Mechanics & Industry **22**, 8 (2021)

https://doi.org/10.1051/meca/2021005

Mechanics & Industry

Available online at:
www.mechanics-industry.org

Regular Article

Open Access

Comparison of six gas turbine power cycle, a key to improve power plants

Ali Akbar Golneshan[1] and Hossain Nemati[2,*]

[1] Thermo-Fluid Department, School of Mechanical Engineering, Shiraz University, Shiraz
[2] Department of Mechanics, Marvdasht Branch, Islamic Azad University, Marvdasht, Iran

Received: 7 September 2020 / Accepted: 20 January 2021

Abstract. It is required to sake methods to improve the power plant performance. Most of the proposed methods can be commenced only in the design stapes. However, the main question of this study is "What can we do to improve the performance of a running power plant?" The first answer to this question is that monitoring the site and periodic overhaul can keep a power plant in its acceptable condition. However, this answer is very qualitatively and needs more precise information like which parameters shall be monitored or which equipment needs more care in the overhaul. In this study, important parameters and the method of their calculations are introduced that must be monitored and compared. Six similar gas turbine power cycles were selected to be compared deeply during a day in this study. In this way, many data were collected every five minutes. Unlike most of the previous studies, this one concerns with maintenance policy and repair strategy. Results of this comparison lead to answer to these questions that which equipment needs special care? Finally, it was shown that in each unit, which equipment needs more attention and which one can be considered as a standard for the others.

Keywords: Gas turbine / power cycle / thermodynamic optimization / overhaul policy

1 Introduction

In spite of the considerable improvement in electricity generation by new and green sources of energy, it is still undeniable that electrical power mainly depends on fossil fuels. On the other hand, it seems that the adverse effect of using fossil fuels on nature and climate besides the depletion of fossil fuel reserves are avoidable. So, it is required to use fossil fuels more efficiently.

Methods used in optimizing power plants may be divided into two distinct categories; methods that may be used during the design and methods which are applied during operation. Several references may be found that proposed the optimization of power plants [1–10]. Most of these works are based on exergy destruction minimization. The Exergy of a system at a specific thermodynamic state is the maximum available work that a system provides when reaches the equilibrium state relative to the surroundings.

It is known that [8,9] boiler systems as well as the turbines are the main sources of exergy destruction. Kaushik et al. [7] concluded that the largest exergy loss occurs during the combustion process. Hou et al. [11] found that in a steam power plant, most of the exergy destructs in steam turbines and in the boilers. In fact, irreversibilities in the combustion process and heat leak to ambient causes a large amount of exergy destruction in the boiler. The main contribution of the combustion process in exergy destruction was approved by Ahmadi and Toghraie [6]. They also showed that in the second stage, the steam turbine is the most considerable source of destroying exergy.

More and less, the same results may be found as researches conclusions. As it can be observed, these valuable results mostly concern with the first category aforementioned above and must be considered in design steps. Because considering these results lead to major changes in power plants and are usually avoided.

The present study, unlike those studies, is concerned with the second category. Maintenance policy and repair strategy have been always big questions during overhaul. Which equipment does need special care? For which one routine inspection or repairmen is enough? These questions are critical questions that can affect the performance of a power plant. However, those aforementioned studies may not answer these questions simply.

In such conditions, comparing similar plants may be a powerful technique to improve the performance of under operation plants. To show the ability of this method, the

* e-mail: H.Nemati@miau.ac.ir

Fars power station is selected as the pilot. The station has six similar gas turbine power cycles. The performance of the main components was compared with each other for one day. Data were gathered every 5 min in one day and thoroughly analyzed. Results of the comparison showed the strength and or weakness of each component. Later, it was deduced that which one was working more efficiently and which one needed maintenance. Based on this comparison, some advice has been suggested for further improvements. Owing to this fact that the comparison took place over one-day length; the impact of the ambient temperature was studied in reality.

This study is classified into five parts. After the introduction, in the second part, plant details are specified. Required equations are derived and presented in part 3. Since the evaporative coolers were at the service on that afternoon, the common assumption of dry air and using low heating value are not applicable. Part 4 is devoted to results and discussions. In this part, the comparisons of similar components from different viewpoints are presented. As a final point, a brief conclusion is presented in part 5. Moreover, complementary information is summarized in the appendices.

2 Plant details

The power station is comprised of 123.4 MW six gas. Combustion chambers are feed with natural gas. Ambient air is fed into the compressor through a filter and a small pressure drop occurs in the filtration process. At the ambient temperature around 40 °C, the evaporative cooler installed after the air filter comes into service. Spraying water droplets into the air stream decrease compressor inlet temperature and increase its humidity.

In the compressor, air pressure increases and is delivered to the combustion chamber. In the combustion chamber, the temperature increases in an approximately isobaric process. Outlet gas generates work in the turbine and then goes to the stack.

3 Analysis

Thermodynamic analysis for the power plant is presented in the following. Unless otherwise specified, the variation of kinetic energy as well as the potential energy is neglected. Also, flow is assumed steady-state. For a steady-state process, the mass balance and energy balance for each component can be written as:

$$\sum_o \dot{m}_o = \sum_i \dot{m}_i \tag{1}$$

$$\sum_i \dot{m}h + \dot{Q} = \sum_o \dot{m}h + \dot{W} \tag{2}$$

The required data like the pressure and temperature of all components of six gas plants were gathered on July, 27th. Data gathering started from 09:00 till 16:00 for every five minutes. A datasheet of measured data for gas unit number 1 is shown in Appendix A.

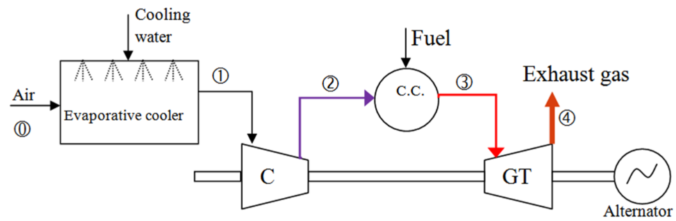

Fig. 1. A schematic diagram of the power plant.

Figure 1 shows a schematic diagram of the power plant. In all components, air or flue gas is considered as an ideal gas.

Using the evaporative coolers, the air moisture content is considerably high. So, fuel low heating value beside the dry air assumption is not applicable, and more precise equations are needed in combustion simulation.

