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Effect of Variable Parameters of the Performance of Gas Turbine Cycle

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ABSTRACT

The gas turbine are being usually used in several application such as aircraft engine and electricity power station plants. To improve its performance many researches has been performed. In this study the thermodynamic analysis has been carried out of a gas turbine power plant located at the sea level in Rasanuf city (north of Libya), based on the conservation of energy to evaluate its efficiency, net power and specific fuel consumption. The analysis of thermodynamic equations of first law performed depending on the variable parameters (atmosphere air temperature, pressure ratio) at fixed turbine inlet temperature (TIT) for 1000 K, 1200 K, and 1500 K, compressor pressure ratio from 4 to 20 and ambient temperature 15 ^{o}C to 45 ^{o}C.

Keywords: *Gas Turbine, Power generation, energy*

1. Introduction

The gas turbine is a machine to convert thermal energy carrying by gases at high pressure to the mechanical energy by converting heat energy into kinetic energy in a fixed blades and then rotating a set of moving blades. Its new aspect has been invented by Sir Charles Parsons in 1884 [1]. It has been replaced the reciprocating steam engine which has a lower thermal efficiency and smaller power-to-weight ratio. The gas turbine generates a rotational mechanical power which is suitable for driving an electrical generator – the gas turbine power plant provides about 60% of all electricity generation in the overall the world [1]. Recently its application in the industrial power generation has more attention than other field. Gas turbine plants are being used in distant oil, pumping stations for gas pipeline, on offshore platforms and in gas fields for more than 70th [1]. Compact gas turbine sets has been used where the water is mingy, supplying fuel is cheap and area requirements fixed, have demonstrated invaluable. Recently the improvement was achieved in the performance of the gas turbine plants for producing electrical energy. The main advance of performance of gas turbine has been noted in the raising of its net power from 50 to 100 MW, rise in its efficiency up to 37 % and falling in the cost of power generation [1]. These advantages of gas turbine of power generation gives more attractive in overall the world.

Several researches have performed to analysis the thermodynamic cycle of gas turbine based on first and second law of thermodynamic to improve its performance, Ankit Kumar, Ankit Singhania, Abhishek Kumar Sharma, Renendra Roy, Bijan Kumar Mandal [3]. Performed an analysis depending on mathematical equations of conservation of energy to investigate The influence of operating parameters (compressor inlet temperature, humidity, Pr, TIT, η_c, η_t) on the gas turbine performance. Sara et al [4]. The Chemcad simulation, exergy and energy analysis on the actual working data to determine the losses in actual conditions have been conducted. In addition to study the effect of the energy and exergy analysis Thamir et al [5]. performed a parametric analysis using actual operating data of the open simple cycle of gas turbine plant, at different intake ambient temperature of compressor, this analysis had showed that exergy analysis give more accurate in determining of the thermal cycle efficiency of the system than the first law of thermodynamic.

In the present work the investigation of Raslanuf power plant was performed out depending on the conservation of energy to study the influence of variable parameters (inlet temperature of compressor, pressure ratio) on its performance at fixed TIT.

2. Gas Turbine power plant and its thermodynamic Cycle

The main parts of gas turbine in power plant are compressor, combustion chamber, gas turbine and generator. The compressor takes air from atmosphere at state 1 and raising its pressure to desired higher pressure at state 2, after that the pressurized air enters the combustion chamber and increasing its temperature by burning it with the fuel to reaches to state 3, then the gases at high pressure and temperature leaves combustion chamber to the gas turbine, in this stage the thermal energy and pressure energy (enthalpy) transfer to the mechanical energy to rotates its shaft power, after expanding of gases in gas turbine the gases leaves it to the atmosphere state 4.

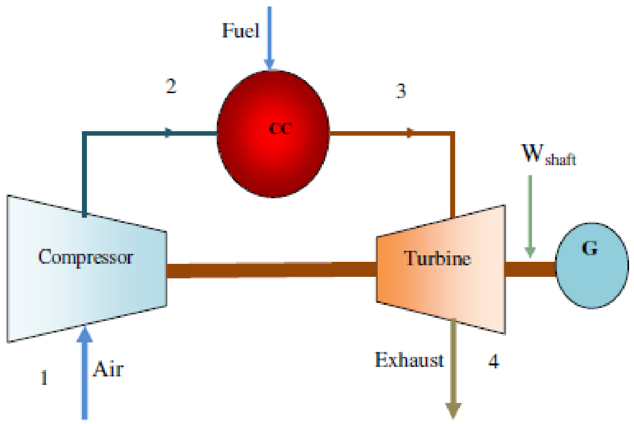

Fig. 1 main components of gas turbine [6]

The thermodynamic cycle of gas turbine consists of four processes is known as Brayton cycle (ideal cycle) as depicted in figure 2. (T-S) diagram shows ideal and actual thermodynamic cycle of gas turbine.

