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Study And Analysis Of Thermal Performance Of Taza Gas Power Plant In Kirkuk –Iraq

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ABSTRACT

In this research, the concept of energy balance was implemented in one of the gas turbine electricity generation units in Iraq (Kirkuk - Taza) K1. The design operating data is taken and compared with the data calculated in the computer model used by Engineering Equation Solutions (EES), and the validity of the model used was confirmed. The results indicate the release of a large amount of thermal energy into the atmosphere due to the open Brayton cycle, as energy losses amounted to 60% of the energy input. The results based on the data of the first unit of the station demonstrated that the theoretical efficiency of the unit is a function of the two variables, which are the temperature of the air entering the compressor and the Turbine inlet temperature: Increasing the compressor inlet temperature leads to a decrease in net power output and first-law efficiency and an increase in the specific fuel consumption rate. Increasing the turbine inlet temperature to $1C^0$ leads to an improvement in both net power output and first-law efficiency by (0.24MW, and 0.04%) respectively. The results also showed that cooling the air entering the compressor for $1C^0$ instead of leads to improves power output and first law efficiency by (0.72MW, and 0.12%), respectively, and reduces specific fuel consumption by 7.8kg/MWh.

1.Introduction

The Kirkuk-Taza gas station in Iraq is a significant energy provider with The three types of gas units (K1, K2, and K3) that make up the Kirkuk gas station have a design power of 260 MW, 65 MW, and 283 MW, respectively. Unit K1, equipped with a V94.3A gas turbine, compressor, combustion chamber, and turbine. A portion of the thermal energy contained in flue gas is transformed by the turbine into mechanical rotational energy, which is then transformed into electrical energy by the electrical turbo-generator. The remainder of the heat energy content of an exhaust gas can either cause thermal loss or be efficiently used in a downstream heat exchanger[1] .The natural gas turbine performance may be impacted since the unique atmospheric conditions in Iraq differ from those required by ISO(1 bar, 15 C^0, and 60% relative humidity). High temperatures will reduce its power since they will cause the air's mass flow to diminish. Additionally, efficiency declines as a result of compressors needing more power in higher temperatures. The first-law in thermodynamics is applied to any analysis of a thermal system. The first law states that energy cannot be destroyed once it enters a thermal system through heat, work, or mass. However, it makes no mention of internal losses or the quality of energy that reaches the boundaries of the thermal system. Instead, it merely presents the concept of energy conservation[2].Numerous scholars have examined and evaluated gas power plants, in addition to the impact of temperature variations on the effectiveness and functionality of gas turbine parts. D. Pinilla Fernandez and et [3].A study compared the performance of a gas power plant under design and actual operational conditions. Higher compressor inlet temperature led to a decrease in power output and thermal efficiency. For every 1C^0 rise above 15 C^0, thermal efficiency drops by 0.06%. Y. Koc and et.[4] An annual exergy study was performed and theoretical calculations were compared with the system's actual characteristics. For both the real and theoretical cases, the cogeneration system's maximum thermal and exergy efficiency were found to be 79.00% and 94.19%, respectively, and 83.92% and 91.64%. A.Mohapatra and et.[5], shows that evaporative cooling improves both The specific work of a turbine by 10.48% and the efficiency by 4.6%, while evaporative compression cooling improves the specific work of the turbine and efficiency (18.4 %) and (4.18%), respectively.gas turbine power by 22. H.O.Egware and et.(2023) [6] In Benin City, Nigeria, the energy available the gas turbine was analyzed.The destructive exergy efficiency was found to be 43.98% in the combustion chamber, 7.78% in the compressor, and 3.67% in the turbine. An increase in ambient air temperature leads to higher destructive exergy efficiency and lower available energy efficiency**.** A. H. Ahmed and et.(2020)[7],Research was conducted on a 150 MW gas turbine station Kirkuk to analyze its energy and available energy using a data flow sheet. First and second law efficiencies for the compressor unit, combustion room, turbine unit, and exhaust gases were calculated as 93.34%, 85.52%, 94.11%, 42.32%, and 93.3%, 89.52%, 95.57%, 82.34% respectively. The overall available energy efficiency is 32.397%, while its thermal efficiency is 33.069%. M.Alhazmy and et. E. Kakaras , A. Doukelis and Laboratory(2004)[8],[9] With different climatic conditions .when the inlet temperature is reduced from (3 C^0 -15 C^0), increases energy by (1-7)%, and improves efficiency by 3%. Cooling the air entering the compressor increases the specific volume of the air, thus reducing the compression ratio and decreasing the work required for the compressor