3.1 Evaporative cooler

Enthalpy of moisture and air mixture is:

$$h_1 = h_{air1} + \frac{h_{water1}}{18\omega_1}. \tag{3}$$

in which ω_1 is the specific humidity of inlet air to the compressor. The mass-weighted average of the specific heat capacity of the mixture is assumed to be only a function of temperature:

$$cp_{1mix} = \frac{cp_{air}(T_1) + cp_{water}(T_1)\omega_1}{1+\omega_1} \tag{4}$$

$$cp_{2mix} = \frac{cp_{air}(\overline{T}_{12}) + cp_{water}(\overline{T}_{12})\omega_1}{1+\omega_1} \tag{5}$$

in which $\overline{T}_{12}$ is the average temperature between T_1 and T_2.

$$R_{mix} = \frac{R_{air} + R_{water}\omega_1}{1+\omega_1} \tag{6}$$

For times that evaporative cooler is shut down, $\omega = \omega_0$ and otherwise $\omega = \omega_1$:

$$\omega_1 = \frac{\omega_0\left[h_{water}(P_w, T_0) - h_{f,water}(T_0)\right] - cp_{air}(\overline{T}_{01})[T_1 - T_0]}{\left[h_{g,water}(T_1) - h_{f,water}(T_0)\right]} \tag{7}$$

h_g,*water* and $h_{f,water}$ are the saturated steam and saturated liquid specific enthalpy, respectively. In Eqution (8), the water partial pressure, P_w, is calculated by [12]:

$$P_w = \varphi_0 P_{sat,water}(T_0) \tag{8}$$

$$\varphi_0 = \frac{\omega_0 P_0}{(0.622+\omega_0)P_{sat,water}(T_0)} \tag{9}$$

Table 1. Fuel content.

	CH_4	C_2H_6	C_3H_8	N2
Molar Fraction %	89.19	4.1	1.21	5.5
Enthalpy of formation [13] (kJ/kmol)	−74873	−84740	−103900	0
Molar mass [13] (kmol/kg)	16.043	30.07	44.097	28.0135

Table 2. The flue gas contents.

	Co_2	H_2o	N_2
Mole	1.0102	1.9552	7.52913
Enthalpy of formation [13] (kJ/kmol)	393522	241826	0
Molar mass [13] (kmol/kg)	44.01	18	28.0135
Gas constant (R) [13] (kJ/kgK)	0.1889	0.4615	0.2968

In which $P_{sat,water}(T_0)$ is the water saturated pressure at temperature T_0. The inlet water vapor mass flow rate to the gas unit is:

$$\dot{m}_{moist,0} = \dot{m}_{dry\_air}\omega_0 \tag{10}$$

Also, the vapor mass flow rate inlet to the compressor is:

$$\dot{m}_{moist,1} = \dot{m}_{dry\_air}\omega_1 \tag{11}$$

Therefore, the total air mass flow rate is:

$$\dot{m}_{air} = \dot{m}_{dry\_air} + \dot{m}_{moist,1} \tag{12}$$

3.2 Compressor

The compressor isentropic pressure ratio can be calculated as:

$$\left(\frac{P_{2s}}{P_1}\right) = \left[\frac{T_2}{T_1}\right]^{cp2mix/R_{mix}} \tag{13}$$

in which P_{2s} is compressor isentropic outlet pressure.

3.3 Combustion chamber

Based on the fuel analysis, used natural gas components are (Tab. 1):

Analysis of exhausted flue gas from the stack shows the complete combustion. Therefore, the flue gas components are (Tab. 2):

The combustion in the combustion chamber can be simulated thermodynamically as:

$$\sum_i \dot{n}_i \left[h_f + \{h(T_3) - h(298.15K)\}\right]_i - \sum_j \dot{n}_j \left[h_f + \{h(T_3) - h(298.15K)\}\right]_j = \dot{Q} \tag{14}$$

In the above equation, h_f is the formation enthalpy and i and j, indicates to the ith component of inlet and the jth component of the outlet of the combustion chamber respectively. $\dot{Q}$ is the released heat for each mole of fuel.

Knowing x_i as mass fraction, J_i as mole fraction and W_i as molar mass of ith component and W_{fuel} as fuel molar mass, the thermodynamic properties of flue gas is:

$$cp_{1flu} = \frac{\left\{\begin{array}{c}(x_{Co_2}\, cp_{Co_2} + x_{h_2o}\, cp_{h_2o} + (x_{N2} + 3.76CW_{N2})cp_{N2} \\ +7.63C\omega_1 W_{H_2o} cp_{h_2o} + (C - J_{O2})W_{O2}cp_{O2})\end{array}\right\}}{(1+\omega_1)(137.33C) + W_{fuel}} \tag{15}$$

All above used properties are at the outlet temperature of the turbine. C is the available air mole (including excess air) for each mole of the fuel. Gas constant (R) of flue gas mixture is calculated as the weighted function of individual components:

$$R_{flu} = \frac{\left\{\begin{array}{c}R_{Co_2}x_{Co2} + R_{H2o}x_{H2o} + (x_{N2} + 3.76CW_{N2})R_{N2} \\ +7.63C\omega_1 W_{H_2o} R_{H2o} + (C - J_{O2})Wo_2 R_{O2}\end{array}\right\}}{(1+\omega_1)(137.33C) + 100W_{Fuel}} \tag{16}$$

in the same way, cp_{2flu} is evaluated at the outlet temperature of the combustion chamber.

3.4 Turbine

The isentropic temperature at the turbine outlet is evaluated as:

$$T_{S,4} = \left(\frac{P_1}{P_2}\right)^{R_{flu}/cp_{1flu}} \left[\frac{T_3}{\overline{T}_{34}}\right]^{cp_{2flu}/cp_{1flu}} \overline{T}_{34} \tag{17}$$

in which $\overline{T}_{34}$ is the average temperature between T_4 and T_3. Consequently, the isentropic efficiency of the turbine is:

$$\eta_T = \frac{cp_{2flu}\left[T_3 - \overline{T}_{34}\right] + cp_{1flu}\left[\overline{T}_{34} - T_4\right]}{cp_{2flu}\left[T_3 - \overline{T}_{34}\right] + cp_{1flu}\left[\overline{T}_{34} - T_{S,4}\right]} \tag{18}$$

It is assumed that $P_2 \approx P_3$ and $P_1 \approx P_4$. Moreover:

See equation (19) below.

in the above equation, all the thermodynamic properties are at T_4 temperature. In the same method, h_3 may is evaluated combustion chamber temperature, T_3. Temperature functionality of all mentioned thermodynamic variables is presented in Appendix B.