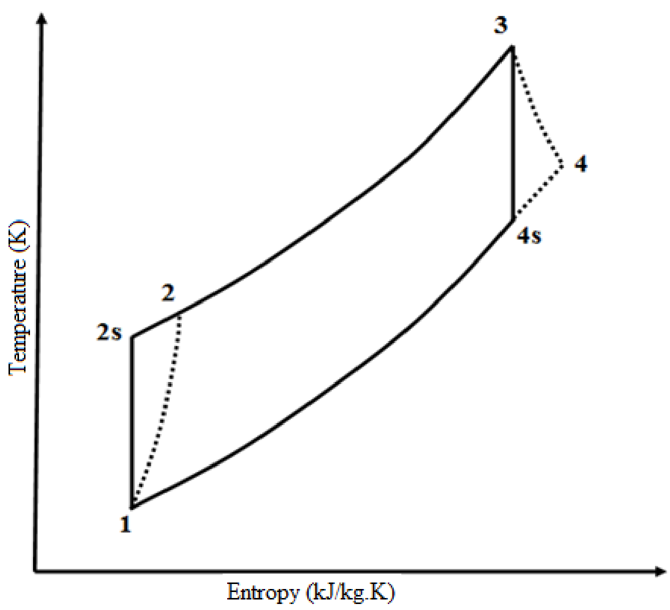

Fig. 2 Isentropic and actual thermodynamic cycle of gas turbine [6]

From state 1 to 2s is adiabatic and reversible process (Isentropic process) in this stage the air is compressed to the higher pressure of cycle by work of compressor, while state 1 to 2 is actual process (irreversible process) .state 2 to 3 is isobaric process, heat is added to the compressed air at constant pressure. Comprehended compression process in this process, the working gas is compressed isentropically from state 1 to state 2s with the help of a compressor. The process is done by keeping the entropy constant and increasing its internal pressure (1-2 actual process). Process 2-3 Isobaric heat addition process in this process, the heat is added at constant pressure i.e. isobaric process. The gas is heated using an external heat source at constant pressure. Process 3-4s Isentropic expansion process in the isentropic expansion process, the heated gas will be expanded isentropically from the state 3 to 4s in a turbine (3-4 actual process). The amount of energy loses of the system is equal to the work have done by the turbine. Process 4-1 Isobaric heat rejection in this process, the expanded gas have rejected its heat from state 4 to state 1. This process will be completed at constant pressure. So, as with any other ideal process, the actual process differs from the ideal process. The ideal process is assumed to make the calculations and setting a specific standard. In the actual cycle, we have seen the compression, as well as expansion process, are isentropic in the ideal Brayton cycle. But in actual it doesn't happen, both the compression and expansion increase the entropy it doesn't remain constant.

3. Raslanuf gas turbine power plant (MS 5001)

The selected gas turbine unit for this study is located at the sea level in Rasanuf city. It began its operation in1986. The power station consists of two units of gas turbine type MS-5001, and each unit of gas turbine producing 26.8 MW. The three main components of gas turbine are, axial compress with seventeen stages, combustor system consists of ten chambers designed for different types of fuel including heavy and light distillates, natural gas, and gas turbine with two stages, high pressure wheel and low pressure wheel. The MS5001 designed for heavy duty, long work life and ease maintenance processes and operation [7].

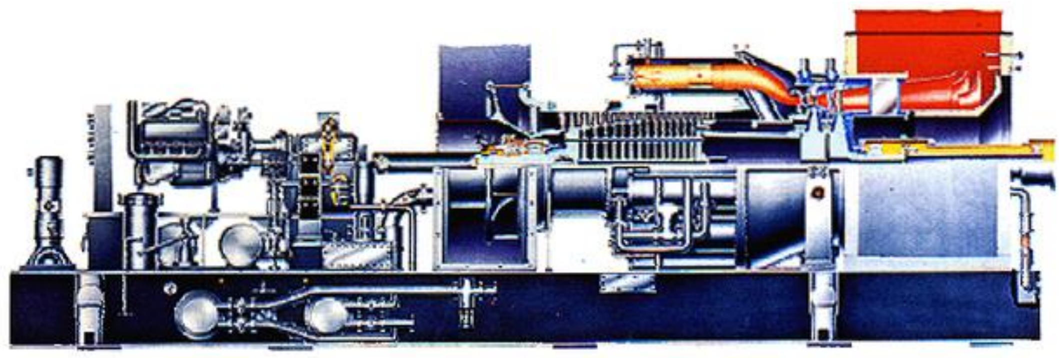

Fig. 3: Heavy Duty Gas Turbine MS 5001

Table (1): Design Thermodynamic Parameters of (MS 5001)

26.8 (MW)	Power
0.85	Efficiency of compressor (η_c)
0.89	Efficiency of turbine (η_T)
28.4	Thermal efficiency (η_{th})
123.1 (kg/s)	Mass flow rate of air (m_a)
125.2 (kg/s)	Mass flow rate of gas (m_g)
12.687 (kJ/kw.h)	Heat rate
943 (°C)	Firing temperature
10.5	Pressure ratio ($\boldsymbol{r_p}$)
1.333	expansion coefficient of gas ($\boldsymbol{\gamma_g}$)
0.93	Efficiency of combustion chamber η_{cc}
2.34 (kJ/kg.k)	Specific heat of fuel c_{pf}
1.4	atmospheric air compression coefficient ($\boldsymbol{\gamma_a}$)

4. Thermodynamic Analysis of Thermal Cycle

4.1 Absorbing power of compressor

$$W_c = \dot{m}_a\,(h_{2s} - h_1) = \dot{m}_a C_{pa}\,(T_{2s} - T_1) \quad \text{for ideal cycle.} \qquad (1)$$

$$\mathrm{W_{ca}} = \dot{\mathrm{m}}_\mathrm{a}\,(\mathrm{h_{2a}} - \mathrm{h_1}) = \dot{\mathrm{m}}_\mathrm{a}\mathrm{C_{pa}}\,(\mathrm{T_{2a}} - \mathrm{T_1}) \quad \text{for actual cycle.} \qquad (2)$$

To find the exit temperature of air from the compressor (T_2) using the following equation (3):

$$\frac{T_{2s}}{T_1} = {r_p}^{\alpha} \qquad (3)$$

Where:

γ_a Specific heat ratio of the air

$\alpha = (\gamma_a - 1)/\gamma_a$

$r_p = \frac{P_2}{P_1}$ Compression ratio

To find the actual exit temperature of air from the compressor (T_{2a}) using the following equation (4.b):

$$\eta_c = \frac{T_{2s} - T_1}{T_{2a} - T_1} \quad (4.a)$$

$$T_{2a} = T_1 . \left\{1 + \frac{{r_p}^{\alpha} - 1}{\eta_c}\right\} \quad (4.b)$$

Where η_c Isentropic efficiency of the compressor.

