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Original scientific paper

Numerical analysis and field study of the time-dependent exergy–energy of a gas–steam combined cycle

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Abstract: In this study, a time-dependent exergy analysis of the Fars Combined Power Plant Cycle was considered. The exergy analysis was used to investigate each part of the actual combined cycle by considering irreversibility from April 2006 to October 2010. Performance analysis was performed for each part by evaluating the exergy destruction in each month. By use of the exergy analysis, the aging of each part was evaluated with respect to time. In addition, the rate of lost work for each month was calculated and the variation of this parameter was considered as a function of aging rate. Finally, the effect of exergy destruction of each part on the exergy destruction of the whole cycle was investigated. The entire analysis was realized for Unit 3 and 4 of the gas turbine cycle that combine with Unit B of the steam cycle in the Fars Combined Power Plant Cycle located in the Fars Province in Iran.

Keywords: exergy analysis; exergy loss; combined power plant.

INTRODUCTION

The rapid improvement of gas turbine technology in the 1990s drove combined cycle thermal efficiency to nearly 60 % with natural gas as the fuel[1] and it will probably go even higher in the future. This high plant efficiency together with low emissions, and competitive capital and running costs made combined cycle gas turbine (CCGT) plants very popular prime movers for electricity generation. This increasing interest in CCGT plants led to more consideration being devoted the behavior of a plant after long service times. As a result, simulation codes were developed to predict the behavior of such power plants and their subsystems on a thermo-fluid dynamic basis.[2,3] Even under normal engine operating conditions, with good inlet filtration systems and using clean fuel, the flow path section in the gas turbine engine will become fouled, eroded, corroded and co-

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vered with rust scale. Any failure or malfunctioning within the gas turbine would be magnified, as it would affect two CCGT plants simultaneously. Therefore, in combined cycles in which the gas turbine is connected to another plant, such as a steam cycle, the deterioration of the performance of the gas turbine engine would be increased. Exergy analysis based on the First and Second laws of Thermodynamics are significant tools to analyze energy systems. It also reveals inefficient thermodynamic processes. On the other hand, the Second Law of Thermodynamics deals with the quality of the energy and determines the maximum amount of work obtainable from an energy resource. Exergy analysis is performed in two main parts.[4] The first one is devoted to evaluating the system and determining inefficient processes based on exergy destruction calculations and the second one is based on making some changes and corrections in the processes based on avoidable and unavoidable exergy destructions. Exergy is defined as the maximum theoretical useful work that can be obtained as a system interacts with an equilibrium state. Exergy is generally not conserved like energy but is destroyed in the system. Exergy calculation shows the place in the system where losses occur and the magnitude of these losses. Exergy efficiency of a combined cycle power plant is usually lower than the thermal efficiency of the same plant.

In 1960, a primitive investigation of a combined cycle power plant was performed by Sieppel and Bereuter.[5] Czermak and Wunsch made a thermodynamic analysis of an actual combined power plant.[6] Wunsch mentioned that the efficiency of a combined cycle power plant is more related to the gas turbine parameters, such as maximum temperature and pressure ratio, than to the steam cycle parameters. Khaliq and Kaushik made a Second Law (exergy) analysis of a Rankine–Brayton cycle with a pre-heater. They also derived some correlation for the First and Second Law efficiency of the entire cycle and the exergy destruction for each part.[7] Ramaprabhu studied a computational model of a combined cycle power plant with an inlet air conditioning (fogging) system and applied it to an actual cycle in Arizona, USA.[8] Arrieta *et al*. investigated an actual combined cycle in Brazil and revealed that the ambient temperature, atmospheric pressure and air humidity have significant effects on the performance of a combined cycle.[9]

Cihan[10] used an exergy and energy analysis for a combined cycle power plant and reported that the combustion chambers, gas turbines and HRSG account for more than 85 % of the exergy losses. Sue and Chuang[11] performed a First and Second Law analysis of a combined cycle power plant with and without cogeneration using dual pressure HRSGs, turbine inlet air-cooling, absorption chilling and fuel gas preheating. Khaliq and Kaushik[12] investigated the effects of the steam pressure and pinch point in a gas turbine cogeneration system and reported higher First and Second Law efficiencies with reheat. The effect of reheat and intercooling on Second Law performance of a gas turbine cogeneration system has not received much attention in the literature, which was also registered by Khaliq and Kaushik. According to them, energy-based performance analyses are often misleading as they fail to identify the deviation from ideality. Ganapathy *et al*. performed an exergy and energy analysis for investigating a 50 MW steam cycle with coal as the fuel and declared that the energy assessment must be made through the energy quantity as well as the quality.[13] Erdem analyzed comparatively the performance of thermal power plants from an exergetic and energetic viewpoint in Turkey. By means of exergetic and energetic analyses, the thermodynamic inefficiencies of each plant were identified.[14] Irreversibility in a combined cycle power plant may be caused by several damages, such as fouling and erosion of the gas turbine blade, fouling in the compressor, deposition in the HRSG and condenser. Several papers in which these damages in combined cycle power plants were studied are available.[15–20]

