THERMODYNAMIC PERFORMANCE OF A 285 MW GAS TURBINE UNIT UNDER VARYING OPERATING CONDITIONS

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الملخص

الأداء الديناميكي الحراري لوحدة التربين الغازي يعتمد أساسا على حالة المحيط الجوي والحمل الواقع على التربين والفرق بين القدرة الحقيقية المنتجة منها والقدرة التصميمية ويظهر هذا الاعتماد عندما تعمل الوحدة في أماكن تكون حالة المحيط الجوي تختلف عن حالة المحيط المشترطة من منظمة المعايير العالمية، أو في حالة التشغيل الجزئي حيث تكون القدرة المنتجة الحقيقية والكفاءة الحرارية مختلفة كثيرا عن القيم التصميمية. تهدف هذه الورقة إلى دراسة تأثير تغير حالة المحيط الجوي من درجة الحرارة والرطوبة النسبية على الأداء الديناميكي الحراري لوحدة التربين الغازي عند أحمال مختلفة. من خلال دراسة أداء وحدة تربين غازي بقدرة مركبة 285 MW واستخدامه. معادلات اتزان الكتلة والطاقة مع مواصفات وحدة التربين الغازي استخدمت لبناء النموذج الديناميكي الحراري الذي تم على أساسه تصميم برنامج حاسوب واستخدامه للتحقق من الأداء للوحدة تحت حالات تشغيل متغيرة للمحيط الجوي والحمل. حيث تم تغير درجة حرارة المحيط من صفر إلى 50 °C والرطوبة النسبية من صفر إلى 100% وحمل وحدة التربين من 25 % إلى 100% مع ثبوت درجة حرارة الغازات الداخلة للتربين عند 1242 °C. ومن بين النتائج المتحصل عليها أتضح أنه عند زيادة درجة حرارة المحيط الجوي درجة مئوية واحدة (1°C) تؤدي إلى انخفاض 0.53 % في القدرة المنتجة و 0.18 % في الكفاءة الحرارية وزيادة بنسبة 0.195 % في استهلاك الوقود النوعي. أما فيما يتعلق بتأثير الرطوبة النسبية فأتضح أن لها تأثير مهم على القدرة المنتجة ولكن ليس لها تأثير مهم على الكفاءة الحرارية. حيث استنتج انه في درجة حرارة المحيط الجوي 40 °C كانت الزيادة 4.15 % في القدرة المنتجة عندما ترتفع قيمة الرطوبة النسبية من صفر إلى 100%. أخيرا، تم التحقق من دقة الحسابات وذلك بمقارنتها بنتائج المصنع. أستنتج من هذه الدراسة بان درجة حرارة المحيط لها تأثير مهم على الأداء الديناميكي الحراري لوحدة التربين الغازي عند الحمل الجزئي والكامل ولهذا دعم وحدة التربين الغازي بتركيب منظومة تبريد لهواء الدخول يجب أن يؤخذ في الاعتبار.

ABSTRACT

The thermodynamic performance of a gas turbine unit mainly depends on the ambient and loading conditions. The difference between the actual power generated by a gas turbine and the design rated power is observed whenever it operates at site ambient conditions that vary from the stipulated ISO conditions or at part-loading operation. Consequently, its actual power output and thermal efficiency are different from its rated design values.

This paper is devoted to study the effects of ambient conditions variations, such as ambient temperature and relative humidity on gas turbine unit performance at different loads. For this work, the power plant chosen is a gas turbine unit of 285 MW installed capacity. Mass and energy balance equations with typical specifications of the selected gas turbine unit have been used to develop the thermodynamic model. Based on this model, a computer program has been established and used to investigate the performance of the unit under varying ambient and loading conditions. These include ambient temperature, ranging from zero to 50 °C, relative humidity, ranging from zero to 100%,

gas turbine load ranging from 25% to 100% with turbine inlet temperature fixed at 1242°C.

The results obtained showed that a degree (1°C) increase in ambient temperature, leads to 0.53% decrease in the net power output, 0.18 % decrease in the thermal efficiency and 0.195 % increase in specific fuel consumption. The relative humidity has a remarkable influence on net power output but almost none on thermal efficiency. At 40°C ambient temperature, the effect of relative humidity produces an increase of 4.15% in the net power output when the RH increases from zero to 100%. Finally, the calculated results were validated against the actual gas turbine unit data provided by the manufacturer. In conclusion, the effect of ambient temperature on the thermodynamic performance of gas turbine power plants will be significant at full and part loads. Therefore, the incorporation of the existing gas turbine plant with inlet air cooling technologies should be considered seriously.

KEYWORDS: Gas Turbine Performance; Ambient Effect; Part-load; Power Production.

INTRODUCTION

Gas turbine (GT) power plants are widely used in Libya and other countries practically during the peak demands and in inland regions. Gas turbines become very popular prime-over, due to attractive properties like high power to weight ratio, compactness and ease of installation [1]. Libya is a country, which has large daily and seasonal ambient condition variations. The weather in far deserts (hot and dry) is different from that of coastal regions in which relative humidity (RH) is considerably higher. Therefore, electricity demand varies considerably from day to night, from summer to winter and from north to south. In Libya, the modern GT power plants are being installed and are used to produce electrical power to cover the peak load demand and in inland regions. They produce about 60% of the total capacity of power generation in Libya [2]. These power plants are used single or integrated with steam power plants to form combined cycle plants [2].

