

# Performance of a Gas Turbine Power Plant

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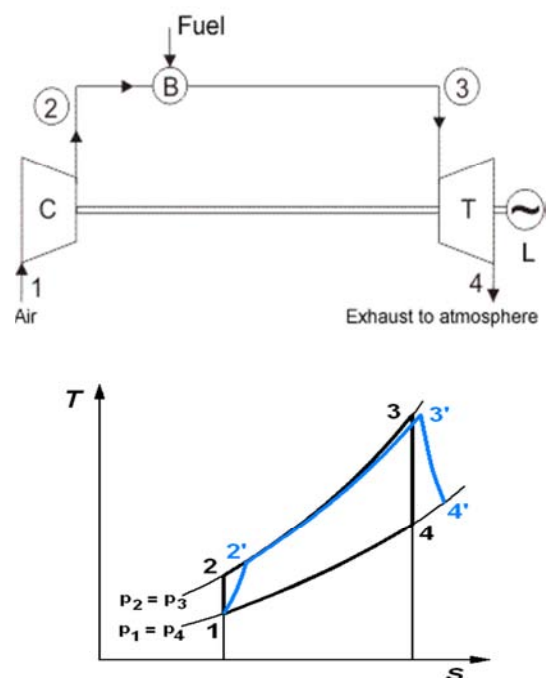
**Abstract:** Population of the world is increasing and so the demand of energy for domestic and industrial use. Worldwide efforts are being meet to produce energy from renewable resources but it is impossible to replace the fossil fuel resource of energy. Gas turbine remains the main energy converter of fossil fuel both for its liquid and gaseous states. This paper analyzes the performance of a unit of an actual power plant situated at Riyadh, Saudi Arabia. The power plant has G.E. gas turbine units which operate as simple gas turbine units at peak demand periods and as a part of combined power plant unit at other periods of operation. The performance of the unit has been evaluated from the actual data from the unit obtained around the year. The study reveals that for gas turbine cycle an increase in compressor inlet temperature increases efficiency of compressor whereas turbine efficiency decreases. Exit temperature of the unit increases with the increase in inlet temperature of compressor that results in enhanced pollution. The exergy and energy efficiencies of the whole unit show dependence on compressor inlet temperature. Plant efficiency and mass of steam produced increase with the increase in the turbine inlet temperature.

**Keywords:** Axial Flow, Compressor, Turbine, Energy, Exergy, Efficiency, Power Plant

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## 1. Introduction

Demand for energy in the world is greater than ever before as it is the primary pillar of infrastructural and industrial growth in any country besides meeting the human energy demand. The energy demand at a location occurs during peak hours that vary according to latitude, altitude and the season of a place. Gas turbine power plants are ideally suited for this operation because of their short time needed for both peak generation and synchronization. Figure 1 shows the main parts of gas turbine power plants along with Brayton cycle which is basis of operation of gas turbines. Components of the gas turbine are compressor (C), combustion system (B), turbine (T) and generator. Brayton cycle thermodynamically represents gas turbines performance. An ideal cycle consists of isentropic compression of air in compressor, isobaric combustion in combustor and isentropic expansion of gases in a turbine. In a real cycle kinetic energy changes and fluid friction occur in its components, compression and expansion processes are non-isentropic and heat transfer in its components becomes appreciable.



**Figure 1.** Simple Gas Turbine Plant and Brayton cycle.

Huge quantities of hot are exhausted at the exit of turbine that contains huge energy. These gases are wasted in units operating on simple cycle and results in environmental degradation. These gases can be used to produce steam for combined power plants, heat for cogeneration and drinkable water in desalination plant. Such a use enhances the energy efficiency of these plants from 20 % for pure gas turbines to 60% for combined plants and 80% for cogeneration units. Most of the power stations in the Kingdom use GE MS7001EA gas turbines. A gas turbine unit has 17 stage axial compressor, 10 symmetrical placed can annular combustion chambers and 3 stage axial compressor. The ISO specifications of this gas turbine are given in table 1. Power plant 9 which is situated near Riyadh has 56 such units. These units are subdivided into 6 blocks. First 16 units comprise block A and block B. Units 17 to 24 are grouped in block E. Block C and block D shear 20 units. The last 12 units (45-56) are placed in block F.

**Table 1.** ISO specifications GE MS7001EA gas turbine.

ISO rated power = 85.4 MW	Heat rate= 10990 kJ/kWh
Efficiency= 32.7%	Rated pressure ratio = 12.6
Exhaust mass flow rate = 292 kg/s	Rated turbine speed= 3600 rpm
Exhaust temperature= 537°C	Compressor stages = 17
Number of combustion canes=10	Ambient Pressure at Sea Level=100 kPa
Ambient Temperature =288K	Relative Humidity = 60%

Pressure of the air at compressor exit and temperature of the hot gases at the entry and exhaust of the turbine are the important parameters that affect the performance of gas turbine. The compressor pressure ratio depends on the tip speed of compressor rotor, axial velocity at the entrance of the compressor and the compressor stage blade geometry [1]. Tip speeds are limited to 350 m/s due to stress consideration whereas axial velocity of 150 m/s and 200 m/s is limits for common industrial gas turbines and aeroderivative engines respectively because of compressibility consideration [2]. Any non-uniformity of the flow at the inlet of compressor enhances the work input compressor, decrease in net output of the unit and decrease in its efficiency [1]. Material considerations restrict the turbine inlet temperature to 1800°C for common material and 2500°C for special materials [3]. The combustion chamber temperature rise depends on fuel air ratio. The stoichiometric fuel air ratio in the primary zone of the combustor of the gas turbine is 1:14.

