have been investigated. It has been indicated by the results that thermal efficacy and exergy for single pressure state are 41.84%, 43.24% and for dual pressure state are 43.06%, 44.39% respectively. It can be inferred from results that in both cycles combustion chamber in the gas section and HRSG in the steam section have the highest exergy losses.

KEYWORDS: Steam cycle, Repowering, Heat Recovery Steam Generator, Exergy analysis, Efficiency

1. INTRODUCTION

Nowadays energy plays remarkable role in all aspects of life as one of the most significant issues of human daily life. In between power plants which generate electricity are major sources of energy generation. Among power plants generating electricity, combined cycle power plants in comparison to gas and steam cycle power plants have greater efficiencies; because both sources of generating power are used in the combined cycle power plants.In this research it has been tried to obtain a new combined cycle using full repowering method. Weaknesses and strengths of the obtained cycle would be revealed using exergy analysis. Repowering includes utilization of exhaust gas of gas turbine(s) to upgrade steam power plant performance. There are two main methods of repowering: full repowering and partial repowering. Full repowering is the most common way of repowering. In this method an old boiler is replaced by a HRSG and a gas turbine (or turbines). This method is beneficial to the power plants with minimum age of 25 years. Idea of using this method has been suggested in 1949 for the first time and has been utilized in 1960 (Stoll et al., 1994). According to old age and low efficiency of country steam power plants idea of repowering has been strengthen. In full repowering various ways can be performed owing to the way of omitting the feed water heater among following methods:

- Omitting low pressure feed water heater.
- Omitting low and high pressure feed water heater.
- Not omitting feed water heaters.

Be'sat is one of Iran's old steam power plants. Its old age has been led to an efficiency reduction in comparison to its utilization time (Ameri, 1999). Modeling steam power plant cycles have been done according to the presented technical documents by General Electric Corporation. In this research steam turbines have been modeled in the first place, and thereafter their specifications have been considered in new concerning full repowering scenarios; feed water heaters have been omitted because of power plant old age. HRSGs and gas turbine cycle have been designed proportional to the steam turbines in the reference cycle. New combined cycles have been designed considering all restrictions. Single pressure boilers without reheat and dual pressure boilers with reheat have been used. Since there is a steam injection line into steam turbines, a recovery boiler has been used in new repowered cycle to be in better conformity with the reference cycle.

Also in the second design a dual pressure HRSG with reheat (according to Bandar Abbas power plant) has been used. Finally it has been tried to analyze exergy efficiency, exergy losses, temperature of the stack exhausting gas etc. by exergy

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analysis of the two repowered cycles. EES thermodynamic software has been used for modeling new cycles.

Some researchers have been conducted in the fields of repowering and exergy analysis of combined cycles in the previous years. For example, full repowering has been done by Hosseinalipou et al for a certain steam power plant; economic and technical specifications of the new cycle have been optimized by them (Hosseinalipou et al., 2011). Also it should be mentioned that various methods of repowering have been investigated by Mehrpanahi et al(2011) in country electricity generating power plants to measure their effect on electricity generating costs (Mehrpanahi et al., 2011).Full repowering of a steam power plant has been performed by Gambini and Guizzi (1989) to generate more power (Gambini and Guizzi, 1989). In a similar research practical restrictions of a steam cycle full repowering have been investigated by Brander and Chase (1992) (Chase, 1992). Using another method of repowering parallel feed water heat recovery have been optimized by Mehrpanahi and Hoseinalipoor (2011)for Shahid Rajaei power plant in Tehran to decrease electricity generating costs (Mehrpanahi and Hoseinalipoor, 2011). In another research three types of HRSG used in combined cycles have been evaluated and compared by Tajik Mansouri and Ahmadi (2012) to analyze effect of vapor pressure level enhancement in power and efficiency increase and reduction in combined cycles exergy destruction (Mansouri and Ahmadi, 2012). Power and efficiency enhancement using a HRSG triple pressure has been investigated by Bassily (2008). Duct burner and reheating have been used by them to increase generating power (Bassily, 2008). It was believed by Franco and Russo (2002) that higher efficiency and power can be obtained by optimizing effective parameters on HRSG and using different HRSGs along with decreasing pinch temperature and increasing levels of heat absorption in the boiler (Francoand Russo, 2002).

2. REFERENCE CYCLE SPECIFICATIONS

Be’sat power plant is an old steam power plant designed by General Electric Corporation with 31.46% efficiency while its current real efficiency is 26.81%. Efficiency reduction has been caused by power plant old lifetime and exhaustion (Ameri, 1999). A v94.2 gas turbine made by Simense Company with power of 160MW has been used in the gas section of the power plant. Steam cycle has a heat boiler, two high pressure turbines, two intermedial turbines and two low pressure turbines. Also a condenser and a set of feed water reheaters can be seen in this cycle. A portion of steam turbines exhaust gas is extracted to the feed waters and reheats the boiler feed water. Figure1 shows the Schematic diagram of Be'sat steam power plant. Thermodynamic properties of steam/water in different points of the reference cycle have been shown in table 1.