The GT plant power output is rated by the International Standards Organization (ISO), which specified the reference ambient air conditions as: 15 °C, 1.013 bar and RH of 60% [3]. A GT often operates at off-design conditions due to change of ambient conditions or load demand. GTs are constant volume machines, at a given shaft speed; they always move the same volume of air. In GTs, since the combustion air is taken directly from the environment, their performance is strongly affected by ambient conditions [1,3,4]. Therefore, when evaluating the overall performance of the GT, it is important to account for all operating conditions that can be encountered. The power output of a GT depends on the cycle pressure ratios and mass flow-rate of air through it. This is the reason why on hot humid days, when air is less dense, the mass flow-rate of air and cycle pressure ratios decreased, power output falls off. Power output can decrease by as much as 30%, with respect to ISO design conditions, when ambient air temperature (AAT) reaches 45 °C [3,4].

The effect of ambient conditions on the performance of GTs has been discussed in various studies. De Sa et al. [5] considered specific GTs installed at the Dewha city (Dudi) power station. They investigated units performance at various AATs and concluded that for every 1 °C rise in AAT above ISO conditions the units loss 0.1% in thermal efficiency and 1.47MW of useful power output. Ibrahim *at al.* [6] tested a simple gas cycle in the central of Saudi Arabia. They reported that a high AAT during summer can cause a 24%

decrease in GT power plant capacity. Ameri and Hejazi [7] have investigated many GTs units installed in Iran; they concluded that for every 1 °C increase in AAT, the power output and the air mass flow rate would decrease by 0.74% and 0.36% respectively. Jaber *at al.* [8] have reported that a drop of about 25% of the rated power capacity of the GT from that at ISO conditions when AAT reached 40 °C. Deng-chern *et al.* [9] have also reported, in a study dealing with a GT performance at varying ambient conditions that the power output decreases by 18% as the inlet air temperature increases from 15 to 30 °C. Kam and priddy [10] have found that the air mass flow-rate in most GTs drops by about 12% as the AT increases from -18 to 15 °C. Zadpoor et al. [11] investigated the performance of a 16.6 MW gas turbine power plant, the power output of the plant increased by 11.3% with decreasing the AT from 34.2 °C to ISO design condition. There is 0.74% decrease in the power output for increment of each 1 °C ambient air temperature. Alhazmy *at al.* [12] evaluated the performance of ABB-11D5 GT operating in hot and humid ambient conditions of Jeddah city of the Saudi Arabia. They reported that, the average power output of the power plant increases by 0.57% for each 1 °C drop in AT. The effects of AAT and RH on the performance of GT have been reported by several authors as Hadik [13], Bassam [14] and Abam *et a.l* [15]. They concluded that, AAT has the greatest effect on GT power output and thermal efficiency, compared with the effects of RH, which has a negligible effect on both power output and thermal efficiency, especially at low ambient air temperatures. Fellah [16] evaluated the performance of a 500 MW combined cycle power plant located in Libya. He reported that a drop of about 20% of the rated power capacity of the combined cycle that at ISO conditions when the AAT reached 40 °C.

Although sufficient literature is available on this topic, the effects of the ambient conditions on the performance of GT with considering the changes on the pressure ratio, pressure losses and efficiency at each GT cycle component has not been studied widely. Therefore, the objective of the present work was to develop a thermodynamic model, which takes into account the effect of ambient and loading conditions on the performance of a typical GT, with considering the changes on the characteristics of the components of the GT cycle. The effects of ambient and loading conditions variations on the GT cycle performance (power output, overall efficiency, specific fuel consumption and heat rate) would be studied and discussed.

POWER PLANT DESCRIPTION

The gas turbine power plant used as a case study for this research is located in Sarir power station, in Libya. Sarir power station consists of three units Siemens gas turbines, type SGT5-PAC 4000F and each unit with 285 MW rated capacity [17]. Generally the GT power plants consist of four major components namely the air compressor (AC), combustion chamber (CC), turbine and electrical generator (EG), as shown in Figure (1).

According to the basic principle of the GT operates on the Brayton cycle, air entering the AC at point 1 is compressed to some higher pressure and the compression raises the air temperature at point 2. Upon leaving the AC, air enters the CC at point 2, where fuel is injected at point 5 and combustion occurs. The combustion process occurs at essentially constant pressure. Although high local temperatures are reached within the primary combustion zone, the combustion system is designed to provide mixing, burning, dilution and cooling. Thus, by the time the combustion leaves the CC and enters the turbine at point 3, it is at a mixed average temperature. Hot gases leaving the CC are expanded in the GT from point 3 to 4 thereby producing mechanical power and finally discharged to the atmosphere. A portion of mechanical power produced in the GT is used to drive the AC and the balance is converted into electrical power. The variable inlet guide vanes (IGVs) percentage angle opening and fuel injected into the CC, are regulated to control the required maximum and exhaust gas temperatures at part-load operation. The design data for GT power plant selected in this research are summarized in Table (1).