Aeroderivative gas turbines use a higher overall air fuel ratio that decreases the flame temperature to suitable levels of the turbines. The subject matter of this study is energetic and exergetic analysis of a unit of a power plant during peak hours, when it runs on a simple Brayton cycle.

Gas turbines are the ideally suited to meet peak demand energy and have small synchronization time of 30 minutes and grid stability [4]. Simple gas turbine plants operating have high unit cost of generated electricity [5]. These plants have efficiency in the range of 20-28% [6]. Unit cost of electricity could be reduced by inlet cooling at the compressor intake. The ambient temperature has a strong influence on the gas turbine performance [1, 2, and 7]. A 25% loss of the rated power capacity of the gas turbine at ISO conditions has been reported [8] as the ambient temperature reaches 40°C. Al-bortmany (9) has shown that an increase in power production up to 20% in winter and 40% in summer is possible if the inlet temperature of a gas turbine power plant unit is decreased to 7°C.

Production cost of electricity and gaseous emissions can be reduced by using a renewable energy source, [10], or by using regenerative cycle, combined cycle and power augmentation [11]. Studies on combined plants have been directed to output maximization [12, 13], economic system optimization [14] and ecological features [15]. Godoy, et al. [16] has covered most of these aspects for a simple HRSG system. Basharat, et al, [17, 18] have carried out the thermodynamic analysis of a cogeneration gas turbine and desalination plant. They found out that the pressure ratio of the compressor as well as the turbine inlet temperature is important parameters that have significant influence on the exergy efficiency of the gas turbine and the desalination unit. Further the specific exergy destruction rate decreases as the performance ratio of the desalination plant increases.

The detailed performance data of a unit of the plant at different inlet temperatures of its compressor procured from company and is presented in table 2-4. The units run at 3600 rpm and the analysis of the table shows that the mean heating value of the fuel used in these units is 40800kJ/kg approximately. The catalogue performance at ISO conditions has also been included against inlet temperature of 15°C. The data reveals that the temperature at turbine exit depends on the turbine inlet temperature and the pressure ratio. A decrease of pressure from compressor exit of the order of 0.5 bars is noticed from the data tables 2-4.

**Table 2.** Variation of Thermo-Fluid-dynamic State of the Turbine with Inlet Temperature.

Compressor Inlet Temperature→ Turbine Parameters↓	10	15	20	26	31	39	50
Turbine Inlet Temperature °C	1105	1105	1105	1105	1105	1105	1105
Turbine Exit Temperature °C	540	537	534.5±53	538±44	541±49	549±44	555
Turbine Flow Rate Kg/s	244.44	244.44	239.577	241.278	241.1	240.775	244.44
Turbine Inlet Pressure bar	10.5	10.5	10.5	10.5	10.5	10.5	10.5
Turbine Exit Pressure bar	0.963	1.000	0.963	0.963	0.963	0.963	0.963
Power Produced Gross MW	69	-	56.65	57.33	59.09	60.93	58
Power Produced Net MW	68.5	85.4	56.46	56.84	58.85	60.71	57.6

Table 3. Details of Compressor with Inlet Temperature.

Compressor Inlet Temperature (°C) → Compressor Parameters ↓	10	15	20	26	31	39	50
Inlet Pressure mbar	957	991	953	957	955	928	957
Inlet pressure Loss mbar	6	9	13	7	9	7	6
Inlet Temperature °C	10	15	20	26	31	39	50
Pressure Ratio Actual	12.7	12.7	10.86	10.76	10.75	10.85	12.7
Compressor Efficiency	0.826	-	0.858	0.848	0.878	0.902	0.979
ΔT °C	336	351	334	335	336	337	321
Exit Pressure bar	12.23	12.7	10.35	10.34	10.27	10.07	12.23
Mass flow rate kg/s	236.11	292	236.11	236.11	236.11	236.11	236.11

Table 4. Thermo-Fluid-dynamic State of the Fuel.

Fuel Parameters ↓	Average Value
Fuel Pressure Inlet bar	23.09±5.7%
Fuel Inlet Temperature °C	27.25±13%
Fuel Pressure Exit bar	21.925±6%
Exit Gas Temperature °C	1105
Gas Constant R kJ/kg-K	46.805±6.9%
Lower Heating Value kcal/kg	40800±8.5%
Mol. Weight Fuel	20.2±6.3%

## 2. Mathematical Modeling of Gas Turbine Power Plant

The Basic equation that governs the power output of the gas turbine power plant is based on the application of First law of thermodynamics to the operation of its components. The Steady flow energy equation for a component of the power plant can be written as,

$$(\dot{Q}_i - \dot{Q}_o) - (\dot{W}_o - \dot{W}_i) = \dot{m}_a \left( (h_o - h_i) - \left( \frac{V_o^2 - V_i^2}{2} \right) - g(z_o - z_i) \right) \quad (1)$$

This leads to the following ideal analysis for the compressor, combustor and the turbine.

$$\text{Compressor: } \dot{W}_C = \dot{m}_a(h_{o2} - h_{o1}) = \dot{m}_a c_p (T_{o2} - T_{o1}) \quad (2)$$

$$\text{Turbine: } \dot{W}_T = \dot{m}_g(h_{o3} - h_{o4}) = \dot{m}_g c_p (T_{o3} - T_{o4}) \quad (3)$$

$$\text{Combustion Chamber: } \dot{m}_g h_{o3} - \dot{m}_a h_{o2} = \dot{m}_f \times q_{fuel} \quad (4)$$

$$\text{Where } \dot{m}_g = \dot{m}_a + \dot{m}_f \quad (5)$$

The ideal Brayton cycle is not totally reversible. It involves external reversibilities and heat transfer losses from different components. Further the accessories in a power plant also consume some power. Therefore in actual case, these expressions become.

$$\dot{W}_{net} = \dot{W}_T - \dot{W}_C - \dot{W}_{ACC} \quad (6)$$

Where,

$$\dot{W}_C = \frac{1}{\eta_c} \dot{m}_a c_p T_1 \pi_c + \dot{Q}_C \quad (7)$$

$$\dot{W}_T = \eta_T \dot{m}_g c_p T_3 \pi_T - \dot{Q}_T \quad (8)$$

In order to improve the performance of the gas turbine unit

usually second law analysis of the unit is carried out and the component where exergy loss is higher is considered for improvement of performance.

