STUDY OF THE EFFECT OF USING DUCT BURNER ON THE FUNCTIONAL PARAMETERS OF THE TWO REPOWERED CYCLES THROUGH EXERGY ANALYSIS

by

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Steam power plants have been extensively used in Iran for a long time, yet no specific step has been taken for promoting their performance. In this regard, full repowering is considered as a way to enhance the performance of steam power plants. Furthermore, because of the continental condition of Iran, duct burners can be used as a common strategy to compensate for power generation shortage caused by environmental conditions. In this study, the effect of using a duct burner on the full repowering of Be'sat Steam Cycle representing both single-and dual-pressure cycles was investigated based on exergy analysis. The results showed that by using the duct burner, due to the increase in the heat recovery steam generator inlet gas temperature, the general thermal efficiency of the combined cycle and the exergy efficiency of the combined cycle and heat recovery steam generator decreased. However, the results revealed an increase in the stack temperature and resulting exergy losses, steam flow and power generation.

Key words: *repowering, heat recovery steam generator, duct burner, exergy, thermal efficiency*

Introduction

Repowering refers to the utilization of exhaust gas of a gas turbine set to upgrade the performance of steam power plants. Repowering methods are classified into two main types: full repowering and partial repowering. Full repowering is the most common method of repowering. In this method, a heat recovery steam generator (HRSG) and gas turbine(s) are used instead of old boilers. This method is useful for power plants with the minimum age of 25 years and aims to convert steam power plants into the combined cycle with promising technical and economic advantages. Because of the impact of continental conditions as well as wide range of temperature spectrum in Iran, the design and modification of new high performance cycles are very important and critical. Today, using additional combustion in HRSG has been widely developed for the purpose of controlling the temperature and generated steam flow in the mode of lowering gas turbine load and/or offsetting ambient changes; so far, 75% of HRSG have been proven to benefit from this approach. The burners that generate this additional combustion in boilers are called duct burners and have the following advantages [1-3]:

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- better control of exhaust thermal power,
- higher efficiency of steam generation,
- keeping the steam generation fixed as the gas turbine load decreases or even in the case of the gas turbine off-design operation,
- compensation for ambient changes, and
- using fuels that are not suitable for combustion in gas turbines.

In addition to the above-mentioned advantages, there are also some disadvantages in using duct burners, including:

- being far from the optimum states in unfired and/or part load modes since the cycle elements should be designed for maximum heat temperature,
- requiring more care about the utilization time to prevent the overheated pipes, and
- increasing manufacturing cost of the unit by 10 to 15%.

Studies carried out on repowering can be generally categorized into solid and non-solid fuel-based repowering options [4-7]. In spite of some differences, all these investigations have been aimed to improve technical, environmental and economic specifications of the system. In this research, Be’sat steam power plant in Tehran (fig. 1) was repowered through two distinct methods using full repowering method. Further, two distinct cycles combined with different HRSG (single- and dual-pressure) were modeled. Be’sat Power Plant is an old steam power plant designed by General Electric Corporation with 31.46% efficiency, however, its current real efficiency is 26.81%. Efficiency reduction has been caused by the power plant’s old lifetime and exhaustion [8]. Since the reference cycle has a steam injection line to the steam turbines, a single-pressure HRSG is used to be in better correspondence with the reference cycle and dual-pressure HRSG is employed in alternative designs to perfectly use the turbine exhaust gas. Using full repowering method and modeling the new combined cycles, this study was done to enhance the efficiency and power generation of the old Be’sat Power Plant. Moreover, considering the design restrictions and using the exergy analysis, the effects of using a duct burner on the functional parameters of the repowered cycles, including thermal and exergy efficiencies as well as exergy efficiency of the HRSG, were studied. Accordingly, the efficiencies of the combined cycle power plants modeled in this research were generally lower than other existing combined cycles. Modeling procedure was conducted by EES software [9].

Previous studies on duct burners have separately considered the cycle modeling, repowering and thermodynamics analysis. However, in this research, some efforts were made to model two combined cycles by using full repowering methods and a comprehensive thermodynamic analysis followed by modeling to evaluate the performance of the cycles with regard to not using duct burners and various modes of using duct burners. Hence, this study is of novelty in terms of investigating the effects of duct burners on repowered cycles using exergy analysis. On the other hand, although the efficiency of heat recovery of a single-pressure cycle is lower than that of a dual-pressure cycle, the main benefit of single-pressure cycle compared with a dual-pressure cycle in the studied power plant was its consistency with the power plant steam turbines. Additionally, due to the necessity of changes in the structures of steam turbines, capital cost for the dual-pressure case is higher than that of the single-pressure cycle. This study can be used as a scale to compare the mentioned cycles with or without duct burners. Duct burners have been of interest for many researchers. For example, Batshon and Backlund [1] introduced the advantages and disadvantages of duct burners and examined using irregular fuels such as the gas produced from sewers, furnaces of steel plants and decomposition of solid urban wastes in duct burners. Ahmadi and Dincer [10] designed a combined cycle equipped with a duct burner by using a genetic algorithm and optimized the objective functions. Efficiency and power improvement using a triple-pressure HRSG was also investigated by Basily [11]. In his study, he used the duct burner and vapor reheating to increase the generation power. Regarding the field of cycle modeling, Ibraham and Rahman [12] presented the thermodynamic analysis of a triple-pressure combined cycle equipped with a duct burner and studied the impacts of changes in the gas cycle functional parameters on the combined cycle efficiency and power. They suggested that *TIT* changes have the most significant effects on these variables. Tajik Mansouri *et al*. [13] evaluated and compared three types of HRSG used in a combined cycle to examine the effects of increasing the steam pressure levels on the power and efficiency improvement and reduced exergy destruction. Repowering cycle was proposed by Hosseinalipour *et al*. [14] for a steam cycle. In his study, the optimization of the presented combined cycle was done by thermo-economic objective functions. With regard to the exergy analysis of the combined cycles, Ameri *et al*. [15] analyzed a combined cycle in Iran and found that the highest exergy loss took place in the gas turbine combustion chamber. Bracco and Siri [8] optimized the performance of a combined cycle with single-pressure HRSG by using exergy analysis and concluded that the optimized cycle pressure provided higher levels of power and efficiency.