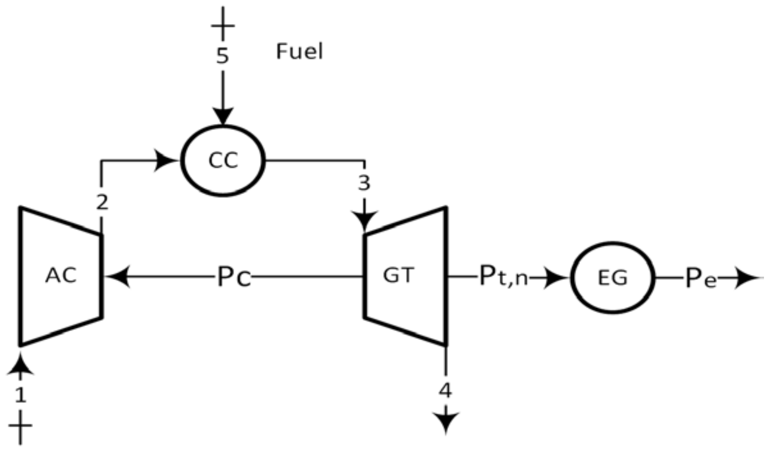


Figure 1: Schematic diagram of the selected gas turbine unit.

Table 1: Gas Turbine Design Parameters [17].

Component	Parameter	Value	Units
Air Compressor	Ambient inlet temperature: T_1	15.0	℃
	Relative humidity: ϕ	60.0	%
	Inlet pressure: p_1	1.013	bar
	Pressure ratio: π_c	17.5	[-]
	Mass flow-rate of air: $\dot{m}_1$	672	kg/s
	Compression efficiency: η_C	88.8	%
	Mechanical efficiency: η_m	98.5	%
	Number of stages	15	[-]
	Speed	3000	rpm
Combustion Chamber	Pressure loss: Δp_{CC}	0.3	bar
	Outlet temperature: T_3	1242	°C
	Specific heat of fuel: Cp_f	2.25	kJ/kg.°C
	Gas constant of fuel: R_f	0.52	MJ/kg
	Lower heat value (NG): LHV	45.784	kJ/kg.°C
	Efficiency: η_{CC}	88.0	%
	Number of burners	24	[-]
Gas turbine	Inlet pressure: p_3	17.43	bar
	Outlet pressure: p_4	1.023	bar
	Expansion efficiency: η_t	91.0	%
	Mechanical efficiency: η_m	98.5	%
	Electrical generator efficiency: η_G	98.4	%
	Number of stages	4	[-]
	Speed	3000	rpm

THERMODYNAMIC MODELING

The performance of an existing gas turbine unit can be predicted rather accurate by matching the characteristics of the compressor and the turbine coupled with a heat balance of the combustor. The characteristics of compressors and turbines are rarely available for people outside the field of manufacturing. In order to make performance predictions one often has to rely on simplified models. In the following such a simplified model is presented. The model rely on knowing a point of operation, typically the design point. The model described in the following uses the principle of relating on two operating conditions by applying simple laws of physics [1,17,18]. The modeling of GT power plant is presented in the following steps.

Basic Assumptions

The following assumptions were made in full and part loads modeling:

- Only temperature and relative humidity are considered for ambient condition.
- Fixed TIT at 1242°C, and it is kept constant during the analysis.
- The fuel injected to the CC is assumed to be natural gas.
- The principle of ideal gas mixture was applied for the humid-air and combustion products.
- The part-load control methods are the control of air mass flow-rate at AC inlet through IGVs associated with the control of the fuel mass flow-rate, to maintain the required TIT.
- Single-shaft with constant operation speed.
- Neglecting the auxiliary electrical power.

The thermodynamic cycle upon which all GTs operate is called the Brayton cycle. Figure (2) shows the temperature-entropy diagram for this cycle. Solid and dashed lines represent the ideal and actual processes, respectively. The governing equations are:

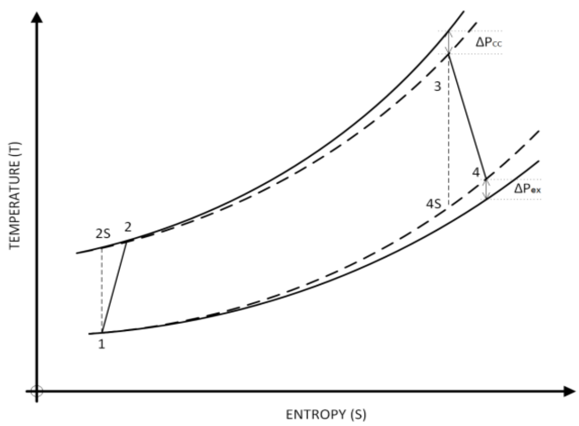


Figure 2: Temperature-entropy diagram for the gas turbine cycle.