The exergy balance for a component of the gas turbine unit is given as

$$\sum \left( 1 - \frac{T_o}{T_k} \right) \dot{Q}_k - \dot{W} + \dot{m}(\Psi_i - \Psi_o) - \dot{X}_{dest} = 0 \quad (9)$$

The exergy destruction relation for a component of the gas turbine unit can be written as

$$\dot{X}_{dest} = T_a S_{gen} = T_o \left( S_o - S_i - \frac{\dot{Q}_i}{T_i} + \frac{\dot{Q}_o}{T_o} \right) \quad (10)$$

$$\dot{X}_o = \dot{m}_o(h_i - h_o - T_o(S_i - S_o)) + \left( 1 - \frac{T_o}{T_i} \right) \dot{Q}_i + \dot{W}_o \quad (11)$$

The efficiencies that are usually referred in the power plants are gross efficiency, net efficiency, isentropic efficiencies of the components and their polytropic or small stage efficiency. Further sometimes actual component efficiencies are also taken into consideration. Both the concepts of energy and exergy have been used to arrive at these efficiency definitions.

Isentropic Efficiency of Compressor,

$$\eta_1 = \eta_{ic} = \frac{T_{2s} - T_1}{T_2 - T_1} \quad (12)$$

Isentropic Efficiency of Turbine,

$$\eta_2 = \eta_{it} = \frac{T_3 - T_4}{T_3 - T_{4s}} \quad (13)$$

Compressor Efficiency (Energy Based)

$$\eta_3 = \eta_{1c} = \frac{W_{rev,inc}}{H_2 - H_1} \quad (14)$$

Compressor Efficiency (Exergy Based)

$$\eta_4 = \eta_{2c} = 1 - \frac{T_o(S_2 - S_1)}{\psi_{21}} \quad (15)$$

Thermal Efficiency of gas turbine unit,

$$\eta_5 = \eta_{cy} = \frac{\text{GrossoutputofPowerPlant}}{\text{Grossheatadded}} \quad (16)$$

Brayton cycle efficiency of gas turbine unit=

$$\eta_6 = \eta_{B1} = 1 - \frac{1}{\left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}}} \quad (17)$$

In a gas turbine power plant the compressor consumes about fifty percent of the energy that is produced by the turbine. Usually the plant accessories need about 10% of the power generated by the turbines which means the net output of the plant is only about 35 to 40 percent of turbine output. Large portion of energy is fed to the atmosphere as waste. The mass flow rate of the air through the compressor has been calculated using energy balance through the unit. Therefore mass flow rate of the air is as under.

$$m_a = \frac{1.1 * W_{net} - m_f * C_{pg} * (T_3 - T_4)}{C_{pg} * (T_3 - T_4) - C_{pa} * (T_2 - T_1)} \quad (18)$$

The analysis as given above and the considerations of axial velocity, tip speed and compressor inlet temperature lead to mass flow rate of air to be around 193 kg/s and fuel rate of 8.333 kg/s, that results in a fuel air ratio of 0.04317 or air fuel ratio of 23.16. This meets the criteria of Fields and Topps [9]. Saravanamuttoo, et al, [19] has opined that the fuel air ratio is a function of combustion chamber temperature increase and the inlet temperature of the air to the combustion chamber. These equations were solved uses EES and Excel programs.

### 3. Results

The variation of energy or first law efficiency of turbine and compressor of the unit with change in the inlet temperature is presented in the figure2. The energy efficiency of compressor decreases with the increase in compressor inlet temperature whereas turbine efficiency shows an increase with the increase in the inlet temperature. The decrease in the energy efficiency is caused due to decrease in the mass flux through the gas turbine that increase the power input to the compressor and hence its causes its efficiency to decrease . Figure 3 presents the variation of exergy or second law efficiency of the compressor and the turbine. The turbine exergy efficiency is seen to increase with the increase in the compressor inlet temperature whereas turbine exergy efficiencies decrease with the increase in compressor inlet temperature. The thermal and Brayton cycle efficiency of the gas turbine unit (figure 4) decrease with the increase in the compressor inlet temperature.

Figure5also depicts that compressor work input, turbine work output and network output decrease with the increase in the inlet temperature of the compressor. This forms the basic reason for the change occurring in changes in the efficiency. Figure 6 shows the energy lost at the exit of the turbine. A huge amount of energy is lost to the atmosphere at the exit of a

simple gas turbine. This loss of energy increases with the increase in the inlet temperature of the compressor. This is due to the fact that the exit temperature of the turbine increases with the increase in the inlet temperature of the compressor as shown in the table 2.