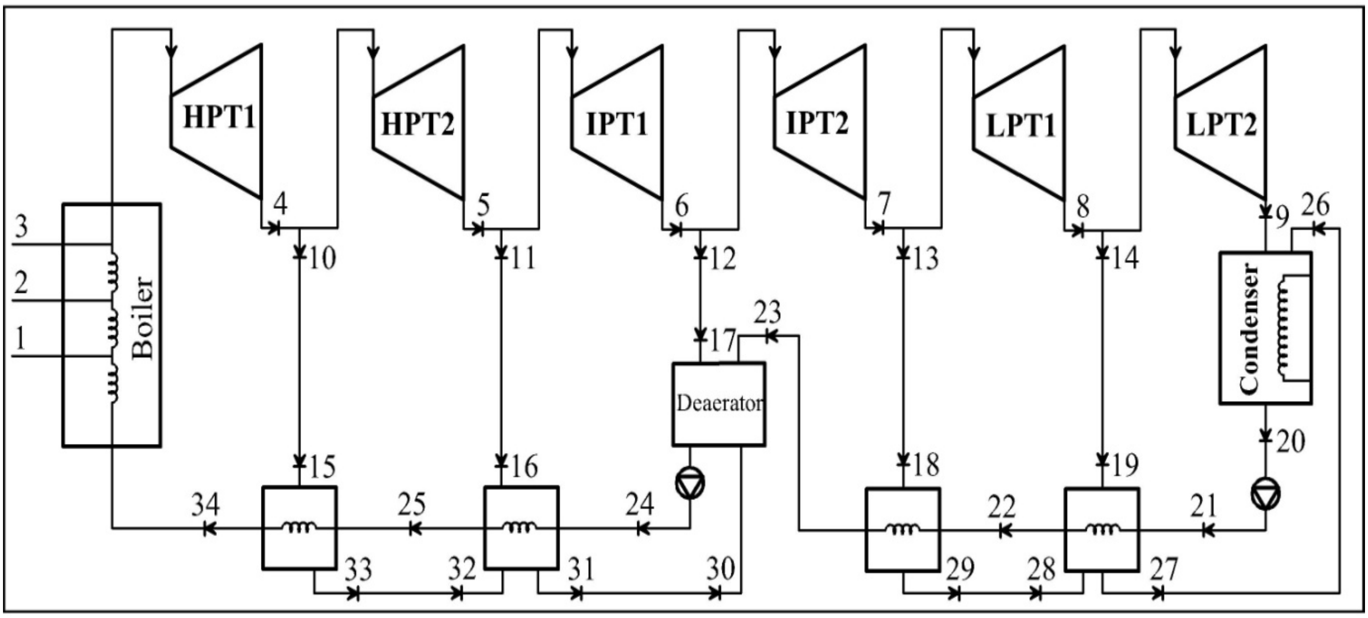

Figure 1. Schematic Diagram Of Be'sat Steam Power plant

Table 1. Water/Steam Properties of Be'sat Plant

point	Temperature (K)	Pressure (bar)	Entropy (KJ/Kg ^{0}K)	Mass Flow (Kg/S)	Enthalpy (KJ/Kg)
1	581.7	96.61	3.334	91.94	1392.5
2	578.0	91.78	5.665	91.94	2739.0
3	783.1	87.20	6.708	91.94	3415.0
4	663.0	33.60	6.826	84.58	3200.6
5	580.4	17.23	6.873	78.58	3047.5
6	476.7	6.50	6.941	73.05	2854.9
7	392.8	1.68	7.211	68.09	2707.0
8	365.6	0.77	7.090	62.67	2533.2
9	315.8	0.08	7.321	62.67	2298.8
10	663.0	33.60	6.826	7.36	3200.6
11	580.4	17.23	6.873	6.00	3047.5
12	476.7	6.50	6.941	5.33	2854.9
13	403.0	2.68	7.001	4.96	2708.6
14	365.6	0.77	7.090	5.42	2533.2
15	635.7	31.26	6.766	7.36	3141.5
16	558.9	16.02	6.825	6.00	3001.5
17	466.5	6.05	6.930	5.33	2834.6
18	400.6	2.50	6.897	4.96	2654.4
19	365.6	0.72	6.979	5.42	2533.2
20	316.2	0.72	0.612	73.05	180.0
21	316.2	6.43	0.610	73.05	180.0
22	358.3	6.24	1.132	73.05	355.9
23	393.6	6.05	1.527	73.05	503.9
24	435.3	108.08	1.952	91.94	690.8
25	471.2	104.84	2.270	91.94	834.2
26	315.8	0.08	0.727	10.38	216.4
27	324.9	0.14	0.725	10.38	216.4
28	363.3	0.71	1.250	4.96	398.0
29	368.3	0.85	1.250	4.96	398.0
30	432.4	6.05	2.061	13.36	726.9
31	444.4	8.16	2.059	13.36	726.9
32	455.6	16.02	2.330	7.36	852.4
33	473.2	18.62	2.330	7.36	852.4
34	508.2	101.69	2.641	91.94	1015.1

3. REPOWERING OPERATION PROCEDURE

- Gas turbine and steam turbines have been considered exactly according to the reference cycle.
- Size of used HRSG and gas turbine have been considered with regard to maximum inject Table steam to the first high pressure turbine in the thermodynamic conditions of the reference cycle (turbine inlet pressure and temperature of the reference cycle) (Hosseinalipou et al., 2011).
- Condenser limited capacity to accommodate current cycle injecting steam should be considered. Measure of the condenser capacity to accommodate extra steam is approximately equal to the amount of extra steam entered to the cycle by extraction deletion (Hosseinalipou et al., 2011).
- Since power plant has a wet cooling tower condenser pressure usual range is 0.068 to 0.136 bars (Mottaghian, 1999). In this research condenser operational pressure which is equal to the condenser pressure in the reference cycle, is considered as a constant and equal to 0.72 bar.
- Selected HRSGs are double pressure with reheat or single pressure without reheat boilers. Since reference cycle has a steam injection line to the steam turbines, a single pressure turbine has been used to be in better correspondence with the reference cycle. A double pressure boiler has been used in alternative design to utilize exhaust gas of gas turbine set perfectly.
- In single pressure repowered cycle steam is entered to the first high pressure turbine in one pressure level. When double pressure boilers with reheat are used

high pressure line is connected to the high pressure turbine and low pressure line is connected to the intermedial pressure turbine. In the double pressure type high pressure exhaust gas is entered to the reheater set and after further heat absorption, it is returned to the intermedial pressure turbine.

- Pressure generated by water pumps in the transfer pipelines will face some losses, so to approach real conditions some pressure losses are considered such as 3.5% in economizer pipes, 3% in superheater, 5% in reheater and 5.5% in water/Steam pipes. There is no pressure loss in evaporator(Escosaand Romeo,2009). Also it should be mentioned that 5% of gas energy has been considered as energy losses related to the gas in the boiler (Hosseinalipou et al., 2011).
- If the boiler exhaust has lower temperature than acidic dew point of the gas mixture, water vapor existing in it would be liquefied on pipes of economizer and pre heater; then it would be blended with other combustion products like CO2 and a corrosive acid would be formed; finally it would cause pipes damage. To prevent this incident 361^0K has been considered for acidic dew point of gas mixture, so boiler exhaust temperature would never be lower than this temperature (Mansouri and Ahmadi, 2012; Ahmadi and Dincer,2011).

4. ENERGY ANALYSIS

Energy balance has been used for control volume in steady state conditions to model cycles(Mansouri and Ahmadi, 2012). Set of gas turbines have been considered according to the simple brayton cycle as can be seen in figure 2.