Reference cycle specifications

Figure 1 shows the schematic diagram of the Be'sat steam power plant which was an old steam power plant designed by General Electric Corporation with 31.46% design efficiency, however, its current real efficiency was 26.81%. Efficiency reduction has been caused by the power's plant old lifetime and exhaustion [8]. A V94.2 gas turbine was used in the gas chamber and the steam cycle had a boiler, two high-, two intermediate-, and two low-pressure steam turbines. In addition, a condenser and a set of feed water reheaters were included in this cycle. In the steam turbines, a part of steam was extracted to the feed waters and reheated the feed water boiler. Thermodynamic specifications of steam/water in different parts of Be'sat power plant are represented in tab. 1.

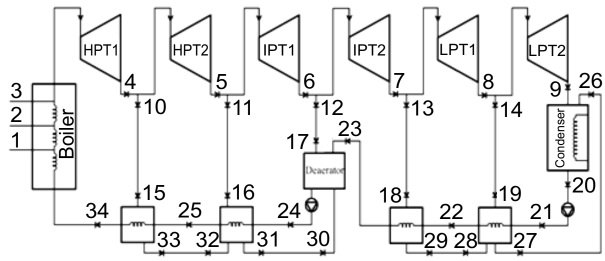

Figure 1. Schematic diagram of Be'sat steam power plant

Table 1. Water/steam properties of Be'sat power plant

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

Modeling conditions of repowered cycles

- In repowered cycles, there exist a steam turbine, a gas turbine, a condenser, a HRSG equipped with a duct burner and feed water pumps. Gas and steam turbines are considered in exact concordance with the reference cycle.
- The HRSG and gas turbine sizes are determined based on the maximum arrival steam to the first high pressure turbine in the thermodynamic conditions of the reference cycle (point 6 in fig. 3 and point 13 in fig. 4).
- To design the cycles, the cooling tower is assumed to be used when the condenser pressure in those cycles ranges between 0.068 and 0.136 bar [16]. In this research, the functional pressure of condenser was considered fixed and equal to 0.72 bar.
- Restrictions on the condenser in receiving the amount of steam fed by present cycles must be carefully specified. To design the condenser, the amount of feeding steam flow is considered as $\dot{m}_{cond}$ = 73.05 kg/s; however, this amount could also increase up to 45% [17].
- The steam produced by HRSG should be compatible with the cycle conditions and the condenser limitations. Thus, the coefficient value k is regarded in studying the designed cycle, indicating the steam flow having arrived at the condenser. In fact, k is the amount of steam generated by the boiler with respect to the performance conditions of the duct burner and combined cycle.
- The pressure generated by the water pump experiences some losses in the transferring pipes. As a result, to have better compliance with the actual states, the pressure losses are taken as 3.5% in economizer pipes, 3% in super heater, 5% in re-heater and 5.5% in water/steam pipes; however, no pressure loss is observed in the evaporator [18]. In addition, 5% of the gas energy is taken as gas energy loss within the HRSG sections (E_{Loss} = 5%) [14].
- In case of the boiler's gas discharge temperature being less than the acid dew point of the gas mix, the water vapor content is distilled on the pre-heater and economizer pipes and generate corrosive acids which damage the pipes in their compositions with other combustion products such as CO_2. In order not to encounter the above-mentioned problem, the dew point temperature of the gas mixture in the HRSG design is considered as 361 K to keep the stack temperature less than the acid dew point [13, 18].
- To design the cycles, the pinch temperature difference should not be less than 10 K because if the pinch temperature is less, higher heat transfer levels will be required to produce the steam [12].
- All processes are performed under steady-state and steady-flow conditions [12, 13].

Energy analysis

The energy balance to control the volume in a steady-state condition has been employed in order to model all elements of gas turbine and steam cycles [14, 15], however, with regard to the procedure adopted in this research, only following equations are mentioned.

Net produced power by gas turbine

$$\dot{W}_{\text{net}} = \dot{W}_{GT} - \dot{W}_{AC} \tag{1}$$

The following parameters are considered constant:
R = 0.2944 kJ/kgK, LHV = 50000 kJ/kg, and η_{CCh} = 99%.

