Energy and Exergy Analysis of Rumaila-Basra Gas Turbine Power Plant During Hot Season

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Abstract- **In this paper, energy and exergy concepts have been carried out on one of the largest gas turbine power plants in Iraq (Rumaila-Basra). Both ISO operating conditions as well as actual operating data recorded for one month in hot season are considered. Results indicate that a lot of heat energy accompanied with remarkable exergy is discharged to the atmosphere. Also, it is found that the combustion chamber has the largest exergy destruction among the plant components. Possibility of cooling the intake air drawn by the compressor and its effects on the plant performance is studied. The required cooling load is found to be in the range 3379 T.R for part load operation to 4723.3 T.R for full load operation.**

Keywords: Power Plant, Gas Turbine, Energy and Exergy Analysis, EES program.Index Terms— Chaos ;Crow Search Algorithm;Chaotic Crow Search Algorithm; Meta-heuristic algorithms.

I. Introduction

Iraq in the past decade has changed dramatically. There was a remarkable rise in the standard of living of the people. Before 2003several homes had air conditioning systems, and nowadays many people starts using this huge power consuming devices. Besides that, there was increase in population as well as the growth of oil industrial plants; all of these factors make the demand for electricity growing at a rapid rate [1]. In this scenario, the using of gas turbine power plants become very attractive world wide because they are light in weight, compact, simple operation, and has a quick starting time. Moreover, the quick installation time is very important factor in Iraq. The gas turbine industry offers a wide range of capacity from micro size to approximately 500 MW. The commonly used fuels to operate gas turbine plants are natural gas, diesel or sometimes heavy fuel oil [2].

The analysis of gas turbine power plant is very essential for the efficient utilization of fuel resources. Although the first law of thermodynamics is the main tool to analysis and assess the overall performance of any system, it cannot show whether the energy type is utilized in the right way or not. Exergy is defined as the maximum useful work that can be obtained when the system interacts and brings to be in equilibrium with the dead state. Unlike energy, exergy is not conserved but generally destroyed in the system due to several reasons. Exergy destruction or irreversibility shows how much useful work is lost in the process [3].

As gas turbine is widely used in the world for electric power production, it has been studied and analysis in a number of research projects purposely to improve the performance. Al-Gburi[4] performed exergy analysis for Al Najaf gas turbine power plant located at Iraq which has a capacity of 115.74MW. According to his results, the combustion chamber is found to be the chief means of irreversibility in the plant. Siddiqui et al. [5] carried out simulation study of 100 MW gas turbine power plant located in Iran. They showed that the firing temperature represents the crucial parameters that affect the exergetic efficiency and exergy destruction of the plant. Igbong and Fakorede [6] performed exergoeconomic analysis on a 100 MW gas turbine power plant at Ughelli located in Nigeria. A gas turbine based power plant located at Tripoli was studied by Fellah et al. [7]. The method used in this study is exergy costing approach.

Rumaila-Basra gas turbine power plant is located in Rumaila area at the north of Basra city, about 44 km northeastern of the city center. The plant has five identical gas turbine generating units (**SGT5–PAC 4000F**) each with a nominal rating of 283.6 MW under ISO conditions. Total installed generating capacity of about 1418 MW. The plant was commissioned in 2013 by Siemens Company. It can use natural gas or liquid fuel in the combustion process. The plant currently receives natural gas fuel from Rumaila oil field located on the Basra production field for which the properties is given in Table 1. The standard model specifications under ISO operating conditions are given in Table 2 [8]. The gas turbine unit is a single shaft arrangement. The 13-stage, axial compressors are supplied with single variable-pitch inlet guide vane row that control the air mass flow rate drawn by the unit. Some of the compressed air is extracted for turbine blade cooling and other usage. The combustion chamber is of ring type and it equipped with 24 burners. The turbine consists of four stages and the unit operates at 3000 r.p.m [9]. Several instrumentations are made in Rumaila-Basra gas turbine power plant. For example, pressure transducers are used to measure the compressor pressure ratio. Also, to measure the temperature of the air flow at the compressor inlet, three dual element resistance thermometers are used. Besides, the exhaust temperature is measured downstream of the turbine using 24 triple element thermocouples that are distributed around the circumference of the exhaust diffuser [9].

The analysis presented in this paper will be applied to single unit in this plant to examine most of the performance characteristics during typical hot season. In addition, the improvement taken place if the inlet air is cooled is to be predicted. The Typical experimental data is selected for July-2015. The plant data are recorded three times a day 6:00AM, 2:00PM, and 10:00PM [8]. The performance characteristics of the plant based on energy and exergy principle will be investigated according to this data.

Table 1. Properties of fuel gas used in Rumaila-Basrah gas turbine power plant [8].

Fuel type	Fuel gas	
Composition % by volume	CH_4	79.19
	C_2H_6	17.3
	C_3H_8	1.04
	nC_4H_{10}	0.02
	iC_4H_{10}	0.01
	nC_5H_{12}	0.03
	iC_5H_{12}	0.02
	CO2	2.4
Density	0.6 kg/m3	
LHV	48.4 MJ/kg	
HHV	53.5 MJ/kg	

*for ISO conditions only 100% CH_4 is considered.