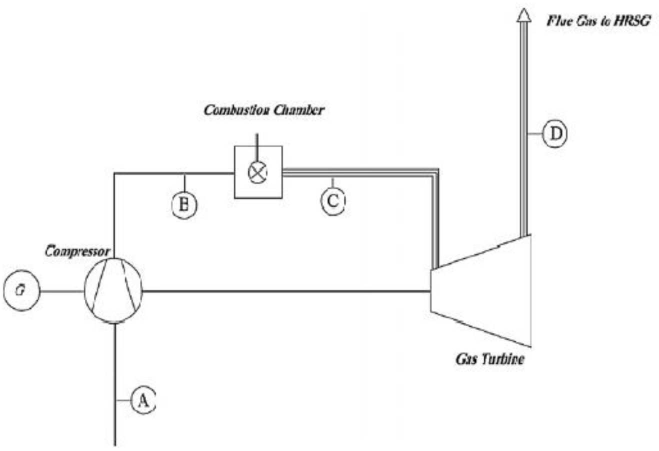

Figure 2. Schematic Diagram Of Gas Turbine Cycle

*4.1. Air Compressor (*Mansouri and Ahmadi, 2012*)*

$$T_B = T_A [1 + \frac{1}{\eta_{AC}} (r_c^{\frac{\gamma_a - 1}{\gamma_a}} - 1)] \quad (1)$$

$$\dot{W} = \dot{m}_a . C_{p,a} (T_B - T_A) \quad (2)$$

$C_{p,a}$ has been defined as a function of temperature(Ahmadi and Dincer, 2011):

$$C_{p,a}(T) = 1.04841 - (\frac{3.8371T}{10^4}) + (\frac{9.4537T^2}{10^7}) - (\frac{5.49031T^3}{10^{10}}) + (\frac{7.9298T^4}{10^{14}}) \quad (3)$$

4.2. Combustion Chamber (Mansouri and Ahmadi, 2012)

$$m_a h_B + \dot{m}_f LHV = \dot{m}_g h_C + (1 - \eta_{AC}) \dot{m}_f LHV \quad (4)$$

4.3. Gas Turbine (Hosseinalipou et al., 2011)

$$T_D = T_C [1 - \eta_{GT} (1 - (\frac{P_C}{P_D})^{\frac{1-\gamma_a}{\gamma_a}})] \quad (5)$$

$$\dot{W}_{net} = \dot{W}_{GT} - \dot{W}_{AC} \quad (6)$$

$$\dot{m}_g = \dot{m}_f + \dot{m}_a \quad (7)$$

$$\dot{m}_a = \frac{\dot{W}_{GT}}{(1 + AF) C_{P,g} (T_3 - T_4) - C_{P,g} (T_2 - T_1)} \quad (8)$$

$$C_{p,g}(T) = 0.991615 - (\frac{6.99703T}{10^4}) + (\frac{2.7129T^2}{10^7}) - (\frac{1.22442T^3}{10^{10}}) \quad (9)$$

Constant values of gases in the products of combustion 0.2944 KJ/Kg^{0}c, losses due to combustion chamber 3%, low heat value of the fuel equals to methane heat value which is 50000 KJ/Kg, combustion chamber efficiency has been considered 97%, gas specific heat capacity and air used have been computed as a function of temperature.

4.4. HRSG

Two kinds of HRSGs have been investigated in this research. Because equations are similar just equations related to the dual pressure HRSG are listed below.

Approach temperature difference has been considered 40 ^{o}K (Braccoand Siri,2010) and15^{o}K, respectively in design of single and dual pressure boilers. Boilers have been designed in a way that pinch temperature difference cannot be lower than 10 ^{o}K in any kind of them.

Following equations can be written using energy equation for steam/water and gas in HRSG various sections(Hosseinalipou et al., 2011; Kumar et al., 2007).

Steam/water:

$$T_{gt,out,pre,eva} = T_{pre,eva} + DELTA\_T_{pre,pinch} \quad (10)$$

$$T_{gt,out,hp,eva} = T_{hp,eva} + DELTA\_T_{hp,pinch} \quad (11)$$

$$T_{gt,out,lp,eva} = T_{lp,eva} + DELTA\_T_{lp,pinch} \quad (12)$$

Gas:

E to F

$$\dot{m}_{g,in,HRSG} . C_{p,gt} . (T_E - T_F)(1 - E_{Loss}) = \dot{m}_{hp}(h_{12} - h_{11} + h_{16} - h_{15}) \quad (13)$$

F to G

$$\dot{m}_{g,in,HRSG} . C_{p,gt} . (T_F - T_G)(1 - E_{Loss}) = \dot{m}_{hp}(h_{11} - h_{10}) \quad (14)$$

G to H

$$\dot{m}_{g,in,HRSG} . C_{p,gt} . (T_G - T_H)(1 - E_{Loss}) = \dot{m}_{hp}(h_{10} - h_5) + \dot{m}_{Lp}(h_7 - h_5) \quad (15)$$

H to I

$$\dot{m}_{g,in,HRSG} . C_{p,gt} . (T_H - T_I)(1 - E_{Loss}) = (\dot{m}_{hp} + \dot{m}_{Lp})(h_5 - h_4) \quad (16)$$

I to J

$$\dot{m}_{g,in,HRSG} . C_{p,gt} . (T_J - T_I)(1 - E_{Loss}) = (\dot{m}_{hp} + \dot{m}_{Lp})(h_4 - h_2) \quad (17)$$

J to k

$$\dot{m}_{g,in,HRSG} . C_{p,gt} . (T_J - T_K)(1 - E_{Loss}) = (\dot{m}_{hp} + \dot{m}_{Lp})(h_2 - h_1) \quad (18)$$

4.5. Steam Turbine

Three classes of high pressure, intermedial and low pressure turbines exist in current cycle. Power generated by these turbines is as follows (Hosseinalipou et al., 2011):

$$\dot{W}_{ST} = \Sigma Stages\ \dot{m}_{ST,in}(h_{ST,in} - h_{ST,out}) \quad (19)$$

At the end thermal efficiency of the cycle can be calculated by following relation (Ameri et al., 2008):

$$\eta_{CC} = \frac{\dot{W}_{GT} - \dot{W}_{AC} + \dot{W}_{Steam}}{\dot{Q}_{in,CC}} \quad (20)$$