Duct burner [16]

The modeling of duct burners is similar to the combustion chamber modeling. The amount of the fuel fed to duct burner is regarded as a variable and the amount of fed fuel should be less than 1 kg/s in order to prevent the overheating of the super heater pipes [14].

$$\dot{m}_g C_{pg} + \dot{m}_{f,DB} LHV = \dot{m}_{g,\text{in},HRSG} C_{p,g,\text{in},HRSG} T_E + \left(1-\eta_{CCh,DB}\right) \dot{m}_{f,DB} LHV \tag{2}$$

HRSG

With regard to the similarities between single and dual-pressure HRSG equations, the dual-pressure HRSG equations are only presented below [13]. In single and dual-pressure boilers (fig. 2), the approach point temperature difference values are considered 40 K [8] and 15 K (fig. 3), respectively. In the design process, the pinch temperature difference is less than 10 K in none of the HRSG. By using the energy equations for water/steam and gas in different parts of HRSG [15, 17-23]:

Steam/water flow

$$T_{gt,\text{out},pre,eva} = T_{pre,eva} + DELTA\_T_{pre,\text{pinch}} \tag{3}$$

$$T_{gt,\text{out},hp,eva} = T_{hp,eva} + DELTA\_T_{hp,\text{pinch}} \tag{4}$$

$$T_{gt,\text{out},lp,eva} = T_{lp,eva} + DELTA\_T_{lp,\text{pinch}} \tag{5}$$

Gas flow

E to F
$$\dot{m}_{g,\text{in},HRSG} C_{p,gt} (T_E - T_F)(1-E_\text{loss}) = \dot{m}_{hp}(h_{12} - h_{11} + h_{16} - h_{15}) \tag{6}$$

F to G
$$\dot{m}_{g,\text{in},HRSG} C_{p,gt} (T_F - T_G)(1-E_\text{loss}) = \dot{m}_{hp}(h_{11} - h_{10}) \tag{7}$$

G to H
$$\dot{m}_{g,\text{in},HRSG} C_{p,gt} (T_G - T_H)(1-E_\text{loss}) = \dot{m}_{hp}(h_{10} - h_5) + \dot{m}_{Lp}(h_7 - h_5) \tag{8}$$

H to I
$$\dot{m}_{g,\text{in},HRSG} C_{p,gt} (T_H - T_I)(1-E_\text{loss}) = (\dot{m}_{hp} + \dot{m}_{Lp})(h_5 - h_4) \tag{9}$$

I to J
$$\dot{m}_{g,\text{in},HRSG} C_{p,gt} (T_J - T_I)(1-E_\text{loss}) = (\dot{m}_{hp} + \dot{m}_{Lp})(h_4 - h_2) \tag{10}$$

J to K
$$\dot{m}_{g,\text{in},HRSG} C_{p,gt} (T_J - T_K)(1-E_\text{loss}) = (\dot{m}_{hp} + \dot{m}_{Lp})(h_2 - h_1) \tag{11}$$

The amount of emerged steam is defined:

$$\dot{m}_\text{steam} = K\dot{m}_\text{cond} \tag{12}$$

Steam turbine

The steam turbines generated power is calculated by eq. (13):

$$\dot{W}_{ST} = \sum\nolimits_\text{stages} \dot{m}_{ST,\text{in}} (h_{ST,\text{in}} - h_{ST,\text{out}}) \tag{13}$$

The thermal efficiency of the repowered cycles is obtained by eq. (14):

$$\eta_{CC} = \frac{\dot{W}_{GT} - \dot{W}_{AC} + \dot{W}_\text{steam}}{\dot{Q}_{\text{in},CC}} \tag{14}$$

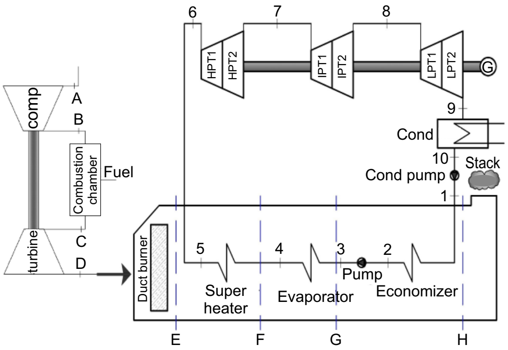

Figure 2. Schematic diagram of steam cycle of single-pressure combined cycle

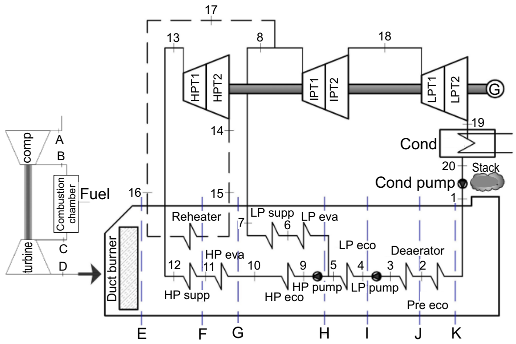

Figure 3. Schematic diagram of steam cycle of dual-pressure reheat combined cycle