Table 2. Gas Turbine Model Specification Siemens (SGT5–PAC 4000F) (Standard and Predicted by the present thermodynamic model)

	Standard Value [9]	Predicted by the present model	%Error
Net power output, MW	283.6	284.1	0.18
Compression ratio	17	17	-
Air mass flow rate, kg/s	670	664	0.9
Fuel mass flow rate, kg/s	14.96	13.45	10.1
Temperature of Exhaust gas,℃	572	567.5	0.79
Firing Temperature, ℃	No records	1254	-
Thermal Efficiency,%	40	42.18	5.45

II. Thermodynamic model

A schematic of a gas turbine power plant cycle is shown in Fig. 1with its representation on T-S diagram. The ambient air at point 1 is drawn into the compressor at point 2 after passing through an air intake. Normally, a percentage portion of the compressed air is extracted at different stages of compressor. This air is used for cooling the turbine rotor and stator blades and any other application needed in the plant. Besides, these extracted air portions plays important role in the control of the plant to ensure safe operation. Now, the compressed air moves to the combustion chamber and reacts with the fuel. The resulted combustion gases stream from the combustion chamber is fed into the turbine to generate power. The modeling of the plant components is based on energy and exergy principles. The following assumptions are assumed in the present analyses:

- The operation is steady state for all plant components.
- In the turbine, compressor, and combustion chamber the processes are adiabatic,
- The potential and kinetic terms concerning energy and exergy terms are neglected.
- Air and combustion gases are modeled as a mixture of ideal gases with variable properties.

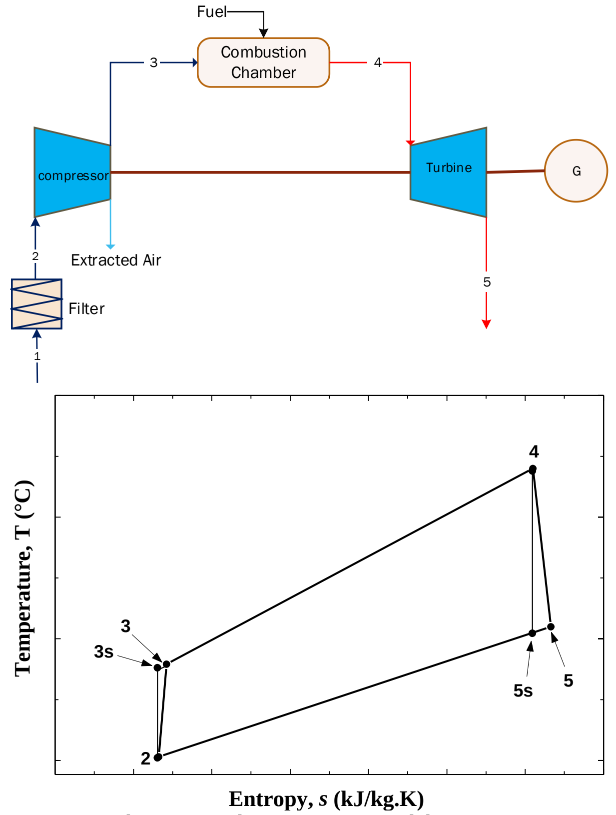

Fig. 1. Gas turbine power plant components and their representation on T-S diagram.

2.1First law analysis (energy analysis)

The general formulation of the first law of thermodynamic applied to any steady flow open system is given by [3]:

$$\sum \dot{Q}_k + \dot{m}_i \left(h_i + \frac{c_i^2}{2} + gZ_i\right) = \dot{m}_e \left(h_e + \frac{c_e^2}{2} + gZ_e\right) + \dot{W} \quad (1)$$

The term $\left(h + \frac{c^2}{2} + gZ\right)$represents the total energy of the flowing stream.

2.1.1Energy balance of the air compressor

In the air compressor, air enters at state P_2,T_2 and leaves at state P_3, T_3. The inlet air is usually drawn to the plant through a duct supplied with filters to remove dust. The inlet pressure to the compressor is given by []:

$$P_2 = P_1 - \Delta P_c \quad (2)$$

In this paper, air is assumed to be a mixture of ideal, non-reactive gases consisting of O_2 and N_2. The mole fraction of each constituent is 0.21for O_2 and 0.79 for N_2. The enthalpy of inlet air is given by [3]:

$$h_{a,2} = \sum_i Y_i \cdot h_i \quad (3)$$

Where Y_i is the mass fraction of the i^th^ component that can be derived directly from the kmole fraction.
Consequently, the entropy of inlet air is given by [3]:

$$s_{a,2} = \sum_i Y_i \cdot s_i \quad (4)$$

After calculating the inlet properties to the compressor, the exit properties must be estimated. First of all, the compressor pressure ratio is defined as $R_p = {}^{P_3}/_{P_2}$. In case of isentropic compression, then the entropy at point 2 is equal to the entropy at point 3. In this case, using this property along with exit pressure P_3, the other isentropic exit properties can be calculated. The actual exit enthalpy is given by:

$$h_{a,3} = h_{a,2} + \left(\frac{h_{a,3,is} - h_{a,2}}{\eta_{c,is}}\right) \quad (5)$$

Again, this property together with the exit pressure P_3 is used to evaluate the other actual properties at point 3. Applying energy balance for the air compressor will result:

$$\dot{W_c} = \frac{\dot{m}_a\left(h_{a,3} - h_{a,2}\right)}{\eta_{c.m}} \quad (6)$$

2.1.2Energy balance of combustion chamber

In the combustion chamber, fuel at T_f, P_f is burned with air at state 3. The air used is always more than that needed to complete the stoichiometric reaction. Since the combustion process is assumed to be adiabatic, then the products of combustion will be at high temperature. Then this high pressure and temperature combustion products are responsible to drive the turbine. As mentioned earlier, in gas turbine power plant, some of compressed air is extracted from the compressor for the sake of turbine blade cooling and other application. By assuming that the percentage of the extracted air to be 8%, then the air supplied to the combustion chamber is reduced to 92%.