The mass flow rate of the dry air is calculated using known fuel component and mass flow rate of the fuel, $\dot{m}_{Fuel}$,:

$$\dot{m}_{dry\_air} = \frac{\dot{m}_{Fuel}}{W_{Fuel}C \times 28.851 \times 4.76} \tag{20}$$

So, the electricity output power is:

$$\dot{W}_E = \dot{m}_{dry\_air}[(h_3 - h_4) - (h_2 - h_1)]\eta_{Gen} \tag{21}$$

η_{Gen} is the efficiency of the power generator.

Unfortunately, combustion chamber exit temperature was not accessible. So, it was calculated indirectly by a try and error algorithm. For this reason, for a first guess of T_3, C and after that, $\dot{m}_{dry\_air}$ were calculated at and based on these values, $\dot{W}_E$ was calculated employing equtaion (21). This method was continued using a better guess for T_3, till the measured value of $\dot{W}_E$ and the calculated value was the same.

4 Results and discussions

4.1 The evaporative coolers

Ambient and inlet compressors temperatures are shown in Figure 2. According to this figure, the range of the ambient temperature is between around 30 °C at 9:00 and around 40 °C at noon. Interestingly, before starting an evaporative cooler, the air inlet temperature to the compressor of unit 1 is about 3– 4 °C warmer than other units. This shows that the location of the compressor unit 1 causes that air enters the compressor with higher temperature and this can be an object of the unit improvement. At 12:00, evaporative coolers start working. These evaporative coolers cool the air entrance temperature and reduce compressors consumption power. Worthwhile to note that, all evaporative coolers do not work in the same level of performance. Because there are a considerable differences among coolers outlet temperatures. The evaporative cooler of the 1st unit reduces the temperature from 40 to 22 °C, while cooler number 6 or number 3, can decrease temperature from 38 to 30 °C. So, this difference in outlet temperature is subject to question.

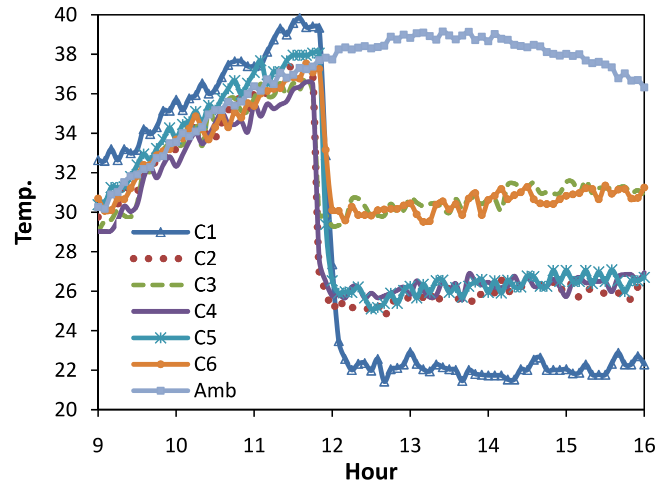

Fig. 2. Comparison of compressors inlet temperature and ambient temperature through one day.

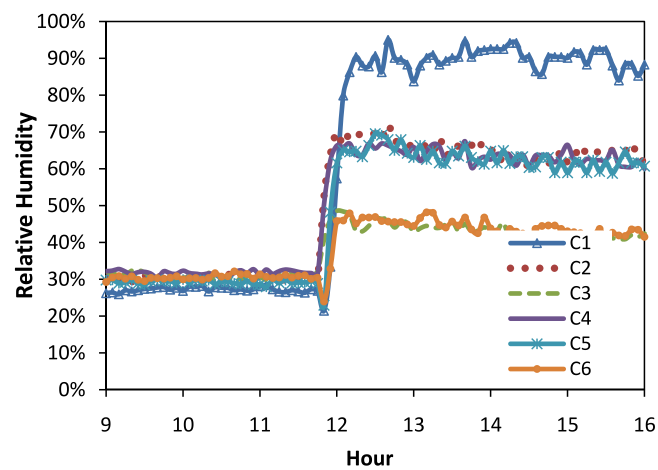

Fig. 3. Relative humidity of inlet air to compressors in one day.

The differences in cooler performances shall be investigated through their structure since the air inlet temperature to all the coolers is approximately the same. For this reason, the relative humilities of coolers outlet air are presented in Figure 3. It can be observed that, for cooler number 1, the outlet relative humidity is around 93% while for cooler numbers 2, 4, and 5, this value something between 60% and 70%. For cooler 3 and 6 the condition is even worse and the relative humidity is around 45%.

$$h_4 = \frac{J_{Co2}h_{Co2} + J_{H2o}h_{H2O} + (C - J_{O2})h_{O2} + (J_{N2} + 3.76C)h_{N2} + C\omega_1 \frac{137.33}{18} h_{H2O}}{137.33C} \tag{19}$$

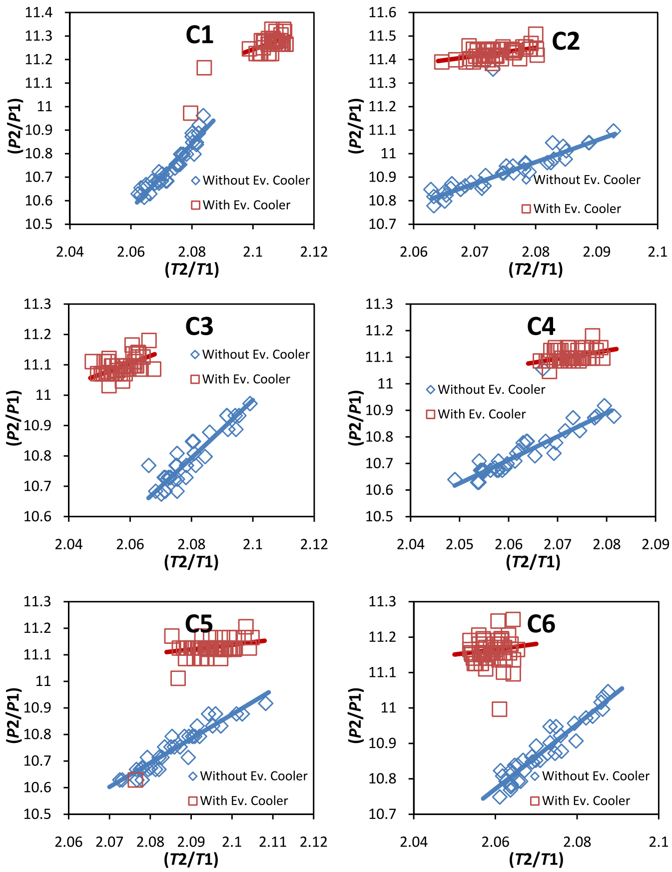

Fig. 4. Compression ratio against temperature ratio.