Air Compressor Model

Variable inlet guide vanes operation of the compressor enables reduction of the air mass flow at part loads. Using the relations for ideal gas and knowing the air inlet temperature, pressure ratio (π_c) and compressor efficiency (η_c), the outlet temperature (T_2) can be calculated from:

$$T_2 = T_1 \times \left[1 + \frac{\pi_c^{\left(\frac{\gamma_h - 1}{\gamma_h}\right)} - 1}{\eta_c}\right] \tag{1}$$

The compressor efficiency η_c at varying operating conditions is approximated by equation as a function of compressor pressure ratio π_c [4]:

$$\eta_c = \eta_{c,r} \times [1 - 0.23 \times \left(1 - \frac{\pi_c - 1}{\pi_{c,r} - 1}\right)^2] \quad (2)$$

The reference condition with index (r) is considered as the design condition in this study. The pressure ratio π_c across the compressor is given as:

$$\pi_c = p_2/p_1 \quad (3)$$

The isentropic exponent γ_h for humid-air is given as;

$$\gamma_h = Cp_h/(Cp_h - R_h) \quad (4)$$

where, Cp_h is mean specific heat of humid-air, determined as a function of the average temperature (T) across the compressor, and calculated as following [13];

$$Cp_h = Cp_a + \omega \times Cp_v \quad (5)$$

where Cp_a and Cp_v are the mean specific heats of dry-air and vapor respectively, which considered in this study as a temperature variable function and can be fitted by equations (6) and (7) for the range of 200 K < T < 800 K [18]:

$$Cp_a = \left[28.11 + \frac{1.967 \times T}{10^3} + \frac{4.802 \times T^2}{10^6} - \frac{1.966 \times T^3}{10^9}\right]/28.97 \quad (6)$$

$$Cp_v = \left[32.24 + \frac{1.923 \times T}{10^3} + \frac{1.055 \times T^2}{10^5} - \frac{4.187 \times T^3}{10^9}\right]/18.015 \quad (7)$$

the values: 28.97 and 18.015 are the molecular weights of dry-air and vapor respectively.

The gas constant (R_h) of humid-air can be calculated as:

$$R_h = \frac{8.3143}{MMW} \quad (8)$$

where, MMW is the molecular weight of humid-air, and is given by:

$$MMW = \frac{1}{\left(\frac{VMF}{18.015}\right) + \left(\frac{AMF}{28.97}\right)} \quad (9)$$

VMF is the vapor mass fraction, and is given by:

$$VMF = \frac{\omega}{(1+\omega)} \quad (10)$$

and, AMF is the dry-air mass fraction, and is given by:

$$AMF = 1 - VMF \quad (11)$$

The specific humidity (ω) is defined as the ratio of vapor mass to dry-air mass:

$$\omega = 0.622 \times \frac{p_V}{p_a} = \frac{\phi \times p_{sat}}{(p_h - p_v)} \qquad (kg_v/kg_a) \quad (12)$$

$$p_V = \phi \times p_{sat} \quad (13)$$

Where, p_h is the total pressure of humid- air, p_{sat} is the saturation pressure at the temperature of the humid- air (T_a=T_v=T_1), p_v and p_a are the partial pressures of dry-air and vapor in the humid-air respectively, and ϕ is the relative humidity of the humid-air.

The compressor power (P_c) working with humid-air between points 1 and 2 is calculated from mass flow-rate of humid-air and enthalpy change across the compressor [19, 20]:

$$P_c = \frac{1}{\eta_c} \times [\dot{m}_a \times Cp_a \times (T_2 - T_1) + \dot{m}_v \times (h_{v,2} - h_{v,1})] \quad (14)$$

where, $\dot{m}_a$ is the dry-air mass flow-rate and for a given operating condition it is equals to:

$$\dot{m}_a = \frac{V_r \times \rho_h}{(1+\omega)} \tag{15}$$

V_r is the volume flow-rate of humid-air at reference condition, and ρ_h is the density of humid-air and it is a function of inlet ambient conditions. Using the ideal gas relations, they can be evaluated as following [18]:

$$V_r = \frac{\dot{m}_{h,r} \times R_{h,r} \times T_{a,r}}{p_{h,r}} \tag{16}$$

$$\rho_h = \frac{p_h}{R_h \times T_h} \tag{17}$$

The mass flow-rate of water vapor $\dot{m}_v$ in the humid-air is calculated by:

$$\dot{m}_v = \dot{m}_a \times \omega \tag{18}$$

The total mass ($\dot{m}_h$) of working fluid (humid-air) flowing through the compressor equal to:

$$\dot{m}_h = \dot{m}_a + \dot{m}_v = \dot{m}_a \times (1+\omega) \tag{19}$$

h_{v1} and h_{v2} in equation (14) are the enthalpies of water vapor at inlet and outlet of the compressor, respectively, and evaluated approximately as in equation (20) [19]:

$$h_{V,i} = 2501.3 + 1.8723 \times T_i \tag{20}$$

Where (i) refers to state points 1 , 2 or 3, and T is the temperature of vapor in °C.