5. ENERGY ANALYSIS

Exergy analysis is defined on the basis of first and second laws of thermodynamic. Exergy analysis is a tool to analyze and determine system inefficiencies. Exergy is defined as follows (Sarabchiand Nabati, 2000): If a system has n subset with temperature T and pressure P and also mole fractions as y=1,2,…,n, exergy is defined as the maximum obtainable theoretical work in process from state(P , T , Y_i) to dead state of (P_0, T_0, Y_i).It should be mentioned that dead state $(P_0, T_0, Y_i)$is a system state which there is perfect balance between system and its environment with temperature T_0 and pressure P_0in this state is considered as the reference state. Exergy is divided into four sections (four kinds of exergy can be mentioned as): physical exergy, chemical exergy, kinematic exergy and potential exergy. In most exergy analysis just physical and chemical exergies are considered. In the current analysis kinematic and potential exergies have been ignored. Using first and second laws of thermodynamics, general exergy equation can be written as follows (Kumar et al., 2007; Francoand Russo, 2002):

$$\dot{E}_Q + \sum \dot{m}_i e_i = \sum \dot{m}_e e_e + \dot{E}_W + \dot{I} \quad (21)$$

e is unit exergy and $\dot{I}$ is exergy losses.

$$\dot{E}_Q = (1 - \frac{T_0}{T_i})\dot{Q}_i \quad (22)$$

$$\dot{E}_W = \dot{W} \quad (23)$$

Chemical and physical exergy of gas mixture:

$$e^{ch} = [\Sigma_{i=0}^{n} X_i e^{ch_i} + RT_0 \Sigma_{i=0}^{n} X_i LnX_i + G^E] \quad (24)$$

$$E^{ph} = (h - h_0) - T_0(S - S_0) \quad (25)$$

$G^E$is related to the amount of Gibbs free energy which can be ignored in low pressures of the gas mixture. 0indexesare used to show reference state and T_0 is the reference environment temperature (Kumar et al., 2007).Relation (26) and (27) are defined for fuel with C_XH_y Compositions and fuel exergy respectively.

$$\cdot \xi = 1.033 + 0.0169 \frac{y}{x} - \frac{0.0698}{x} \quad (26)$$

$$\xi = \frac{e_f}{LHV_f} \quad (27)$$

Amount of fuel exergy is shown by e_f . Combined cycle exergy and HRSG efficiencies are defined as follows (Sarabchiand Nabati, 2000).

$$\eta_{ex,CC} = \frac{\dot{W}_{GT} - \dot{W}_{AC} + \dot{W}_{Steam}}{\dot{E}_f} \quad (28)$$

$$\eta_{ex,HRSG} = \frac{\dot{E}_{Steam,out} - \dot{E}_{water,in}}{\dot{E}_{flue,gas,in} - \dot{E}_{flue,gas,out}} \quad (29)$$

6. REPOWERED CYCLES

Using reference cycle and its related data, two cycles have been considered for full repowering. Steam injection is just possible through one pressure line in one of the cycles while in other cycle steam can be injected with two different pressures. In both cycles two high pressure turbines, two intermedial turbines, two low pressure turbines, one gas turbine, one condenser and several feed water pumps have been used. In these cycles natural gas has been used as a fuel which its composition is presented in table.2 (Ameri et al., 2008; Mansouri and Ahmadi, 2012). Specifications of used turbines in both cycles are presented in the table.3 individually. Composition of the Combustion chamber exhaust gas has been shown in table.4. (Ameri et al., 2008).

Table 2. Volume Fraction Of The Natural Gas Component

Component of natural gas	Volume fraction (%)
Methane	98.57
Ethane	0.63
Propane	0.1
Butane	0.05
Pentane	0.04
Nitrogen	0.6
Carbon dioxide	0.01

Table 3. Some Characteristic Data of Gas Turbine

Parameter	Unit	Combined cycle configuration	
		single-pressure	dual-pressure
Fuel flow(LHV)	MW	50	50
Pressure ratio	-	15.98	15.98
TIT	k	1373	1353
Gas turbine outlet temperature	k	788.5	793.2
Power output	MW	160	160

Table 4. Combustion Product Mole Composition

Component	Molar composition (%)
N_2	74.463
O_2	12.623
CO_2	3.715
H_2O	8.303
Ar	0.897

Gas properties in different points of the repowered cycles are shown in tables 5.

Table 5. Stream Characteristics Of The Gas Turbine Cycle

point	Temperature (K)	Pressure (bar)	Mass Flow (Kg/S)	Specific exergy (KJ/Kg)	Exergy (MW)
A	298.15	1.013	544.9	0	0
B	733.2	16.19	544.9	417.3	227.3
C	1353	15.71	555.4	1013.6	562.8
D	793.2	1.013	555.4	235.2	130.6

It should be noted that similar gas turbines are used in both cycles.. Reference environment in this modeling is air with thermodynamic conditions of P=1.013 bar, T=298.15 ^{0}k and its molar composition is presented in table 6 (Sirinivas et al., 2008).

It can be observed in Figure 3 and Figure 4 that steam is entered in one pressure level (87.21bar) into the high pressure turbine in single pressure cycle. Steam is entered into high pressure and intermedial turbines in dual pressure cycle.

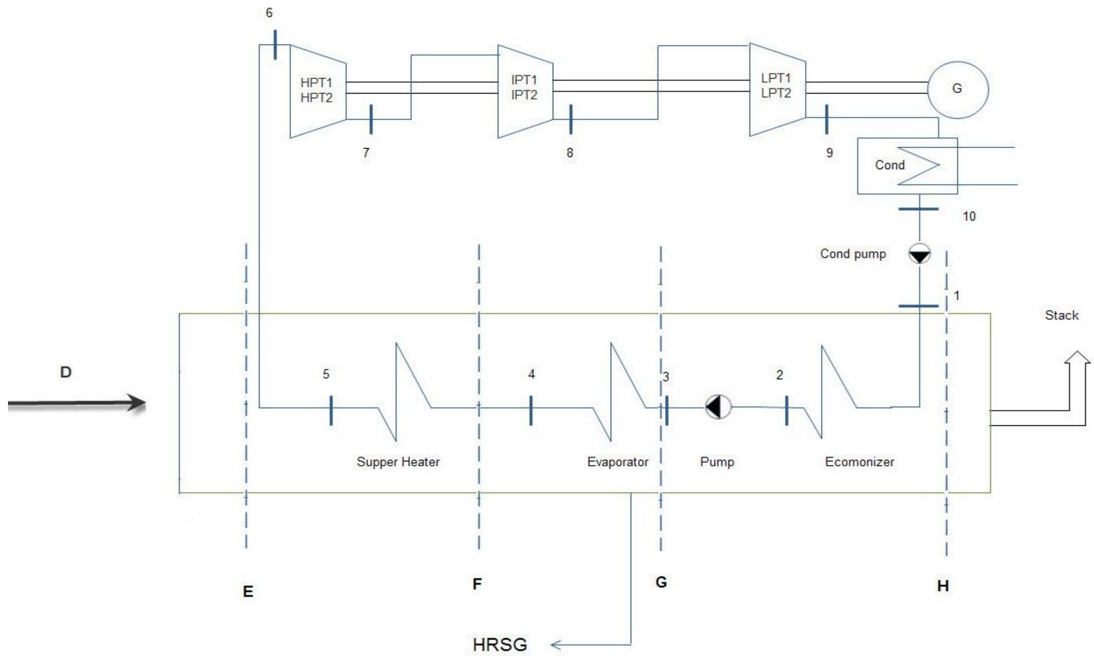

Figure 3. Schematic Diagram Of Steam Cycle Of Single Pressure Reheat Combined Cycle