2.Methodology to energy analysis

The evaluation of the energy analysis for every plant component, as displayed in Figure (1), is carried out in the following manner:

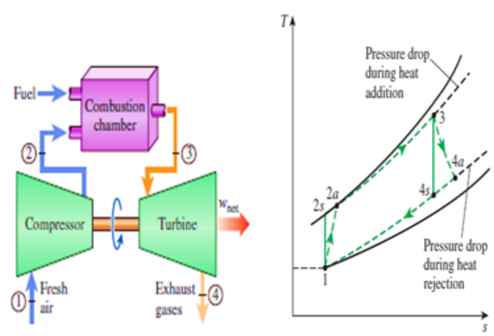

Fig 1. Diagrammatic representation of the gas turbine and related Brayton cycle

2.1 Energy balance of the air compressor

The pressure ratio(rp) for an ideal cycle is as follows: [10,11] .

$$rp = \frac{P_2}{P_1} = \frac{P_3}{P_4} \tag{1}$$

based on the isentropic relation between pressure and temperature:

$$\frac{T_{2,is}}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} = \frac{T_3}{T_4} \tag{2}$$

In states 1-2,is , the compressor processes air through an isentropic compression process, and work input per unit mass(KJ/Kg) provided by:

$$W_{1-2,is} = W_{c,is} = (h_{2,is} - h_1) = cp(T_{2,is} - T_1) \quad (3)$$

In states 1-2,a , the compressor processes air through an actual compression process, and work input per unit mass is provided by:

$$W_{1-2,a} = W_{c,a} = (h_{2,a} - h_1) = cp(T_{2,a} - T_1) \quad (4)$$

To calculate Compressor isentropic efficiency ($\eta_{C,is}$) We apply the following relationship:

$$\eta_{c,is} = \frac{W_{c,is}}{W_{c,a}} = \frac{h_{2,is} - h_1}{h_{2,a} - h_1} = \frac{cp(T_{2,is} - T_1)}{cp(T_{2,a} - T_1)} \quad (5)$$

The work rate of the compressor (KW) can be calculated by knowing the air mass flow rate.

$$\dot{W}_{C,is} = \dot{m}_a W_{C,is} \quad (6)$$

$$\dot{W}_{C,a} = \dot{m}_a W_{C,a} \quad (7)$$

The state equation to the ideal-gas can be expressed as:

$$Pv = mRT \quad (8)$$

But:

$$\rho = \frac{m}{v} \quad (9)$$

The density of air can be calculated from:

$$\rho_{air} = \frac{P_1}{RT_1} \quad (10)$$

The mass flow rate of air can be obtained from:

$$\dot{m}_{air} = \rho_{air}\dot{V} \quad (11)$$

2.2. Energy balance of the combustion chamber(C-C)

In an ideal process, heat is added to a combustion chamber at constant pressure.