Combustion Chamber Model

In the CC, inlet fluids are the humid-air coming from the compressor and the fuel added for the combustion process. The exit fluids are the flue gas and water vapor (combustion products). The basic principle of operation of a CC is based on the energy balance principle. The energy balance on the CC gives [20]:

$$\dot{m}_a(h_2 - h_1) + \dot{m}_v h_{v2} + \dot{m}_f LHV \eta_{cc} + \dot{m}_f h_f = (\dot{m}_a + \dot{m}_f)(h_3 - h_1) + \dot{m}_v h_{v3} \tag{21}$$

after manipulating and dividing equation (21) by the mass flow-rate of dry-air ($\dot{m}_a$), the fuel mass flow-rate ($\dot{m}_f$) is expressed as:

$$\dot{m}_f = \frac{\dot{m}_a \cdot (Cp_g\, T_3 - Cp_a T_2) + \dot{m}_v(\mathrm{h}_{v3} - \mathrm{h}_{v2})}{(LHV \eta_{cc} + Cp_f T_f - Cp_g(T_3 - T_1))} \tag{22}$$

Hence the total heat supplied by fuel (Q_f) in the CC is given by:

$$Q_f = \dot{m}_f \times LHV \tag{23}$$

where, η_{CC} is the combustor efficiency, h_{w3} is the enthalpy of saturated vapor at the outlet of the CC and Cp_g is mean specific heat of combustion gases, determined as a function of the average temperature (T) across the turbine, and calculated as following [20];

$$Cp_g = \left[\left(\frac{9.91615}{10}\right) + \left(\frac{6.99703 \times T}{10^5}\right) + \left(\frac{2.7129 \times T^2}{10^7}\right) - \left(\frac{1.22442 \times T^3}{10^{10}}\right)\right] \tag{24}$$

The losses inside the combustion chamber, which arise due pressure drop is taken into account and estimated by introducing the following expression [1]:

$$\Delta p_{cc} = \Delta p_{cc,r} \times \left(\frac{\dot{m}_3}{\dot{m}_{3,r}}\right)^{1.8} \times \left(\frac{T_3 \times p_{3,r}}{T_{3,r} \times p_3}\right)^{0.8} \tag{25}$$

where $\dot{m}_3$ is the total mass flow-rate of combustion products entering to GT and is given by:

$$\dot{m}_3 = \dot{m}_4 = \dot{m}_a + \dot{m}_v + \dot{m}_f \quad (26)$$

Turbine Model

The flue gases temperature T_4 at turbine outlet can be written as [18]:

$$T_4 = T_3 \times \left[1 - \eta_t \times \left(1 - \frac{1}{\pi_t^{\left(\frac{\gamma_g - 1}{\gamma_g}\right)}}\right)\right] \quad (27)$$

where, π_t is the turbine expansion ratio and defined as:

$$\pi_t = p_3/p_4 \quad (28)$$

and the turbine efficiency η_t at varying operating conditions is approximated by equation as a function of turbine pressure expansion ratio π_c [4]:

$$\eta_t = \eta_{t,r} \times [1 - 0.015 \times \left(1 - \frac{(1/\pi_t) - 1}{(1/\pi_{t,r}) - 1}\right)^2] \quad (29)$$

The isentropic exponent γ_g for flue gases is given as;

$$\gamma_g = Cp_g/(Cp_g - R_g) \quad (30)$$

The gas constant of the flue gases (R_g) can be obtained using the following expression [22]:

$$R_g = 287.05 + 212.85 \times 1/AFR - 197.89 \times (1/AFR)^2 \quad (31)$$

where AFR is the air to fuel ratio. For the part-load analysis, the turbine inlet pressure (p_3) for considered choked condition is calculated from an improved Flugel formula as following [23]:

$$p_3 = \left[\sqrt{\left[\frac{\left(\frac{\dot{m}_3}{\dot{m}_{3,r}}\right)}{\left(\frac{T_{3,r}}{T_3}\right)^{0.5}}\right]^2 \times \left(\pi_{t,r}^2 - 1\right) + 1}\right] \times p_4 \quad (32)$$

and, the compressor outlet pressure (p_2) is given by:

$$p_2 = p_3 + \Delta p_{cc} \quad (33)$$

The total output mechanical power of the GT (P_t) is expressed as:

$$P_t = (\dot{m}_a + \dot{m}_v + \dot{m}_f) \times Cp_g \times (T_4 - T_3) \times \eta_m \quad (34)$$

Hence the net mechanical power output ($P_{t,n}$) of the GT is:

$$P_{t,n} = P_t - P_c \quad (35)$$

The net electrical power output (P_e) produced from GT is equal to:

$$P_e = P_{t,n} \times \eta_G \quad (36)$$

Plant Overall Performance

The overall energy efficiency of the GT power plant (η_{GT}):

$$\eta_{GT} = \left(\frac{P_{t,n}}{Q_f}\right) \times 100\% \quad (37)$$

heat rate (HR) of the GT power plant:

$$HR = \frac{3600}{\eta_{GT}} \quad kJ/kwh_e \quad (38)$$

and the specific fuel consumption (SFC) of the GT power plant:

$$SFC = \frac{HR}{LHV} \quad kg_f/kWh_e \quad (39)$$

THE COMPUTER PROGRAM

A computer program is written in FORTRAN language. At the beginning, the values of load, ambient air temperature and relative humidity are assumed, and the compressor outlet pressure (p_2) is estimated. With this estimate all thermodynamic properties are calculate. By setting up and solving equations (1) to (39), improved value for p_2 is obtained. These steps are repeated until the solution reaches the required accuracy. The program contains thermodynamic data for water/steam. The computational procedure of the program is shown in Figure (3).