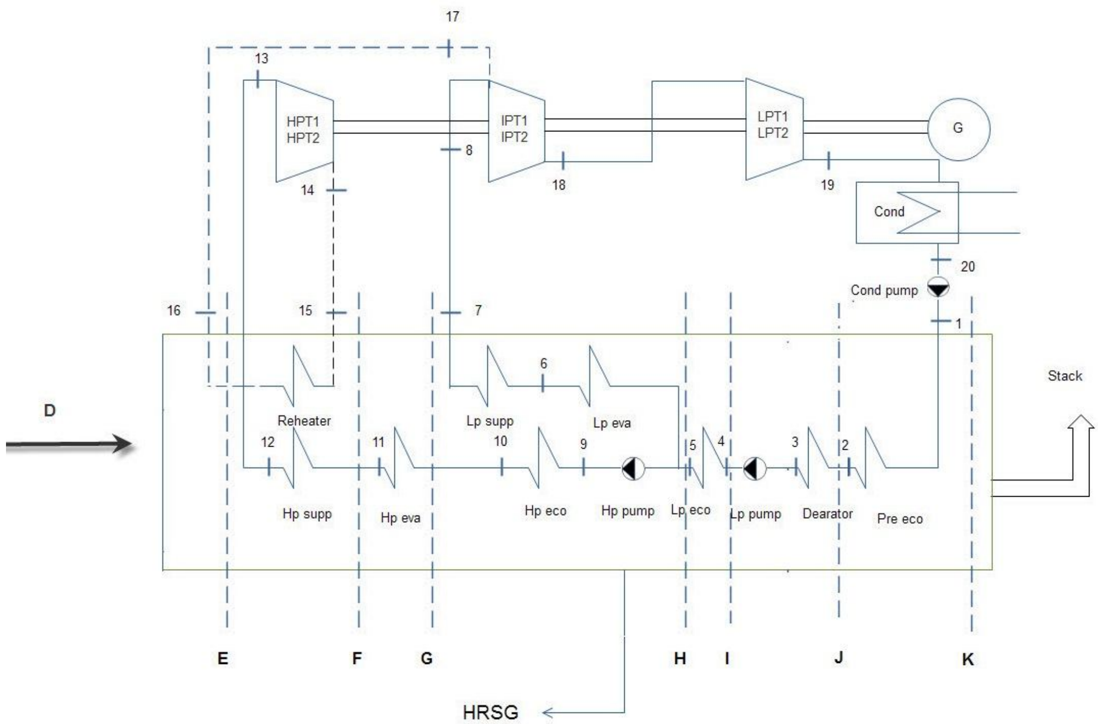

Figure 4. Schematic Diagram Of Steam Cycle Of Dual Pressure Reheat Combined Cycle

Table 6. Reference Environment Model Of Air

Air components	Molar fraction (%)
N_2	75.67
O_2	20.35
H_2O	3.03
CO_2	0.0345
CO	0.0007
SO_2	0.0002
H_2	0.00005
Others	0.91455

In dual pressure cycle high pressure exhaust gas after returning to the boiler and further reheating, would be mixed with LP pressure which is 9.827bar and would enter intermedial turbines. Basic and reference data to model combined cycles is listed according to the reference cycle properties (Table 7).

According to the Figure3 and Figure4 various thermodynamic relations (exergy and energy relations) have been used to find thermodynamic properties of steam/water and gas in all points of the cycle and tables 7-11 have been obtained for single and dual pressure states.

Table 7. Input Data For Thermodynamic Modeling Of The Cycles

Parameter	Unit	single pressure	dual pressure
HP inlet temp.	K	783.1	783.1
HP inlet press	Bar	87.21	87.21
LP inlet temp.	K	-	580[(a)]
LP inlet press.	Bar	-	10.27
Condenser press.	Bar	0.72	0.72

a=Hot reheat temperature

Table 8. Water/Steam Properties Of Single Pressure Combined Cycle

Stream number	Temperature (k)	Pressure (bar)	Entropy (KJ/Kgc°)	Mass flow (Kg/S)	Enthalpy (KJ/Kg)	Exergy (MW)
1	316.2	6.43	0.6127	53.16	180.9	0.138
2	540.2	6.237	2.936	53.16	1168	15.825
3	541.7	98.07	2.935	53.16	1168	15.841
4	580.2	98.07	5.648	53.16	2734	56.083
5	788.2	91.08	6.695	53.16	3422	76.071
6	783.1	87.21	6.708	53.16	3415	75.434
7	580.4	9.77	6.996	53.16	3067	49.78
8	392.8	1.22	7.38	53.16	2714	27.72
9	312.6	0.071	7.555	53.16	2347	5.438
10	316.2	0.72	0.6122	53.16	180.1	0.116

Table 9. Gas Side Properties Of Single Pressure HRSG

Point	Temperature (K)	Pressure (bar)	Mass Flow (Kg/S)	Specific exergy (KJ/Kg)	Exergy (MW)
E	793.2	1.013	555.4	235.17	130.61
F	733.2	1.039	555.4	192.50	106.91
G	590.2	1.026	555.4	98.59	54.75
H	499.9	1.013	555.4	51.42	28.55