4.2 *Adding heat in combustion chamber*

$$Q_{in} = \dot{m}_g \,(h_3 - h_2) \quad (5)$$

$$Q_{in} = \dot{m}_g \,. C_{pg} (T_3 - T_2) \quad (6)$$

Where:

$\dot{m}_g$ gas mass flow rate

$$\dot{m}_g \dot{=} m_a + \dot{m}_f \quad (7)$$

$\dot{m}_a$ air flow rate

$\dot{m}_f$ fuel mass flow rate

4.3 *Power delivered by the turbine*

$$W_t = \dot{m}_g \,(h_3 - h_{4s}) = \dot{m}_g C_{pg} \,(T_3 - T_{4s}) \quad \text{for ideal cycle.} \quad (8)$$

$$W_{ta} = \dot{m}_g \,(h_3 - h_{4a}) = \dot{m}_g C_{pg} \,(T_3 - T_{4a}) \quad \text{for actual cycle.} \quad (9)$$

To find the exit temperature of gases from the gas turbine (T_4) using the following equation

$$\frac{T_3}{T_{4s}} = {r_p}^{\beta} \quad (10)$$

Where:

β specific heat ratio of gases

$\beta = (\gamma_g - 1) / \gamma_g$

r_p Compression ratio

$r_p = \frac{P_3}{P_4}$

To find the actual exit temperature of gases from the gas turbine (T_{4a}) using the following equation

$$\eta_t = \frac{T_3 - T_{4a}}{T_3 - T_{4s}} \qquad (11.a)$$

$$T_{4a} = T_3.\{1 - \eta_t(1 - r_p{}^{\beta})\} \qquad (11.b)$$

η_t Isentropicl efficiency of the turbine

4.4 *Net power output*

$$W_{net} = W_t - W_c \quad \text{for ideal cycle} \qquad (12)$$

$$W_{net,a} = W_{ta} - W_{ca} \quad \text{for actual cycle} \qquad (13)$$

4.5 *Thermal Efficiency*

$$\eta_{th} = \frac{W_{net.a}}{Q_{in}} \qquad (14)$$

$$\eta_{th} = 1 - \left(\frac{1}{r_p^{\beta}}\right) \qquad (15)$$

4.6 *Fuel Mass flow rate*

$$\dot{m}_f = \dot{m}_a \left\{\frac{\boldsymbol{h_3 - h_2}}{(LCV).\eta_{comb.} - \boldsymbol{h_3}}\right\} \qquad (16)$$

Where

$\eta_{comb.}$ Actual efficiency of the e combustion chamber.

LCV Lower Calorific Value of Fuel

4.7 *Specific fuel consumption*

$$SFC = 3600 . \frac{\dot{m}_f}{W_{net,a}} \qquad (17)$$

5. Result and Discussion

The performance of gas turbine is investigated parametrically by Engineering Equations based on the conservation of energy using data design in table .1 with changing of pressure ratio, atmosphere temperature and turbine inlet temperature.

Table 2: Changing of net output power, thermal efficiency and SFC with different values of atmosphere temperature and pressure ratio at TIT = 1000 °C

	T_1 = 15 C			T_1 = 25 C			T_1 = 35 C			T_1 = 45 C		
rp	P(kw)	SFC	η_{th}%	P(kw)	SFC	η_{th}%	P(kw)	SFC	η_{th}%	P(kw)	SFC	η_{th}%
4	16.62	0.414	19.0	15.90	0.424	18.6	15.18	0.434	18.1	14.46	0.445	17.7
6	17.59	0.358	22.0	16.60	0.369	21.3	15.62	0.382	20.6	14.63	0.396	19.8
8	17.21	0.340	23.2	16.02	0.354	22.2	14.82	0.370	21.2	13.62	0.390	20.2
10	16.30	0.336	23.4	14.93	0.354	22.2	13.55	0.376	20.9	12.18	0.403	19.5
12	15.15	0.340	23.1	13.63	0.364	21.6	12.10	0.394	20.0	10.57	0.433	18.2
14	13.89	0.351	22.4	12.23	0.382	20.6	10.56	0.424	18.5	8.90	0.484	16.2
16	12.58	0.368	21.4	10.79	0.411	19.2	9.01	0.472	16.7	7.22	0.568	13.8
18	11.25	0.392	20.0	9.36	0.452	17.4	7.46	0.546	14.4	5.56	0.719	10.9
20	9.93	0.426	18.5	7.93	0.512	15.4	5.93	0.669	11.8	3.93	1.041	7.6