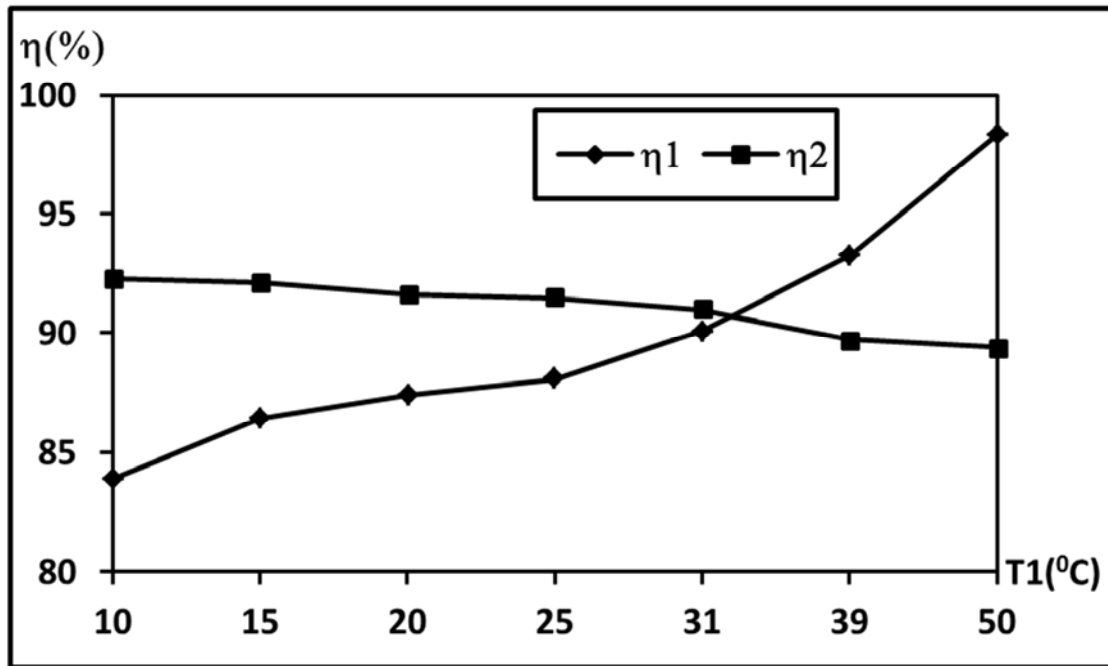
Dincer, et al. [20] have formulated the methodology of obtaining exergy efficiency of Brayton, Rankine, and Otto cycles. A comparison of the current results with the results of Dincer, et al [20] was carried out. Figure 7 shows the variation of exergy efficiency of the unit with compressor inlet temperature for three turbine inlet temperatures and compressor pressure ratios of 12. The variation shows a good agreement with the results of Dincer, et al. [20] at all the turbine inlet temperatures and pressure ratios. The result depicts that the exergy efficiency increases with the increase in turbine inlet temperature and decreases with both the compressor pressure ratio and the compressor inlet temperature. These results depict the relation of exergy efficiency with the work output performance which decreases with the increase in the compressor inlet temperature as has been shown in the results of the actual gas turbine units. Abam and Moses [21] carried out exergy analysis of a 33 MW gas turbine power plant using a computer simulation. Their results showed a remarkable dependence of exergy flow of the power output, exergy efficiency, exergy destruction, heat to power ratio and specific fuel consumption on the compressor inlet temperature and turbine inlet temperature. Figure 8 shows the variations of energy efficiency of compressor, combustion chamber, turbine and unit with the increase in the compressor inlet temperature. The variation shows that all the efficiencies are in full agreement with results of Abam and Moses [21]. The variation of exergy destruction in the various components of the gas turbine unit with compressor inlet temperature is shown in figure 9. The variation depicts that with the increase in compressor inlet temperature the exergy destruction decreases in case of the compressor and turbine whereas it increases in the combustion chamber. The decrease of exergy destruction is a sign of less use of the available energy which results in less turbine output and more power input of the compressor as in observed in figure 5. With higher compressor inlet temperature less energy is needed for the complete combustion which is depicted in the increase in the exergy destruction in the combustion chamber. The contribution of exergy destruction to the overall energy destruction of the unit is shown in figure 10. It shows that the bulk of energy destruction is from the combustion chamber whereas compressor and the turbine contribute to a nearly the same amount. Table 5 shows the exergy destruction rates corresponding to each component of the gas turbine at compressor inlet temperature of 20°C. One can see the rate of exergy destroyed in the combustor chamber is the highest and represents about 79.51% of the whole exergy destruction rates. The compressor and turbine have the lesser rates of irreversibility with respectively 12.77% and 7.72%. Note that the specific exergy of point 1 includes the compressor work in addition to the fluid flow

exergy. About 28% of the total exergy destroyed within the gas turbine is the available energy of the exhausted gases

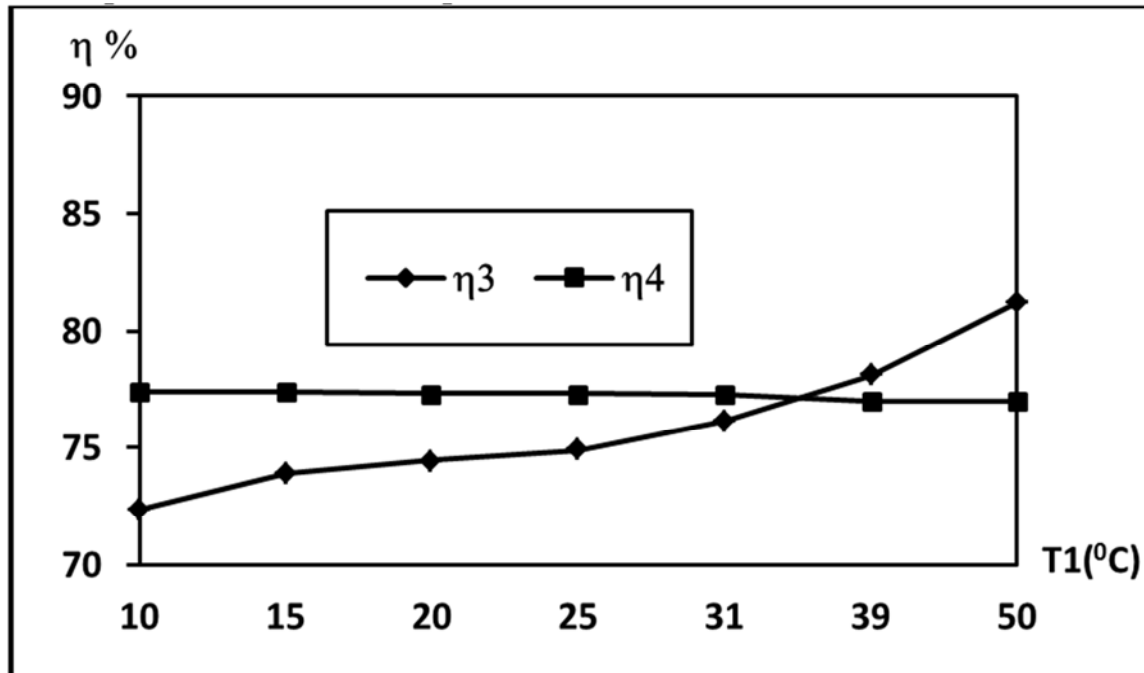
leaving the turbine. It represents a huge amount of energy that should be recovered.