Table 10. Water/Steam Properties Of Dual Pressure Combined Cycle

stream number	Temperature (k)	Pressure (bar)	Entropy (KJ/Kgc°)	Mass flow (Kg/S)	Enthaly (KJ/Kg)	Exergy (MW)
1	316.2	6.43	0.6127	73.05	180.9	0.207
2	418.5	6.237	1.795	73.05	612.5	5.99
3	433.5	6.237	1.947	73.05	677.3	7.41
4	433.6	11.61	1.947	73.05	678.1	7.46
5	443.3	11.27	2.043	73.05	720.2	8.44
6	458.3	11.27	6.545	17.17	2782	14.34
7	584.1	10.87	7.121	17.17	3072	16.36
8	580.4	10.27	7.135	17.17	3065	16.18
9	445	98.59	2.049	55.88	732.3	7.04
10	565.9	95.63	3.178	55.88	1302	20.09
11	580.9	95.63	5.641	55.88	2732	58.95
12	788.2	92.29	6.692	55.88	3422	79.96
13	783.1	87.21	7.211	55.88	3415	79.29
14	552.1	11.38	6.976	55.88	3001	51.73
15	552.1	11.38	6.976	55.88	3001	51.73
16	584.1	10.81	7.124	55.88	3072	53.24
17	580.4	10.27	7.135	73.05	3065	52.67
18	392.8	1.682	7.211	73.05	2730	42.74
19	318.7	0.098	7.402	73.05	2344	10.34
20	316.2	0.72	0.6122	73.05	180.1	0.16

Table 11. Gas side properties of dual pressure HRSG

Point	Temperature (K)	Pressure (bar)	Mass Flow (Kg/S)	Specific exergy (KJ/Kg)	Exergy (MW)
E	793.2	793.2	555.4	235.2	130.6
F	723.6	723.6	555.4	185.3	102.9
G	586	586	555.4	96.69	53.7
H	467.9	467.9	555.4	38.61	21.45
I	462.5	462.5	555.4	36.21	20.1
J	454.5	454.5	555.4	32.92	18.3
K	398.6	398.6	555.4	14.63	8.1

7. EXERGY ANALYSIS RESULTS

In the current research exergy efficiency and exergy losses have been presented for every single part of both cycles. Amount of heat absorption by HRSG components for both states of repowerment has been investigated. Finally thermal and exergy efficiencies of combined cycle along with power generated by steam turbines have been evaluated. This point should be kept in mind that these cycles have not similar overall designs and also pinch and approach temperatures. Every cycle has been designed and analyzed according to its specific restrictions.

7.1. Single Pressure Repowered Cycle

HRSG in this cycle has only three parts including economizer, evaporator and superheater. It is obvious in figure5 that combustion chamber has the maximum loss of exergy in gas cycle section which is due to chemical reactions and great temperature difference between igniter and injecting fuel (Ameri et al., 2008).HRSG also has the maximum exergy loss in the steam generation cycle.

Amount of heat absorption related to the HRSG components has been shown in Figure 6 As it can be seen evaporator which generates steam has the maximum amount of heat absorption since process of converting saturated water received from evaporator drum into saturated vapor exiting from evaporator pipes needs great amount of heat.

In fact this amount of heat absorption in HRSG components has been absorbed by economizer, evaporator and super heater and heat released by passing gas through space between pipes is used optimally. It should be noted that methods such as adding more level of heat absorption (for example reheater and pre heater feed water entering to the economizer), using pipes with fins, dividing economizer, evaporator and superheater levels along the boiler into multi sections would lead into added heat absorption and reduced heat losses due to exhaust gas of HRSG (Sharifi, 2011)

Amount of heat absorption from gas and its consequent temperature reduction would never continue to such an amount that water vapor in the gas would begin to liquefy; this phenomenon is avoided because it could help forming corrosive acids which can damage pipes. In single pressure cycle, temperature of stack exhaust gas (point H in the table.9) is greater than dew point of gas mixture and the phenomenon mentioned above is prevented. Temperature of the stack exhaust gas is very greater than considered limit $T_{stack} > 361^{o} K$ and shows great exergy losses through stack. According to Figure 5 exergy losses through stack is about 28.51 MW. In Figure7 exergy efficiency of the every single component of the cycle along with total exergy efficiency of the cycle is presented.

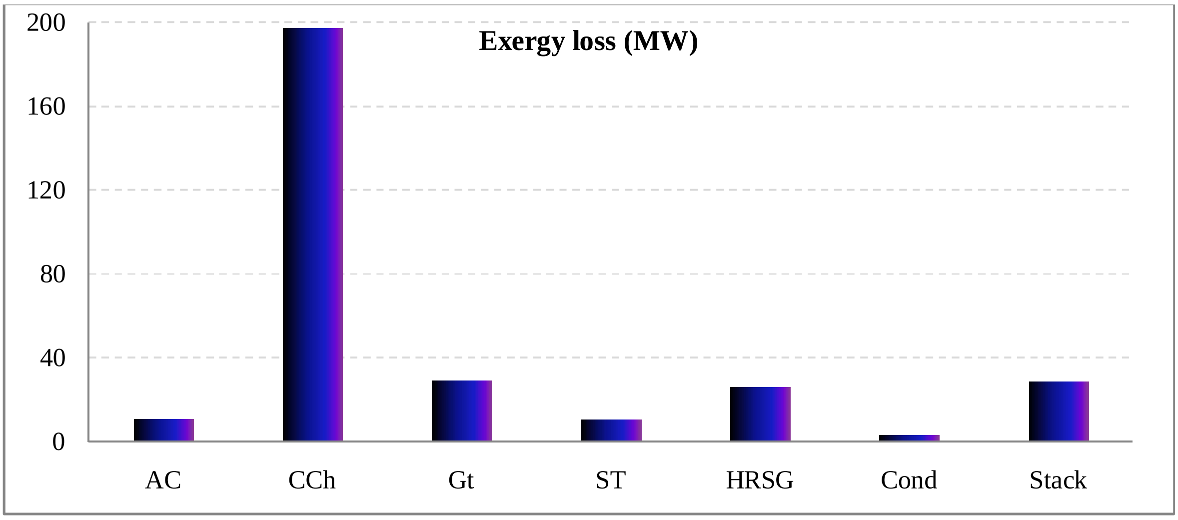

Figure 5. Exergy Losses For Single Pressure Combined Cycle Power Plant Components