There have been several studies on degradation in combined cycle in recent years but no study has been performed on a time dependent exergy analysis with modeling of aging. The purpose of this work was the use of time dependent exergy analysis for investigating each part of the Fars combined cycle plant by considering irreversibility. In addition, performance analysis was realized for each part by evaluating the exergy destruction during April 2006 to October 2010.

THEORY AND METHOD

Exergy can be divided into four distinct components. The two important ones are physical exergy and chemical exergy. In this study, the other two components, *i.e.*, kinetic exergy and potential exergy, were assumed negligible as the elevation and speed have negligible changes. The physical exergy is defined as the maximum theoretical useful work obtained as a system interacts with an equilibrium state. The chemical exergy is associated with the departure of the chemical composition of a system from its chemical equilibrium. The chemical exergy is an important part of exergy in combustion processes. Availability for any thermodynamic state can be calculated as:

$$av_{\mathrm{i}} = h_{\mathrm{i}} - h_{\mathrm{o}} - T_{\mathrm{o}}(s_{\mathrm{i}} - s_{\mathrm{o}}) \tag{1}$$

The physical exergy for that state is:

$$\dot{E}x_{\mathrm{ph,i}} = \dot{m}_{\mathrm{i}}\, av_{\mathrm{i}} \tag{2}$$

The chemical exergy of air, fuel and exhaust gas can be written, respectively, as:

$$\dot{E}_{\mathrm{ch,air}} = \dot{m}_{\mathrm{air}}\left(\frac{\bar{R}}{M_{\mathrm{air}}}\right)T_{\mathrm{o}}\left[y_{\mathrm{O_2,air}}\ln\left(\frac{y_{\mathrm{O_2,air}}}{y^{\mathrm{e}}_{\mathrm{O_2,air}}}\right) + y_{\mathrm{N_2,air}}\ln\left(\frac{y_{\mathrm{N_2,air}}}{y^{\mathrm{e}}_{\mathrm{N_2,air}}}\right)\right] \tag{3}$$

$$\dot{E}_{\mathrm{fuel,ch}} = \left(\frac{\dot{m}_{\mathrm{fuel}}}{M_{\mathrm{CH_4}}}\right)\left[g_{\mathrm{CH_4}} + 2g_{\mathrm{O_2}} - g_{\mathrm{CO_2}} - 2g_{\mathrm{H_2O(g)}} + RT_{\mathrm{o}}\ln\left(\frac{(y^{\mathrm{e}}_{\mathrm{O_2}})^2}{y^{\mathrm{e}}_{\mathrm{CO_2}}(y^{\mathrm{e}}_{\mathrm{H_2O}})^2}\right)\right] \tag{4}$$

$$\dot{E}_{\text{ch,exhaust}} = \dot{m}_{\text{exhaust}} \left(\frac{\bar{R}}{M_{\text{exhaust}}}\right) T_{\text{o}} \times$$

$$\times \left[y_{CO_2} \ln\left(\frac{y_{CO_2}}{y^{e}_{CO_2}}\right) + y_{H_2O} \ln\left(\frac{y_{H_2O}}{y^{e}_{H_2O}}\right) + y_{N_2} \ln\left(\frac{y_{N_2}}{y^{e}_{N_2}}\right) + y_{O_2} \ln\left(\frac{y_{O_2}}{y^{e}_{O_2}}\right) \right] \quad (5)$$

where y_i^e is the equivalent mole ratio of constitute i of a gas mixture and for air, the values are given in Table I. $\bar{R}$ is the gas constant and g_i is the Gibbs function of a gas constitute. The Gibbs function of some gases at 298 K and 1 atm are:

$$g_{CH_4} = -50790,\ g_{O_2} = 0,\ g_{CO_2} = -394380 \text{ and } g_{H_2O} = -228590$$

Table 1. Molar mass and equivalent mole ratio of air components

Product	M / g mol^{-1}	y_e
N_2	28.17	0.7567
O_2	32	0.2035
H_2O gas	18	0.0303
CO_2	44	0.0003

Each device in the power plant constitutes a control volume and the associated equations of energy and exergy analysis are given below:

The Continuity Equation is expressed as:

$$\sum_{i} \dot{m}_{i} = \sum_{e} \dot{m}_{e} \quad (6)$$

where $\dot{m}$ is the mass flow rate and the subscripts i and e refer to the inlet and exit conditions, respectively.