The stoichiometry reaction deals with reactants and products within a given chemical reaction when there is no excess air supplied. Usually more air is supplied to any combustion reaction to ensure complete reaction. In gas turbine combustion chamber, this situation is important to decrease the combustion products temperature which is restricted to specified limit. The chemical reaction equation represents stoichiometeric reaction for a given fuel is [3]:

$$C_xH_y + a_{stoi}\left[x_{O_2}O_2 + x_{N_2}N_2\right] \longrightarrow b_{stoi}CO_2 + c_{stoi}H_2O + d_{stoi}N_2 \quad (7)$$

Two important calculations are gained from the above equation. The first one is calculating the stoichiometeric fuel to air ratio F_{stoi}. This term is defined as the mass of fuel consumed divided by the mass of air needed just to complete the combustion process given that there is no extra oxygen found in the products of combustion. For the previous stoichiometric reaction it is given by [3]:

$$F_{stoi} = \frac{(M_f)}{(a_{stoi}\ M_a)} \quad (8)$$

Where the stoichiometric kmoles of air a_s is found by usual atomic balance which results:

The other important calculation is evaluation the enthalpy of stoichiometric products of combustions. Actually, supplying extra air for the combustion process will not affect the amount of stoichiometric products resulted per kg of fuel supplied. The enthalpy of stoichiometric product gases h_{sp} can be calculated using Eq. (3). The mass fraction of ith component Y_i is also found directly from the kmole fraction.

The molecular weight of the stoichiometric product mixture is given by:

$$M_{stoi} = \sum x_i M_i \quad (9)$$

As mentioned previously, in the actual reaction the amount of air entered the reaction is always more than that needed in the stoichiometric reaction. Usually, the amount of excess air is linked with that required for stoichiometeric combustion. The equivalence ratio ϕ represents the ratio between the actual fuel to air ratio F_{act} to the stoichiometeric one i.e [10]:

$$F_{act} = \phi \cdot F_{stoi} \quad (10)$$

Once the fuel analysis is known and equivalence ratio is calculated, then the fuel mass flow rate required to attain a specified firing temperature.

The following is the actual combustion reaction equation written for 1 kg or entered air, where there is some excess air entered the reaction [3]:

$$F_{act}\ kg\ of\ Fuel + 1\ kg\ of\ ai\ r \rightarrow (F_{act} + \phi)stoi.products + (1 - \phi)Excess\ air \quad (11)$$

The value of equivalence ratio ϕ is calculated based on the total enthalpy concept. Appling this principle gives the following relation for the equivalence ratio ϕ[10]:

$$\phi = \frac{h_{a,4} - h_{a,3}}{(h_{a,4} - h_{a@15}) - \left[(1 + F_{stoi})(h_{stoi,4} - h_{stoi@15})\right].. \\ .. + \left[(\eta_{cc} \cdot F_{stoi})\left(LHV + h_{f@T_f} - h_{f@15}\right)\right]} \quad (12)$$

This equation is very important to estimate the ratio of fuel to air required for attaining the required firing temperature.

By knowing the fuel gas lower heat value LHV, and fuel mass flow rate $\dot{m}_f$, the heat input to the plant is defined as:

$$\dot{Q}_{in} = \dot{m}_f\left(LHV + h_{f@T_f} - h_{f@15}\right) \quad (13)$$

2.1.3Energy balance of turbine:

The flue gases leaving the combustion chamber enter the turbine at a high firing temperature. The high pressure and temperature gases expand in the turbine and thus developing useful work which converted to electric power in the generator [3]. Assuming a pre-defined pressure drop for combustor ΔP_{cc} and exhaust gases duct ΔP_{exh}, then the pressure at the inlet and outlet of turbine can be calculated as [2]:

$$P_4 = P_3 - \Delta P_{cc} \quad (14)$$

$$P_5 = P_1 + \Delta P_{exh} \quad (15)$$

In case of isentropic expansion, then the entropy of combustion products at turbine inlet is equal to the entropy at turbine exit i.e $s_{p,5,is} = s_{p,4}$

By using this property along with exit pressure P_5, the other isentropic exit properties can be calculated. The actual exit enthalpy is given by [3]:

$$h_{p,5} = h_{p,4} - \eta_{t,is}\left(h_{p,4} - h_{p,5,is}\right) \quad (16)$$

Following the same manner done in compressor analysis, this property together with the exit pressure is used to evaluate the other isentropic actual properties. The output power from the turbine is calculated from energy balance across the gas turbine control volume and it is given by:

$$\dot{W}_t = \frac{\left(0.92\,\dot{m}_a + \dot{m}_f\right)\left(h_{p,4} - h_{p,5}\right)}{\eta_{m,t}} \quad (17)$$

2.2 Second law analysis (exergy analysis)

In order to get a deeper understand for the thermal performance of Rumaila-Basrah gas turbine power plant, an exergy analysis based on second law of thermodynamic has been carried out. In general, for steady state flow open system, the exergy balance equation is given as [3]:

$$\Psi_w = \sum_k \left(1 - \frac{T_o}{T_k}\right) Q_k + \sum\left[(\dot{m}\psi)_{in} - (\dot{m}\psi)_e\right] - T_o S_{gen} \quad (18)$$

Where Ψ_w is the useful work done on/or by the system, the first term on the right hand side is the exergy due to heat transfer, while the term $\dot{m}\psi$ represents the flow exergy of the working fluid. The third term $T_o S_{gen}$represents the irreversibility or the exergy distraction in the system. The flow exergy is defined as [3]:

$$\psi = \dot{m}\left[(h - h_o) - T_o (s - s_o)\right] \quad (19)$$

Where s_o and h_o represent the entropy and enthalpy at the reference dead state.