*Table 5. Exergy destruction distribution gas turbine unit at 20°C.*

Point	T (°C)	P (kPa)	h (kJ/kg)	ex (kJ/kg)	Specific exergy destruction (kJ/kg)	Exergy destruction (kW)	Exergy efficiency (%)
1	25.4	96.4	299	343.2			
2	364	1044	646.5	1069	27.16	4888.8	12.77
3	1105	1044	1489	899.4	169.1	30438	79.51
4	524	96.4	819.1	883	16.41	2953.8	7.72
Total exergy destruction rate				3194.6	212.67	38280.6	100



*Figure 2. Variation of Energy Efficiency ( $\eta_1$  and  $\eta_2$ ) of Compressor and ( $\eta_3$  and  $\eta_4$ ) of Turbine with Compressor Inlet Temperatures.*



*Figure 3. Variation of Exergy Efficiency of compressor ( $\eta_3$ ) and of Turbine ( $\eta_4$ ) with Compressor Inlet Temperatures.*

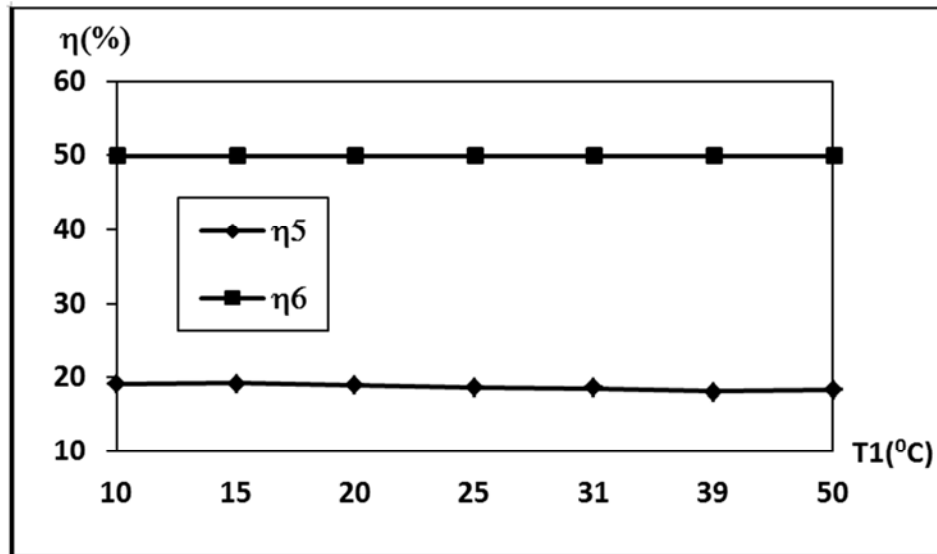


Figure 4. Variation of Thermal Efficiency ( $\eta_5$ ) and Brayton Cycle Efficiency ( $\eta_6$ ) of Gas Turbine Unit with Compressor Inlet Temperatures.

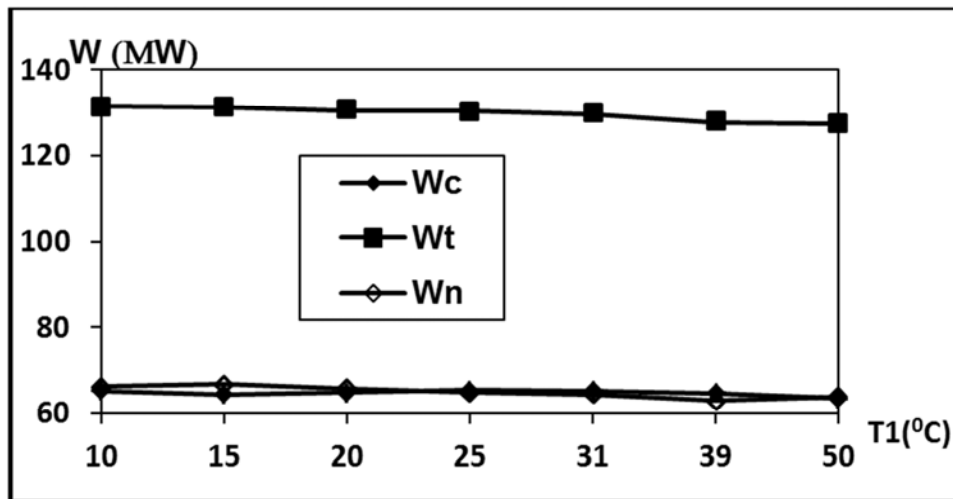


Figure 5. Variation of Power Output from the Turbine, Power input to the compressor, Net Work output with the Inlet Temperature of Compressor.