$$P_3 = P_2 \quad (12)$$

mass flow rate of flow gas($\dot{m}_g$) as in the equation:

$$\dot{m}_g = \dot{m}_{air} + \dot{m}_f \quad (13)$$

The supply of heat (KW) For a cycle is provided by;

$$\dot{Q}_{in,cv} = \dot{m}_g (h_3 - h_2) = \dot{m}_g cp(T_3 - T_2) \quad (14)$$

The combustion of heat rate $\dot{Q}_{in}$ (kw) For a cycle is provided by

$$\dot{Q}_{in} = \dot{m}_f LHV \quad (15)$$

To calculate Combustion chamber isentropic efficiency ($\eta_{c,is}$) We apply the following relationship:

$$\eta_{cc} = \frac{\dot{Q}_{in,cv}}{\dot{Q}_{in}} \quad (16)$$

The air fuel ratio equation (AF):

$$AF = \frac{\dot{m}_{air}}{\dot{m}_f} \quad (17)$$

2.3. Energy balance of the turbine

In states 3-4,is the turbine processes air through an isentropic expansion process, and Work out put (KW) provided by: [12],[13]

$$\dot{W}_{3-4,is} = \dot{W}_{T,is} = \dot{m}_g (h_3 - h_{4,is}) = \dot{m}_g cp(T_3 - T_{4,is}) \quad (18)$$

In states 3-4,a , the turbine processes air through an actual expansion process, and work out Put (K) is provided by

$$\dot{W}_{3-4,a} = \dot{W}_{T,a} = \dot{m}_g (h_3 - h_{4,a}) = \dot{m}_g cp(T_3 - T_{4,a}) \quad (19)$$

To calculate turbine isentropic efficiency($\eta_{T,is}$) We apply the following relationship:

$$\eta_{T,is} = \frac{W_{T,a}}{W_{T,is}} \quad (20)$$

Heat rejected can be calculated by:

$$\dot{Q}_{out} = \dot{m}_g (h_4 - h_1) \quad (21)$$

2.4. Energy balance of the cycle

It is possible to calculate the open ideal gas turbine cycle's efficiency as: [14],[15]

$$\eta_1 = \frac{NetWorkOutput}{HeatSupplied} = \frac{W_T - W_C}{Q_{in}} \tag{22}$$

" The fraction of the turbine work used to drive the compressor is called the back work ratio " [17]

$$B.W.R = \frac{W_C}{W_T} x100\% \tag{23}$$

To estimate of the specific consumption of fuel (SFC) Kg/kw.h is as follows: [16].

$$SFC = \frac{\dot{m}_f x3600}{\dot{W}_{net}} \tag{24}$$

3. Results and discussion:

All equations utilized in this study were solved using computer software "Engineering equations solving" (ESS). The values generated in the software and the standard design values for the Taza-Kirkuk gas station in ISO circumstances, where the error rate was modest and acceptable not to exceed 2%, were compared in Table 2 of the research. The energy distribution among the plant equipment is depicted in Figure 2. These distribution is limited to ISO operating conditions for full load. The energy and energy sources provided to the plant are represented by the fuel input. 40% of the energy intake is accounted for by the power output. The plant's first-law efficiency is defined by this percentage. 60% of the fuel energy is released into the atmosphere by the plant as exhaust gasses. This implies that if there is a chance to improve plant performance, it will undoubtedly occur in this area of the unit.

Table1 properties for the fuel-gas used in a gas-turbine power plant at Taza-Kirkuk.

Fuel type	**fuel gas**	
Composition % by volume	METHANE	85.930%
	ETHANE	12.033%
	PROPANE	1.8020%
	N_BUTANE	0.1235%
	I_BUTANE	0.0790%
	N_PENTANE	0.0107%
	I_PENTANE	0.0121%
	H_2S	0.0007%
LHV	37.7 MJ/m^3	
DENSITY	0.6 kg/m^3	

*for ISO conditions only 100% CH4 is considered

Table 2 Siemens (SGT5-4000F, V94.3)Gas Turbine Model Specification (Standard and Predicted by the Present Thermodynamic Model).