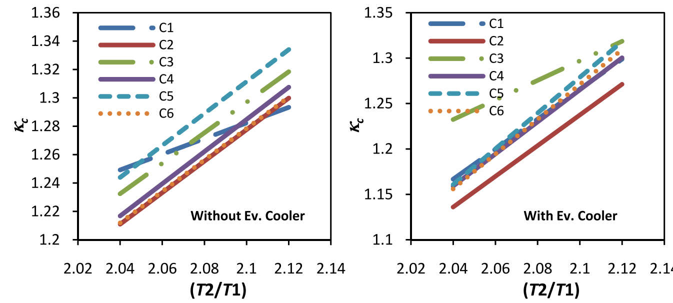

Fig. 5. Variations of κ_c against temperature ratio.

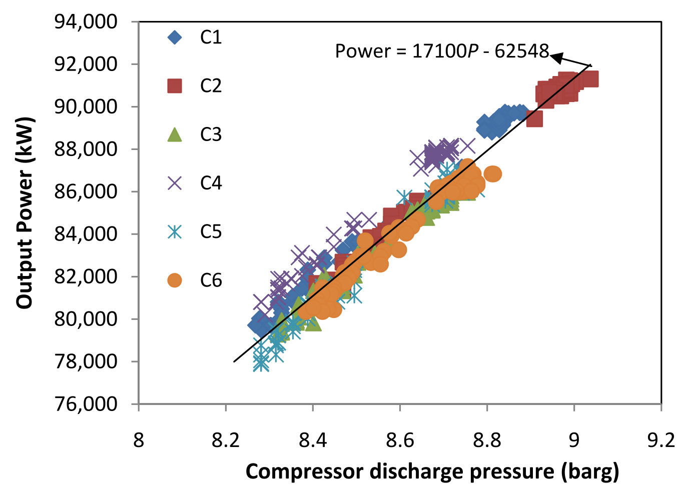

Fig. 6. Plants power outputs against compressor discharge pressure.

4.2 Compressors

Variations of compression ratio for all six compressors against outlet to inlet temperature ratio are presented in Figure 4. Aggregated points are related to times when evaporative coolers are in service and the compressors air inlet temperatures are independent of ambient temperature and are relatively constant. In the same manner, scatter points are related to the morning working condition. At first glance, it clear that decreasing the inlet temperature increases the compressor discharge pressure. Moreover, comparing the pressure ratio for the same temperature ratio in both conditions (with and without evaporative cooler), the pressure ratio when the evaporative cooler is in the service is much more.

However, this variation is not the same for all compressors. Although the inlet conditions of compressors are not the same, some parts of these deviations are due to dissimilarities in operation despite this fact that the compressors are the same.

To find out, which compressor works better and also to separate the contribution of inlet condition; κ_c may be a good parameter for judgment. κ_c is introduced as the ratio of isentropic to real compressor outlet pressure. Consequently, it shows how far the compressor outlet pressure is from isentropic outlet pressure:

$$\kappa_c = \frac{P_{2s}}{P_2} \tag{22}$$

It is clear that in an ideal compressor, the κ_c is equal to unity. Larger κ_c shows that the compressor is far from its ideal isentropic operation. Variations of κ_c against temperature ratio are shown in Figure 5 for both conditions, when evaporative coolers are in the service and when they are out of service. Figure 5 shows that, when the evaporative coolers are off, compressors numbers 2, 4, and 6 work efficiently in comparison to others.

Also, compressor number 2 performance is superior to the others in both working conditions. So, it is worth wide to compare other compressors with compressor number 2 during the overhaul.

All in all, it can be claimed that if all the coolers worked like cooler compressor number 2 and all the compressors like compressor number 1, then the compressor discharge pressure in all units would reach more than 9.1 bars; which means a considerable increase in output power. This judgment can be supported by Figure 6. This figure shows generated power outputs against the compressor discharge pressure. Based on Figure 6, an increase of 5% in compressor number 6 discharge pressure results in 7%

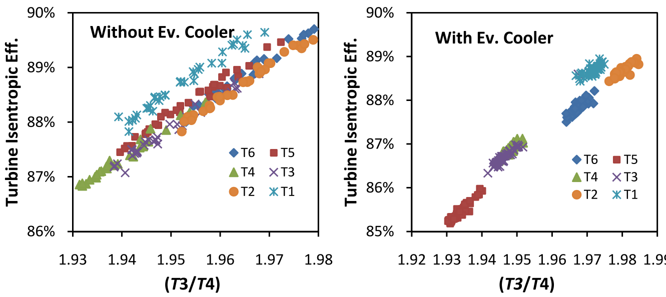

Fig. 7. Turbines isentropic efficiencies against temperature ratio.

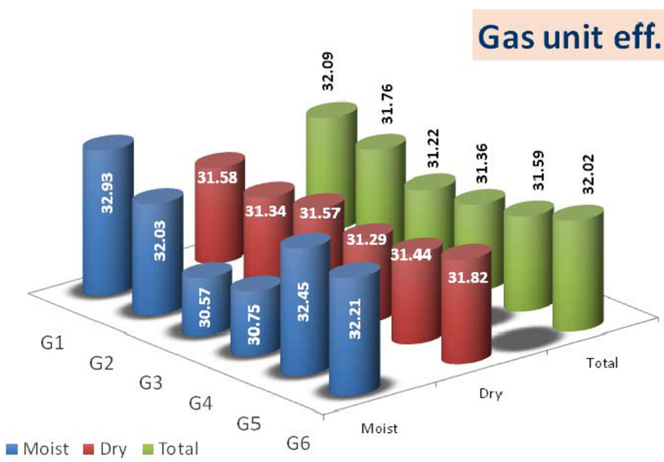

Fig. 8. Gas units efficiencies.

increase in generated power, or on the other hand, 7% increase in discharge pressure of the compressor number 2, increases output power 15%. These examples show how the output power is sensitive to compressor discharge pressure.