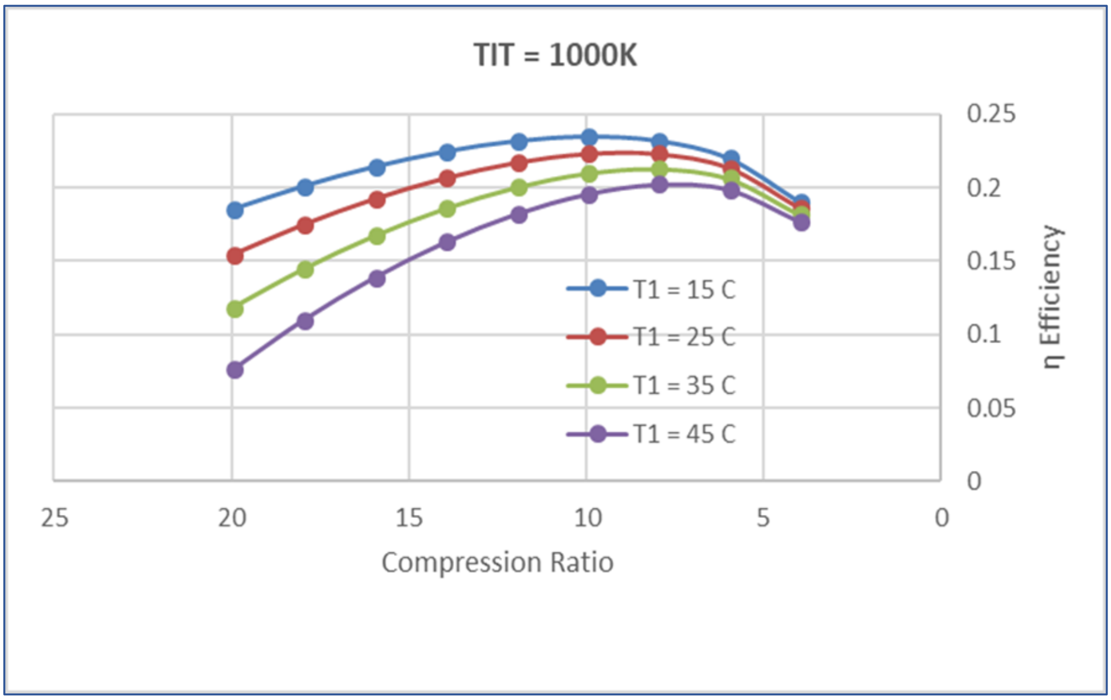

Fig. 4 Variation of thermal efficiency with pressure ratio of different inlet temperature of compressor at turbine inlet temperature (TIT) T_3 = 1000 k

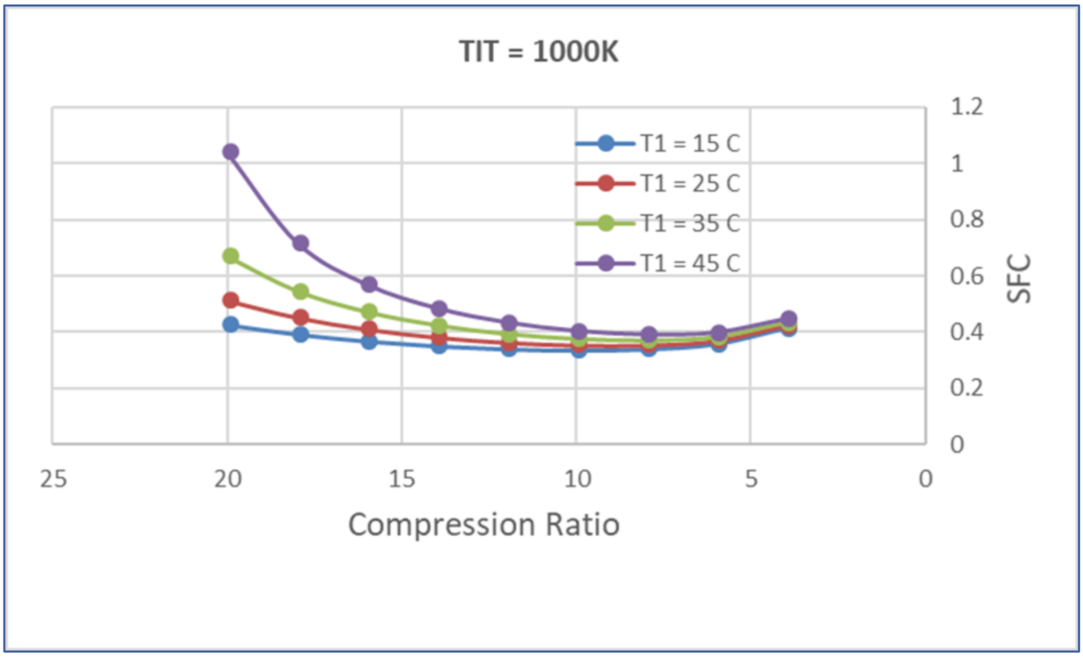

Fig. 5 Variation of SFC with pressure ratio of different inlet temperature of compressor at turbine inlet temperature (TIT) T_3 = 1000 k

