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Technical Analysis of Conversion of A Steam Power Plant to Combined Cycle, Using Two Types of Heavy Duty Gas Turbines

S. Nikbakht Naserabad, K. Mobini*, A. Mehrpanahi, M. R. Aligoodarz

Department of Mechanical Engineering, Shahid Rajaee Teacher Training University, Tehran, Iran

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A B S T R A C T

Due to long life of steam power plants in Iran, transformation of steam cycles to combined cycles is under consideration. Bandar-Abbas steam power plant with capacity of 320 MW has been analyzed in this work. This old plant is located near the harbor city of Bandar-Abbas in south of Iran. Exergy analysis method is used to study the current and the repowered systems. Optimum state of the repowered cycle is also obtained using exergy analysis. In this work, a point by point analysis of Bandar-Abbas steam power plant is performed for different modes of full repowering, using exergy analysis method. For this purpose, V94.2 and V94.3A gas turbines are used and effect of duct burner is investigated for each case. The results show that at the best repowering mode, power plant efficiency is 34.5% higher than the design efficiency of the current steam plant. Minimum rate of exergy destruction rate is 6711 MW at this mode and the heat rate reduces by 26.6%. According to our results, increasing fuel consumption in duct burner and use of V94.2 gas turbine are not recommended for repowering of Bandar-Abbas power plant.

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NOMENCLATURE

CHP	combined heat and power
C_p	specific heat (kJ.kg^{-1}.K^{-1})
dA	surface element (m^2)
E	specific exergy (kJ.kg^{-1})
$\dot{E}^D$	exergy destruction (MW)
$\dot{E}^W$	exergy of work (MW)
E_l	heat loss (%)
H	efficiency
H	enthalpy (kJ.kg^{-1})
HR	heat rate (kJ.kW^{-1}.h^{-1})
HRSG	heat recovery steam generator
LHV	lower heating value (kJ.kg^{-1})
$\dot{m}$	mass flow rate (kg.sec^{-1}
N	number of gas turbines
P	pressure (Bar)
$\dot{Q}$	heat transfer rate (MW)
R	universal gas constant (kJ.kg^{-1}.K^{-1}
r_p	pressure ratio
S	specific entropy (kJ.kg^{-1}.K^{-1})
T	temperature (K)
$\dot{W}$	power (MW)
SUPERSCRIPTS	
CH	chemical
ex	exargy
KN	kinetic
PH	physical
PT	potential
SUBCRIPTS	
a	air
Ac	air compressor
Cch	combustion chamber
Cond	condenser
DB	duct burner
Eco	economizer
Eva	evaporator
f	fuel
fw	feed water
g	gas
GT	gas turbine
HP	high pressure
IP	intermediate pressure
ise	isentropic
LP	low pressure
Pre	preheater
RC	repowered cycle
ST	steam turbine
Sup	superheater
t	turbine
th	thermodynamic
0	dead state

*Corresponding Author's Email: k.mobini@srttu.edu (K. Mobini)

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1. INTRODUCTION

Since the advances of technology and energy-dependent processes rely mainlyon electricity, it is forecasted that there will be 40-45% increase in demand of electricity by 2050 [1]. Fossil fuels are the most important source of energy used to produce electricity[2], and they build approximately 90% of the global energy consumption. This fact makes two problems: increasing environmental pollution and faster reduction of fossil fuels resources. The ever-growing demand of electricity production requires construction of new thermal power plants. The main fuel consumed in thermal power plants is fuel oil, which contains a high percentage of sulfur and a high ratio of carbon to hydrogen. Therefore, the exhaust gases entering in the atmospheric air are highly pollutant, and the problem of environmental pollution is increased [2]. Even if a cleaner fuel was used, increase of CO_2 emission and its greenhouse effect, still persists. Today, there are researches in the context of carbon capture and storage technology (CCS) that aims 90 percent reduction of CO_2 emissions. But this technology is still in the research phase and is not expected to be available until 2025-2030 [3].

Considering the annual growth of electricity demand in Iran which is 4.5%[1], repowering of the steam power plants is an appropriate solution. Repowering increases the power plant efficiency and the net outlet power. It also reduces the formation of pollutants. Repowering of a plant means use of the heat exhausted from one or more gas turbine(s) in a steam cycle. As a result, efficiency of the power plant is highly increased, which means reduction of the fuel consumption and the air pollution. It was found that repowering leads to reduction of 10-30% of CO_2 emissions [3]. Repowering is an opportunity to optimize and improve the capabilities of a steam power plant. Currently, the best repowering method is using the gas turbine cycles which work with natural gas [2] . The idea of optimizing a steam power plant with gas turbine(s) is not a new one. In a report published 42 years ago in the UK Magnox power plant, there is a proposal to establish a gas turbine and use the outlet heat to increase the temperature of the steam power plant [4].The main aims of repowering can be summarized as follows [5, 6]:

- Increasing the plant lifespan,
- Increasing the plant efficiency,
- Increasing the plant net power output,
- Increasing the plant availability and reliability,
- Reducing the repair and maintenance costs,
- Increasing the plant operation flexibility,
- Reducing emission of the pollutants.

Repowering of a steam plant may increase thermal efficiency by 20-30% compared to its initial condition. It may also increase the net power output by 150-200% [7, 8]. There are two general repowering methods: minor repowering and full repowering. Minor repowering is applied to the modern power plants [9, 10]. The most typical methods of minor repowering are [11, 12]:

- Feed Water Heating method
- Hot Wind Box method
- Supplemental Boiler method

Full repowering is applied to the old steam plants. The average lifespan of these plants is more than 25 years. This method is the most common and the easiest method for plant repowering, and it is the best option to maximize the efficiency with the least cost [13].

This study investigates full repowering of Bandar-Abbas steam power plant with three different flow rates of feed water to the heat recovery steam generator:

1. The feed water flow rate through the heat recovery steam generator equals to the water flow rate through the condenser in the base power plant, when the base power plant works in the nominal load.
2. The boiler feed water flow rate through the heat recovery steam generator equals to the water flow rate through the condenser in the base power plant, when the base power plant works in the maximum load. In this case, the water flow rate is approximately 20% more than the nominal load.
3. The feed water flow rate through the heat recovery steam generator equals the water flow through the condenser in the base power plant, when all of the extractions of the steam turbine(s) are closed in the base power plant.

Each of the above three repowering conditions is investigated for two cases of using either V94.2 or V94.3A gas turbines. The effect of using a duct burner in each case is also investigated. As a result, the cycle has been analyzed in 24 different repowering cases. Finally, the best case is selected based on the highest efficiency using exergy analysis.