	Standard value [18]	Predicted by the present model	Error %
Compression ratio(rp)	17	17	-
Thermal Efficiency %	39.8	39.77	0.07
Net power output. MW	265	264.4	0.23
Volumetric flow rate . m^3/sec	555 (max.)	540	2.7
Air mass flow rate. kg/s	650 @250MW	661.8	1.82
Fuel mass flow rate. kg/s	14.5 @250MW	14.77	1.86
Firing Temperature,C^0	1200-1390	1248	-
Temperature of Exhaust gas. C^0	577	567	1.7

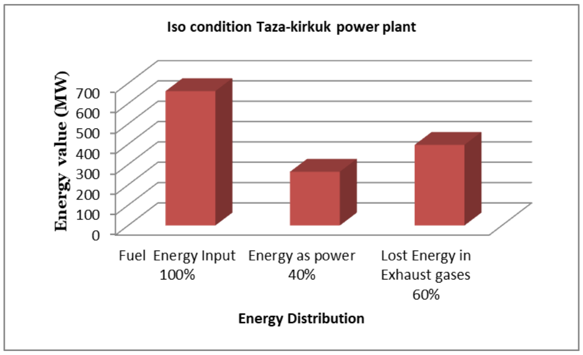

Fig. 2 Energy (MW) Distribution in Taza-kirkuk gas turbine power plant components operating.

In Fig. 3 The real data produced from the station and the values estimated in the software display the inverse connection between the temperature of the air entering the compressor and the power production over 2023. The greatest power production recorded in February, at 10.2 °C, was 218.9 MW for the computed data and 216.7 MW for the real data. The computed and actual data's poweroutput figures dropped to 198.129 MW and 196.88 MW, respectively, in August, when the compressor's intake air temperature was 37.8 C. The compressor requires more effort from the net because a rise in temperature causes a drop in density and an increase in specific volume. These outcomes are comparable to those seen in:[19],[20].

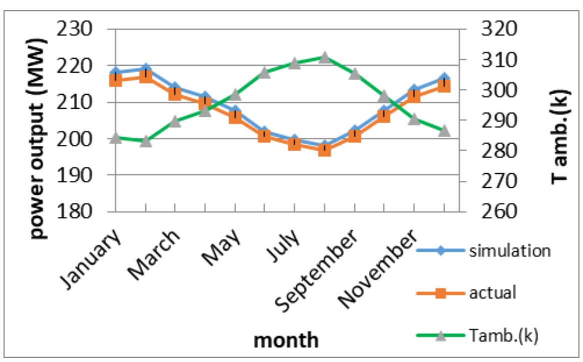

Fig. 3 Power output with ambient temperature for 2023 (actual and simulation data).

Fig. 4 shows demonstrates the relationship between the ambient temperature and the thermal efficiency (actual and simulation) which is the true measure of the turbine's performance, is equal to the quotient of the amount of energy produced by the fuel consumed According to equation(21)In Februaryat atemperature of 10.2 C^0 , the highest thermal efficiency of the gas turbine was 36.38%, 36.01% for the actual and calculated data, respectively. And in the summer, when the temperature rose to 37.8 C^0, the efficiency decreased to 32.7%, 32.91% for the actual and calculated data respectivelyThese outcomes are comparable to those seen in : [7]

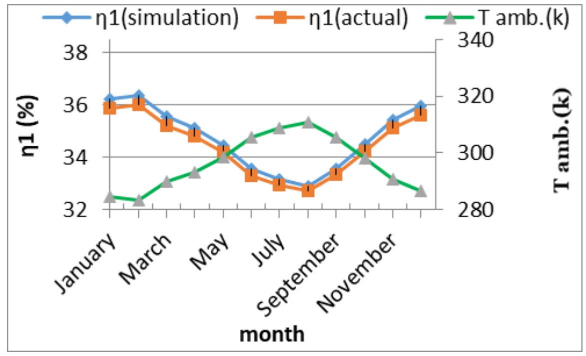

Fig. 4 Thermal efficiency (η_1) with ambient temperature for 2023 (actual and simulation data).

Fig.5 shows the correlation between the specific fuel consumption (SFC) which is obtained in equation (24) and the air temperature entering the compressor. An increase in the compressor's intake air temperature causes the mass flow rate at the compressor input to drop at a constant compression ratio, which raises SFC. February is when the value hits (0.198,0.199) Kg/Kw.h. during the winter. It increases to (0.219, 0.22) Kg/Kw.h for (real and simulation data respectively) in the summer (August). These values are nearly identical to the ones found in:[21].