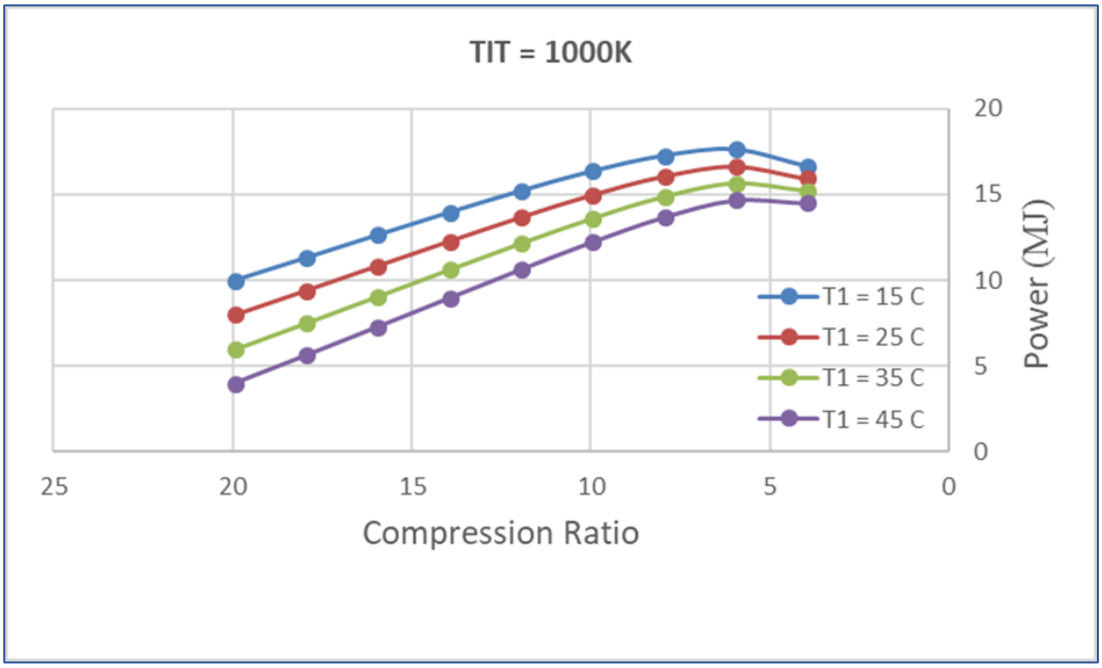

Fig. 6 Variation of net output power with pressure ratio of different inlet temperature of compressor at turbine inlet temperature (TIT) T_3 = 1000 k

Table 3: Variation of thermal efficiency, net output power and specific fuel consumption (SFC) with different values of ambient temperature and pressure ratio at turbine inlet temperature(TIT) T_3 = 1200

	T_1 = 15 C			T_1 = 25 C			T_1 = 35 C			T_1 = 45 C		
rp	P(kw)	SFC	η_{th}%	P(kw)	SFC	η_{th}%	P(kw)	SFC	η_{th}%	P(kw)	SFC	η_{th}%
4	16.62	0.414	19.0	15.90	0.424	18.6	15.18	0.434	18.1	14.46	0.445	17.7
6	17.59	0.358	22.0	16.60	0.369	21.3	15.62	0.382	20.6	14.63	0.396	19.8
8	17.21	0.340	23.2	16.02	0.354	22.2	14.82	0.370	21.2	13.62	0.390	20.2
10	16.30	0.336	23.4	14.93	0.354	22.2	13.55	0.376	20.9	12.18	0.403	19.5
12	15.15	0.340	23.1	13.63	0.364	21.6	12.10	0.394	20.0	10.57	0.433	18.2
14	13.89	0.351	22.4	12.23	0.382	20.6	10.56	0.424	18.5	8.90	0.484	16.2
16	12.58	0.368	21.4	10.79	0.411	19.2	9.01	0.472	16.7	7.22	0.568	13.8
18	11.25	0.392	20.0	9.36	0.452	17.4	7.46	0.546	14.4	5.56	0.719	10.9
20	9.93	0.426	18.5	7.93	0.512	15.4	5.93	0.669	11.8	3.93	1.041	7.6

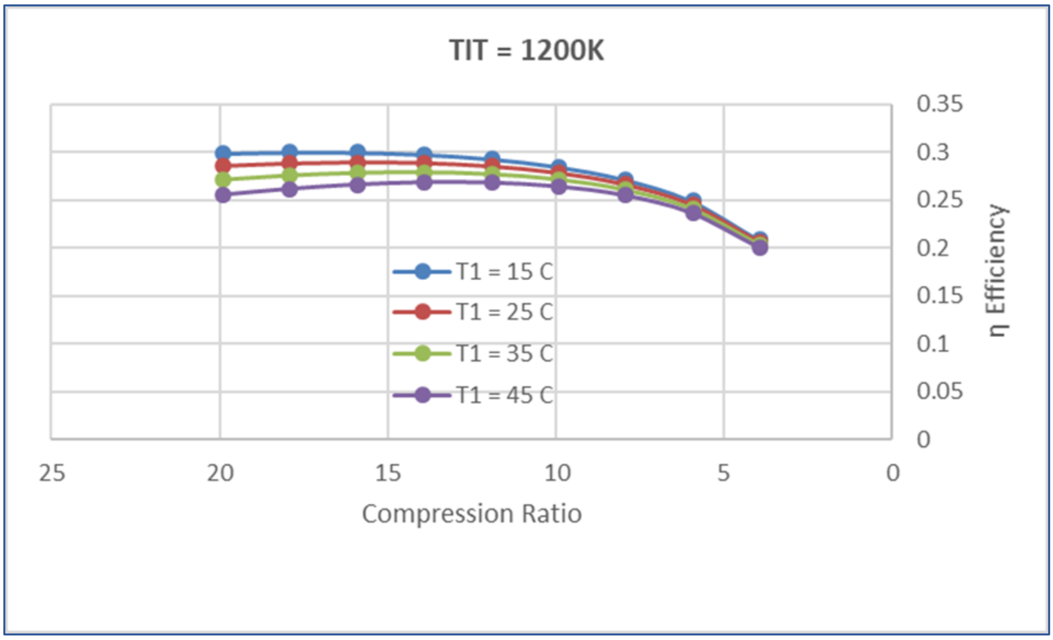

Fig. 7 Variation of net thermal efficiency with pressure ratio of different inlet temperature of compressor at turbine inlet temperature (TIT) T_3 = 1200 k