Exergy analysis

Exergy analysis is defined according to the first and second laws of thermodynamics to analyze and determine the system inefficiencies. Exergy is defined as follows [14]: If a system contains *n* subsets of temperature *(T)* pressure *(P)* and the mole fractions Y_i ($i = 1,2,...,n$), exergy is defined as the maximum obtainable theoretical work in process from state (P, T, Y_i) to dead state (P_0, T_0, Y_i). It should be mentioned that the dead state (P_0, T_0, Y_i) is a system state in which there is a perfect balance between the system and its environment with temperature T_0 and pressure P_0. This state is also considered as the reference state. Exergy is divided into four types: physical exergy, chemical exergy, kinematic exergy, and potential exergy. Physical and chemical exergies have just been of interest in most exergy analyses. In the analysis employed in this study, the kinematic and potential exergies were ignored. Using the first and second laws of thermodynamics, some equations are adopted to model different parts of both gas and steam cycles [8, 14, 24]. Exergy efficiencies of the new combined cycles and HRSGs are evaluated using the following equations:

$$\eta_{ex,CC} = \frac{\dot{W}_{GT} - \dot{W}_{AC} + \dot{W}_{\text{steam}}}{\dot{E}_f} \tag{15}$$

$$\eta_{ex,HRSG} = \frac{\dot{E}_{\text{steam,out}} - \dot{E}_{\text{water,in}}}{\dot{E}_{\text{flue,gas,in}} - \dot{E}_{\text{flue,gas,out}}} \tag{16}$$

Designed cycles

Two combined cycles were designed by using two different HRSG. In the single-pressure repowered cycle, the steam injection was just possible through one pressure line; however, the steam could be injected with two different pressures in dual-pressure mode. In these cycles, natural gas was used as the fuel [14, 16]. The specifications of the gas turbine used in both cycles are listed in tab. 2. The conditions of the condenser and steam turbines were considered the same for both cycles in terms of temperature and the entry steam pressure based on the reference steam cycle. These conditions for single- and dual-pressure cycles are listed in tab. 3. Gas properties at different points of the gas turbine cycle are represented in tab. 4. It must be noted that a similar type of gas turbine was inserted in both cycles. The modeling reference ambient was air containing the following thermodynamic conditions: $P = 1.013$ bar, $T = 298.15$ K [24].

Table 2. Some characteristics of the gas turbine

Parameter	Unit	Amount
Fuel flow (LHV)	MW	50
Pressure ratio	–	15.98
Gas turbine inlet temperature	K	1353
Gas turbine outlet temperature	K	793.2
Power output	MW	160

Table 3. Input data to model the cycles

Parameter	Unit	Amount
HP inlet temperature	K	783.1
HP inlet pressure	bar	87.21
LP inlet temperature	K	580[a]
LP inlet pressure	bar	10.27
Condenser pressure	bar	0.72

a = Hot reheat temperature

Table 4. Steam characteristics of the gas turbine cycle

Point	Temperature [K]	Pressure [bar]	Mass Flow [kgs^{-1}]	Specific exergy [kJkg^{-1}]	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

In continuation, the thermodynamic conditions of water/steam and gas at all the points of the repowered cycle were investigated for both fired and unfired duct burner. In the fired mode of the single- and dual-pressure cycles, the specific mode $\dot{m}_{f,DB} = 0.5$ kg/s was selected. The coefficient K in the single pressure for both with and without the duct burner was taken as 0.814 and 0.727, respectively, to make the performance conditions of the combined cycle in terms of pinch point temperature difference and amount of generated steam become close to the optimum. For the dual-pressure states, the coefficients *were* 1 and 1.093, respectively, (tabs. 5-8). The duct burner conditions might change with respect to the cycle demands during each time interval by changes made in feeding fuel flow. For example, the amount of steam generation increased as the flow of fuel fed to duct burner and the gas temperature increased. This is shown in the following tables and diagrams and, with such a trend, the amount of K is also increasing. In tabs. 9 and 10, the information on the performance of repowered cycles in different states of using the duct burners is listed.

Table 5 .Water/steam properties in fired and unfired modes for single-pressure repowered cycle

	Unfired case load 100% T_{amb}= 298.15 K					Fired case $\dot{m}_{f,DB}$ = 0.5 kg/s load 100% T_{amb} = 298.15 K				
Point	Temperature [K]	Pressure [bar]	Mass flow [kgs^{-1}]	Enthalpy [kJkg^{-1}]	Exergy [MW]	Temperature [K]	Pressure [bar]	Mass flow [kgs^{-1}]	Enthalpy [kJkg^{-1}]	Exergy [MW]
1	316.2	6.43	53.16	180.9	0.138	316.2	6.43	61.29	180.9	0.159
2	540.2	6.237	53.16	1168	15.825	540.2	6.237	61.29	1168	18.24
3	541.7	98.07	53.16	1168	15.841	541.6	98.07	61.29	1168	18.26
4	580.2	98.07	53.16	2734	56.083	580.2	98.07	61.29	2734	64.66
5	788.2	91.08	53.16	3422	76.071	788.2	91.08	61.29	3422	87.70
6	783.1	87.21	53.16	3415	75.434	783.1	87.21	61.29	3415	86.97
7	580.4	9.77	53.16	3067	49.78	556.1	10.94	61.29	3064	57.39
8	392.8	1.22	53.16	2714	27.72	392.8	1.371	61.29	2712	31.96
9	312.6	0.071	53.16	2325	5.438	315.3	0.080	61.29	2337	7.33
10	316.2	0.72	53.16	180.1	0.116	316.2	0.72	61.29	180.1	0.134