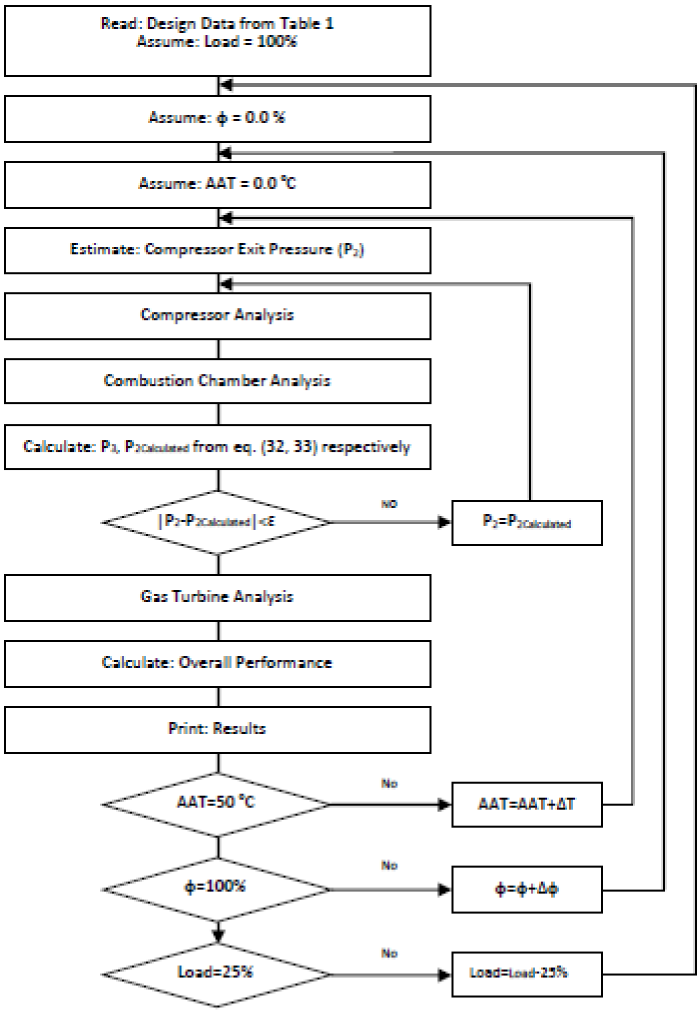


Figure 3: Flow chart of the performance prediction program for the GT unit.

RESULTS AND DISCUSSION

The effect of ambient conditions (temperature and relative-humidity) on gas turbine thermodynamic performance has been studied. For this purpose a full computer program is constructed using the thermodynamic model derived from the previous section. The input design parameters are taken from the Table (1). The ranges of the study for the AAT, RH and part-loading are considered from: zero to 50 ^{o}C, zero to 100% and 25% to 100% respectively. The power output, thermal efficiency, specific fuel consumption and heat rate are calculated for fixed value of turbine inlet temperature. Both part and full loads were considered. At full-load compressor IGVs are fully open, while at part-load the compressor IGVs are partly open. The obtained results would be presented and discussed as following;

EFFECT OF AMBIENT TEMPERATURE

The effect of AAT on the performance of the GT power plant is shown in Figures (4 to 8). Initially, the reference condition is fixed at the ambient conditions (ISO) 15 ºC and 60% RH, and the TIT and air mass flow-rate were taken as 1242 °C 672 kg/s respectively. After, the effect of AAT on GT performance is verified at fixed TIT and full-load operation. Figures 4 and 5 show the turbine, compressor and net turbine powers, thermal efficiency and heat supply at various AATs. With an increase in AAT, the net power output decreases. This is because, for a GT with constant speed, the volumetric flow-rate is constant; the air density inlet to compressor decreases with increase of the AAT resulting in a reduced mass flow-rate through the compressor and turbine.

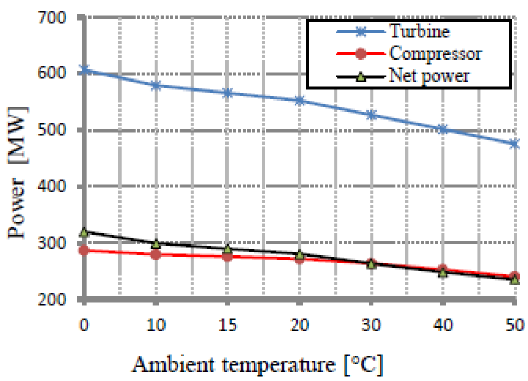


Figure 4: Effect of ambient temperature on power (ϕ=60%).

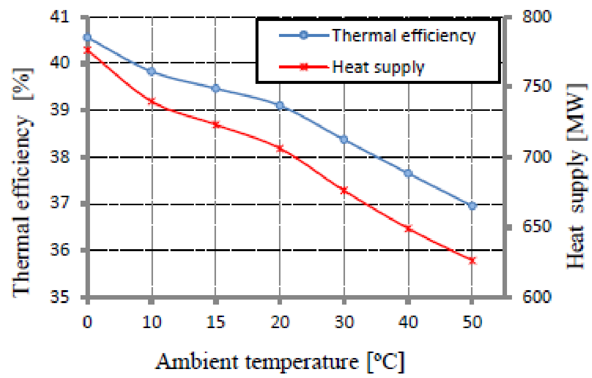


Figure 5: Effect of ambient temperature on thermal efficiency and heat supply.