First Law of Thermodynamics is in the following form:

$$\sum_{i} \dot{m}_{i} h_{i} + \dot{Q} = \sum_{e} \dot{m}_{e} h_{e} + \dot{W} \quad (7)$$

where $\dot{Q}$ is the heat transfer rate to the control volume, $\dot{W}$ is the work given out per unit time and h is the enthalpy. The kinetic and potential energy changes are omitted since they are negligibly small compared to the changes in enthalpy.

The exergy balance is presented as follow:

$$\dot{E}_{Q} - \dot{E}_{W} = \sum_{e} \dot{m}_{e} e_{e} - \sum_{i} \dot{m}_{i} e_{i} + \dot{E}_{L} \quad (8)$$

where e is the specific exergy and E_L is the exergy loss rate. E_Q and E_W are the exergy rates due to the heat input and mechanical energy, respectively, which are defined as:

$$\dot{E}_{Q} = \left(1 - \frac{T_{o}}{T_{j}}\right) \dot{Q}_{j} \quad (9)$$

$$\dot{E}_{W} = \dot{W} \quad (10)$$

where T is the absolute temperature and the subscripts j and o refer to the surface and environmental conditions, respectively. Exergy destruction in each component of a combined cycle, which is schematically shown in Fig. 1, can be calculated as below:

Exergy destruction for compressor is presented as:

$$\dot{E}_{\mathrm{L_{Comp}}} = \dot{W}_{\mathrm{C}} - (\dot{E}_2 - \dot{E}_1)\,, \tag{11}$$

for the combustion chamber as:

$$\dot{E}_{\mathrm{L_{C,Ch}}} = \dot{E}_{\mathrm{fuel}} - (\dot{E}_3 - \dot{E}_2)\,, \tag{12}$$

for the gas turbine as:

$$\dot{E}_{\mathrm{L_{GT}}} = (\dot{E}_3 - \dot{E}_4) - \dot{W}_{\mathrm{GT}}\,, \tag{13}$$

for the HRSG as:

$$\dot{E}_{\mathrm{L_{HRSG}}} = (\dot{E}_4 - \dot{E}_9) - (\dot{E}_5 - \dot{E}_8)\,, \tag{14}$$

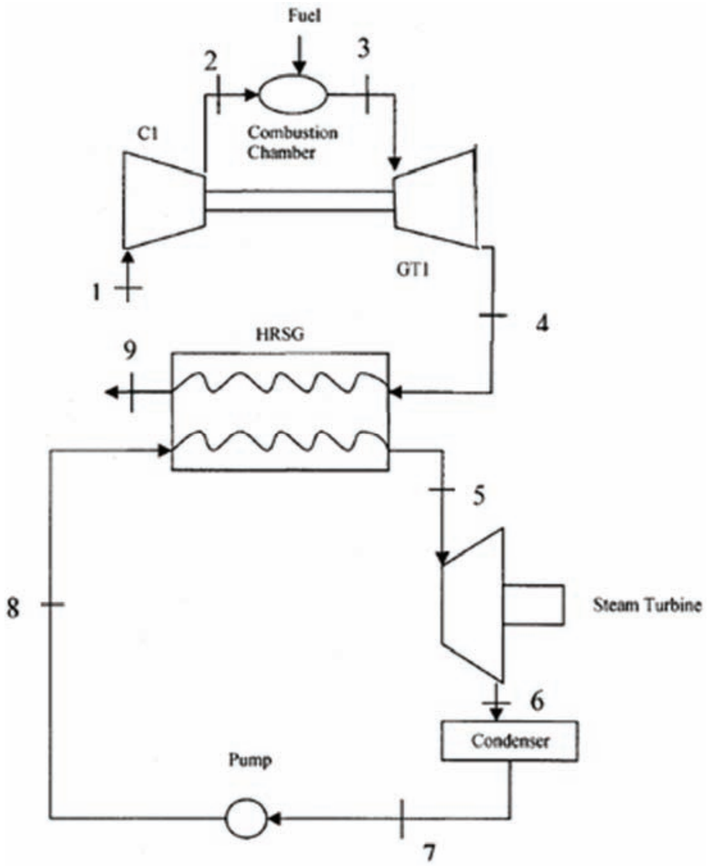

Fig. 1. The components of a combined cycle.

for the steam turbine as:

$$\dot{E}_{\mathrm{L_{ST}}} = (\dot{E}_5 - \dot{E}_6) - \dot{W}_{\mathrm{ST}}\,, \tag{15}$$

and for the condenser as:

$$\dot{E}_{\mathrm{L_{Cond}}} = (\dot{E}_7 - \dot{E}_6) \tag{16}$$

Finally, the Second Law efficiency could be written as follow:

$$\eta_{\mathrm{II}} = \frac{\sum_{\mathrm{k}} \dot{E}_{\mathrm{W_k}}}{\dot{E}_{\mathrm{fuel}}} \tag{17}$$

The heat recovery steam generators (HRSG) of Unit 3 and 4 in the Fars Combined Cycle Power Plant, which combine Unit 3 and 4 of the gas cycle and Unit B of the steam cycle are shown in Fig. 2.