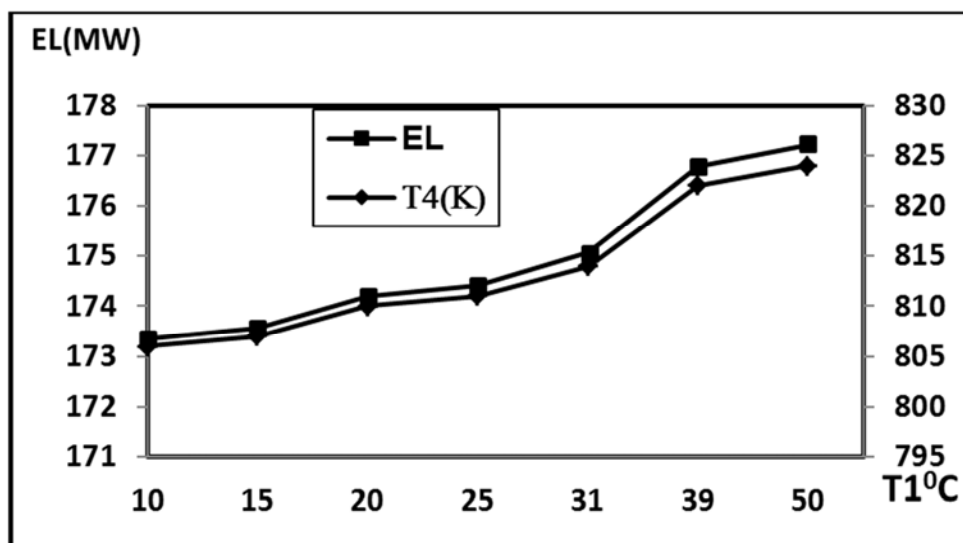


Figure 6. Variation of the Exit Energy Loss and Turbine Exit Temperature (K) with different Compressor Inlet Temperature.

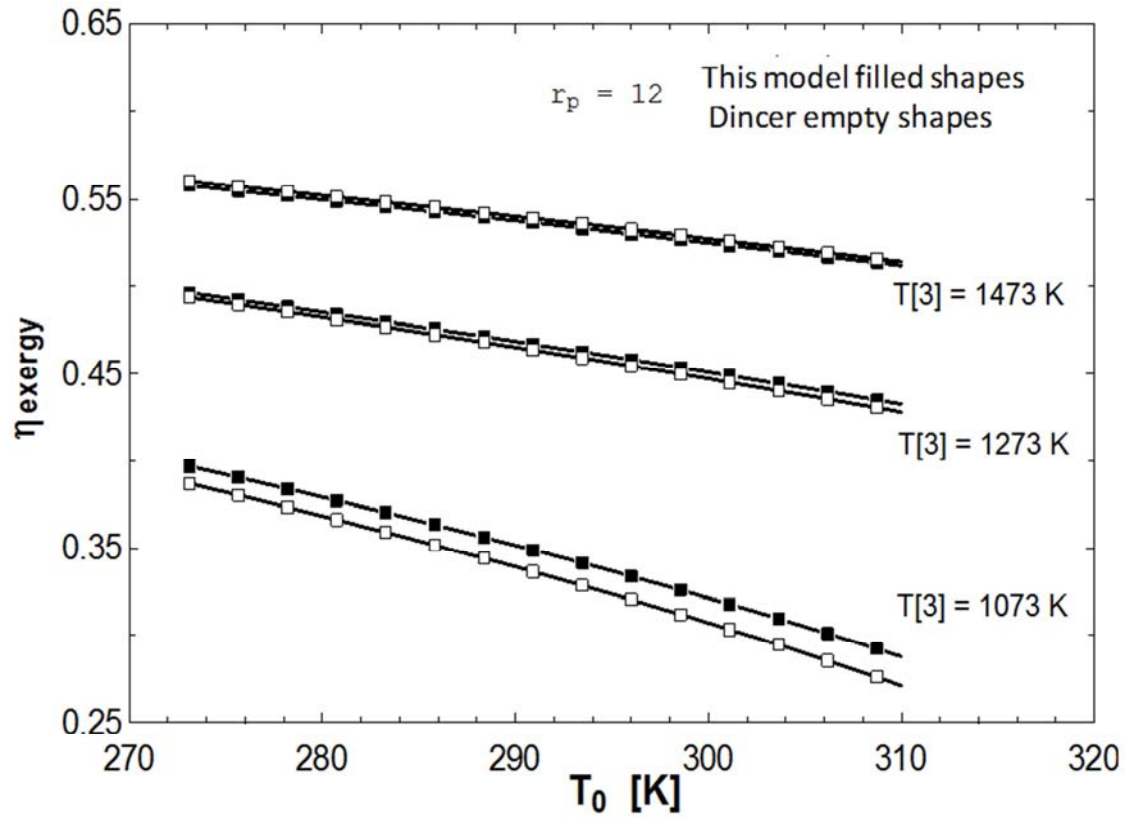


Figure 7. Variation of Exergy Efficiency of the unit with Compressor Inlet Temperature and Turbine Inlet Temperatures.