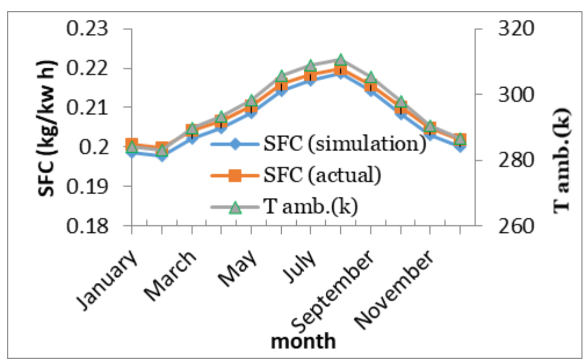

Fig. 5 specific fuel consumption (SFC) with ambient temperature for 2023 (actual and simulation data).

When summertime temperatures are 36 C^0, the compressor's air conditioning system is activated. The hottest temperatures ever recorded in 2023 happened in August, and they peaked between 10 a.m. and 2 p.m. The system operates to lower the temperature at the compressor inlet at a rate of 11.1C^0–15.2C^{0}when the air temperature varies between 36C^{0}and 49C^0, according to average real readings recorded between 8 a.m. and 4 p.m.

It was discovered that decreasing 1 C^0 increases power output by (0.72, 0.75) MW and raises the first law's efficiency by (0.12%, 0.13%) for the actual and computed data, respectively, as in Figs. 6- (a&b)

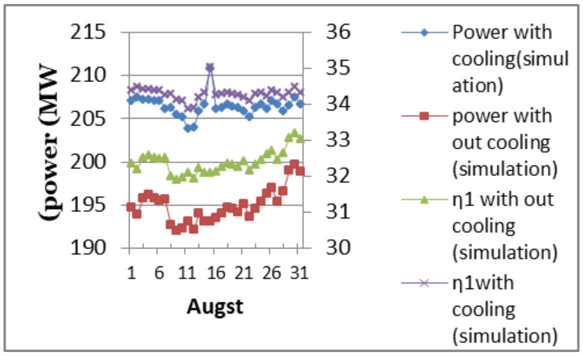

Fig.6-a Gas turbine net power output and η_1 with and without cooling system for Augst 2023.(simulation data).

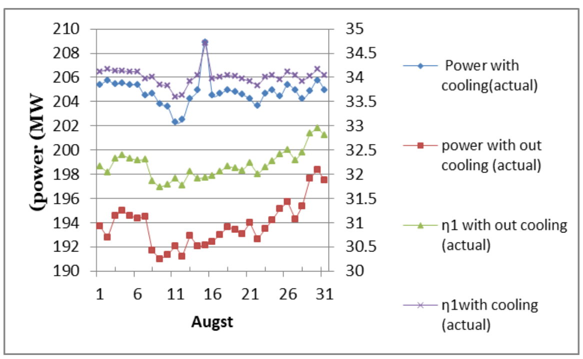

Fig.6-b Gas turbine net power output and η_1 with and without cooling system for Augst 2023.(actual data) **.**

Figs.7-(a&b) They show that the average specific fuel consumption decreases by (0.0078, 0.00815) kg/kWh for the actual and calculated data, respectively, for August, for each decrease of 1 C^0.

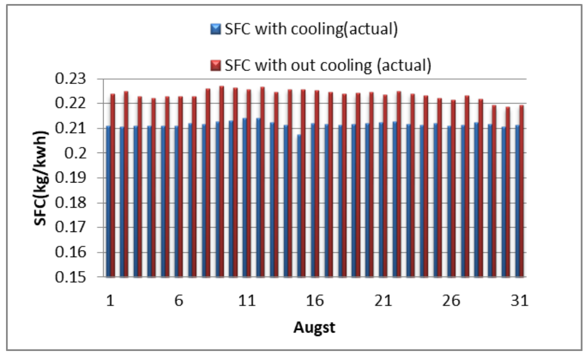

Fig.7-a SFC with and without cooling (actual data)