For calculations of the exergy loss of the components, the inlet and outlet states of each component are required. The inlet and outlet compressor pressure and temperature are included in gas cycle data sheets, therefore the inlet temperature and pressure of the combustion chamber are known. The outlet temperature and pressure of the gas turbine could also be obtained from the gas cycle data sheets. By neglecting the pressure drop in the combustion chamber, the combustion chamber outlet pressure has the same value as the combustion chamber inlet pressure. To obtain the combustion chamber outlet temperature, the energy balance in the combustion chamber can be used:

$$\dot{m}_{\mathrm{air}} h_2 + \dot{m}_{\mathrm{f}}\, LHV = \dot{m}_{\mathrm{mix}} h_3 \tag{18}$$

where LHV is the low heating value of the fuel, which for natural gas is 50020 kJ kg^{-1} In Eq. (18), m_{air}, m_{mix} and h_3 are unknown. $\dot{m}_{mix}$ can be related to m_{air} and m_{f} as follows:

$$\dot{m}_{\mathrm{mix}} = \dot{m}_{\mathrm{f}} + \dot{m}_{\mathrm{air}} \tag{19}$$

By estimating m_{air} in Eq. (19), m_{mix} can be obtained. By using this value, Eq. (18) can be solved. T_3 can be obtained from combustion equation as below:

$$a\mathrm{CH_4} + x(\mathrm{O_2} + 3.76\mathrm{N_2}) \rightarrow a\mathrm{CO_2} + b\mathrm{H_2O} + c\mathrm{N_2} + d\mathrm{O_2} \tag{20}$$

where

$$\begin{aligned} &a = \frac{\dot{m}_{\mathrm{fuel}}}{M_{\mathrm{fuel}}},\ x = \frac{\dot{m}_{\mathrm{air}}}{M_{\mathrm{air}} \times 4.76} \\ &c = 3.76x \\ &b = 2a \\ &d = x - 2a \end{aligned} \tag{21}$$

In this case, the mole fraction of each component in the mixture that entered the gas turbine can be calculated as follow:

$$y_{\mathrm{CO_2}} = a/n, \quad y_{\mathrm{H_2O}} = b/n, \quad y_{\mathrm{N_2}} = c/n, \quad y_{\mathrm{O_2}} = d/n \tag{22}$$

where:

$$n = a + b + c + d \tag{23}$$

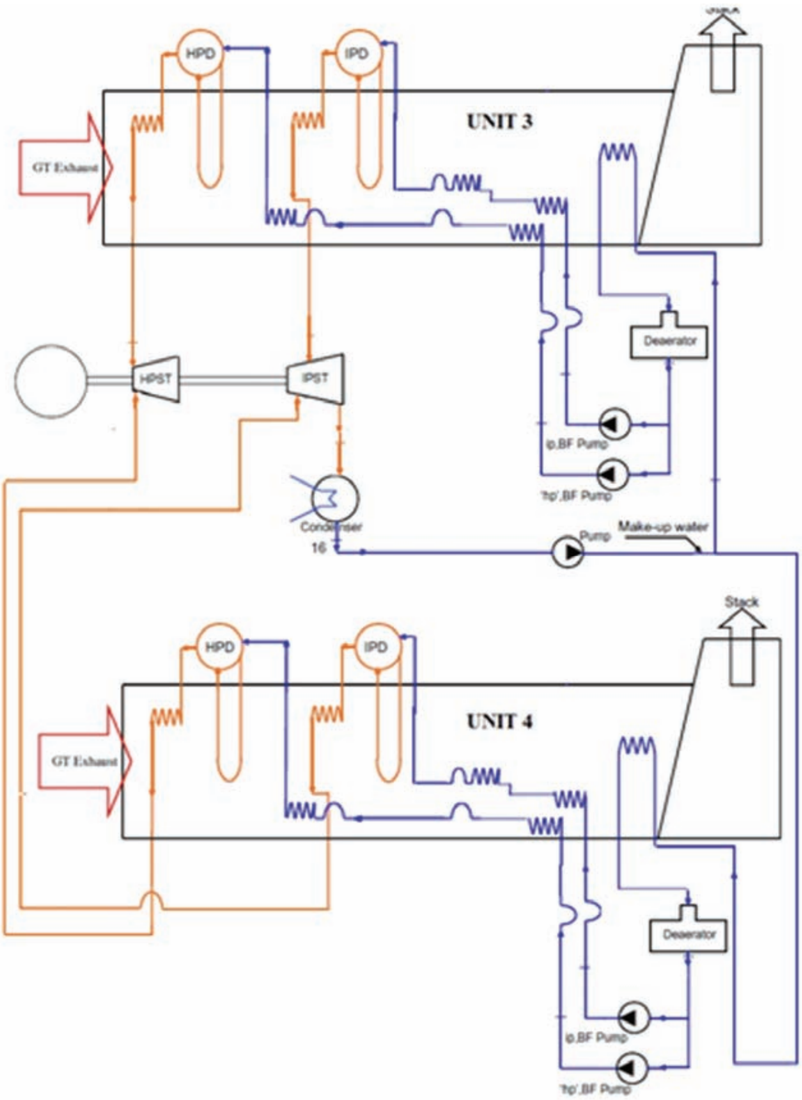

Fig. 2. HRSG of Unit 3 and 4 in the Fars Combined Cycle Power Plant.

$\bar{h}_3$, the combustion chamber outlet molar enthalpy, can be obtained by multiplying $\bar{h}_3$ and molar mass of the mixture, which is presented in Eq. (24):

$$\overline{h_3} = h_3 M_{\text{mix}} \tag{24}$$

M_{mix} can be obtained by Eq. (24):

$$M_{\text{mix}} = \sum_i y_{\text{i}} M_{\text{i}} \tag{25}$$

The combustion chamber outlet molar enthalpy can also be calculated using the Gas Law as follow:

$$\bar{h}_3 = \sum_i y_i \bar{h}_i(T_3) \tag{26}$$

The molar enthalpy for each component of the mixture can be obtained from thermodynamic tables. By using the try and error process, T_3 can be obtained from Eq. (26).

The accuracy of the primitive estimation for m_{air} could be ensured from the output work correlation, which presented by Eq. (27):

$$W_{net} + M_{air}\left(h_2 - h_1\right) = M_{mix}\left(h_3 - h_1\right) \tag{27}$$

If this equation is correct, the primitive estimation for m_{air} is also correct. The inlet and outlet state of the steam cycle components can be obtained from data that is held in the ANNEX room of the Fars Combined Power Plant complex.

Plant description

In 2003, one of the notable power plant projects in Iran was inaugurated. This project, which includes 22 Combined Cycle Power Plants, has not yet been completed.[21] One of these power plants, *i.e.*, the Fars Combined Power Plant, is located near the city of Shiraz. This power plant has six gas Units. Each two-gas unit combines with a steam unit by using HRSGs. As a result, this plant has six gas cycles and three steam cycles. The Siemens GE Frame 9 gas turbines of this combined cycle were installed in 1981. Each gas cycle output power, at 100 % load, is 100 MW. In each gas cycle, air is compressed to approximately 10 bars by an axial compressor from the ambient pressure of 0.86 bars. The compressed air enters into the combustion chamber. The flue gas from the combustion chamber enters the turbine at approximately 1550 K. The exhaust gas at 550 °C enters the HRSG without supplementary firing. Each dual pressure HRSG generates HP and LP at approximately 500 and 200 °C, respectively. The rated steam turbine power output is 130 MW at 100 % load. The expanded steam enters a condenser and transfers its heat to water. Then, the water flows through a cooling tower and releases its heat. The condensate water drops to the hot well at the bottom of the condenser and is pumped to the deaeriator by means of condensate extraction pumps. The feed water to the LP evaporator and HP first and second economizers is fed by a common two-pressure HP/LP pump, which take their suction from the feed water storage tank. The HP water discharge of the pump passing through HP economizers goes to the HP drum and, after evaporation, the separated saturated steam in the drum passes through the superheater. Finally, this steam goes to the steam turbine HP section and completes the cycle.

RESULTS AND DISCUSSION

In present study, an exergy analysis was realized for Units 3 and 4 gas turbine and Unit B steam turbine of the Fars Combined Power Plant. The data for the analysis were obtained from resources such as the archives of the power plant, log sheets of the gas cycle, daily reports of the steam and gas units and the daily fuel consumption of whole power plant. Results of the power output and exergy loss for Units 3 and 4, the entropy generation of the whole cycle by means of the power output and the Second Law efficiency, exergy loss for gas, steam and the whole cycle for each month and the exergy loss per MW for each component of the cycle are presented in this section.