4.3 Turbine

Turbine is entropic efficiency is defined by equation (18). To have good judgment about the performances of gas turbines, the efficiency of each turbine has been shown in Figure 7 against its compressor temperature ratio, T_3/T_4, in both working conditions, when evaporative coolers are working and when they are out of service. This figure shows how far the efficiency of a turbine is from the ideal point. First of all, it shows that in both working conditions, by increasing the temperature ratio, T_3/T_4, the turbine efficiency gets closer to its ideal efficiency. Moreover, a meaningful difference may be observed between Turbine 1 with the others. So, it can be recommended that all turbines get compared deeply with turbine 1 during overhaul.

4.4 Power plants efficiencies

The efficiency of each power plant was calculated. The efficiency is a good index to compare units, overall. The results of this comparison are presented in Figure 8. Since evaporative coolers are not in the service all day, units efficiencies are classified into three groups:

- When the evaporative coolers are in service (in the afternoon),
- when the evaporative coolers are off (in the morning)
- and one for one day (the combination of two above).

Based on Figure 8, except for unit 3, the performances of all other units are approximately the same. However, the overall efficiency of unit 3 is slightly lower. Moreover, since the evaporative cooler in unit one has a higher performance when the evaporative coolers are in service, this unit has a higher efficiency than the others.

5 Conclusion

In this study, the required parameters for monitoring a power plant was introduced and calculated for six similar gas turbine. These parameters were compared in one day. As a result, it was cleared that which equipment works well and which one needs more care.

Some concluded results in these comparisons are:

- All evaporative coolers do not show the same performance. Cooler number 1 works better than the others while cooler number 3 or cooler number 6 is not as well as cooler number 1.
- Reducing the inlet temperature of the compressor increases the compressor discharge pressure.
- The compressor number 1 shall be verified thoroughly during overhaul.
- Performance of the compressor number 2 is better than others. So, it is worthwhile to compare other compressors with compressor number 2 during the overhaul.
- For a compressor in the same temperature ratio, the pressure ratio when the evaporative cooler is in the service is much more in comparison to the condition in which the evaporative cooler is off.
- It must be tried to reform all compressors to reach the discharge pressure of compressor number 2 and all coolers work like cooler number 1. Under this condition, compressor discharge pressure would reach 9.1 bars and consequently, a considerable gain in power.
- Since turbine 1 acts better than the other turbines, it is recommended that all turbines get compared deeply with turbine 6 during overhaul.
- Unit number 4 has lower efficiency among all units and unit numbers 6 and 1 have the best efficiencies.
- Due to the high performance of evaporative cooler number 1, the overall efficiency of unit 1, when the cooler is in service, is the highest.
- Using the evaporative coolers increase the plant efficacies significantly.

The above findings show the benefits of comparing similar cycles. These results simply show the ability of this method in cycle performance improvement. Valuable repair or maintenance guidelines can be in hand if one casts periodically this method.

Nomenclature

$cp(T)$	Specific heat capacity at temperature T (kJ/kg K)
h	Specific enthalpy (kJ/kg)
J	Mole fraction
$\dot{m}$	Mass flow rate (kg/s)
$\dot{Q}$	Rate of heat transfer (kW)
R	Gas constant (kJ/kmol K)
T	Temperature (K or °C)
W	Molar mass (kg/ kmol)
$\dot{W}$	Power (kW)
x	Mass fraction

Greek symbols

φ	Relative humidity
ω	Specific humidity
η_T	Turbine isentropic efficiency
η_{Th}	Unit thermodynamic efficiency

Subscripts

C	Compressor
Gen	Generator
i	Inlet
mix	Mixture
$moist$	Moisture
o	Outlet
S	Isentropic
sat	Saturated
T	Turbine

References

[1] R. Bontempo, M. Manna, Efficiency optimisation of advanced gas turbine recuperative-cycles, Proceedings of the Institution of Mechanical Engineers, Part A: Journal of Power and Energy (2019)

[2] A.A. Golneshan, H. Nemati, Energy and exergy comparison of three similar combined power cycles, Mechanics & Industry **20**, 504 (2019)

[3] S.M. Bapat, G.D. Gokak, Exergetic evaluation and optimization of Combined Heat and Power (CHP) plant of 20.7 MW capacities under varying load conditions: a case study exergetic evaluation and optimization of Combined Heat and Power (CHP) plant of 20.7 MW capacities under varying load conditions: a case study, European Journal of Electrical Engineering **21**, 207–219 (2019)

[4] A.A. Golneshan, H. Nemati, Evaluation of six gas turbine evaporative cooling for Fars-Iran, Annales de Chimie, Science des Materiaux 42, 281 (2018)

[5] A.S. Karakurt, Performance analysis of a steam turbine power plant at part load conditions, Journal of Thermal Engineering **3**, 1121–1128 (2017)

[6] G.R. Ahmadi, D. Toghraie, Energy and exergy analysis of Montazeri Steam Power Plant in Iran, Renewable and Sustainable Energy Reviews **56**, 454–463 (2016)

[7] S. Kaushik, V.S. Reddy, S. Tyagi, Energy and exergy analyses of thermal power plants: a review, Renewable and Sustainable Energy Reviews **15**, 1857–1872 (2011)

[8] P. Regulagadda, I. Dincer, G.F. Naterer, Exergy analysis of a thermal power plant with measured boiler and turbine losses, Applied Thermal Engineering **30**, 970–976 (2010)

[9] I.H. Aljundi, Energy and exergy analysis of a steam power plant in Jordan, Applied Thermal Engineering **29**, 324–328 (2009)

[10] M. Ameri, P. Ahmadi, S. Khanmohammadi, Exergy analysis of a 420 MW combined cycle power plant, International Journal of Energy Research **32**, 175–183 (2008)