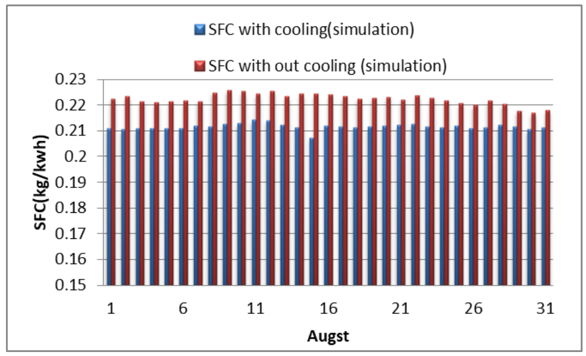

Fig.7-b SFC with and without cooling (simulation data).

The Fig.8 Illustrates how the first law's efficiency and net power production are directly correlated with the temperature at the inlet of the turbine. An increase of one degree Celsius in the turbine's intake temperature results in a corresponding increase of 0.24 MW and 0.04 %) in net power production and efficiency.

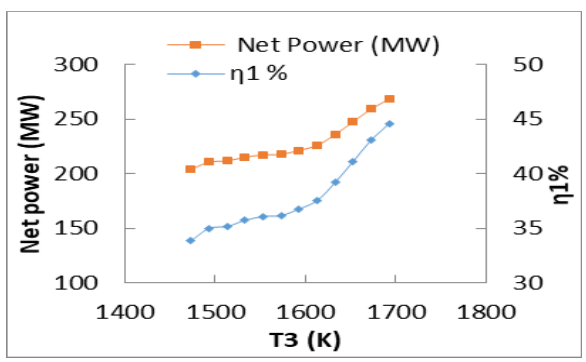

Fig.8 Turbine inlet temperature with η_1 and net power output during 2023 (simulation data).

CONCLUSIONS:

The main conclusions of the present study could be summarized as follows:

- The energy balance in the Kirkuk-Taza gas station was able to measure the amount of energy losses, which amounted to 396.6 megawatts (60%), which goes with the exhaust gases in the open cycle.
- A rise in the compressor inlet temperature causes the specific fuel consumption rate to rise along with a fall in net power output and first-law efficiency.
- Reducing specific fuel consumption by 7.8kg/MWh and increasing power output and first law efficiency by 0.72MW and 0.12%, respectively, are the results of cooling the compressor's intake air by $1C^0$.
- The first law efficiency and net power production both improve by 0.24 MW and 0.04%, respectively, with a $1C^0$ increase in the turbine inlet temperature.

NOMENCLATURE

Cp Specific heat at constant pressure(kJ/kg.K)

h Specific enthalpy, (kJ/kg)

rp pressure ratio

T Temperature, C^0

P pressure , kpa

$\boldsymbol{w}$ Work, kJ/kg

$\dot{\boldsymbol{W}}$ Power or work rate(kw)

$\dot{\boldsymbol{m}}$ Mass flow rate, (kg/s)

R Gas constant [kJ/mol – K]

m mass (kg)

v volume (m^3)

$\dot{\boldsymbol{v}}$ volumetric flow rate(m^3/s)

L H V Lower heating value (kJ/kg)

$\dot{\boldsymbol{Q}}$ Heat rate , (kw)

B.W.R back work ratio

SFC specific fuel consumption(kg/kw.h)

Greek symbols

$\boldsymbol{\rho}$ Density, (kg/$\mathbf{m^3}$)

$\boldsymbol{\gamma}$ Adiabatic index

η_1 First law efficiency

Subscripts

amb Ambient

in Input

out Outlet

cc Combustion chamber

c Compressor

is Isentropic

a Actual

T Turbine

f Fuel

$\boldsymbol{CV}$ Control volume

g Flue gasses

1 At compressor inlet

2 At compressor outlet

3 At gas turbine inlet

4 At exhaust

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