2.2.1 Exergy balance of air compressor

Applying Eq.(18), the exergy balance for the air compressor is given by:

$$T_o S_{gen,c} = \dot{W}_c + \dot{m}_a\left(\psi_{a,2} - \psi_{a,3}\right) \quad (20)$$

The irreversibility occurring in the compressor is:

$$I_c = T_o S_{gen,c} = \dot{m}_a T_o\left(s_{a,3} - s_{a,2}\right) \quad (21)$$

The second law efficiency of the compressor is defined as the ratio of minimum work that is needed for the compressor which is represented by the reversible work to the actual work, i.e.:

$$\eta_{II,c} = \frac{\dot{m}_a\left(\psi_{a,3} - \psi_{a,2}\right)}{\dot{W}_c} = 1 - \frac{I_c}{\dot{W}_c} \quad (22)$$

2.2.2 Exergy balance of combustion chamber

The exergy balance of the combustion chamber is:

$$T_o S_{gen,cc} = 0.92\dot{m}_a\psi_a + \dot{m}_f\psi_f - \left(0.92\dot{m}_a + \dot{m}_f\right)\psi_p \quad (23)$$

The fuel exergy is consist of two parts. The first one is the physical exergy, which is due to its own enthalpy. The second part, which is more imperative, is the chemical exergy which is due to its chemical energy. Several methods are available in the literature to determine the chemical exergy of fuel. These methods give close results for fuel gas. In this study, the chemical exergy is approximated by the lower heating value LHV multiplied by a factor of 1.04[12]. In this case the total fuel exergy is given by:

$$ex_f = \dot{m}_f\left[1.04LHV + \left(\psi_{f@T_f} - \psi_{f@15}\right)\right] \quad (24)$$

The second law efficiency of the combustion chamber is defined as the ratio of the output exergy in terms of product gases to the input exergy in terms of fuel exergy and hot air exergy supplied by the compressor, [3]:

$$\eta_{II,cc} = \frac{\left(0.92\dot{m}_a + \dot{m}_f\right)\psi_p}{0.92\dot{m}_a\psi_a + ex_{fuel}}$$

$$= 1 - \frac{T_o S_{gen}}{0.92\dot{m}_a\psi_{a,2} + ex_{fuel}} \quad (25)$$

2.2.3Exergy balance of turbine

The exergy balance for the gas turbine is give by:

$$T_o S_{gen,t} = \left(0.92\dot{m}_a + \dot{m}_f\right)\left(\psi_{p,4} - \psi_{p,5}\right) - \dot{W}_t \quad (26)$$

The irreversibility is given by:

$$I_t = T_o S_{gen,t} = T_o\left(0.92\dot{m}_a + \dot{m}_f\right)\left(s_{p,4} - s_{p,5}\right) \quad (27)$$

The second law efficiency of the turbine is defined as the ratio of the useful work output to the maximum work that can be obtained from the turbine which is represented by the reversible work, i.e.:

$$\eta_{II,t} = \frac{\dot{W}_t}{\left(0.92\dot{m}_a + \dot{m}_f\right)\left(\psi_{p,4} - \psi_{p,5}\right)}$$

$$= 1 - \frac{I_t}{\left(0.92\dot{m}_a + \dot{m}_f\right)\left(\psi_{p,4} - \psi_{p,5}\right)} \quad (28)$$

2.3 Overall plant performance characteristics:

The net output power from the plant is calculated as:

$$\dot{W}_{net} = \eta_{gen}\left(\dot{W}_t - \dot{W}_c\right) \quad (29)$$

Thermal efficiency of a gas turbine power plant is the ratio of the net output work to the input heat i.e.:

$$\eta_{th} = \frac{\dot{W}_{net}}{Q_{in}} \quad (30)$$

The second law exergy efficiency for the plant can be written as:

$$\eta_{II,plant} = \frac{\dot{W}_{net}}{ex_f} \quad (31)$$

This definition measures the fraction of input exergy converted to useful work.

The exhaust heat rejected to the environment is:

$$Q_{out} = Q_{in} - \dot{W}_{net} \quad (32)$$

Similarly, the input exergy is represented by the fuel chemical exergy plus physical exergy. The difference between the input exergy and output useful work equals to the rate of exergy destruction inherent in the plant component plus exergy lost from the plant due to ,mainly the exhaust gases, [13]:

$$ex_{lost} = ex_f - W_{net} - I_c - I_{cc} - I_t \quad (33)$$

III. Results and discussion

All the equations stated in this study were arranged in a computer program with “Engineering Equation Solver (EES)”. Before introducing the results obtained, it must be stated that in the plant data gained no records are found for air mass flow rate and firing temperature. However, other useful data is recorded. In view of that, two main iterations are made in the computer program, one for the air mass flow rate and the other for firing temperature. The net power output from the plant is used to determine the air mass flow rate and the exhaust temperature is used to estimate the firing temperature. Any assumption and reference data taken are given in Table 3.

Table 3. Assumptions and Reference Data

ISO Conditions	15 ℃, 101.325 kPa
Fuel Temperature and pressure	15 ℃, 2500kPa
$\eta_{c,m} = \eta_{t,m} = \eta_{gen}$	0.95
$\eta_{c,is}$	0.87
$\eta_{t,is}$	0.89
η_{cc}	0.98
$\Delta P_c = \Delta P_t$	10 kPa
ΔP_{cc}	0.04* P_{a3}
Turbine Cooling Air Fraction	8%

Table 2 shows the measured and predicted values of performance characteristics for Rumaila-Basra GT power plant based on ISO operating conditions. All the results show good agreement except for that of fuel mass flow rate which has the maximum error obtained of 10%. This error percentage is acceptable given that the fuel mass flow rate is too low compared to that of air.