Most of the published researches on repowering are about partial repowering and there are only a few works on full repowering which have studied the general effect of repowering. However, in this work full repowering of the plant for the above 25 cases has been studies using exergy analysis. Moreover, most of the previous studies are based on a simple exergy analysis only, while in this work exergy dissipation of each element and its heat rates are also investigated. Presentation of the point to point data for steam and gas is an effective method for introduction of the repowering structure.

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[1] Energy Balance Sheet of Iran (EBSI)., Macro Programming Office of Power and Energy, Power Ministry, Iran, (2008).

[2]. Pace S., Graces11 D., Stenzel., Strategic Assessment of Repowering. *Interim Report*, June (1997)

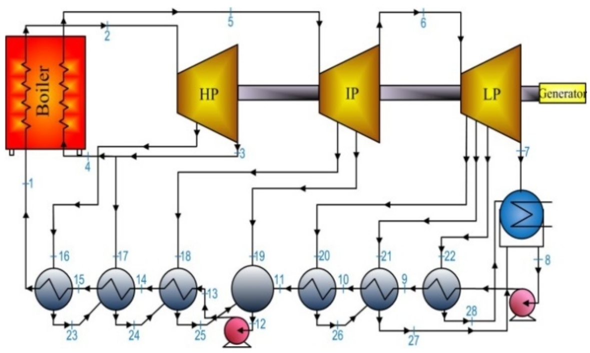


Figure 1. Schematic diagram of Bandar-Abbas power plant

2. BANDAR-ABBAS POWER PLANT SPECIFICATION

Selection of a proper repowering method is much dependent on the technical specification of the base power plant, thus the complete insight of the operation of the plant is necessary before repowering. The Bandar-Abbas steam power plant is located at the Persian Gulf coast near the city of Bandar-Abbas. It includes 4 units of 320 MW. As shown in Figure 1, each unit consists of steam turbines, condenser, feed water heaters, boiler and pumps. The outlet steam from the low pressure turbine enters the condenser and after condensation it is pumped into the feed water heaters. The heat required for the feed water comes from the steam turbine extractions to the heaters. There are three high pressure closed water heaters, three low pressure closed water heaters and one intermediate pressure open heater. The intermediate pressure open heater not only heats the feed water, but also it is the deaerator of the system. After being heated, the feed water enters the boiler, exits as dry steam with high pressure and then goes through steam turbines (see Figure 1).

3. MODELING OF BANDAR-ABBAS STEAM POWER PLANT

In this study, each section is modeled separately and point-to-point. Since repowering must be based on the ideal steam conditions for the steam turbines, the results from thermodynamic and exergy analysis of the base power plant are necessary for repowering. Table1 shows the results of modeling of the base cycle at design mode and nominal load. This modeling is performed using EES software. The numbers related to the points in Table 1 are related to the locations shown in Figure 1.

Table 2 shows the comparison of the results from modeling and the experimental data obtained from the plant. It is clear that the modeling results have good agreement with the experimental data.

TABLE 1. Results of modeling of Bandar-Abbas steam power plant cycle

Point	Temperature (K)	Pressure (bar)	Mass flow rate(kg.sec^{-1})	Enthalpy(kJ.kg^{-1})	Entropy(kJ.kg^{-1}.K^{-1})	Exergy(kJ.kg^{-1})
1	564.10	177.27	288.740	1287.27	3.1315	352.760
2	811.00	170.00	288.740	3392.86	6.4000	1477.32
3	599.30	38.780	246.010	3034.11	6.4998	1088.63
4	598.85	38.540	222.295	3033.61	6.5015	1087.59
5	811.00	34.920	222.295	3536.95	7.2669	1361.21
6	596.00	7.2300	207.434	3105.94	7.3634	901.231
7	341.00	0.1000	178.608	2401.13	8.1760	112.135
8	315.54	0.0860	207.940	177.500	0.6041	1.60100
9	339.21	7.6580	207.940	277.124	0.9062	10.5500
10	362.11	7.4800	207.940	373.089	1.1800	24.3300
11	398.51	7.3300	207.940	526.946	1.5849	56.6670
12	437.90	7.1300	288.740	696.346	1.9902	104.403
13	440.05	178.04	288.740	715.434	1.9910	123.250
14	475.37	177.74	288.740	869.160	2.3268	176.186
15	517.83	177.49	288.740	1061.11	2.7135	252.074
16	702.10	79.530	30.3900	3217.85	6.4816	1277.83
17	598.85	38.540	22.8500	3033.61	6.5015	1087.59
18	707.50	17.170	13.8200	3327.13	7.3111	1138.13
19	593.90	7.3500	13.7300	3101.25	7.3480	901.152
20	486.30	2.5400	12.8200	2857.50	7.3703	650.716
21	373.30	0.7700	7.34300	2679.10	7.4885	436.835
22	353.00	0.3000	8.31100	2545.31	7.5312	289.420
23	522.83	77.150	39.3930	1083.76	2.7807	254.539
24	480.37	37.140	53.2390	885.715	2.3954	172.147
25	445.05	16.660	67.0510	728.060	2.0596	115.275
26	367.11	2.5600	12.8200	393.735	1.2380	27.5600
27	344.20	0.7500	20.1640	297.442	0.9677	12.4010
28	341.21	0.2900	8.31100	284.886	0.9312	10.8010

TABLE 2. Comparison between the results from modeling and the steam power plant data

Modeling parameter	Unit	Modeling result	Power plant data	Error (%)
Total power	MW	325.3	320	1.66
Efficiency	%	39.36	38.70	1.71

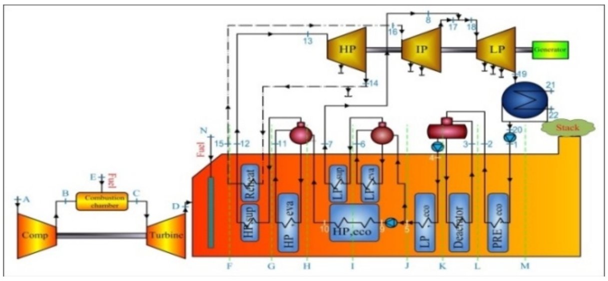


Figure 2. Exergy destruction at the repowered cycle components

4. REPOWERING OF STEAM POWER PLANT

Considering the working life of Bandar-Abbas steam power plant and its technical specifications, full repowering is the best approach for its repowering. In full repowering, the main boiler is replaced by a heat recovery steam generator in which the required heat is provided by one or more gas turbines. Considering the fact that the three steam turbines work in three different pressure levels and considering the conventional heat recovery steam generators produced in Iran for combined-cycle power plants, a double-pressured heat recovery steam generator with reheat is chosen to heat feed water and produce superheated steam. In order to provide the required heat for the heat recovery steam generator, two types of the conventional gas turbines have been tried: V94.2 and V94.3A. The structure of Bandar-Abbas steam power plant after repowering is shown in Figure 2.