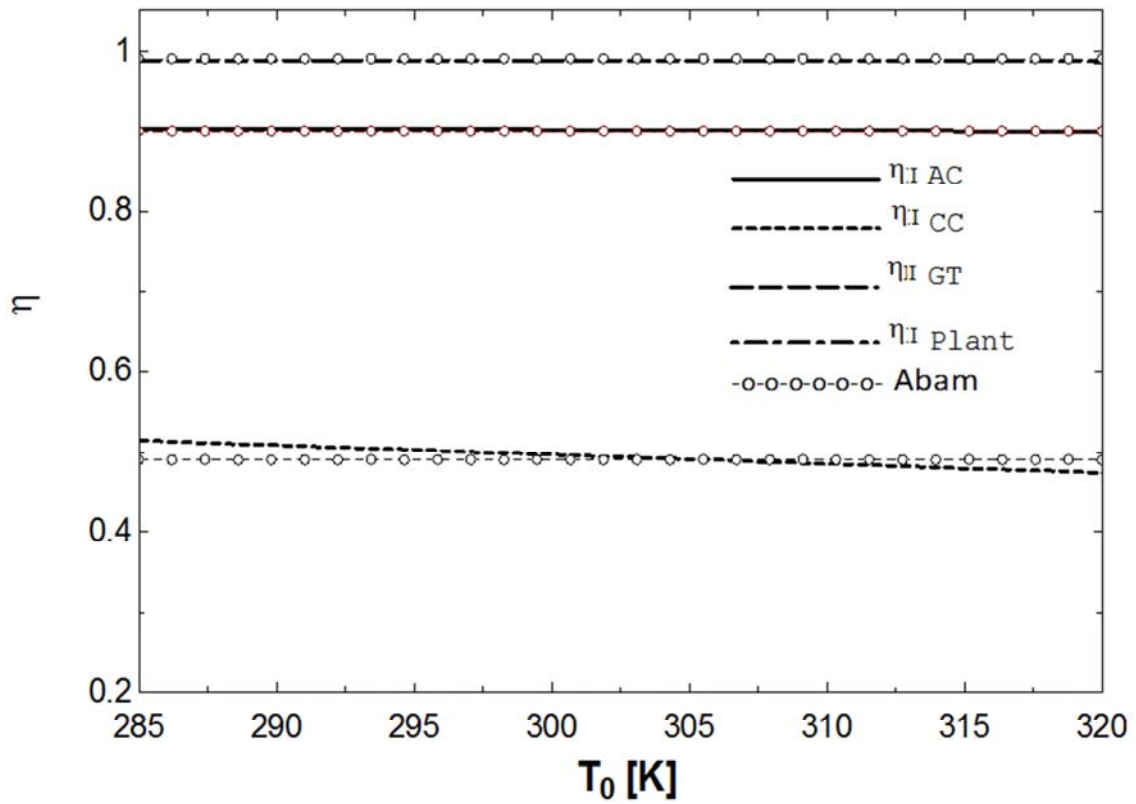


Figure 8. Variation of Efficiencies of Compressor, Combustor, Turbine and Unit with Compressor Inlet Temperature.

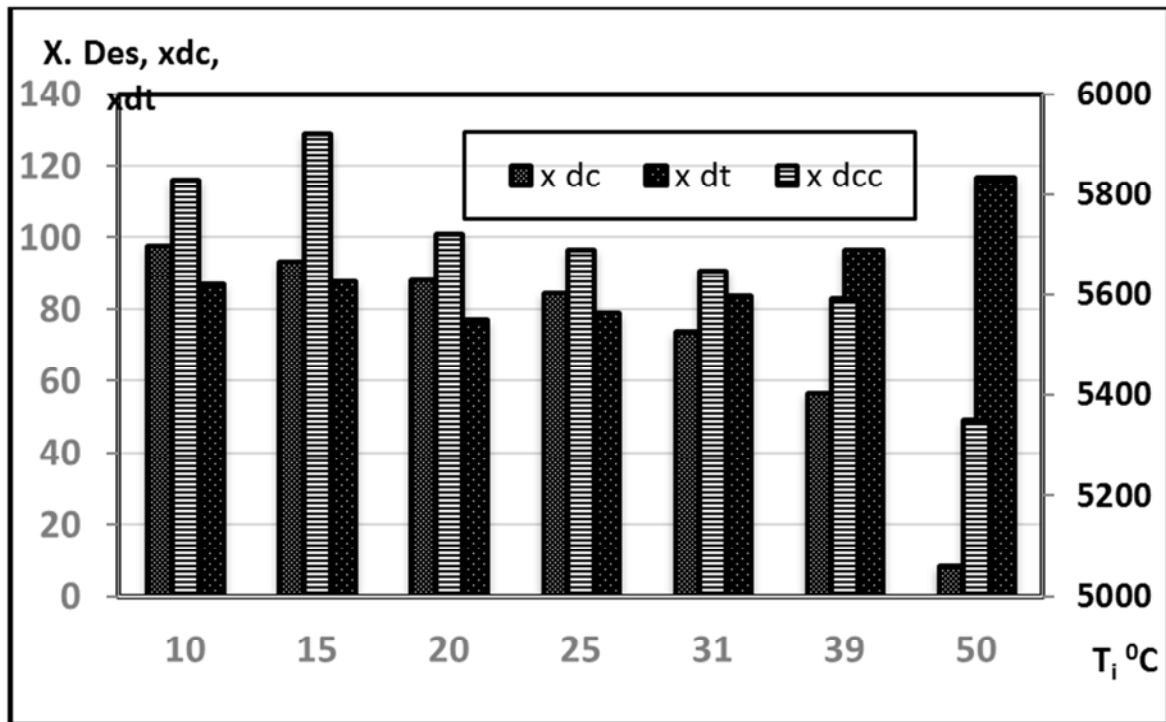


Figure 9. Variation of Exergy Destruction in Compressor, Combustor and Turbine with Compressor Inlet Temperature.