Figures 2 and 3 show the energy and exergy distribution among the plant equipments. These distributions are restricted to the full load ISO operation conditions. The fuel input represents the energy and exergy sources supplied to the plant. The power output accounts42.2% from the input energy and 36.6% from the input exergy. These two percentages are actually the definition of first and second law efficiencies of the plant. The plant discharge 57.8% of the fuel energy to the environment as exhaust gases. The maximum exergy destroyed (irreversibility) occurs in the combustion chamber which accounts 39.4%. That is mean, if there is any opportunity to develop the plant performance, then this will certainly be in this part of the unit. This is due to the high temperature in this component and no work interaction is found. This fact explains the great interest directed to the fuel cell technology nowadays. Lower values of exergy destroyed are occurs in compressor and turbine. These values accounts about 2% from the total input exergy. This is due to no heat interaction between these components and environment. The losses occurred are merely internal due to friction. The exergy lost to the environment accounts 20.1%.

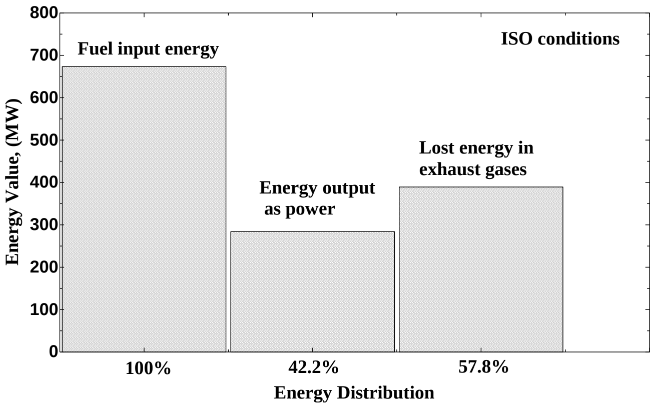

Fig. 2. Energy (MW) Distribution in Rumaila-Basrah gas turbine power plant components operating at ISO conditions.

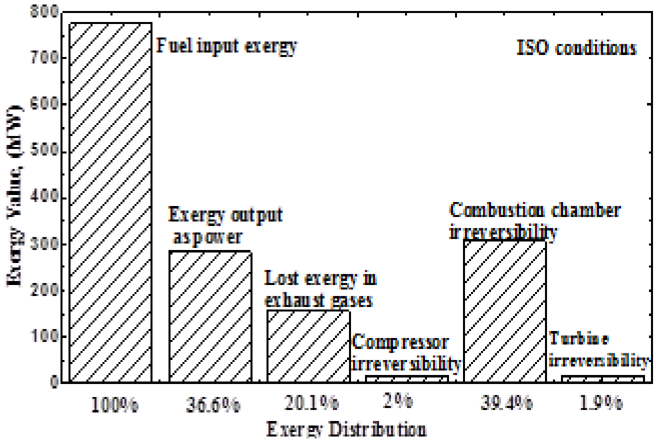

Fig. 3 Exergy (MW) Distribution in Rumaila-Basrah gas turbine power plant components operating at ISO conditions.

This exergy is available at high temperature and thereby it represents good opportunity to develop the plant performance. Commonly, Energy flow throughout any thermal system is represented by Sankey diagram and exergy flow is represented by Grassman diagram. These two diagrams are both drawn for Basrah-Rumaila gas turbine power plant and are shown in Figures 4 and 5 respectively.

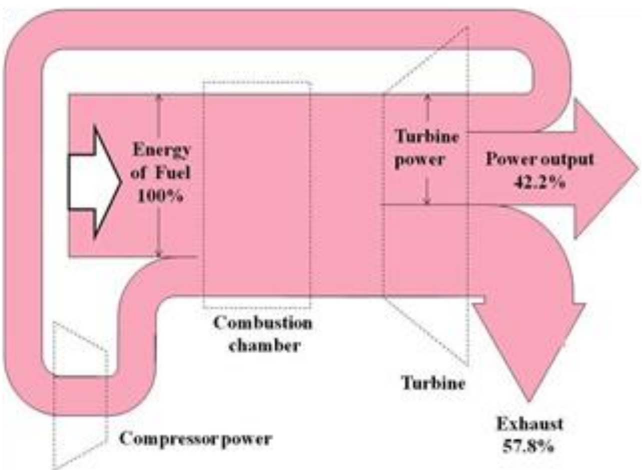

Fig. 4 Sankey diagram showing the energy flow in Rumaila-Basrah gas turbine power plant.

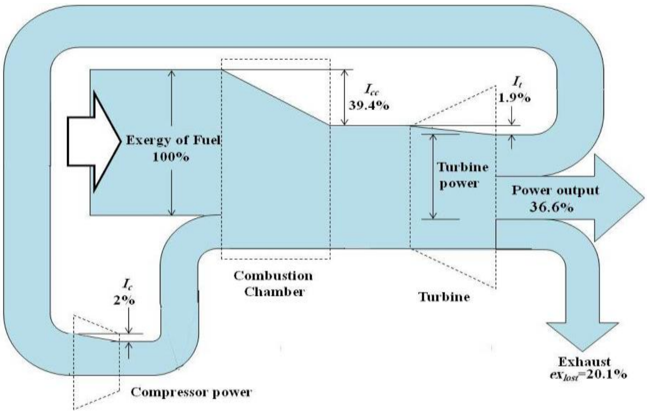

Fig. 5 Grassmann diagram showing the exergy flow in Rumaila-Basrah gas turbine power plant.