5.GOVERNING EQUATIONS IN THERMODYNAMIC ANALYSIS

Modeling of the repowered cycle is separately done for each section. The governing equations of each section are as following [14, 15]:

5. 1. Compressor The outlet temperature is obtained assuming an adiabatic process[16]:

$$T_B = T_A.[1+\frac{r_{p,Ac}^{\left(\frac{k_a-1}{k_a}\right)}-1}{\eta_{ise,AC}}] \quad (1)$$

The specific heat of air at different temperatures is calculated using the following equation:

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

Equation (3) gives the required power for the compressor:

$$\dot{W}_{AC} = \dot{m}_a.\left[\frac{C_{p,a}.(T_B - T_A)}{\eta_{mec,AC}}\right] \quad (3)$$

The outlet pressure of the compressor is calculated by:

$$\frac{p_B}{p_A} = r_{p,Ac} \quad (4)$$

5. 2. Combustion Chamber In order to obtain the temperature of the exhaust gases from the combustion chamber, the energy balance equation is written as bellow:

$$\dot{m}_a C_{p,B} T_B + \dot{m}_f LHV + \dot{m}_f C_{p,f} T_f = \dot{m}_g C_{p,C} T_C + (1-\eta_{Cch})\dot{m}_f LHV \quad (5)$$

The gas specific heat at each temperatures is calculated using the following equation [16]:

$$C_{p,g} = 0.991615 + 6.99703.\frac{T_g}{10^5} + 2.7129.\frac{T_g^2}{10^7} - 1.22442.\frac{T_g^3}{10^{10}} \quad (6)$$

Air, fuel and gases mass flow rates are related by Equations (7) and (8):

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

$$\dot{m}_g = \dot{m}_a + \dot{m}_f \quad (8)$$

Equation (9) gives the pressure of the exhaust gases from the combustion chamber:

$$\frac{p_C}{p_B} = 1 - \Delta p_{Cch} \quad (9)$$

5. 3. Turbine The outlet temperature from the turbine is found from the following equation[17]:

$$T_D = T_C\cdot\left[1-\eta_{ise,t}\cdot\left(1-r_{pt}^{\left[\frac{1-k_g}{k_g}\right]}\right)\right] \quad (10)$$

The turbine pressure ratio $r_{p,t}$ is calculated by the following equation:

$$r_{p,t} = r_{p,Cch}.r_{p,Ac} \tag{11}$$

The net gas turbine power is related to the turbine and compressor powers as below:

$$\dot{W}_{net,GT} = \dot{W}_t - \dot{W}_{AC} \tag{12}$$

The net gas turbine power is calculated by the following equation:

$$\dot{W}_{net,GT} = \dot{m}_a.\begin{bmatrix} (1+FA).\eta_{mec,t}.\left(C_{p,g}.\left(T_C - T_D\right)\right) \\ -\left(\dfrac{C_{p,a}.\left(T_B - T_A\right)}{\eta_{mec,AC}}\right) \end{bmatrix} \tag{13}$$

The thermal efficiency of the gas turbine is defined as following:

$$\eta_{th,GT} = \frac{\dot{W}_{net,GT}}{\dot{m}_f LHV} \tag{14}$$

5. 4. Heat Recovery Steam Generator (HRSG) The governing equations of the heat recovery steam generator are obtained by energy balance between the inlet gases and the feed water. The inlet gases to this section include the outlet gases from the turbine(s) and the gases produced by the duct burner. The implemented duct burner at the inlet of the heat recovery steam generator has the task of making stability in the power plant at different conditions. Fuel flow rate at the duct burner cannot be more than $2kg.s^{-1}$[18].

The feed water entering the heat recovery steam generator comes from the condenser after pumping. After turning into superheated steam, it goes to the high pressure and low pressure turbines. The equations of heat recovery steam generator are as following:

$$\begin{aligned} &\dot{m}_{fw}.(h_{out,\,fw} - h_{in,\,fw}) \\ &= \dot{m}_g.C_{p,g}.(T_{out,g} - T_{in,g})(1 - E_l) \end{aligned} \tag{15}$$

$$\Delta T_{pinch} = T_{g,out,eva} - T_{fw,in,eva} \tag{16}$$

$$\Delta T_{approach} = T_{fw,eva} - T_{out,eco} \tag{17}$$

The pinch temperature difference is the temperature difference between the gases and the feed water during evaporation in the evaporator [19]. The reduction of this temperature difference to less than 5K is not economically efficient. The temperature difference between the evaporation temperature in the evaporator and the temperature of the condensed liquid entering the evaporator is called the approach temperature difference. This temperature difference is necessary to prevent evaporation in the economizer. However, increasing this temperature difference may cause thermal shock in the evaporator. Therefore, the proper range for this temperature difference is between 5K and 15K [20].

Two important parameters in the heat recovery steam generator are k and y. k is the ratio of the current feed water flow rate to the flow rate of the feed water in the base power plant at nominal load. y is the ratio of the flow rate of the steam passing through the high pressure section of the heat recovery steam generator to the flow rate of the total inlet feed water. The equations of these two parameters are as following:

$$K = \frac{\dot{m}_{fw,new}}{\dot{m}_{fw,nom,base,pp}} \tag{18}$$

$$y = \frac{\dot{m}_{hp,line}}{\dot{m}_{fw,new}} \tag{19}$$

5. 5. Steam Turbine When the extractions of the steam turbines are closed, the flow rate and state of the steam will change at each section. Therefore, the steam turbines will no longer work at the designed condition. There are three steam turbines with different operating pressures, and each turbine should be separately analyzed. The following equation is called Stodela equation which relates mass flow rates, temperatures and pressures for the first and the new cycles [21]:

$$\frac{\dot{m}_{new}}{\dot{m}_{first}} = \frac{p_{out,new}}{p_{out,\,first}}.\sqrt{\frac{T_{in,\,first}}{T_{in,new}}}.\sqrt{\frac{1-\left(\dfrac{p_{in,new}}{p_{out,new}}\right)^2}{1-\left(\dfrac{p_{in,\,first}}{p_{out,\,first}}\right)^2}} \tag{20}$$

After change of mass flow rate, the efficiency is changed. Having the initial efficiency, the new efficiency can be found using the following equation [21]:

$$\begin{aligned} \frac{\eta_{ise,st,new}}{\eta_{ise,st,\,first}} &= -1.0176.\left(\frac{\dot{m}_{new}}{\dot{m}_{fiest}}\right)^4 + 2.4443.\left(\frac{\dot{m}_{new}}{\dot{m}_{fiest}}\right)^3 \\ &-2.1812.\left(\frac{\dot{m}_{new}}{\dot{m}_{fiest}}\right)^2 + 1.0535.\left(\frac{\dot{m}_{new}}{\dot{m}_{fiest}}\right) + 0.701 \end{aligned} \tag{21}$$

Two general equations in the steam turbines are:

$$\eta_{ise,st} = \frac{(h_{in,st} - h_{out,st})}{(h_{in,st} - h_{out,st,ise})} \tag{22}$$

$$\dot{W}_{ST} = \dot{m}_{ST}.(h_{in,ST} - h_{out,ST}) \tag{23}$$

After closing the steam turbines extractions, the amount of steam flowing through the high pressure turbine increases. Therefore, the thermodynamic efficiency of this turbine highly decreases. Hence, in this study it is supposed that the low pressure turbine is flexible with the changes of operating conditions and the desired technical conditions can be achieved by applying changes in the new system.