Table 6. Gas side properties in fired and unfired cases for single-pressure repowered cycle

	Unfired case load 100% T_{amb} = 298.15 K					Fired case $\dot{m}_{f,DB}$ = 0.5 kg/s load 100% T_{amb} = 298.15 K				
Point	Temperature [K]	Pressure [bar]	Mass flow [kgs^{-1}]	Specific exergy [kJkg^{-1}]	Exergy [MW]	Temperature [k]	Pressure [bar]	Mass flow [kgs^{-1}]	Specific exergy [kJkg^{-1}]	Exergy [MW]
E	793.2	1.013	555.4	235.17	130.61	823.6	1.013	555.9	255.19	144.08
F	733.2	1.039	555.4	192.50	106.91	755	1.039	555.9	208.4	115.84
G	590.2	1.026	555.4	98.59	54.75	590.2	1.026	555.9	98.62	54.82
H	499.9	1.013	555.4	51.42	28.55	486.2	1.013	555.9	45.45	25.26

Table 7. Water/steam specifications in fired and unfired modes for dual-pressure repowered cycle

	Unfired case load 100% T_{amb} = 298.15 K					Fired case $\dot{m}_{f,DB}$ = 0.5 kg/s load 100% T_{amb} = 298.15 K				
Point	Temperature [K]	Pressure [bar]	Mass flow [kgs^{-1}]	Enthalpy [kJkg^{-1}]	Exergy [MW]	Temperature [K]	Pressure [bar]	Mass flow [kgs^{-1}]	Enthalpy [kJ^{-1}kg]	Exergy [MW]
1	316.2	6.43	73.05	180.9	0.207	316.2	6.43	79.84	180.9	0.22
2	418.5	6.237	73.05	612.5	5.99	418.5	6.237	79.84	612.5	6.55
3	433.5	6.237	73.05	677.3	7.41	433.5	6.237	79.84	677.3	8.1
4	433.6	11.61	73.05	678.1	7.46	433.7	12.7	79.84	678.2	8.16
5	443.3	11.27	73.05	720.2	8.44	447.3	12.31	79.84	737.2	9.7
6	458.3	11.27	17.17	2782	14.34	462.3	12.31	18.76	2785	15.9
7	584.1	10.87	17.17	3072	16.36	584.1	11.88	18.76	3069	18.08
8	580.4	10.27	17.17	3065	16.18	580.4	11.23	18.76	3063	17.88
9	445.0	98.59	55.88	732.3	7.04	449.0	98.59	61.08	749.8	8.06
10	565.9	95.63	55.88	1302	20.09	565.9	95.63	61.08	1302	21.96
11	580.9	95.63	55.88	2732	58.95	580.9	95.63	61.08	2732	64.43
12	788.2	92.29	55.88	3422	79.96	788.2	92.29	61.08	3422	87.4
13	783.1	87.21	55.88	3415	79.29	783.1	87.21	61.08	3415	86.67
14	552.1	11.38	55.88	3001	51.73	557.8	12.44	61.08	3010	57.53
15	552.1	11.38	55.88	3001	51.73	557.8	12.44	61.08	3010	57.53
16	584.1	10.81	55.88	3072	53.24	584.1	11.82	61.08	3070	58.84
17	580.4	10.27	73.05	3065	52.67	580.4	11.23	61.08	3063	58.22
18	392.8	1.682	73.05	2730	42.74	392.8	11.83	79.84	2707	45.9
19	318.7	0.098	73.05	2344	10.34	320.4	0.107	79.84	2366	12.31
20	316.2	0.72	73.05	180.1	0.16	316.2	0.72	79.84	180.1	0.174

Table 8. Gas side properties in fired and unfired cases for dual-pressure repowered cycle

	Unfired case load100% T_{amb} = 298.15 K					Fired case $\dot{m}_{f,DB}$ = 0.5 kg/s load 100% T_{amb} = 298.15 K				
Point	Temperature [K]	Pressure [bar]	Mass flow [kgs^{-1}]	Specific exergy [kJkg^{-1}]	Exergy [MW]	Temperature (K)	Pressure [bar]	Mass flow [kgs^{-1}]	Specific exergy [kJkg^{-1}]	Exergy [MW]
E	793.2	1.013	555.4	235.2	130.6	823.6	1.013	555.9	259.9	144.47
F	723.6	1.037	555.4	185.3	102.9	749.3	1.037	555.9	204	113.4
G	586.0	1.028	555.4	96.69	53.7	599.6	1.028	555.9	104.4	58.35
H	467.9	1.021	555.4	38.61	21.45	473.4	1.021	555.9	40.8	22.68
I	462.5	1.018	555.4	36.21	20.1	465.0	1.018	555.9	37.19	20.67
J	454.5	1.015	555.4	32.92	18.3	456.3	1.015	555.9	33.58	18.66
K	398.6	1.013	555.4	14.63	8.1	295.2	1.013	555.9	13.74	7.63