All the above discussion is restricted to the ISO operating conditions. As a matter of fact, the plant normally works at part load and many parameters are not at their standard values. Fig. 6 shows the actual variation of output power and compressor pressure ratio with time. The figure reveals that the power variation follows the same style as that for compressor pressure ratio. This manner is attributed to the effect of air mass flow rate drawn by the unit. So, decreasing air mass flow rate will directly affect the power output, as it directly proportional to it. In the other hand, decreasing air mass flow rate passing through the compressor at constant rotational speed will certainly decrease the pressure ratio according to the axial compressor characteristics map. Actually, this behavior cannot be shown in the usual parametric studies since it is based constant operational parameters like pressure ratio.

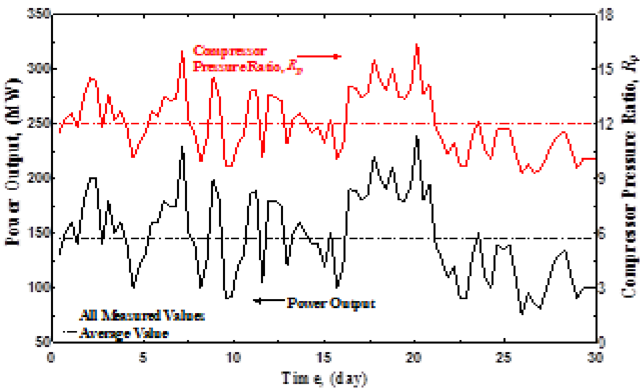

Fig. 6 Variation of power output and compressor pressure ratio of Rumaila -Basrah gas turbine power plant during July 2015.

Fig. 7 shows the variation of temperature at inlet and outlet of the unit equipments. At first, compressor inlet temperature suffers a wide range of variation away from the standard ISO conditions. The minimum recorded environmental temperature is 30ºC which occurs on the 9th day at 6:00AM while the maximum one is 50.5 ºC that occur on 29th day at 2:00PM and the mean value is 39.9ºC. This variation along with the load demand on the plant (characterized with the IGV position) is responsible for the whole performance specifications described in this section. The turbine exhaust temperature is almost constant at 569ºC as this is the main control system task. This target is reached by controlling the fuel mass flow rate which depends on the air mass flow rate drawn to the unit. The later affects the pressure ratio, and then this situation will certainly vary the firing temperature so that it will always be in the safe side.

The firing temperature is found to be in the range 1031-1239 ºC with average value of 1151 ºC. As explained previously, the main cause of this variation is the pressure ratio at part load operation condition. So, as the pressure ratio increases, due to higher air mass flow rate (in response to higher power demand), then the firing temperature must increase so that the exhaust temperature be constant. In the other hand, when the pressure ratio decreases due to decreasing the air mass flow rate (in response to lower power demand), then the firing temperature must decrease so that the exhaust temperature will be the same. This situation also affects compressor exit temperature at the same manner. By overlaying the temperature and entropy calculated for the Basra gas turbine power plant during July 2015.

Plant in a T-S diagram, the cycle is shown in Fig. 8 for both ISO conditions and typical operating data in the 15th day at 2:00PM. It is clear from the figure that compressor pressure ratio decreases at part load which, for keep the constant exhaust temperature, the firing temperature must be decreased.

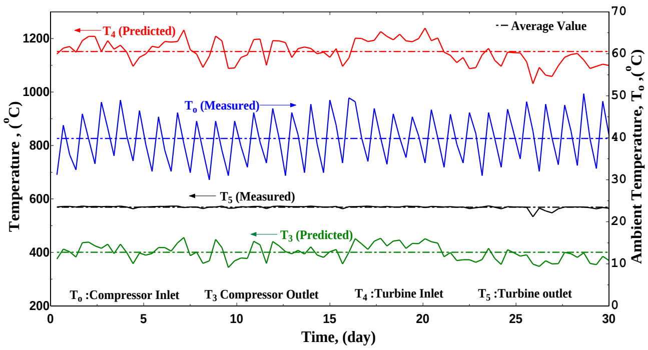

Fig. 7. Variation of operating temperatures of Rumaila

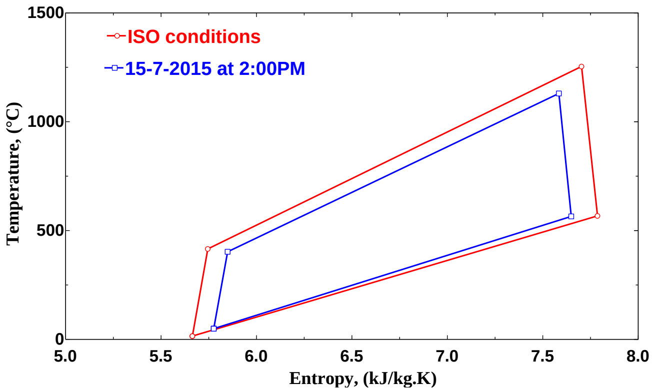

Fig. 8. T-S diagrams for Rumaila-Basrah gas turbine power plant one during 15-7-2015 and one for ISO operating conditions.

Variation of power input and output from the plant is shown in Fig. 9. As seen from figure, all the curves follow the manner of variation with time. That means that there is main factor which affects these parameters. Again, this parameter is the air mass flow rate. As illustrated before, this parameter is used to control the power output from the plant since power is directly proportional to it. Accordingly, fuel mass flow rate will increase nearly at the same ratio in order to keep the exhaust temperature constant. That is why the heat input to the cycle change at the same manner as power output is.