Power output and exergy loss for the components of Units 3 and 4

The power output and exergy loss for different parts of Units 3 and 4 are presented, respectively, in Figs. S1 and S2 (Supplementary material to this paper). For a proper comparison, a factor of the exergy loss for each component was considered to obtain values in the same order of magnitude. Thus the exergy loss of the compressor, gas turbine, combustion chamber and HRSG were divided by 100, 250, 2000 and 500, respectively. From Figs. S1 and S2, the exergy loss of each part showed an approximately ascending trend with some oscillations. As observed in the figures, these oscillations were related to variations in the power output; therefore, the power output and the exergy loss for each part have a similar trend. This similarity was obtained due to the essence of exergy loss, which is directly related to power output and irreversibilities. In addition, it could be observed that with decreasing output power, the exergy loss of each part also decreased. The decreasing power output was caused by reduction in the fuel consumption, which descended the input exergy of the whole unit. This phenomenon affected directly the exergy loss of each part.

Entropy generation for the whole cycle based on the power output and Second Law efficiency

The entropy generation *vs*. power output is shown in Fig. S3 (Supplementary material to this paper). As can be seen, the entropy generation increased with increasing power output of the combined cycle. The entropy generation based on the Second Law efficiency is presented in Fig. S4 (Supplementary material to the paper), from which it can be seen that with increasing the Second Law efficiency, a greater part of the input fuel exergy is transformed to mechanical energy. Accordingly, the exergy losses and irreversibility become a minor part of the input exergy. This conclusion could be confirmed by Eq. (17). Two distinct trends can be seen in Fig. S4. The upper trend related to the time when Units 3 and 4 were working together and a lower when one of the Units was under maintenance. Based on the fact mentioned above in relation to Fig. S3, by decreasing the power output, lower entropy generation could be obtained for the whole cycle.

Exergy losses for Units 3 and 4 gas turbine, Unit B steam turbine and whole cycle based on time duration

Exergy losses *vs*. time duration for the whole cycle, Units 3 and 4 gas turbine and Unit B steam turbine are presented in Fig. 3. MI, CCI and HGP imply major inspection or overhaul of each unit, combustion chamber inspection and high temperature gas path, respectively. The black line is related to the exergy loss of the whole cycle, the blue line to the Unit B steam turbine, the green and orange lines present the exergy loss of Unit 3 and 4 gas turbine, respectively. From diagrams in Fig. 3, the exergy loss value for the steam cycle is much lower than the value in gas cycle. This event is due to the high exergy loss in the components of the gas cycle, such as combustion chamber, gas turbine and HRSG, which is in good correlation with previous studies. In Fig. 3, it can be seen that the whole cycle exergy loss has a rapid decrease caused by maintenance of some units. In addition, the exergy loss for each unit had an increasing trend with some oscillations. These oscillations were caused by fluctuations in the power output of each unit. In Fig. S5 (Supplementary material of the paper), non- dimensional analysis was realized by dividing the exergy loss by the power output of each unit, in order to eliminate power output effects. As a result, an increasing trend of exergy loss per MW for each unit was obtained. It is very obvious that after the maintenance period of each unit, the exergy loss per MW decreased, this reduction can be seen from the results for the whole cycle.