[11] D. Hou, S. Shao, Y. Zhang, S.L. Liu, Y. Chen, S.S. Zhang, Exergy analysis of a thermal power plant using a modeling approach, Clean Technologies and Environmental Policy **14**, 805–813 (2012)

[12] C. Borgnakke, R.E. Sonntag, Fundamentals of thermodynamics, Wiley Global Education, 2016

[13] M.J. Moran, H.N. Shapiro, D.D. Boettner, M.B. Bailey, Fundamentals of Engineering Thermodynamics, Wiley, 2010

[14] H. Kurt, Z. Recebli, E. Gedik, Performance analysis of open cycle gas turbines, International Journal of Energy Research **33**, 285–294 (2009)

Cite this article as: A.A. Golneshan, H. Nemati, Comparison of six gas turbine power cycle, a key to improve power plants, Mechanics & Industry **22**, 8 (2021)

Appendix A: Sample of measuring data for gas unit number 1

			G1										
			Temperature (°C)				Pressure bar		$\dot{m}$ Fuel	Elec. Power	Power gen. Eff.	C.C. Eff.	HRSG Eff.
Time	Ambient Temp.	Ambient ω	Comp. Inlet	Comp. Outlet	Turb. Outlet	Water is added?	Comp. Inlet	Comp. Outlet	(Kg/s)	(kW)			
9:00:00	30.1	0.0107	32.6	364.0	553.8	No	0.86	8.529	5.77	84180	0.95	0.98	0.8
9:05:00	30.4	0.0109	32.6	363.5	554.2	No	0.86	8.495	5.77	83655	0.95	0.98	0.8
9:10:00	30.7	0.0111	33.2	364.1	555.1	No	0.86	8.495	5.77	83655	0.95	0.98	0.8
9:15:00	31.0	0.0113	32.6	363.5	554.5	No	0.86	8.495	5.77	83310	0.95	0.98	0.8
9:20:00	31.3	0.0115	33.2	364.1	554.3	No	0.86	8.495	5.77	83640	0.95	0.98	0.8
9:25:00	31.6	0.0117	33.0	364.1	555.1	No	0.86	8.456	5.77	83250	0.95	0.98	0.8
9:30:00	31.9	0.0119	33.2	364.0	555.1	No	0.86	8.456	5.77	83250	0.95	0.98	0.8
9:35:00	32.2	0.0121	34.2	366.4	556.3	No	0.86	8.456	5.77	82890	0.95	0.98	0.8
9:40:00	32.5	0.0123	34.0	364.8	555.6	No	0.86	8.422	5.77	82605	0.95	0.98	0.8
9:45:00	32.8	0.0125	34.3	366.8	555.6	No	0.86	8.422	5.77	82710	0.95	0.98	0.8
9:50:00	33.0	0.0127	35.1	366.5	556.9	No	0.86	8.388	5.77	82320	0.95	0.98	0.8
9:55:00	33.3	0.0128	35.1	367.5	555.7	No	0.86	8.422	5.77	81885	0.95	0.98	0.8
10:00:00	33.6	0.0129	35.7	368.1	556.5	No	0.86	8.422	5.77	81885	0.95	0.98	0.8
10:05:00	33.8	0.0120	35.2	367.1	556.5	No	0.86	8.422	5.77	81885	0.95	0.98	0.8
10:10:00	34.0	0.0120	35.8	368.2	556.7	No	0.86	8.422	5.77	81885	0.95	0.98	0.8
10:15:00	34.3	0.0121	35.9	368.3	556.6	No	0.86	8.388	5.77	81375	0.95	0.98	0.8
10:20:00	34.5	0.0122	36.5	368.4	556.5	No	0.86	8.354	5.77	80880	0.95	0.98	0.8
10:25:00	34.7	0.0124	36.0	368.4	556.1	No	0.86	8.388	5.77	81450	0.95	0.98	0.8
10:30:00	35.0	0.0125	36.2	368.6	557.1	No	0.86	8.388	5.77	81450	0.95	0.98	0.8
10:35:00	35.2	0.0127	36.8	368.6	557.5	No	0.86	8.354	5.77	81075	0.95	0.98	0.8
10:40:00	35.4	0.0129	37.5	369.2	558.2	No	0.86	8.354	5.77	80805	0.95	0.98	0.8
10:45:00	35.6	0.0120	37.6	369.9	557.8	No	0.86	8.32	5.77	80670	0.95	0.98	0.8
10:50:00	35.8	0.0120	37.6	369.9	556.7	No	0.86	8.354	5.77	80670	0.95	0.98	0.8
10:55:00	36.0	0.0120	37.4	370.2	557.3	No	0.86	8.32	5.77	80670	0.95	0.98	0.8
11:00:00	36.2	0.0120	37.4	369.7	557.9	No	0.86	8.32	5.77	80475	0.95	0.98	0.8
11:05:00	36.4	0.0120	37.4	370.2	557.9	No	0.86	8.281	5.77	80475	0.95	0.98	0.8
11:10:00	36.5	0.0120	38.1	370.9	558.0	No	0.86	8.315	5.77	80295	0.95	0.98	0.8
11:15:00	36.7	0.0120	38.8	371.5	557.2	No	0.86	8.315	5.77	80295	0.95	0.98	0.8
11:20:00	36.9	0.0120	39.3	371.5	557.9	No	0.86	8.315	5.77	80025	0.95	0.98	0.8
11:25:00	37.0	0.0120	39.0	371.8	558.5	No	0.86	8.315	5.77	80025	0.95	0.98	0.8
11:30:00	37.2	0.0120	39.6	371.8	558.8	No	0.86	8.281	5.77	79485	0.95	0.98	0.8
11:35:00	37.3	0.0121	39.8	373.1	558.6	No	0.86	8.315	5.77	79830	0.95	0.98	0.8
11:40:00	37.4	0.0121	39.4	372.2	558.3	No	0.86	8.281	5.77	79710	0.95	0.98	0.8
11:45:00	37.6	0.0121	39.4	372.1	558.1	No	0.86	8.281	5.77	79710	0.95	0.98	0.8
11:50:00	37.7	0.0121	39.3	371.6	559.0	No	0.86	8.281	5.77	79710	0.95	0.98	0.8
11:55:00	37.8	0.0121	32.9	363.3	553.3	No	0.86	8.61	5.77	84525	0.95	0.98	0.8
12:00:00	37.9	0.0121	27.4	353.1	549.9	Yes	0.86	8.704	6.13	87165	0.95	0.98	0.8
12:05:00	38.0	0.0121	23.5	349.4	549.7	Yes	0.86	8.704	6.13	88800	0.95	0.98	0.8
12:10:00	38.1	0.0122	22.6	348.9	548.8	Yes	0.86	8.704	6.13	89295	0.95	0.98	0.8
12:15:00	38.2	0.0122	22.0	348.6	548.5	Yes	0.86	8.704	6.13	89685	0.95	0.98	0.8
12:20:00	38.3	0.0122	22.3	348.9	548.5	Yes	0.86	8.704	6.13	89685	0.95	0.98	0.8
12:25:00	38.4	0.0122	22.4	349.4	548.5	Yes	0.86	8.742	6.13	89670	0.95	0.98	0.8