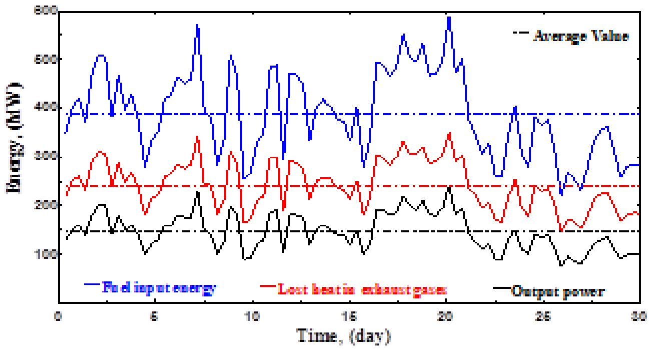

Fig. 9. Varation of energy distribution for Rumaila-Basrah gas turbine power plant operating during July 2015.

In light of the last discussion, Fig. 10 shows how exergy input, output, and destroyed are varied with time. The same discussion stated for power distribution is valid for exergy distribution. However, the ambient temperature plays important role in the way the exergy destroyed is varied. The percentage of the average value of each item is close to that calculated at the ISO conditions. The average output power accounts 32% from the average input exergy. The average lost exergy to the environment amounts 19% . Exergy destroyed for the combustion chamber amounts 45% and that for the turbine and compressor each of 2%.

Fig. 11 shows the variation of various efficiency definitions. The figure reveals the style of variation just like the past illustrated ones. That's mainly due to changing air mass flow rate accompanied with changing pressure ratio discussed before. The average value of thermal efficiency is 37.3%. This indicates a penalty of 13% decreasing from that for ISO conditions.

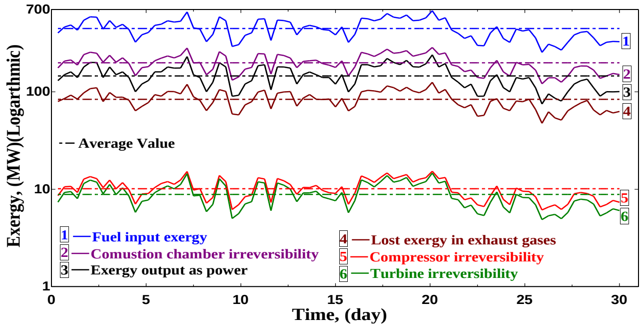

Fig. 10. Variation of exergy distribution for Rumaila-Basrah gas turbine power plant operating during July 2015

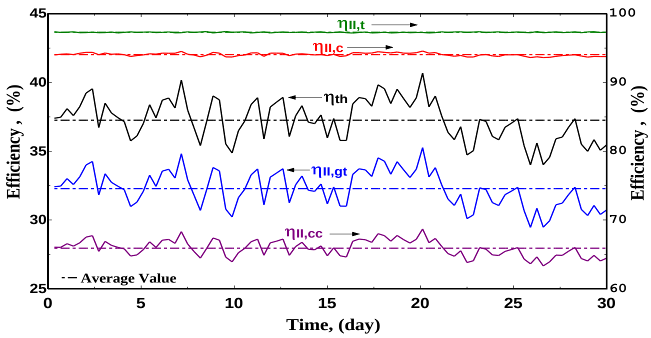

Fig. 11. Variation of various second law efficiencies for Rumaila-Basrah gas turbine power plant operating during July 2015

IV. Enhancement opportunity of Rumaila-Basra gas turbine power plant:

The last discussion reveals that the air mass flow rate is very important parameter that governs the whole gas turbine unit performance. Essentially, there are two factors influencing the mass flow rate drawn by the unit. The first one is the air density that depends mainly on air temperature (given that the atmospheric pressure is constant). The other factor is the value of volume flow rate which is controlled by the IGV as a response for power demand. That is mean at a specified operating point recorded by the plant (characterized by IGV position), the intake air mass flow rate is certainly lower than that of the ISO conditions. As a result, two options for enhancement the plant by inlet air cooling could be studied:

1-Enhance the intake air mass flow rate back to the ISO operating conditions at full load (100% opened IGV).

2-Enhance the intake air mass flow rate back to ISO operating temperature 15 ºC while keeping the load demand constant (partially opened IGV).

Generally, the improvement in mass flow rate caused by recovering intake temperature bake to 15 ºC (doesn't matter full or part load) is given by:

$$(\dot{m}_a)_{after\ cooling} = (\dot{m}_a)_{before\ cooling}\frac{\rho_{a,@15}}{\rho_{a,@T_o}} \qquad (\dots 46)$$

The first studying strategy is impractical; since the machine is always operating at part load and hence the second one is more realistic.

Fig. 12 represents the prediction of cooling load required for the plant. First of all, it must be stated that the recorded values for relative humidity in July-2015 are from 1.2% to 46% , which are all lower than ISO conditions of 60%. This indicates that there is no latent cooling load needed. Instead, this indicates the possibility of adopting evaporative cooler in the plant region. The figure shows only the sensible cooling load needed to bring the compressor inlet temperature back to ISO temperature of 15 ºC. Two cases are studied, which are the full and part load condition as explained in the previous sections. Results show that the average cooling load needed is 4723.3 T.R for full load operation and 3379 T.R for part load operation. After estimation the required cooling load, the improved air mass flow rate at part load condition is re-entered to the simulation program to appreciate the gain from this improvement. In case of improvement to full load, the plant will be simply restored to its ISO conditions.

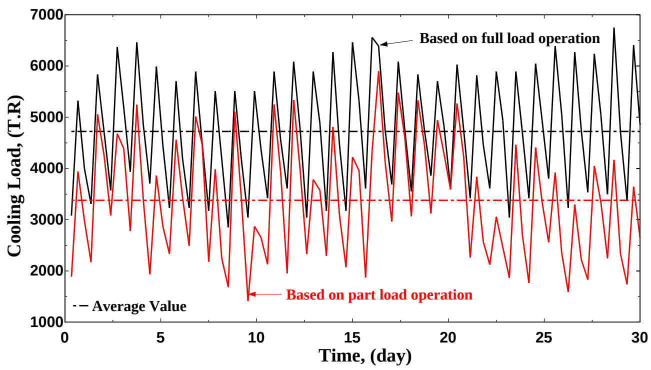

Fig. 12. Variation of cooling load required for Rumaila-Basrah gas turbine power plant operating during July 2015.