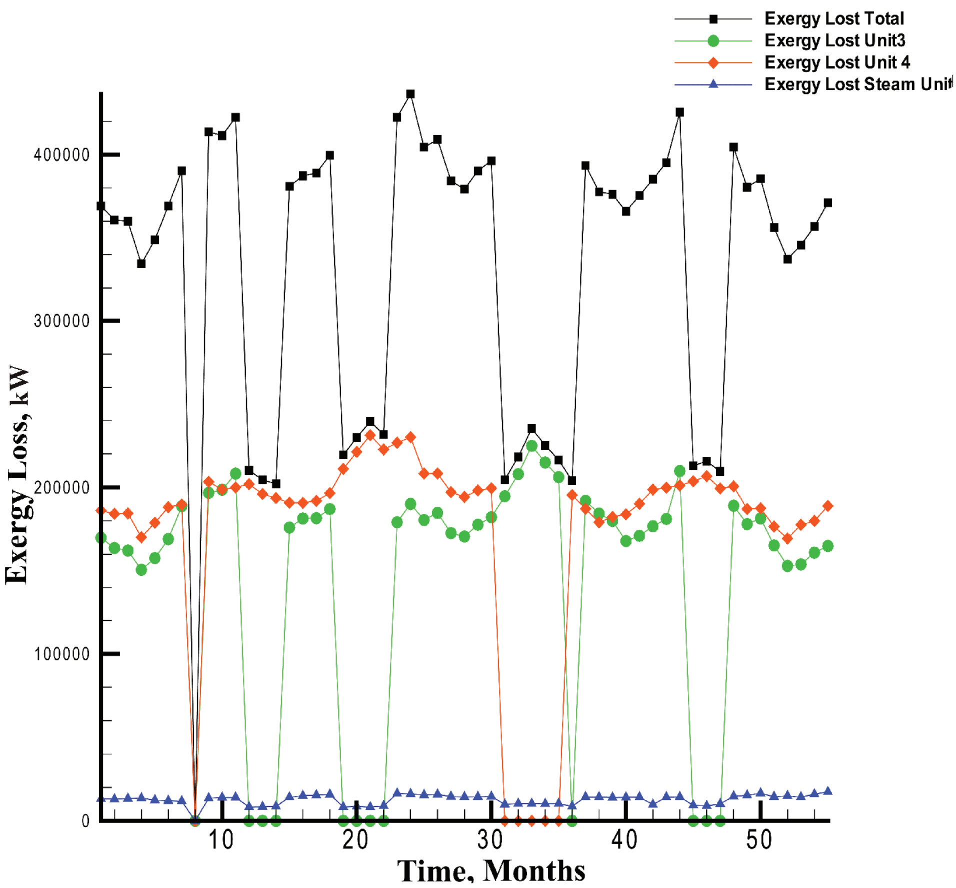

Fig. 3. Exergy losses *vs*. time for the whole cycle, Units 3 and 4 gas turbines and Unit B steam turbine.

Exergy loss per MW for each component in Unit 3, 4 and Unit B

In this section, exergy loss per MW for each component in Units 3, 4 and B is presented. Due to the low exergy loss in the condenser and pump, these parts are neglected in the exergy analysis. As a result, the exergy loss per MW for the compressors, gas and steam turbines, combustion chambers and HRSG were obtained and calculated.

Exergy loss per MW for compressors. The exergy losses per MW for the compressors are presented in Figs. S6 and S7 (Supplementary material to the paper). Based on the maintenance schedule, a major inspection (MI) of Unit 3 occurred from Oct 2007 to Jan 2008 and of Unit 4 from Oct 2008 to Feb 2009. As can be seen from Fig. S6, the exergy loss for the compressor of Unit 3 increased because of the MI because either the MI was not realized at the proper moment or improper maintenance was performed. In Fig. S7, a large reduction in exergy loss for compressor of Unit 4 could be noticed after the MI period because the MI was performed at the proper moment. In addition, the exergy loss for the compressor of Unit 3 is greater than that for the compressor of Unit 4; for this reason, the performance of Unit 4 is better than that of Unit 3.

Exergy loss per MW for the combustion chambers. Exergy losses per MW for the combustion chambers are presented in Figs. S8 and S9 (Supplementary material to the paper). As could be seen, exergy losses for the combustion chambers were greater than those from the other components. From the results presented Figs. S8 and S9, it could be stated that the MI process performed at the right moment, when the exergy losses had their highest values. In combustion chamber, CCI, MI and HGP process have the great effects on realizing a reduction of exergy loss for both Unit 3 and 4.

Exergy loss per MW for gas turbines and HRSGs. Exergy losses per MW for the gas turbines and HRSGs are presented in Figs. S10–S13 (Supplementary material to the paper). In all diagrams, after MI and the CCI process, the exergy loss for each component was reduced significantly. In addition, by means of the HGP process, the exergy loss was decreased because the HGP process improved the isentropic efficiency of the gas turbine, which reduced the irreversibility of the gas turbine and HRSG. As in the previous data, it is noticeable that exergy loss decreased after the maintenance process. In addition, Unit 3 had a faster trend in increasing the exergy loss than Unit 4.

Exergy loss per MW for the steam turbine. The exergy loss per MW for the steam turbine is presented in Fig. S14 (Supplementary material to the paper). Unexpected increasing trends in the exergy loss could be seen. This occurred when one of the gas units was under maintenance. In Fig. S14, a significant increase in exergy loss took place from month 31 to 35 when Unit 4 was under the MI process. Other increasing trend was related to maintenance of Unit 3. It is obvious that when Unit 4 was under overhaul, the exergy loss in the steam turbine was extensively increased. This event is the result of the mentioned proper operation of Unit 4 in the whole cycle. In addition, after the HGP period of Unit 3, an extreme reduction could be seen in exergy loss; hence, the high temperature gas path of Unit 3 might be the reason for the large exergy loses. Finally, by neglecting the maintenance periods, the exergy loss per MW during time has an ascending trend.