5. 6. Condenser The vapor entering the condenser is condensed using sea water. The following equation is applied:

$$\dot{m}_{19}.h_{19} = \dot{Q}_{Cond} + \dot{m}_{20}.h_{20} \tag{24}$$

5. 7. Pump The condition of the outlet feed water from the pump depends on three parameters of the inlet fluid condition, the pump power and its efficiency. It can be found from the following equations:

$$\dot{m}_{fw}.h_{fw,out,Pump} = \dot{m}_{fw}.h_{fw,in,Pump} + \dot{W}_{Pump} \tag{25}$$

$$\eta_{ise,Pump} = \frac{h_{ise,\ fw,out,Pump} - h_{fw,in,Pump}}{h_{fw,out,Pump} - h_{fw,in,Pump}} \tag{26}$$

6. EXERGY ANALYSIS

6. 1. Exergy and Its Governing Equations In the science of thermodynamics, the first law of is based on energy and the second on exergy. Exergy is defined by some researchers as the part of energy which is convertible to other forms of energy [22]. In other words, exergy indicates the quality of energy and it is the maximum useful work obtained from a specific amount of energy[23]. The main aim of exergy is to determine the place and amount of the irreversible productions along a process[20].

Ignoring the nuclear, magnetic and electrical energies and the surface tension, exergy is comprised of four components in a thermal system: physical exergy (E^{PH}), kinetic exergy (E^{KN}), potential exergy (E^{PT}) and chemical exergy (E^{CH}) [15, 20]. Hence, the total exergy E is found by the following equation:

$$E = E^{PH} + E^{KN} + E^{PT} + E^{CH} \tag{27}$$

Similarly, the specific exergy of a system is calculated by the following equation:

$$e = e^{PH} + e^{KN} + e^{PT} + e^{CH} \tag{28}$$

In a steady process when the system is at rest relative to the environment, there is no change in the height and speed. Therefore, kinetic and potential exergies are negligible. In this case, physical exergy, also known as thermodynamic exergy [21], would be defined as the maximum theoretical useful work. The maximum theoretical useful work is the amount of work obtained from the interaction between a system and its environment until the system reaches the dead state [24].The physical exergy for steam is defined by the following [15, 16, 23]:

$$e^{PH} = (h - h_0) - T_0 (s - s_0) \tag{29}$$

Chemical exergy is the amount of useful work produced during the process of a system from limited dead state (at temperature T_0 and pressure P_0) to complete dead state in which the system is in total equilibrium with the environment (mechanical and thermodynamic equilibrium). Generally, exergy of gases consist of two parts. The first part is related to temperature and pressure difference between the system and the environment (with temperature T_o and pressure Po) which is the physical exergy and is defined as follows [15, 16]:

$$e^{PH} = C_{p(T)}\left[T - T_0 - T_0 \ln\left(\frac{T}{T_0}\right)\right] + RT_0\left(\frac{p}{p_0}\right) \tag{30}$$

The second part of exergy is the chemical exergy which is related to the difference between the partial pressure of gases and the environment pressure. For a mixture of gases at temperature T_o and total pressure of P_o, all the components are at temperature T_o, but their partial pressures are different from P_o. Therefore, they are not in mechanical equilibrium with the environment. The chemical exergy of a mixture of k gases is obtained from the following equation [25]:

$$\bar{e}^{CH} = -RT_0 \sum_{i=1}^{k} x_i \ln\frac{x_i^e}{x_i} \tag{31}$$

In this equation, x_i and x_i^e are the mole fractions of the component i relative to the gas mixture and the environment respectively. Chemical exergy of a fuel which is a mixture of k gases is calculated by the following equation [26]:

$$\bar{e}^{CH} = \sum_{i=1}^{k} x_i \bar{e}_i^{CH} + RT_0 \sum_{i=1}^{k} x_i \ln x_i + G \tag{32}$$

In this equation, G is the Gibbs free energy that can be ignored for the fuels with low pressure and the term $\bar{e}_i^{CH}$ is the standard chemical exergy of the component i [25].Due to the complexity of Equation (32), the chemical exergy of hydrocarbon fuels with chemical formula of C_aH_b is calculated by the simplified Equations (33) and (34)[15, 26, 27]:

$$e_f^{CH} = LHV.\gamma_f \tag{33}$$

$$\gamma_f = 1.033 + 0.0169\frac{b}{a} - \frac{0.0698}{a} \tag{34}$$

In the above equations, γ_f is the fuel exergy grade function [15].The thermodynamic and exergy efficiency of a system is defined by the following equations[23]:

$$\eta_{th,RC} = \frac{\dot{W}_{net,RC}}{\dot{Q}_f} \tag{35}$$

$$\eta_{ex,RC} = \frac{\dot{W}_{RC}}{\dot{E}_f} \tag{36}$$

$\dot{Q}_f$ and $\dot{E}_f$ are the fuel energy and exergy respectively and are obtained from the following equations [28]:

$$\dot{Q}_f = \left(N.\dot{m}_{f,Cch} + \dot{m}_{f,DB}\right).LHV \tag{37}$$

$$\dot{E}_f = \dot{E}_{f,Cch} + \dot{E}_{f,DB} = \left(N.\dot{m}_{f,Cch} + \dot{m}_{f,DB}\right).\gamma_f.LHV \tag{38}$$

In Equations (35) and (36), the total work is defined as following:

$$\dot{W}_{RC} = \dot{W}_{net,GT} + \dot{W}_{net,ST} \tag{39}$$

Heat rate of a system is defined as the amount of the heat required to produce one unit of net work. It is inversely proportional to thermal efficiency. For the repowered cycle, heat rate is defined as following:

$$HR_{RC} = \frac{\dot{m}_f LHV_f}{\dot{W}_{RC}}.3600 \tag{40}$$

TABLE 3. Exergy balance and exergy efficiency of the repowered power plant components