Fig. 13 shows the variation of output power before and after applying the intake air cooling. Results indicates that the percentage increase in power output in case of part load operation varies according to the load demand (IGV position) and ambient condition. The percentage improvement range is from 11.3% to 28.9% in case of part load operation and 18-279% for full load condition. As stated before, this case is impractical; since the plant is always a part load operated.

Without doubt, the intake cooling technique also affects thermal and second law efficiencies of the plant. This is shown in Fig. 14. Results indicate that the thermal efficiency improvements ranged from 2.1% to 5.2% at part-load operation. The second law efficiency has the same improvement range.

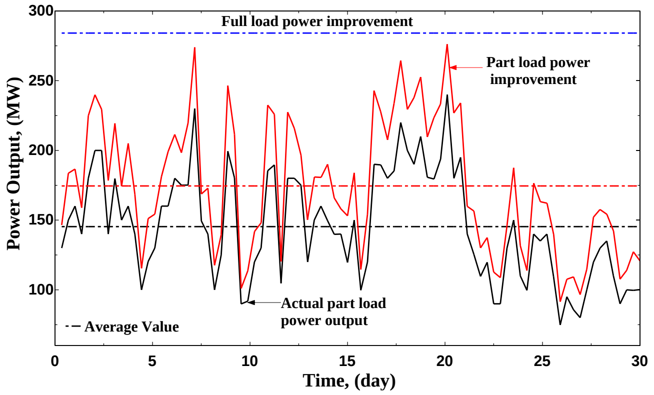

Fig. 13. Prediction the improvement in power output of Rumaila-Basrah gas turbine power plant when cooling inlet air is applied.

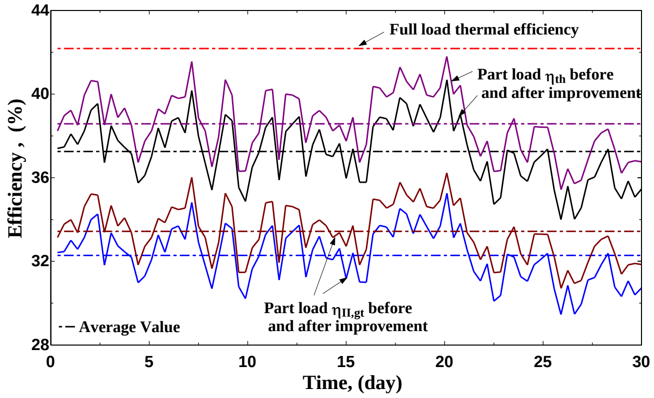

Fig. 14. Prediction of the improvement in efficiency of Rumaila-Basrah gas turbine power plant when cooling inlet air is applied.

V. Conclusions:

Carrying out of an energy and exergy balance of Basra-Rumaila gas turbine power plant enables to quantify the energy and exergy losses in the plant. Furthermore, the analysis helps to predict the improvement that could be achieved when inlet air cooling technique is adopted. Results show that the required cooling load capacity ranged from 3379 to 4723.3 TR for part load and full load operation respectively. The largest exergy plenty for Rumaila-Basra GTPP is found in the combustion chamber (as irreversibility) followed by the exergy lost to the environment which is accompanied to the exhaust gas. The irreversibility in both the compressor and turbine are small and close to each other.

VI. Acknowledgments:

The authors would like to acknowledge the staff of Rumaila-Basra gas turbine power plant for their support in providing the actual operating data used in this study.

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VIII. Nomenclature

English symbols

$a, b, c,$ and d	Number of kmoles (kmole).
ex	Exergy term (kW).
F	Fuel to air ratio (kg_f/kg_a)
g	Gravitational acceleration (m/s^2).
h	Enthalpy (kJ/kg).
HR	Heat Rate (kJ/kWh)
I	Exergy destruction (kW)..
HHV	Higher heating value (kJ/kg_f).
LHV	Lower heating value (kJ/kg_f)..
M	Molecular weight (kg/kmole).
$\dot{m}$	Mass flow rate (kg/s).
P	Pressure (kPa).
Q	Heat power (kW).
Rp	Pressure ratio (-).
s	Entropy (kJ/kg.K)..
S_{gen}	Entropy generation (kJ/kg.K).

T	Temperature (K).
$\dot{W}$	Power (kW).
x	Mole Fraction (-).
Y	Mass Fraction (-).

Greek symbols

η	Efficiency (-).
Δ	Difference (-).
ϕ	Equivalence ratio (-).
ψ	Flow physical exergy(kJ/kg).
Ψ_w	Exergy due to work (kW).

Subscripts

$1,2 \ldots 5$	Points at gas turbine cycle components.
II	Second law.
a	Air.
act	Actual.
exh	Exhaust.
f	Fuel.
c	Compressor.
cc	Combustion chamber.
gen	Generator.
i	Denotes to the *i*'th component.
in, e	Inlet and exit.
is	Isentropic.
m	Mechanical
net	Net value.
o	Denotes to the dead state.
p	Products of combustion.
$stoi$	Stoichiometric.
sp	Stoichiometric products.
t	Turbine.
th	Thermal.
x, y	Number of carbon and hydrogen in fuel.

Abbreviations:

EES	Engineering equation solver.
IGV	Inlet guide vans
ISO	International standards organization.