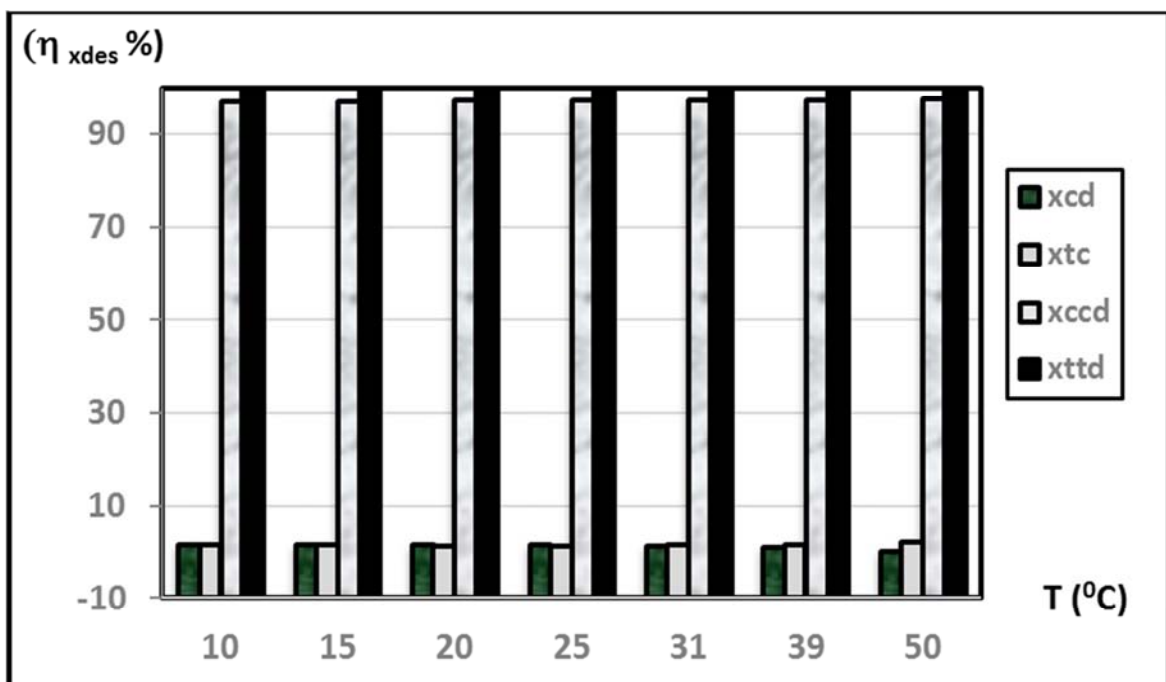


Figure 10. Variation of Exergy Destruction Efficiency in Compressor, Combustor and Turbine and Gas Turbine unit with Compressor Inlet Temperature.

## 4. Conclusions

A detailed study of the performance of a gas turbine unit has been carried out. The main emphasis of the study was the changes in energy and exergy efficiency of the unit and its components with the change in the temperature at the inlet of the compressor. The study concludes that the energy efficiency of compressor decreases whereas turbine

efficiency increases with the increase in compressor inlet temperature. The exergy efficiency of compressor increases whereas turbine efficiency decreases with in compressor inlet temperature. It has been found that the thermal efficiency and Brayton cycle efficiency of a gas turbine unit decrease with the increase in compressor inlet temperature. The network output, gross work output and compressor work input decreases with increase in the compressor inlet temperature. The turbine exit temperature increases with the increase in



the temperature at the inlet of the compressor. Hence more energy is lost to the atmosphere at higher compressor inlet temperature and further the exergy destruction in the compressor and turbine decrease with the increase in the compressor inlet temperature.

## Nomenclature

$C_{pa}$	Average specific heat at constant pressure for air, kJ/kg K
$C_{pg}$	Average specific heat at constant pressure for gases, kJ/kg K
$\chi_{Des}$	Exergy destruction rate, kW
$\dot{\chi}$	Exergy rate, kW
$h$	Enthalpy, kJ/kg
$h_0$	Stagnation value of enthalpy, kJ/kg
$k$	Ratio of specific heats
$\dot{m}$	Mass flow rate, kg/s
$\dot{m}_a$	Mass flow rate of air, kg/s
$\dot{m}_f$	Mass flow rate of fuel, kg/s
$\dot{m}_g$	Mass flow rate of gas, kg/s
$P$	Pressure, kPa
$\dot{Q}$	Rejected heat transfer rate in the condenser, kW
$\dot{Q}_C$	Heat transfer rate in the compressor, kW
$\dot{Q}_{in}$	Heat transfer rate in to the component, kW
$\dot{Q}_{out}$	Heat transfer rate out of the component, kW
$\dot{Q}_T$	Heat transfer rate in the turbine, kW
$q_i$	Heat transfer into component, kW
$q_{fuel}$	Input heat transfer rate in the combustion chamber, kW
$q_o$	Heat transfer from the component, kW
$s$	Entropy, kJ/kg
$T$	Temperature, °C
$T_k$	Hot reservoir temperature, °C
$T_o$	Dead state/ambient temperature, °C
$V$	Velocity, m/s
$V_{in}$	Velocity at the entrance of a component, m/s
$V_{out}$	Velocity at the exit of a component, m/s
$\dot{W}_{ACC}$	Net power of the gas turbine accessories, kW
$\dot{W}_C$	Net power of the Compressor, kW
$\dot{W}_{in}$	Power input to the component kW
$\dot{W}_{out}$	Power output from a component kW
$\dot{W}_{Net}$	Net work of the turbine, kW
$\dot{W}_T$	Net work of the turbine, kW
$Z$	Elevation, m
$Z_{in}$	Elevation at entrance to a component, m
$Z_{out}$	Elevation at the exit of a component, m

## Greek Letters

$\eta$	efficiency
$i$	inlet location for a component
$o$	Outlet location for a component
1	Compressor inlet
2	Compressor outlet
3	Turbine inlet
4	Turbine outlet

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