Table 9. Operation parameters of single pressure repowered cycle with changes in the fuel of duct burner

State	$\dot{m}_{f,DB}$ [kgs^{-1}]	K	$\dot{m}_{steam}$ [kgs^{-1}]	$\dot{W}_{CC}$ [MW]	$T_{in,HRSG}$ [K]	$T_{out,stack}$ [K]	$T_{HP,pinch}$ [K]	$E_{loss,HRSG}$ [MW]	$\eta_{loss,stack}$ [MW]	$\eta_{Th,CC}$ [%]	$\eta_{ex,CC}$ [%]
1	0	0.727	53.16	221.1	793.2	499.9	10	26.86	28.42	0.4192	0.4066
2	0.111	0.729	53.28	221.3	795.4	499.4	10.37	27.04	28.12	0.4312	0.4183
3	0.250	0.761	55.62	224	803.9	495.1	10.06	28.48	27.08	0.4306	0.4177
4	0.361	0.782	57.18	225.8	810.7	493.5	11.04	29.64	26.69	0.4294	0.4166
5	0.444	0.804	58.75	227.5	815.8	490	10.22	30.51	25.86	0.4293	0.4165
6	0.500	0.814	59.53	228.4	819.2	489.2	10.70	31.10	25.67	0.4287	0.4159
7	0.638	0.847	61.87	231	827.6	484.8	10.34	32.57	24.65	0.4280	0.4152
8	0.750	0.868	63.44	232.7	834.4	483.1	11.29	33.78	24.27	0.4268	0.4140
9	0.888	0.900	65.78	235.2	842.8	478.7	10.90	35.28	23.27	0.4259	0.4131
10	1	0.921	67.34	236.8	849.5	477.1	11.82	36.52	22.89	0.4245	0.4118

Table 10. Operation parameters of dual-pressure repowered cycle with changes in the fuel of duct burner

State	$\dot{m}_{f,DB}$ [kgs^{-1}]	K	$\dot{m}_{steam}$ [kgs^{-1}]	$\dot{W}_{CC}$ [MW]	$T_{in,HRSG}$ [K]	$T_{out,stack}$ [K]	$T_{HP,pinch}$ [K]	$T_{IP,pinch}$ [K]	$T_{pre,pinch}$ [K]	$E_{loss,HRSG}$ [MW]	$E_{loss,stack}$ [MW]	$\eta_{Th,CC}$ [%]	$\eta_{ex,CC}$ [%]
1	0	1	73.05	234.2	793.2	398.6	5.129	9.623	21	42.42	8.103	0.4439	0.4306
2	0.15	1.020	74.51	235.3	802.3	400.7	10.79	12.72	24.21	44.57	8.428	0.4396	0.4265
3	0.25	1.047	76.48	236.6	808.4	396.7	11.81	10.10	21.71	45.79	7.828	0.4380	0.4249
4	0.35	1.067	77.94	237.4	814.5	395.5	14.23	9.925	21.58	47.14	7.649	0.4355	0.4225
5	0.5	1.093	79.84	238.3	823.6	395.2	18.67	11.06	22.76	49.26	7.615	0.4312	0.4184
6	0.6	1.113	81.30	238.9	829.6	394	21.08	10.93	22.63	50.64	7.439	0.4284	0.4156
7	0.7	1.133	82.77	239.3	835.7	392.7	23.50	10.84	22.51	52.03	7.265	0.4253	0.4126
8	0.8	1.153	84.23	239.5	841.7	391.5	25.92	10.76	22.40	53.43	7.093	0.4220	0.4093
9	0.9	1.173	85.69	215.3	847.6	390.3	28.34	10.71	22.28	54.84	6.923	0.3759	0.3647
10	1	1.193	87.15	215.7	853.6	389.1	30.76	10.69	22.17	56.27	6.756	0.3734	0.3623

Results and discussion

In this study, the effects of using a duct burner on the performance parameters of the repowered cycle, such as thermal and exergy efficiencies of repowered cycle, exergy efficiency of HRSG, exergy loss from stack, flow of produced steam and the production power of combined cycle, were studied.

According to tabs. 9 and 10, it can be observed that the amount of steam generation increased in both repowered cycles using the duct burner because of an increase in the HRSG inlet gas temperature. In steam turbines, this issue led to the increased generation power. On the other hand, the exergy increase of the duct burner gas was absorbed by the HRSG and was consumed to generate more steam. As a result, the temperature of stack outlet gas and, subsequently, its exergy loss decreased. The exergy and thermal efficiencies were also reduced. It is also worth noting that those cycles were not similar in terms of general design and the pinch and approach temperatures and they were designed and studied with respect to their specific restrictions. The designed cycles were only similar in some cases such as condenser function conditions, steam turbines and thermodynamic conditions of steam generation in boilers. Thus, the results related to the exergy analysis of each cycle were studied in separate diagrams. Figures 5 and 6 reveal that the temperature of the HRSG inlet gas increased by increasing the gas burn in duct burners. To prevent the overheating of the super-heater pipes, the temperature must not exceed a certain point, depending on the type of super-heater design and its texture material. The effects of the amount of duct burner's fuel flow and the amount of produced steam flow on the temperature of HRSG outlet gas (stack temperature) are shown in figs. 4 and 5. In each performance state of the duct burner, the stack temperature decreased with increasing the amount of k (k as the determining factor for the amount of produced steam), implying the absorption of more heat and better use of exergy by HRSG.