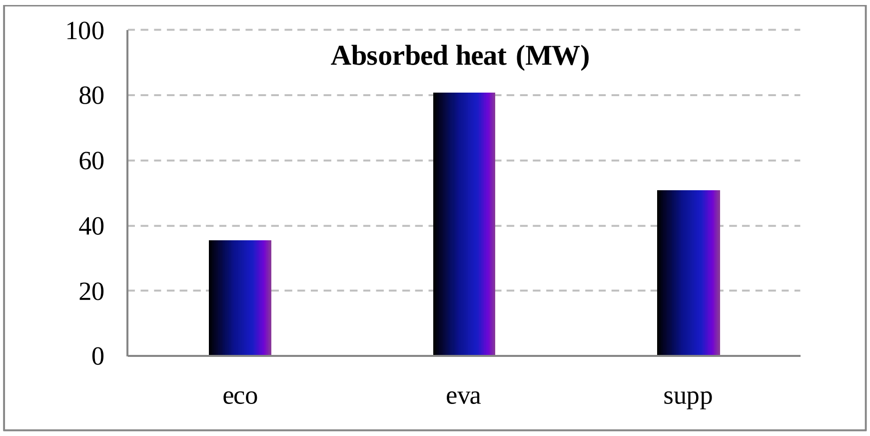

Figure 6. Absorbed Heat Rate Of Single-Pressure HRSG Components

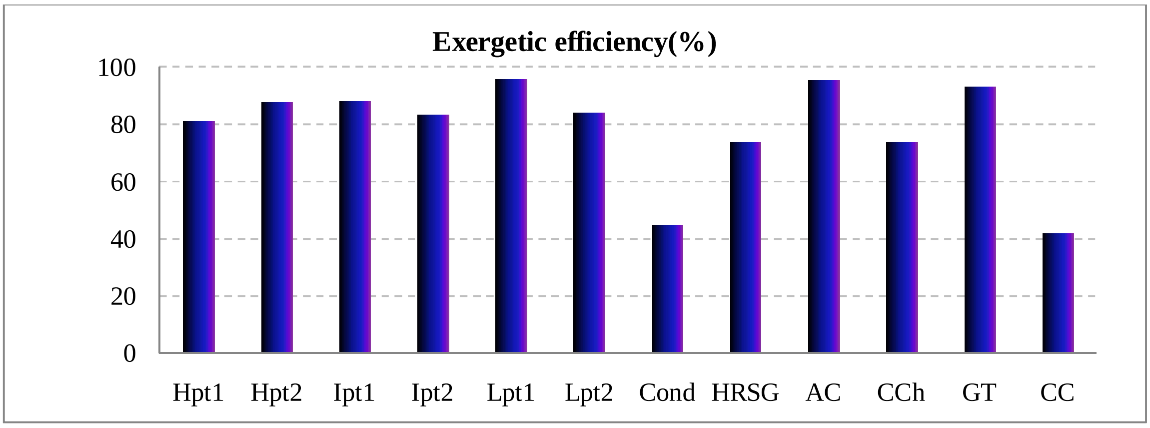

Figure 7. Exergetic Efficiency Of The Single-Pressure's Combined Cycle Components

As it can be seen in the gas cycle combustion chamber has the minimum exergy efficiency due to its great exergy losses which has been discussed above. In steam cycle condenser and intermedial turbine has the minimum and the maximum exergy efficiencies, respectively. Remarkable point is greatness of the turbine efficiency in the gas cycle, while according to the figure5 this component has significant exergy losses. This great efficiency is due to the remarkable amount of power generated by the gas turbine (Ameri et al., 2008).Combined cycle thermal and exergy efficiencies in this case are 43.24% and 41.85%, respectively.

7.2. Dual Pressure Repowered Cycle

In model of this cycle a dual pressure boiler with reheat unit has been utilized to use exhaust gas exergy in a better way. This boiler is composed of one pre heater feed water, two economizers, two evaporators, two superheaters and one reheat unit. Exregy losses of the combined cycle components has been shown in Figure 8.In the gas section just like single pressure cycle combustion chamber has the maximum exergy loss and in the steam section maximum exergy loss is related to HRSG. Considering HRSG results in both cycles, it would be necessary to design a suitable HRSG to decrease gas exergy losses. In fact amount of steam generation and thermodynamic properties of generated steam are directly affected by arrangement of HRSG components and its design. To have a better use of exhaust gas exergy, stack exergy losses has been reduced in comparison to single pressure HRSG. Amount of these losses in repowered dual pressure cycle is 8.103 MW. This value is less than concerning value for single pressure cycle which shows that boiler exhaust gas temperature would be lower in this state. This temperature in the repowered dual pressure cycle is equal to 398.6^{0}K (point K in table 11). Exhaust gas temperature is greater than acidic dew point of mixture in this cycle. Amount of heat absorbed by each component of HRSG is shown in figure9. It can be seen that maximum absorption occurs in HP evaporator and HP superheater. By the way in LP steam generating section, evaporator which generates steam, has the maximum amount of heat absorption. Remarkable point is absorption of 3.977MW heat by reheater unit and 36.26 MW heat by preheater and deaerator.