Therefore, the operating pressure ratios of compressor and turbine also decrease to accommodate the reduced mass flow-rate (Figure 6). Consequently, both the compressor and the turbine powers decrease (*denser air requires more energy to compress and turbines are basically mass flow machines the more mass that can be pushed through the unit the more power it can make and vise-versa*). However, since the decrease in compressor power is less than the decrease in turbine power, the power output and thermal efficiency of the cycle decrease. The thermal efficiency also decreases with increase the exhaust gases temperature (Figure 7).

For a GT with a TIT of 1242 °C, RH of 60% and full-load operation, the reductions of 46.5 MW in compressor power, 131.2 MW in turbine power, 84.7 MW in net turbine power output and 3.6% in terms of thermal efficiency are observed when operating at AAT of 50 °C over that of 0.0 °C. For every degree, (1°C) rise in ambient temperature above ISO conditions the GT loses 0.53% of its net power output and 0.18 % in thermal efficiency. These results are in a good agreement with findings reported [1, 5,15,17,19].

The increase in AAT would reduce the air and fuel mass flow-rates to gas cycle (Figures 6 and 7). Hence the heat supply to the gas cycle decreases with the increase in the AAT as shown in Figure (5). The heat supplied decreases from 776 MW to 626 MW as the AAT increases from 0ºC to 50ºC. The decrease is 0.4% for every 1 ºC rise in ambient temperature.

Figure (7) shows the influence of AAT on exhaust and compressor outlet temperatures and fuel mass flow-rate. It can be seen from the figure that as the AAT increases the compressor outlet temperature increases while the fuel mass flow-rate decreases. This is because; increasing the AAT increases the temperature of air entering the CC. In this way less fuel is needed to reach the desired TIT. The increase in exhaust temperature is attributed to the decrease in the pressure ratio across the turbine when the TIT is fixed. Increasing the AAT by 50 °C decreases the fuel mass flow-rate by 19.3% and increases the exhaust and compressor outlet temperatures by 6% and 10.4% respectively.

Figure (8) presents the variations of SFC and HR for GT cycle with AAT. Since, the increase in the AAT would reduce the net power output and thermal efficiency, thus both the SFC and HR increase with an increase in AAT. It is seen that the SFC and HR increase by 0.195% and 0.185 % respectively, with an increase in ambient temperature 1 ºC.

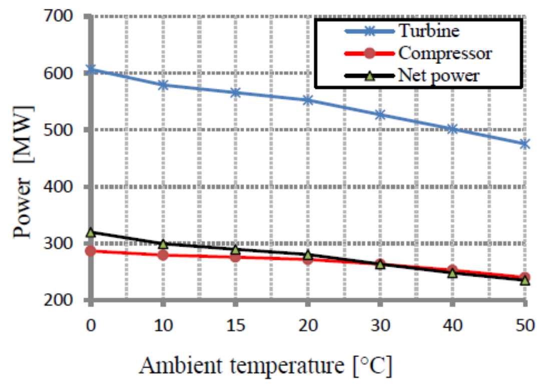


Figure 6: Effect of AAT on pressure ratio and air mass flow-rate (ϕ=60%).

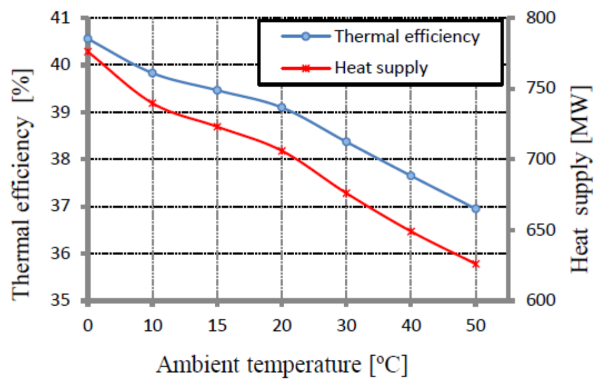


Figure 7: Effect of AAT on exhaust and compressor outlet temperatures (ϕ=60%).