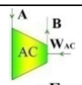
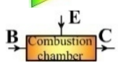
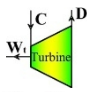
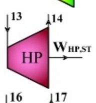
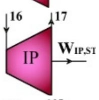
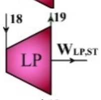
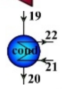
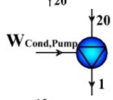
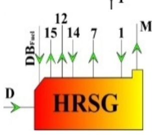
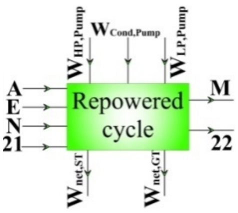
Component	Schematic	Exergy balance	Exergy efficiency
Compressor		$\dot{E}_A^a + \dot{E}_{AC}^W - \dot{E}_B^a = \dot{E}_{AC}^D$	$\eta_{AC}^{ex} = \frac{\dot{E}_B^a - \dot{E}_A^a}{\dot{W}_{AC}}$
Combustion chamber		$\dot{E}_B^a + \dot{E}_E^f - \dot{E}_C^g = \dot{E}_{Cch}^D$	$\eta_{Cch}^{ex} = \frac{\dot{E}_C^g}{\dot{E}_B^a + \dot{E}_E^f}$
Turbine		$\dot{E}_C^g - \dot{E}_D^g - \dot{E}_t^W = \dot{E}_t^D$	$\eta_t^{ex} = \frac{\dot{W}_t}{\dot{E}_C^g - \dot{E}_D^g}$
HP steam turbine		$\dot{E}_{13}^{fw} - \dot{E}_{14}^{fw} - \dot{E}_{HP,ST}^W = \dot{E}_{HP,ST}^D$	$\eta_{HP,ST}^{ex} = \frac{\dot{W}_{HP,ST}}{\dot{E}_{13}^{fw} - \dot{E}_{14}^{fw}}$
IP steam turbine		$\dot{E}_{16}^{fw} - \dot{E}_{17}^{fw} - \dot{E}_{IP,ST}^W = \dot{E}_{IP,ST}^D$	$\eta_{IP,ST}^{ex} = \frac{\dot{W}_{IP,ST}}{\dot{E}_{16}^{fw} - \dot{E}_{17}^{fw}}$
LP steam turbine		$\dot{E}_{18}^{fw} - \dot{E}_{19}^{fw} - \dot{E}_{LP,ST}^W = \dot{E}_{LP,ST}^D$	$\eta_{LP,ST}^{ex} = \frac{\dot{W}_{LP,ST}}{\dot{E}_{18}^{fw} - \dot{E}_{19}^{fw}}$
Condenser		$\left(\dot{E}_{19}^{fw} - \dot{E}_{20}^{fw}\right) - \left(\dot{E}_{22}^{sw} - \dot{E}_{21}^{sw}\right) = \dot{E}_{Cond}^D$	$\eta_{Cond}^{ex} = \frac{\dot{E}_{22}^{sw} - \dot{E}_{21}^{sw}}{\dot{E}_{19}^{fw} - \dot{E}_{20}^{fw}}$
Condenser pump		$\dot{E}_{20}^{fw} + \dot{E}_{Cond,Pump}^W - \dot{E}_1^{fw} = \dot{E}_{Cond,Pump}^D$	$\eta_{Cond,Pump}^{ex} = \frac{\dot{E}_1^{fw} - \dot{E}_{20}^{fw}}{\dot{W}_{Cond,Pump}}$
HRSG		$\left(N.\dot{E}_D^g - \dot{E}_M^g\right) + \dot{E}_{DB}^f - (\dot{E}_7^{fw} + \dot{E}_{12}^{fw} - \dot{E}_1^{fw})$ $-(\dot{E}_{15}^{fw} - \dot{E}_{14}^{fw}) + \dot{E}_{LP,Pump}^W + \dot{E}_{HP,Pump}^W = \dot{E}_{HRSG}^D$ $\dot{E}_{HRSG}^{Product} = (\dot{E}_7^{fw} + \dot{E}_{12}^{fw} - \dot{E}_1^{fw}) + (\dot{E}_{15}^{fw} - \dot{E}_{14}^{fw})$ $\dot{E}_{HRSG}^{Consumable} = N.\dot{E}_D^g + \dot{E}_{DB}^f + \dot{E}_{LP,Pump}^W + \dot{E}_{HP,Pump}^W - \dot{E}_M^g$	$\eta_{HRSG}^{ex} = \frac{\dot{E}_{HRSG}^{Product}}{\dot{E}_{HRSG}^{Consumable}}$
Repowered cycle		$N.(\dot{E}_A^a + \dot{E}_E^f) + \dot{E}_N^f + \dot{E}_{21}^{sw}\dot{E}_{Cond,Pump}^W + \dot{E}_{LP,Pump}^W$ $+\dot{E}_{HP,Pump}^W - \dot{E}_M^g - \dot{E}_{22}^{sw} - \dot{E}_{net,ST}^W - N.\dot{E}_{net,GT}^W = \dot{E}_{RC}^D$ $\dot{E}_{RC}^{production} = \dot{E}_{net,ST}^W + N.\dot{E}_{net,GT}^W$ $\dot{E}_{RC}^{Consumable} = N(\dot{E}_A^a + \dot{E}_E^f) + \dot{E}_N^f + \dot{E}_{21}^{sw} - \dot{E}_{22}^{sw}$ $-\dot{E}_M^g + \dot{E}_{Cond,Pump}^W + \dot{E}_{LP,Pump}^W + \dot{E}_{HP,Pump}^W$	$\eta_{RC}^{ex} = \frac{\dot{E}_{RC}^{production}}{\dot{E}_{RC}^{Consumable}}$

TABLE 4. Comparison of the results obtained from modeling and the measurements

Parameter (* Input data)	unit	V94.3A Modeling	V94.3A Roodshoor	V94.3A Error (%)	V94.2 Modeling	V94.2 Sabalan	V94.2 Error (%)
* Air inlet temperature	°C	21.7	21.7	-	15	15	-
Turbine inlet temperature	°C	1192	1206	1.16	1028	1040	1.15
* Fuel temperature	°C	26	26	-	30	30	-
AC outlet temperature	°C	432.5	423.5	2.13	347.15	337.9	2.73
Exhaust Temperature	°C	578.9	576	0.50	539.85	545	0.90
* Air Inlet pressure	bar	0.884	0.884	-	0.86	0.86	-
* Fuel pressure	bar	24.4	24.4	-	20.5	20.5	-
Exhaust flow	$Kg.sec^{-1}$	574.3	556.42	3.21	498.1	514	3.09
Air flow	$Kg.sec^{-1}$	560.6	543.94	3.06	487.8	504.25	3.26
* LHV	$Kj.kg^{-1}$	46503	46503	-	49434	49434	-
*Shaft net power	MW	217.6	217.6	-	159	159	-
Thermal efficiency	%	35.38	34.87	1.46	32.47	33	1.6
* $r_{p,Ac}$	%	16.58	16.58	-	11.37	11.37	-
* $r_{p,Cch}$	%	0.97	0.97	-	0.975	0.975	-