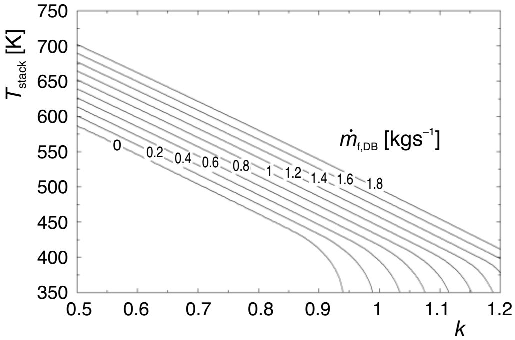

Figure 4. Effects of the amount of duct burner's fuel and produced steam on single-pressure HRSG outlet gas temperature

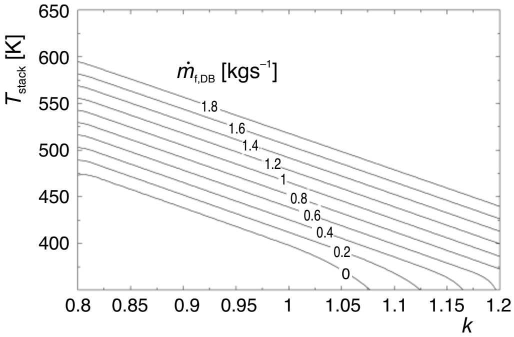

Figure 5. Effects of the amount of duct burner's fuel and produced steam on dual-pressure HRSG outlet gas temperature

In order to produce constant steam flow per k increase in different performance modes of the duct burner and with increasing the flow of fed fuel to the duct burner, the stack temperature increased and the exergy loss went up. Therefore, for producing more steam, temperature and exergy were increased in the duct burner. Such conditions could increase temperature and exergy of the heat exiting from the boiler an, consequently, increased exergy loss. For example, if the steam flow were $\dot{m}_{ST}$ = 73.05 kg/s (k = 1) in the dual-pressure combined cycle, the stack temperature would be T_{stack} = 398.6, 452.3, and 505 K in the three states of $\dot{m}_{f,DB}$= 0, 0.8, and 1.6 kg/s, respectively. The comparison of the two HRSG shows that the steam generation potential of the dual-pressure HRSG was greater. For example, the maximum amount of generation occurred at k = 1.076 and k = 0.94 for the dual- and single-pressure cycles with $\dot{m}_{f,DB}$ = 0, respectively. The effect of the amount of fuel flow of the duct burner and amount of steam production flow on exergy loss by stack are presented in figs. 6 and 7. With increasing the flow of the produced steam in each performance condition of the duct burner, the gas heat absorption increased and the exergy loss of the stack outlet gas decreased.

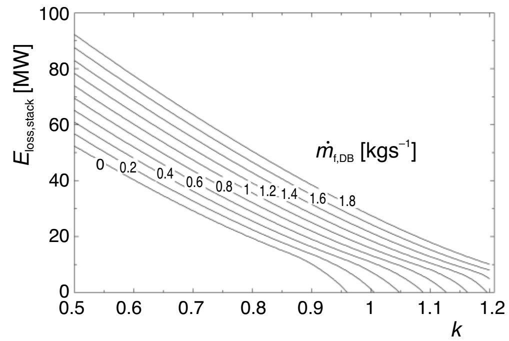

Figure 6. Effects of the amount of duct burner's fuel and produced steam on exergy loss of single-pressure HRSG stack

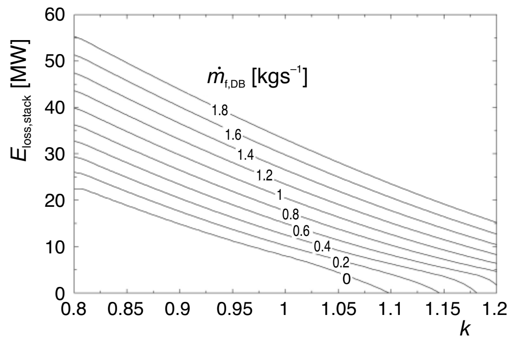

Figure 7. Effects of the amount of duct burner's fuel and produced steam on exergy loss of dual-pressure HRSG stack

To produce a similar steam flow in different functional states of the duct burner, the exhaust gas temperature increased and the exergy loss by the stack increased through increasing the fuel flow of the duct burner. In both cycles, by increasing the steam flow generation in each state of the duct burner activation, the exergy efficiency of the HRSG, the thermal efficiency of the combined cycle and the exergy efficiency of the combined cycle increased due to the increased gas heat absorption. However, to generate the same steam flow in different performance states of the duct burner, these values decreased with increasing the flow of fuel being fed to the duct burner, which was due to the increased HRSG inlet gas temperature and increased exergy loss caused by gas. On the other hand, the fuel in the duct burner was consumed in the Rankin cycle. The efficiency of the Rankine cycle is less than combined cycle; therefore, the efficiency of the combined cycle decreased [3], figs. 9-11.