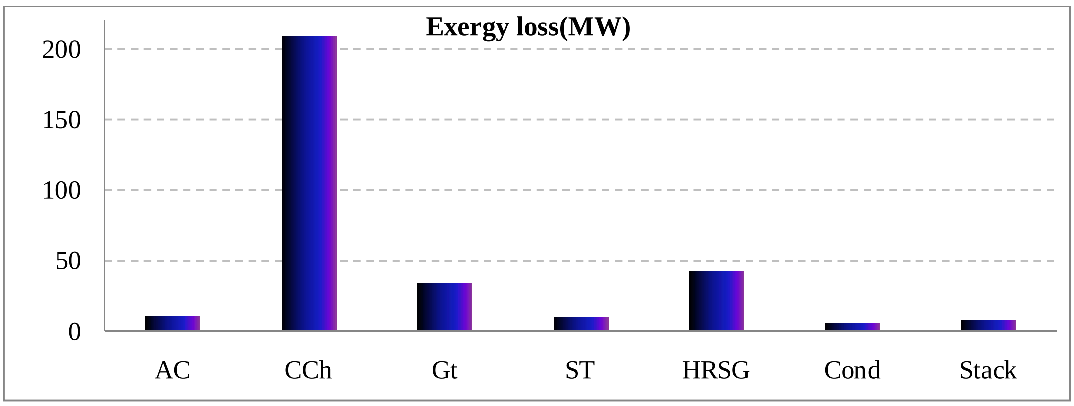

Figure 8. Exergy Losses For Dual Pressure Combined Cycle Power Plant Component

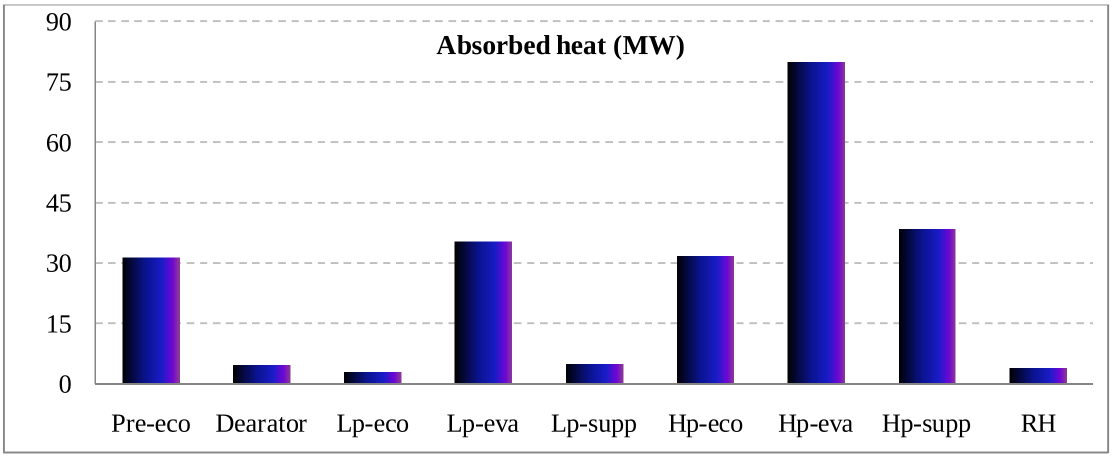

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Figure 9. Absorbed Heat Rate of Dual Pressure HRSG Components

In this cycle also combustion chamber has the minimum level of exergy efficiency in the gas section. In the steam generation section condenser has the minimum efficiency and the first low pressure turbine has the maximum exergy efficiency (Figure 10). Thermal and exergy efficiencies of the combined cycle in this case are 43.24% and 41.85% respectively. Power generated in dual pressure cycle turbines is greater than single pressure cycle because in dual pressure HRSG inlet steam mass flow to the steam turbines is 73.05kg/s while concerning value for single pressure HRSG is 51.72 kg/s. The greater steam mass flow the better would be use of turbine hot gas by dual pressure boiler than single pressure boiler. This fact is confirmed by higher thermal efficiency and exergy of dual pressure cycle in comparison to single pressure (Table 12).

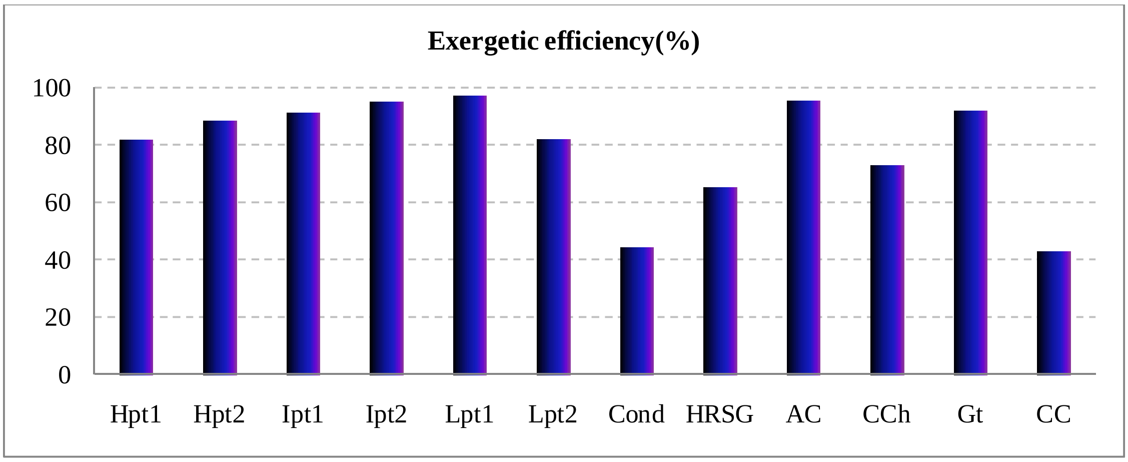

Figure 10. Exergetic Efficiency Of The Dual Pressure's Combined Cycles Components

Table 12. Energy And Exergy Indices Comparison Of Combined Cycle Configurations

Parameter	Unit	Combined cycle configuration	
		single-pressure	dual-pressure
Net power output of GT cycle	MW	160	160
Net power output of steam cycle	MW	59.54	74.23
Net power output of the plant	MW	219.54	234.23
Thermal efficiency of the plant	%	43.24	44.39
Exergetic efficiency of the plant	%	41.94	43.06

8. CONCLUSION

In this research full repowering operation of Be'sat steam power plant has been designed using two different types of HRSGs to enhance efficiency by converting the reference cycle to a combined cycle. Efficiency in single pressure state in comparison to the reference cycle has been increased up to 16.43% using this method, while this value for dual pressure state is 17.58%. It has been shown by exergy analysis of repowered combined cycles that maximum exergy losses are related to combustion chamber and HRSG. Increasing pressure levels of heat absorption in the boiler, heat absorption from turbine hot gas would be increased and boiler exhaust gas temperature would be decreased.

NOMENCLATURE

CCH	Combustion chamber
HRSG	Heat Recovery Steam Generator
TIT	Turbine Inlet Temperature
C_P	Specific heat at constant pressure (KJ/Kg ^{0}K)
LP	Low Pressure
HP	High Pressure
eco	
eva	Evaporator
supp	Superheater
pre-eco	Pre-heater economizer
Cond	Condenser
HPT	High Pressure Steam Turbine
IPT	Intermediate Pressure Steam Turbine
LPT	Low Pressure Steam Turbine
K	Generated Steame Coefficient
CC	Combined Cycle
Comp	Compressor
	Efficiency
e	Specific exergy (KJ/ Kg k)
E	Exergy (KJ)
GT	Gas turbine
G^E	Excess free Gibbs energy (KJ)
I	Exergy loss (KJ)
P	Pressure (bar)
Q	Heat transfer (KJ)
R	Gas constant (KJ Kg^{-1} K^{-1})
S	Specific entropy (KJ Kg^{-1} K^{-1})
T	Temperature
W	Work (KJ)
x	Molar fraction
S	Second
h	specific heat (KJ/Kg)
LHV	Lower heating value (KJ/Kg)
m	Mass flow (Kg)
S	Specific entropy(KJ/Kg ^{0}K)
r_{Comp}	Compressor pressure ratio

Subscripts and superscripts

g	Gas
t	Thermal
ex	Exergy
ST	Steam
pre	Pre-heater
ch	Chemical
e	Exit condition
f	Fuel
i	Inlet condition
ph	physical
a	Air
Comp	Air Compressor
Eva	Evaporator
g	Combustion gasses
gt	Gas turbine
Mix	Mixture
CV	Control volume
t	thermal
0	Dead state
s	Steam/water

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