TABLE 5. Thermodynamic properties of air or gases at different points along the V94.2 & V94.3A gas turbines

Point	V94.2 $\dot{m}(kg.s^{-1})$	V94.2 T(k)	V94.2 P(bar)	V94.2 $e(kj.kg^{-1})$	V94.3A $\dot{m}(kg.s^{-1})$	V94.3A T(k)	V94.3A P(bar)	V94.3A $e(kj.kg^{-1})$
A	530.8	300.15	1.0134	0.000038	677.3	300.15	1.0134	0.000038
B	530.8	642.0.	11.520	329.7	677.3	718.20	16.810	410.10
C	541.4	1353.0	11.230	947.9	693.6	1558.0	16.390	1236.0
D	541.4	783.00	1.0134	212.8	693.6	894.70	1.0134	298.20
E	10.56	303.15	20.500	51227	16.33	303.15	24.400	51238

6. 2. Exergy Loss and Exergy Dissipation

Both exergy loss and exergy dissipation are related to the exergy which a system loses due to irreversibilities. Their difference goes back to selection of the system boundaries. The following exergy balance equation can be written for a control volume at steady state [25]:

$$\dot{E}_{in} = \dot{E}_{out} + \dot{E}_D + \dot{E}_L \tag{41}$$

In this equation, $\dot{E}_L$ is the exergy loss and $\dot{E}_D$ is the exergy dissipation. Because of the steady state of the control volume, the difference between the inlet and outlet exergies remains constant. Therefore, the sum of exergy loss and exergy dissipation must remain constant. The exergy loss of a system having heat transfer with the environment is calculated by the following equation [25]:

$$\dot{E}_L = \int_{in}^{out}\left(1-\frac{T_0}{T_b}\right)q'_A dA \tag{42}$$

In this equation, T_b is the system boundary temperature. Thus, if the system boundaries are selected so that they completely overlap with the boundaries of the environment, then T_o and T_b will be the same, and the exergy loss will be zero. In this case, the change of exergy inside the control volume is the exergy dissipation. Table 3 shows the exergy balance and exergy efficiency equations for all of the components of the repowered cycle [22, 25, 27]. In this table, the boundaries of the control volume and the environment are the same, and hence the exergy loss is zero.

7. RESULTS AND DISCUSSION

In the modeling of the gas turbine(s), compressor, combustion chamber, and turbine are separately analyzed and the results are compared with the data obtained from the control room of the two gas stations of Roodeshoor and Sabalan having V94.3A and V94.2 gas turbines respectively (Table 4). The empirical data are taken from two internal reports available in Persian in the above mentioned power stations. The comparison shows that the computational results are accurate.

Table 5 shows the amount of some properties including exergy at different points along the two gas turbines when they are located in the city of Bandar-Abbas. The net power outputs are taken as 160MW and 293MW for V94.2 and V94.3A gas turbines, respectively. The fuel used in these gas turbines is the natural gas sample 1. The volumetric analysis of this fuel is shown in Table 6.Due to the high capacity of the steam turbines used in the power plant, at least three V94.2 gas turbines or two V94.3A 2 gas turbines are required to provide the necessary steam for the steam turbines. When the steam flow rate increases to increase the power plant capacity, the number of the required gas turbines is increased. The results of the repowering of Bandar-Abbas power plant at three different feed water flow rates, which were discussed in section 1, are shown in Table 7.The data in Table 7 show that the use of V94.3A gas turbine is more appropriate for repowering as compared withV94.2. This is because the efficiency and the heat rate of the power plant are improved more in case of V94.3A. Currently, due to the high working life, the efficiency of Bandar-Abbas power plant is approximately 25%. Since the design mode was considered in the modeling of the repowered plant and in practice, the power plant has lower efficiency and higher heat loss, the improvement of the heat rate by repowering will be much higher than what is shown in Table 7.The use of V94.2 gas turbines for repowering of this power plant has two problems: First, using this type of gas turbine has a little effect in optimization of the efficiency and the heat rate of the power plant. Second, placement of 4 or 5 gas turbines has technical problems.

According to Table 7, the best repowering mode occurs when the feed water flow rate for repowering equals the flow rate of the base plant working at the nominal load and the duct burner is deleted and two V94.3A gas turbines are used. This mode has the highest exergy efficiency and the highest heat rate improvement. Therefore, the heat losses due to the exhaust smoke and the exergy dissipation of the power plant have the minimum values. The properties of the working fluid at different points along the cycle for the best repowering mode are shown in Table8. Table 9 shows the gas thermodynamic properties at different points along the heat recovery steam generator for the best repowering mode.

The followings are the requirements for selecting a repowering mode for a power plant:

- The steam entering the steam turbines should have the proper conditions.
- The pinch and approach temperature differences should be between the minimum and maximum limits.
- The steam temperature at the outlet of the heat recovery steam generator should be less than the maximum temperature limit.

According to Table 8 the temperature of the steam which goes out of the heat recovery steam generator and enters the high pressure steam turbine (point 13) is 812.5K and the temperature of the steam which goes out of the heat recovery steam generator and enters the low pressure steam turbine (point 8) is 596.9K. These temperatures are almost equal to the steam temperatures at the steam turbine inlets in the main power plant. The minimum pinch temperature difference is equal to 5.2K which occurs at the high pressure evaporator. It is the difference between the temperatures at point 11 in Table 8 and point H in Table 9.This value is higher than the minimum permitted value of the pinch temperature difference which is 5K. The approach temperature difference is taken equal to 5K in repowering, which is between the limits of 5K and 15K. The approach temperature difference for the high pressure evaporator is the difference between the temperatures at point 10 and 11 in Table 8.Table 9 shows that the outlet temperature of the gases in the heat recovery steam generator is equal to 399.3K which is higher than the dew point of the exhaust gases which is 393K [3, 29]. Figure 3 shows the exergy destruction in different parts of the repowered power plant. It shows that the highest exergy destruction occurs in the combustion chamber. The exergy dissipation in every system depends on the system irreversibilities. There are three factors for irreversibilities: chemical reaction, heat transfer, and friction. All of these three factors exist in the combustion chamber, but chemical reaction is the main source of exergy destruction. Chemical reactions are the main source of irreversibility. Thus, the exergy destruction in a chemical reactor is significant. Increasing the inlet temperature of a system increases the inlet exergy, and therefore, the exergy dissipation increases. Similarly, the inlet exergy to the combustion chamber increases by preheating the inlet air. Increasing the air-fuel ratio also increases the system temperature, and hence the exergy destruction [25]. These facts are shown in Figure 4. According to Figure 3, the second source of exergy destruction is the heat recovery steam generator. The irreversibilities in the HRSG are due to heat transfer and friction. In Figure 5, exergy destruction rates in different components of the HRSG are compared. It can be seen that at the components such as the evaporators, the high pressure superheater and the reheat line, in which heat transfer is higher, irreversibilities and hence exergy destruction are higher. This figure also shows that a large amount of the system's exergy enters the surrounding environment through the stack. This is inevitable due to the exhaust temperature limitations in the stack.