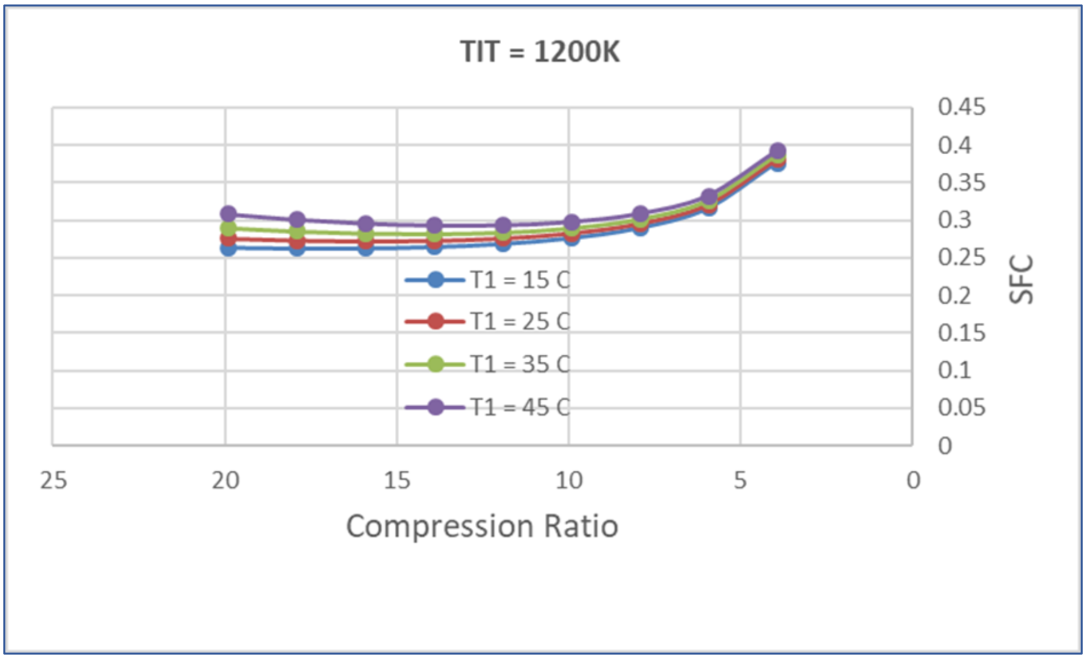

Fig. 8 Variation of SFC with pressure ratio of different inlet temperature of compressor at turbine inlet temperature (TIT) T_3 = 1200 k

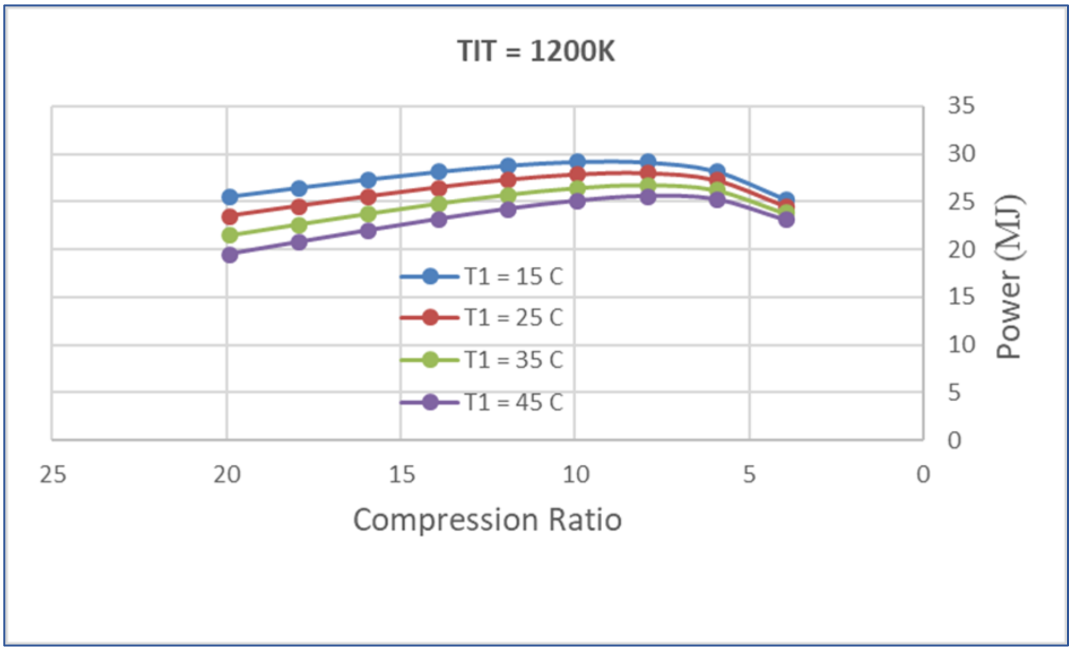

Fig .9 Variation of net output power with pressure ratio of different inlet temperature of compressor at turbine inlet temperature(TIT) T_3 = 1200 k

Table 4: Variation of thermal efficiency, net output power and specific fuel consumption (SFC) with different values of ambient temperature and pressure ratio at turbine inlet temperature(TIT) T_3 = 1500