			G1										
			Temperature (°C)				Pressure bar		$\dot{m}$ Fuel (Kg/s)	Elec. Power (kW)	Power gen. Eff.	C.C. Eff.	HRSG Eff.
Time	Ambient Temp.	Ambient ω	Comp. Inlet	Comp. Outlet	Turb. Outlet	Water is added?	Comp. Inlet	Comp. Outlet					
12:30:00	38.5	0.0122	22.0	349.6	547.6	Yes	0.86	8.777	6.13	89670	0.95	0.98	0.8
12:35:00	38.6	0.0123	22.6	349.7	548.2	Yes	0.86	8.742	6.13	89715	0.95	0.98	0.8
12:40:00	38.6	0.0123	21.4	348.5	548.3	Yes	0.86	8.708	6.13	89715	0.95	0.98	0.8
12:45:00	38.7	0.0123	22.0	349.1	548.9	Yes	0.86	8.708	6.13	89715	0.95	0.98	0.8
12:50:00	38.7	0.0123	22.1	349.2	548.6	Yes	0.86	8.708	6.13	89715	0.95	0.98	0.8
12:55:00	38.8	0.0123	22.3	348.3	548.5	Yes	0.86	8.708	6.13	89715	0.95	0.98	0.8
13:00:00	38.8	0.0124	22.9	349.4	548.9	Yes	0.86	8.708	6.13	89355	0.95	0.98	0.8
13:05:00	38.9	0.0124	22.3	349.4	548.5	Yes	0.86	8.708	6.13	89580	0.95	0.98	0.8
13:10:00	38.9	0.0124	22.0	349.1	548.8	Yes	0.86	8.742	6.13	89580	0.95	0.98	0.8
13:15:00	38.9	0.0125	21.9	348.4	548.9	Yes	0.86	8.708	6.13	89580	0.95	0.98	0.8
13:20:00	38.9	0.0125	22.3	349.4	548.5	Yes	0.86	8.708	6.13	89580	0.95	0.98	0.8
13:25:00	39.0	0.0125	22.2	349.2	548.2	Yes	0.86	8.742	6.13	89745	0.95	0.98	0.8
13:30:00	39.0	0.0125	22.0	349.1	548.5	Yes	0.86	8.708	6.13	89745	0.95	0.98	0.8
13:35:00	39.0	0.0126	22.0	349.1	548.5	Yes	0.86	8.742	6.13	89520	0.95	0.98	0.8
13:40:00	38.9	0.0126	21.5	347.5	548.8	Yes	0.86	8.708	6.13	89520	0.95	0.98	0.8
13:45:00	38.9	0.0126	22.0	348.5	548.6	Yes	0.86	8.674	6.13	89520	0.95	0.98	0.8
13:50:00	38.9	0.0127	21.8	348.9	548.9	Yes	0.86	8.708	6.13	89520	0.95	0.98	0.8
13:55:00	38.9	0.0127	21.8	348.3	548.8	Yes	0.86	8.708	6.13	89520	0.95	0.98	0.8
14:00:00	38.9	0.0127	21.7	348.3	549.1	Yes	0.86	8.708	6.13	89520	0.95	0.98	0.8
14:05:00	38.8	0.0128	21.7	348.3	548.6	Yes	0.86	8.708	6.13	89130	0.95	0.98	0.8
14:10:00	38.8	0.0128	21.7	348.8	548.8	Yes	0.86	8.742	6.13	89700	0.95	0.98	0.8
14:15:00	38.7	0.0129	21.5	348.1	548.8	Yes	0.86	8.708	6.13	89550	0.95	0.98	0.8
14:20:00	38.7	0.0129	21.5	348.6	548.6	Yes	0.86	8.708	6.13	89280	0.95	0.98	0.8
14:25:00	38.6	0.0129	22.0	348.6	549.3	Yes	0.86	8.674	6.13	89280	0.95	0.98	0.8
14:30:00	38.6	0.0130	22.0	349.1	549.0	Yes	0.86	8.708	6.13	88980	0.95	0.98	0.8
14:35:00	38.5	0.0129	22.5	348.5	549.5	Yes	0.86	8.708	6.13	88965	0.95	0.98	0.8
14:40:00	38.4	0.0129	22.6	349.1	548.8	Yes	0.86	8.708	6.13	89340	0.95	0.98	0.8
14:45:00	38.3	0.0130	22.0	348.6	548.8	Yes	0.86	8.708	6.13	89340	0.95	0.98	0.8
14:50:00	38.2	0.0120	22.0	348.5	549.1	Yes	0.86	8.708	6.13	89190	0.95	0.98	0.8
14:55:00	38.1	0.0120	22.0	349.6	548.8	Yes	0.86	8.708	6.13	89190	0.95	0.98	0.8
15:00:00	38.0	0.0120	22.0	349.6	549.0	Yes	0.86	8.747	6.13	89190	0.95	0.98	0.8
15:05:00	37.9	0.0120	21.8	349.5	549.0	Yes	0.86	8.747	6.13	89325	0.95	0.98	0.8
15:10:00	37.8	0.0120	21.9	348.9	549.1	Yes	0.86	8.708	6.13	89325	0.95	0.98	0.8
15:15:00	37.7	0.0120	22.3	348.9	549.1	Yes	0.86	8.708	6.13	88905	0.95	0.98	0.8
15:20:00	37.6	0.0129	21.8	349.4	548.6	Yes	0.86	8.708	6.13	89085	0.95	0.98	0.8
15:25:00	37.4	0.0128	21.8	348.9	548.9	Yes	0.86	8.708	6.13	89490	0.95	0.98	0.8
15:30:00	37.3	0.0127	21.8	348.9	548.7	Yes	0.86	8.708	6.13	89265	0.95	0.98	0.8
15:35:00	37.1	0.0126	22.3	348.8	549.1	Yes	0.86	8.674	6.13	89265	0.95	0.98	0.8
15:40:00	37.0	0.0125	22.9	348.8	549.4	Yes	0.86	8.674	6.13	88905	0.95	0.98	0.8
15:45:00	36.8	0.0124	22.3	349.4	549.1	Yes	0.86	8.708	6.13	89415	0.95	0.98	0.8
15:50:00	36.7	0.0123	22.3	348.8	549.5	Yes	0.86	8.674	6.13	88890	0.95	0.98	0.8
15:55:00	36.5	0.0123	22.7	348.8	549.4	Yes	0.86	8.674	6.13	88890	0.95	0.98	0.8
16:00:00	36.3	0.0120	22.3	348.8	548.9	Yes	0.86	8.708	6.13	88890	0.95	0.98	0.8