Figure 6 compares the exergy efficiency of the main components of the repowered power plant. It shows that the maximum exergy efficiency is related to the turbine of the gas-turbine cycle, which is slightly higher than the efficiency of the steam turbines. Condenser, on the other hand, has the lowest exergy efficiency which is related to its high level of heat transfer to the surrounding.

TABLE 6. Components of the natural gas and their volume fractions [27]

Component of natural gas	Volume fraction (%)
Methane (CH_4)	98.57
Ethane (C_2H_6)	0.63
Propane (C_3H_8)	0.1
Butane (Iso-C_4H_{10})	0.05
Pentane (Iso-C_5H_{12})	0.04
Nitrogen+Argon (N_2+Ar)	0.6
Carbon dioxide (CO_2)	0.01

TABLE 7. The results from modeling of the repowered power plant at different conditions

Type of Gas Turbine	Power plant load	Numbers of Gas Turbines	Duct burner	k	y	η_{exergy} (%)	Exhaust heat loss (MW)	Exergy destruction (MW)	Heat rate (kJ/kW.h)	Heat rate Improvement (%)	Total power (MW)
V94.3A	Nominal	2	Exist	1	0.88	52.00	176.7	828.4	6843	25.2	875.2
			Deleted	1	0.83	52.04	144.7	815.8	6711	26.6	866.0
		3	Exist	1	0.88	45.87	644.3	1228	7614	16.8	1168
			Deleted	1	0.88	46.81	602.1	1218	7461	18.4	1168
	Overload	2	Exist	1.2	---	---	---	---	---	---	---
			Deleted	1.2	---	---	---	---	---	---	---
		3	Exist	1.2	0.88	47.94	488.5	1153	7285	20.4	1221
			Deleted	1.2	0.88	48.92	446.8	1222	7139	21.9	1221
	Removed Extractions	2	Exist	1.373	---	---	---	---	---	---	---
			Deleted	1.373	---	---	---	---	---	---	---
		3	Exist	1.373	0.88	49.56	357.4	1235	7046	23.0	1262
			Deleted	1.373	0.83	50.57	316.2	1219	6905	24.5	1262
V94.2	Nominal	4	Exist	1	0.83	42.19	431.4	1187	8279	9.48	929.4
			Deleted	1	0.72	42.28	397.9	1166	8260	9.69	909.9
		5	Exist	1	0.88	40.21	767.5	1459	8685	5.04	1102
			Deleted	1	0.88	40.97	663.3	1431	8524	6.80	1102
	Overload	4	Exist	1.2	0.69	43.19	308.5	1203	8086	11.6	951.5
			Deleted	1.2	0.60	43.27	275.7	1180	8071	11.8	931.2
		5	Exist	1.2	0.86	41.93	566.4	1472	8329	8.94	1149
			Deleted	1.2	0.77	42.06	532.0	1451	8304	9.21	1131
	Removed Extractions	4	Exist	1.373	0.57	43.65	211.6	1214	8001	12.5	961.6
			Deleted	1.373	---	---	---	---	---	---	---
		5	Exist	1.373	0.75	42.65	459.2	1488	8189	10.5	1169
			Deleted	1.373	0.67	42.76	425.5	1465	8168	10.7	1150

TABLE 8. Results of modeling the repowered plant at the optimum mode for different points along the cycle

point	Temperature (K)	Pressure (bar)	Mass flow rate(kg.s^{-1})	Enthalpy (kJ.kg^{-1})	Entropy (kJ.kg^{-1}.K^{-1})	Specific exergy (kJ.kg^{-1})
1	315.8	2.000	208.05	178.6	0.6069	1.838
2	387.4	1.940	208.05	479.5	1.4660	45.03
3	392.4	1.940	208.05	500.7	1.5200	49.91
4	392.5	6.677	208.05	501.3	1.5200	50.44
5	430.0	6.477	208.05	662.1	1.9110	93.80
6	435.0	6.477	35.550	2760	6.7340	744.1
7	600.8	6.250	35.550	3118	7.4500	887.5
8	596.9	5.937	35.550	3108	7.4610	874.0
9	436.8	191.2	172.50	702.5	1.9580	120.2
10	627.7	185.4	172.50	1697	3.8140	557.4
11	632.7	185.4	172.50	2487	5.0630	972.6
12	817.8	178.9	172.50	3402	6.3900	1489
13	812.5	170.0	172.50	3397	6.4050	1480
14	567.2	25.87	172.50	2991	6.5970	1016
15	816.4	24.57	172.50	3559	7.4530	1327
16	811.1	23.35	172.50	3548	7.4630	1314
17	617.4	5.937	172.50	3154	7.5330	898.7
18	613.9	5.937	208.02	3146	7.5200	894.6
19	321.9	0.116	208.02	2464	7.7090	155.5
20	315.7	0.090	208.02	178.3	0.6067	1.642

TABLE 9. Results of modeling the repowered plant at the optimum mode for different points along HRSG

Point	Temperature (K)	Mass flow (kg.s^{-1})	Enthalpy (kJ.kg^{-1})	Specific exergy (kJ.kg^{-1})
F	894.7	1387	1059	315.7
G	730.2	1387	832.8	186.3
H	637.9	1387	711.1	124.3
I	574.8	1387	631.3	87.45
J	470.6	1387	504.4	38.00
K	446.9	1387	476.4	29.07
L	443.9	1387	472.9	28.02
M	399.3	1437	421.3	14.23