	T_1 = 15C			T_1 = 25C			T_1 = 35C			T_1 = 45C		
rp	P(kw)	SFC	η_{th}%	P(kw)	SFC	η_{th}%	P(kw)	SFC	η_{th}%	P(kw)	SFC	η_{th}%
4	38.70	0.351	22.4	37.98	0.353	22.3	37.27	0.356	22.1	36.55	0.359	21.9
6	44.83	0.289	27.2	43.84	0.292	26.9	42.85	0.295	26.7	41.86	0.298	26.4
8	47.80	0.262	30.0	46.60	0.264	29.7	45.40	0.267	29.4	44.20	0.270	29.1
10	49.33	0.246	32.0	47.96	0.249	31.6	46.58	0.252	31.3	45.20	0.255	30.9
12	50.07	0.235	33.4	48.55	0.238	33.0	47.02	0.242	32.6	45.49	0.245	32.1
14	50.35	0.228	34.5	48.69	0.231	34.0	47.02	0.235	33.5	45.36	0.238	33.0
16	50.32	0.223	35.3	48.54	0.226	34.7	46.75	0.230	34.2	44.97	0.234	33.6
18	50.09	0.219	35.9	48.20	0.223	35.3	46.30	0.227	34.7	44.40	0.231	34.1
20	49.73	0.216	36.4	47.73	0.220	35.7	45.73	0.224	35.1	43.73	0.229	34.4

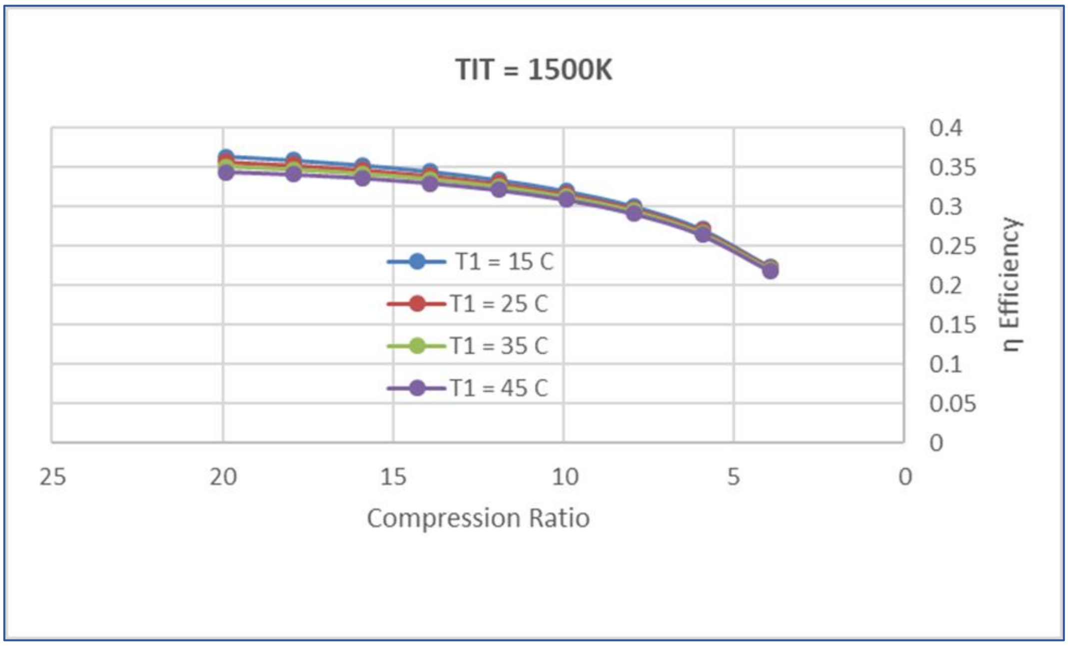

Fig. 10 Variation of thermal efficiency with pressure ratio of different inlet temperature of compressor at turbine inlet temperature(TIT) T_3 = 1500 k

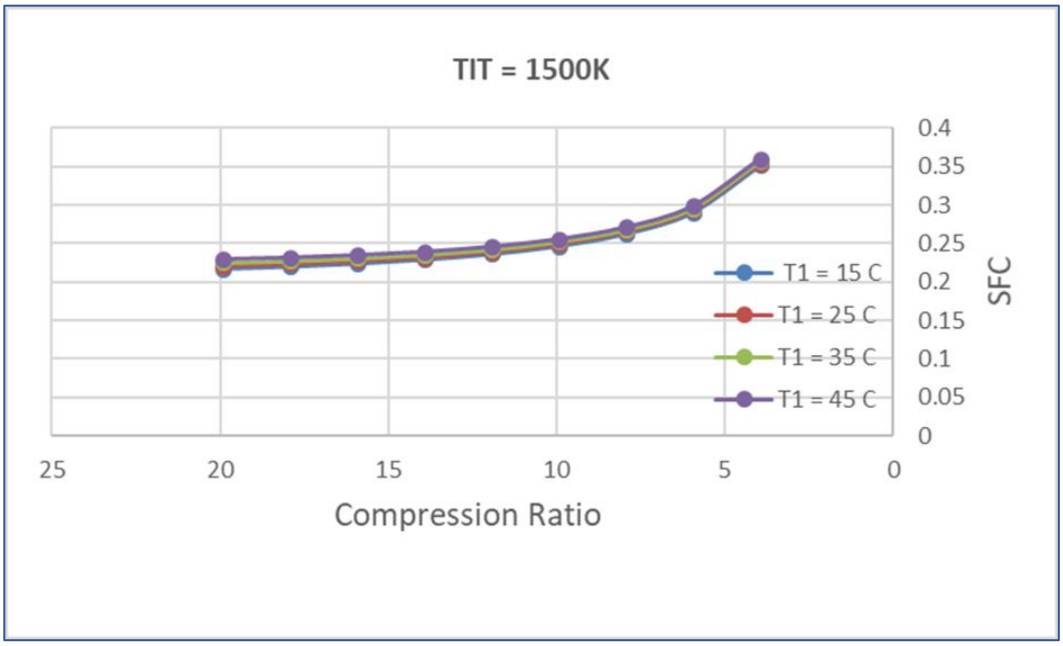

Fig. 11 Variation of SFC with pressure ratio of different inlet temperature of compressor at turbine inlet temperature(TIT) T_3 = 1500 k