Appendix B: Temperature functionality of thermodynamic variables

(T in Kelvin)

$$\theta = T/100$$

for air [10,14]

$$cp_{air} = (1.04841 - 3.8371\frac{\theta}{10^3} + 9.4537\frac{\theta^2}{10^5} - 5.4931\frac{\theta^3}{10^7} + 7.9218\frac{\theta^4}{10^{10}}\ (\text{kJ/kgK}) \tag{B1}$$

for other gases [12]

$$cp_{N2} = \left[39.06 - 512.79\theta^{-1.5} + 1072.7\theta^{-2} - 820.4\theta^{-3}\right]/28.013\ (\text{kJ/kgK}) \tag{B2}$$

$$cp_{O2} = \left[37.432\ +\ 0.0201020\theta^{1.5} - 178.57\theta^{-1.5} + 236.88\theta^{-2}\right]/31.999\ (\text{kJ/kgK}) \tag{B3}$$

$$cp_{CO2} = \left[-3.7357\ +\ 30.529\theta^{0.5} - 4.1034\theta + 0.024198\theta^2\right]/44.01\ (\text{kJ/kgK}) \tag{B4}$$

$$cp_{H2O} = \left[143.05 - 183.54\theta^{0.25} + 82.751\theta^{0.5} - 3.6989\theta\right]/18.015\ (\text{kJ/kgK}) \tag{B5}$$

$$cp_{CH4} = \left[-672.87\ +\ 439.74\theta^{0.25} - 24.875\theta^{0.75} + 323.88\theta^{0.5}\right]/16.04\ (\text{kJ/kgK}) \tag{B6}$$

$$cp_{C2H6} = \left[6.895\ +\ 17.26\theta - 0.6402\theta^2 + 0.00728\theta^3\right]/30.07\ (\text{kJ/kgK}) \tag{B7}$$

$$cp_{C3H8} = \left[-4.042 + 30.46\theta - 1.571\theta^2 + 0.03171\theta^3\right]/44.09\ (\text{kJ/kgK}) \tag{B8}$$

$$cp_{C4H10} = \left[3.954 + 37.12\theta - 1.833\theta^2 + 0.034980\theta^3\right]/58.124\ (\text{kJ/kgK}) \tag{B9}$$

Regression based on thermodynamic table [12],

$$h_{air} = 4.666416 + 0.968316\theta + 3.961\ \times\ 10^{-5}\theta^2 + 3.2648\ \times\ 10^{-8}\theta^3\ (\text{kJ/kg}) \tag{B10}$$

$$h_{N2} = \begin{cases} -40848385.458 + 136669978\theta^{1.0024978}/\left(4711.4676 + \theta^{1.0024978}\right) & \text{for}\ \ \theta\ \leq\ 4 \\ -8670 + 2923.5942\theta - 12.0449984\theta^2 + 2.1364334\theta^3 & \text{for}\ \ \theta < 10 \\ \exp\left(8.0789435 - \dfrac{4.8921249}{\theta} + 1.0352864\ln(\theta)\right) & \text{for}\ \ \theta \geq 10 \end{cases}\ (\text{kJ/kmol}) \tag{B11}$$

$$h_{O2} = \begin{cases} -8683.2986 + 2915.58557\theta - 11.791232\theta^2 + 3.69226661\theta^3 & \text{for}\ \ \theta < 4 \\ -8349 + 26672\theta + 43\theta^2 & \text{for}\ \ \theta\ \leq\ 4 \\ -7939.7229 + 2451.347\theta + 80.132035\theta^2 - 1.88383384\theta^3 & \text{for}\ \ \theta < 10 \\ -8879.333 + 2882.75507\theta + 31.329828\theta^2 - 0.282527785\theta^3 & \text{for}\ \ \theta \geq 10 \end{cases}\ (\text{kJ/kmol}) \tag{B12}$$

$$h_{CO2} = \begin{cases} -9363.977 + 2939.30760\theta - 83.011986\theta^2 + 50.5774680\theta^3 & \text{for} \quad \theta < 4 \\ -9525 + 2646\theta + 184\theta^2 & \text{for} \quad \theta \leq 4 \\ -9584.9286 + 2538.8571\theta + 240.08333\theta^2 - 6.416667\theta^3 & \text{for} \quad \theta < 10 \\ \exp\left(8.6546661 - \dfrac{5.9395707}{\theta} + 1.0233514\ln(\theta)\right) & \text{for} \quad \theta \geq 10 \end{cases} \text{(kJ/kmol)} \tag{B13}$$

$$h_{H_2O} = \begin{cases} -9903.9842 + 3250.3251\theta + 42.806253\theta^2 - 6.21838320\theta^3 & \text{for} \quad \theta < 4 \\ -9598 + 3094\theta + 42\theta^2 & \text{for} \quad \theta \leq 4 \\ -9494.7937 + 3083.6892\theta + 31.208332\theta^2 + 1.95370380\theta^3 & \\ -0.041666670\theta^4 & \text{for} \quad \theta < 10 \\ -10205.43 + 2943.4614\theta + 73.011089\theta^2 - 0.639090980\theta^3 & \text{for} \quad \theta \geq 10 \end{cases} \text{(kJ/kmol)} \tag{B14}$$