CONCLUSIONS

The combined power plant is one of the most important power generators in Iran and special consideration must be given for this type of power plant. In this study, an exergy analysis was performed for each component of Unit 3 and 4 gas cycle and Unit B steam cycle based on data from the Fars Combined Power Plant. The results showed that, the exergy loss in the components of Unit 3 and 4 increased with increasing power output and also increased with operation time. In addition, to consider time effect on exergy loss, the exergy loss was divided by MW to eliminate the MW effect. Finally, entropy generation of whole power plant based on MW and the Second Law efficiency were considered and calculated.

SUPPLEMENTARY MATERIAL

Figs. S1–S14 are available electronically from http://www.shd.org.rs/JSCS/, or from the corresponding author on request.

И З В О Д

НУМЕРИЧКА АНАЛИЗА И ИНДУСТРИЈСКА ИСПИТИВАЊА ВРЕМЕНСКЕ ЗАВИСНОСТИ ЕКСЕГИЈА–ЕНЕРГИЈА КОД КОМБИНОВАНОГ ЦИКЛУСА ЗА ПРОИЗВОДЊУ ЕЛЕКТРИЧНЕ ЕНЕРГИЈЕ

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Истраживање дато у овом раду анализира комерцијални комбиновани циклус за производњу електичне енергије у области Фарс у Ирану, употребом ексергетске анализе, при чему је промена ексергије разматрана у зависности од времена. Ексергетска анализа је употребљна да би се истражио сваки део комбинованог циклуса узимањем у обзир неповратности процеса који се дешава у сваком делу опреме у периоду од априла 2006 до октобра 2010. Анализа сваког дела опреме спроведена је прорачунавањем трансформације ексергије у анергију за сваки месец. Употребом ексергетске анализе смањење ефикасности опреме разматрано је у зависности од времена. Такође, израчуната је брзина умањења рада за сваки месец, при чему је умањење разматрано као функција времена употребљавања опреме. На крају, разматрано је како умањење ексергије за сваки део опреме утиче на умањење ексергије целог циклуса. Анализа је изведена за комерцијални комбиновани циклус у области Фарс у Ирану, у коме се електрична енергија производи употребом гасних турбина у постројењима 3 и 4, док се издувни гасови из гасних турбина комбинују са парним постројењем Б.

(Примљено 8. јула 2011, ревидирано 9. фебруара 2012)

REFERENCES

1. M. S. Briesch, R. L. Bannister, *J. Eng. Gas Turbines Power* **117** (1995) 734
2. M. R. Erbes, R. R. Gay, in *Proceeding of the Winter Annual Meeting of the ASME*, San Francisco, CA, USA, 1989, p. 223
3. J. Roy-Akins, *Proc. Inst. Mech. Eng. Part A: J. Power Energy* **209** (1995) 281
4. T. J. Kotas, *The Exergy Method of Thermal Plant Analysis*, Butterworth, Stoneham, MA, USA, 1985
5. C. Sieppel, R. Bereuter, *Brown Boveri Rev.* **47** (1960) 83
6. H. Czermak, A. Wunsch, in *Proceedings of ASME*, 1982, USA, paper No. 82 GT-323
7. A. Khaliq, S. C. Kaushikb, *Appl. Energy* **78** (2004) 179
8. V. Ramaprabhu, R. P. Roy, *J. Energ. Resour. - ASME* **126 (**2004) 231
9. F. R. P. Arrieta, L. E. E. Silva, *Appl. Energy* **80** (2005) 261
10. A. Cihan, K. Kahveci, *Int. J. Energy Res.* **30** (2006) 115
11. D. C. Sue, C. C. Chuang, *Energy* **29** (2004) 1183
12. A. Khaliq, S. C. Kaushik, *Appl. Therm. Eng.* **24** (2004) 1785
13. T. Ganapathy, N. Alagumurthi, R. P. Gakkhar, K. Murugesan, *J. Eng. Sci. Tech*. **2** (2009) 123
14. H. H. Erdem, A. V. Akkaya, *Int. J. Therm. Sci*. **48** (2009) 2179
15. A. Hamed, W. Tabakoff, D. Singh, *Int. J. Rotating Mach.* **4** (1998) 243
16. M. Morini, M. Pinelli, *J. Eng. Gas Turbines Power **132*** (2010) 032401-1
17. A. Hamed, W. Tabakoff, *J. Turbomach.* **127** (2005) 445
18. P. R. Spina, in *Proceedings of ASME Turbo Expo*, Amsterdam, Netherlands, 2002, ASME P. GT-2002-30275
19. D. E. Muir, H. I. H. Saravanamuttoo, D. J. Marshall, *J. Eng. Gas Turbines Power* **111** (1989) 244
20. A. R. Howell, R. P. Bonham, *Proc. Inst. Mech. Eng., C* **163** (1950) 235
21. M. Ameri, P. Ahmadi, S. Khanmohammadi, *Int. J. Energy Res.* **32** (2008) 175.