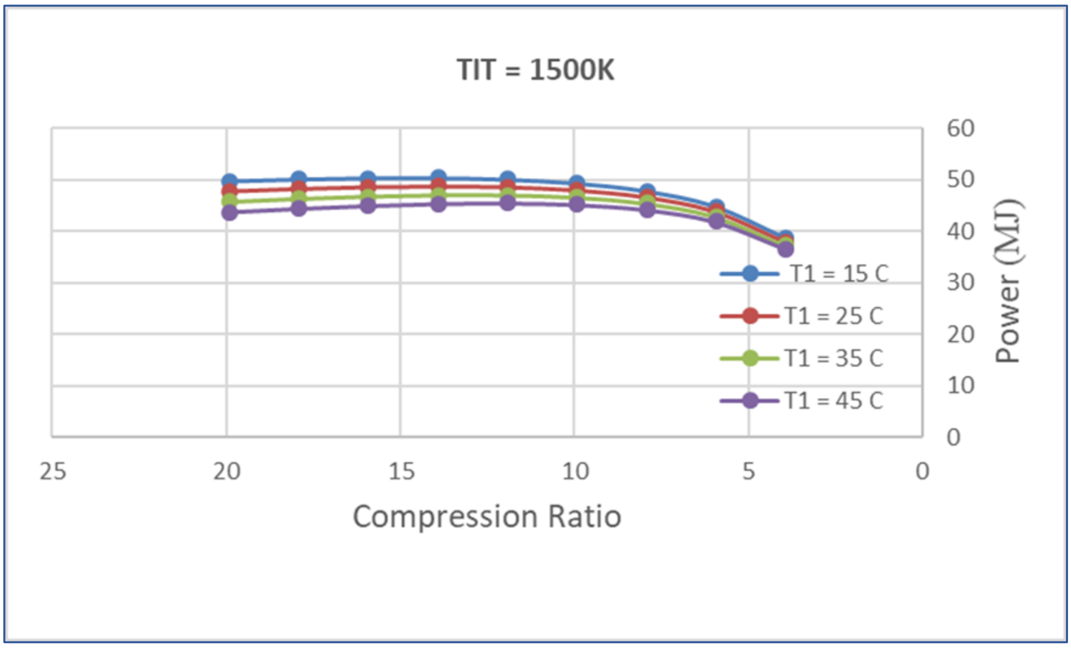

Fig .12. Variation of net output power with pressure ratio of different inlet temperature of compressor at turbine inlet temperature(TIT) T_3 = 1500 k

As illustrated In the figure (4), the curves of thermal efficiency of gas turbine at TIT = 1000 K showed that, a little pit increasing of thermal efficiency with a higher inlet temperature of compressor at the same pressure ratio, but as increasing of pressure ratio there are three different behaviour of curves at this inlet temperature of gas turbine, increasing until reaches to the maximum after that it slowly dropped and then faster decreasing in the thermal efficiency. Whereas this behaviour is completely different as increasing of inlet temperature of gas turbine, as depicted in fig (7, 10), the thermal efficiency increases with higher pressure ratio.

The set curves of SFC at TIT =1000 K with different value of inlet temperature of compressor have been illustrated in fig (5), it has been slightly increased as inlet temperature of compressor increased, and when changing the pressure ratio the curve is slightly dropped and then go up with increasing the pressure ratio. Whilst for increasing inlet temperature of turbine with raising pressure ratio, it has been started to significantly dropped until reaches to the minimum, after that it barely goes up as it has been showed in figure (8), but the behaviour at TIT= 1500 K is somewhat different in contrast with TIT =1200 K as depicted in figure (11).

Figure (6) has been depicted that ,the net output power has been reduced with raising the inlet temperature of compressor, and with changing the pressure ratio by increasing it the thermal efficiency go up until certain value and after that it significantly lowering it, As the inlet temperature of turbine raising to the higher value the behaviour will be changing and after reaching to its maximum value the affected of pressure ratio can be negligible as in fig (8, 12).

Conclusions

Based on the previous results and discussion of performance of gas turbine by using thermodynamic analysis of conservation of energy one can conclude the following:

- The influence of pressure ratio on the efficiency and net output power depending on TIT at any inlet temperature of compressor. the highest TIT the best efficiency and net power output.
- The net power output and thermal efficiency decrease as increasing of ambient temperature in the cycle at any level of pressure ratio and TIT.
- Although the thermal efficiency and net power output at TIT = 1500K is higher but also losses due to irreversibility is maximum as compared to others TIT.
- As the inlet temperature of compressor increased in the hot weather that’s lead to the higher SFC and it is an independent of TIT. While the effect of

the pressure ratio indicated that, as a higher pressure ratio as a minimum of SFC until middle level of pressure ratio (10-12), after this level of pressure ratio the SFC will be up. Whereas at the highest TIT(1500) as increasing pressure ratio as lowering the SFC.

Nomenclature

Symbol		**unit**
h	specific enthalpy	(kJ/kg)
HR	heat rate	(kJ/kWh)
LHV	lower heating value	(kJ/kg)
ṁ	mass flow rate	(kg/s)
p	pressure	(bar)
P	power	(kW)
Q	heat supply	(kW)
R	gas constant	(kJ/kg.°C)
SFC	specific fuel Consumption	(kg/kW)
T	temperature	(°C or K)
V	volume flow rate	(m^3/s)

Abbreviations

AFR -air to fuel ratio

AAT - ambient air temperature

CC - combustion chamber

EG - electrical generator

GT - gas turbine

IGV - inlet guide vanes

TIT - turbine inlet temperature

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