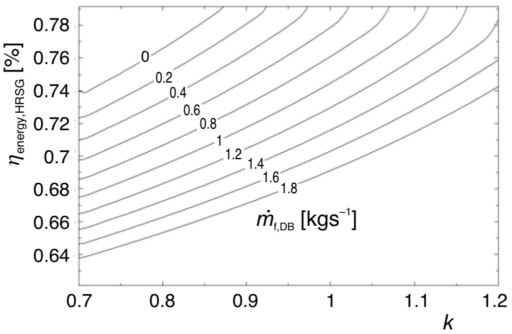

Figure 8. Effects of amount of duct burner's fuel and produced steam on the exergy efficiency of single-pressure HRSG

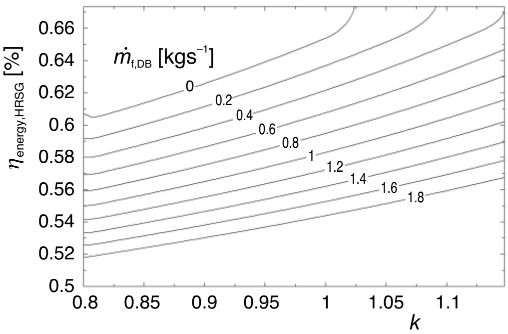

Figure 9. Effects of amount of duct burner's fuel and produced steam on the exergy efficiency of dual-pressure HRSG

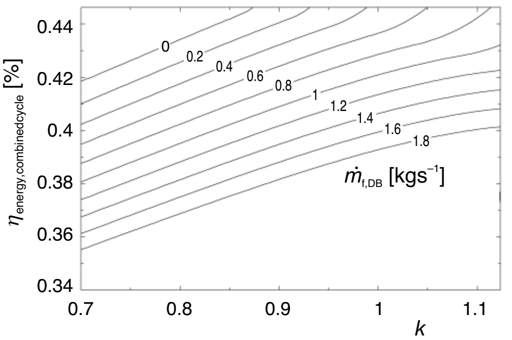

Figure 10. Effects of amount of duct burner's fuel and produced steam on the exergy efficiency of single-pressure repowered cycle

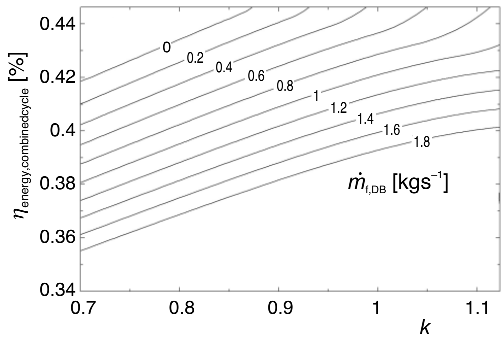

Figure 11. Effects of amount of duct burner's fuel and produced steam on the exergy efficiency of dual-pressure repowered cycle

Conclusions

In this study, two different combined cycles with HRSG (single-pressure and dual-pressure) were designed to investigate the effects of using the duct burner on their functional parameters. The designed cycles were two different models used to repower Be'sat Steam Power Plant in Tehran, Iran. By the exergy analysis of the cycles, the different functional modes of the duct burner were examined. To sum, although the exergy efficiency of HRSG as well as the exergy and thermal efficiencies of the repowered cycle decreased after using the duct burner and increasing the fuel flow fed to it, the temperature of the stack gas, the stack exergy loss and the generated steam flow and generation power of the combined cycle increased.

Nomenclature

C_p – specific heat at constant pressure, [kJkg^{-1}]
C_v – pecific heat at constant volume, [kJkg^{-1}K^{-1}]
CC – combined cycle
CCH – combustion chamber
Comp – compressor
Cond – condenser
E – exergy, [kJ]
e – specific exergy, [kJkg^{-1}K^{-1}] *eco* – economizer
eva – evaporator
G^E – excess free Gibbs energy, [kJ]
GT – gas turbine
HP – high pressure
HPT – high pressure steam turbine
HRSG – heat recovery steam generator
h – specific heat [kJkg^{-1}]
I – exergy loss, [kJ]
IPT – intermediate pressure steam turbine
K – generated steam coefficient
LHV – lower heating value, [kJkg^{-1}]
LP – low pressure
LPT – low pressure steam turbine
m – mass flow, [kg]
P – pressure, [bar]
pre- eco – pre-heater economizer
Q – heat transfer, [kJ]
R – gas constant, [kJkg^{-1}K^{-1}]
r_{Comp} – compressor pressure ratio
S – specific entropy, [kJkg^{-1}K^{-1}]
supp – superheater
T – temperature [K]
TIT – turbine inlet temperature
W – work [kJ]
x – molar fraction
η – efficiency

Subscripts and superscripts

Amb – ambient
a – air
CV – control volume
Comp – air compressor
ch – chemical
eva – evaporator
e – exit condition
ex – exergy
f – fuel
Gt – gas turbine
g – gas
g – combustion gases
i – inlet condition
Mix – mixture
Ph – physical
pre – pre-heater
ST – steam
s – steam/water
t – thermal
0 – dead state
$\cdot$ – rate

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