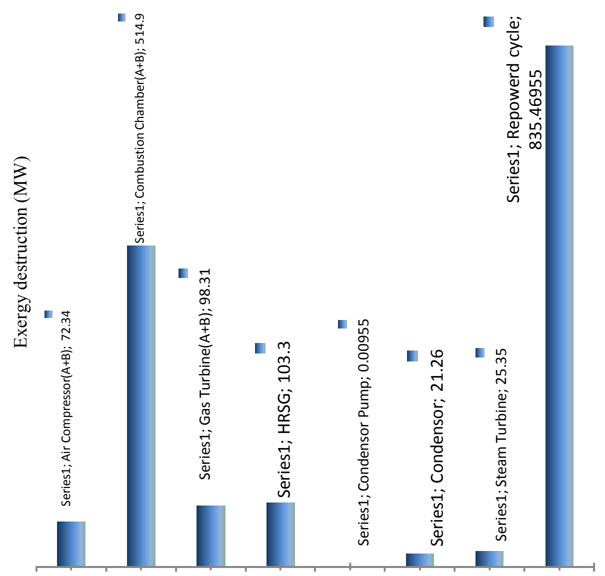


Figure 3. Exergy destruction at the repowered cycle components

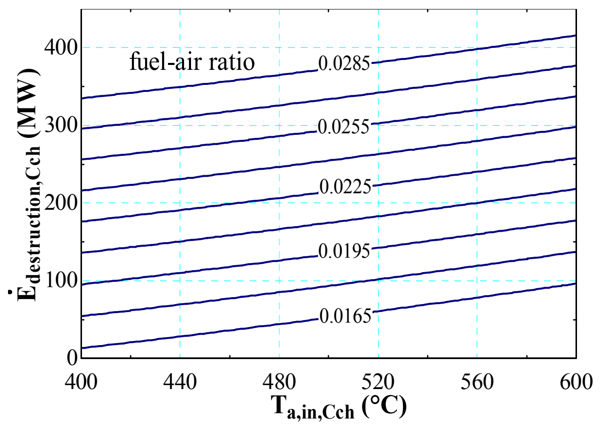


Figure 4. Effect of the inlet air temperature and fuel-air ratio on exergy destruction in the combustion chamber

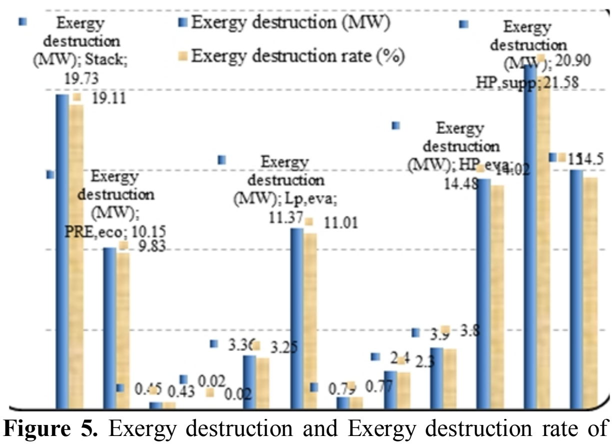


Figure 5. Exergy destruction and Exergy destruction rate of HRSG components

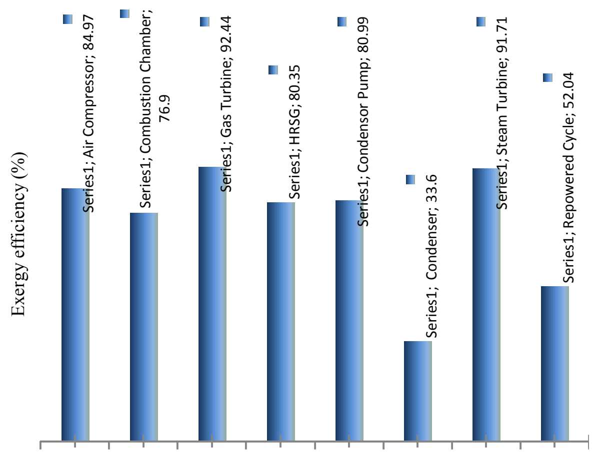


Figure 6. Exergy efficiency of the repowered power plant components

8. CONCLUSION

Full repowering is an effective method to improve the efficiency and to increase the productive lifespan of an old steam power plant. Using exergy analysis method, full repowering of the Bandar-Abbas power plant is investigated in this work. The repowered power plant has been analyzed in 24 different modes. The best repowering mode at which the maximum exergy efficiency and the minimum heat rate and exergy dissipation rate are obtained and the required steam conditions and limitations are satisfied, has been chosen for further analysis. This mode is when the feed water flow rate for repowering equals the flow rate of the base plant working at the nominal load and the duct burner is deleted and two V94.3A gas turbines are used. In this condition, the heat rate reduces by 26.6% relative to the design mode and the exergy efficiency increases by 34.5 and 108.2% relative to the design mode and to the current condition, respectively. Addition of the duct burner causes higher outlet power, but it has negative effect on the overall efficiency and the heat rate. According to this analysis, the V94.2 gas turbines are not recommended for repowering of Bandar-Abbas power plant.

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Technical Analysis of Conversion of A Steam Power Plant to Combined Cycle, Using Two Types of Heavy Duty Gas Turbines

S. Nikbakht Naserabad, K. Mobini, A. Mehrpanahi, M. R. Aligoodarz

Department of Mechanical Engineering, Shahid Rajaee Teacher Training University, Tehran, Iran

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چکیده

با توجه به عمر طولانی نیروگاه‌های بخار در ایران، تغییر ساختار این سیکل‌ها به سیکل ترکیبی تحت بررسی می‌باشد. در پژوهش حاضر، نیروگاه بخار بندرعباس با ظرفیت ۳۲۰ مگاوات بررسی شده است. این نیروگاه قدیمی در بندرگاه شهر بندرعباس و در جنوب ایران واقع شده است. به منظور مطالعه سیستم در دو حالت کنونی و بازتوانی شده، روش آنالیز اگزرژی به کار گرفته شده است. حالت بهینه سیکل بازتوانی شده نیز با استفاده از تحلیل اگزرژی به دست می‌آید. برای دست‌یابی به این هدف، از دو توربین گاز V94.2و V94.3Aاستفاده و در هر حالت تاثیر داکت برنر ملاحظه شده است. نتایج نشان می‌دهد که در بهترین ساختار بازتوانی، راندمان نیروگاه ۳٤/۵ درصد بیشتر از راندمان طراحی نیروگاه موجود می‌باشد. حداقل نرخ انهدام اگزرژی در این حالت ٦۷۱۱ مگاوات بوده و نرخ حرارت نیروگاه تا ٢٦/٦ درصد کاهش یافته است. مطابق نتایج به دست آمده، افزایش سوخت مصرفی داکت برنر و همچنین استفاده از توربین گازV94.2 برای بازتوانی نیروگاه بخار بندرعباس توصیه نمی‌گردد.

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