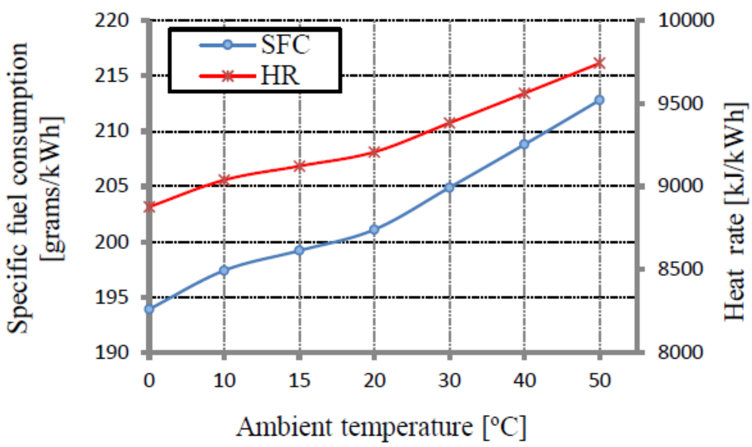
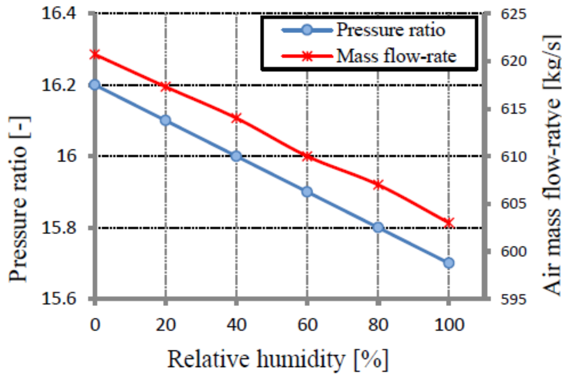


Figure 8: Effect of ambient temperature on SFC and heat rate (ϕ=60%).

EFFECT OF RELATIVE HUMIDITY

The atomic mass of the H_2O is less than N_2 and O_2. Due to that reason, mass of humid-air is less than the mass of the dry air for the same volume. Therefore, humid-air has less density than dry air. Because of low-density air, the amount of humid-air mass flow-rate entering into the compressor of gas cycle reduces [3]. Thus, the pressure and temperature of air at compressor outlet reduced while the mean specific heat increases as shown in Figures (9 and 10).



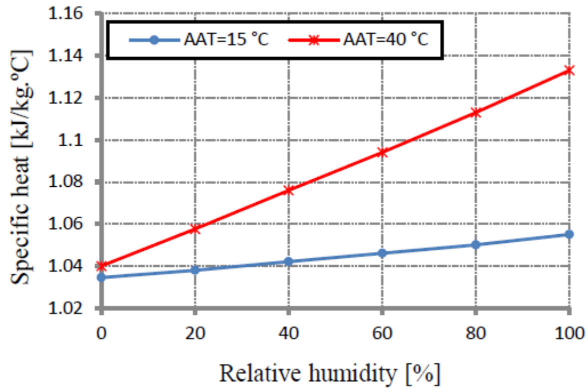
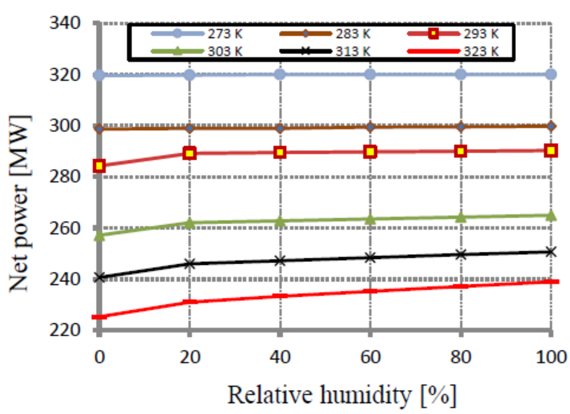


Figure 9: Effect of relative humidity on pressure ratio and air mass flow-rate. (AAT=40 °C).

Figure 10: Effect of relative humidity on specific heat capacity.

At full-load operation, the effects of RH on power output and thermal efficiency were studied at different inlet AATs and shown in Figures. (11 and 12). These Figs. show that the net power output increases and thermal efficiency decreases with RH. The effect of RH on the thermal efficiency is small, and it has only slight effect when inlet AAT is increased. Where, the RH has a remarkable influence on net power output when the inlet AAT is increased.

The increase of the RH would increase the net power output. This is expected since the decrease in turbine power (16.3 MW) is less that the decrease in compressor power (22.4 MW) as shown in Figure (13). This in turn will tend to increase the net power output of the GT cycle. The decrease in the compressor and turbine powers is attributed to the decrease in the air mass flow-rate and pressure ratios across the compressor and as explained at Figure (9). The thermal efficiency decreases as more fuel (Figure 14) has to be combusted to heat the dry air-vapor mixture up to the desired TIT. An increase of 4.15% in net power output was observed, when the GT operated at an AAT of 40 °C and 100% R.H over that of 40 °C and 0.0% RH.



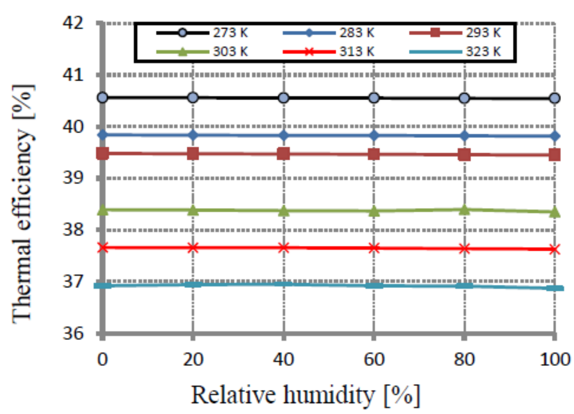


Figure 11: Effect of relative humidity on net power output with various AATs

Figure 12: Effect of relative humidity on thermal efficiency with various AATs

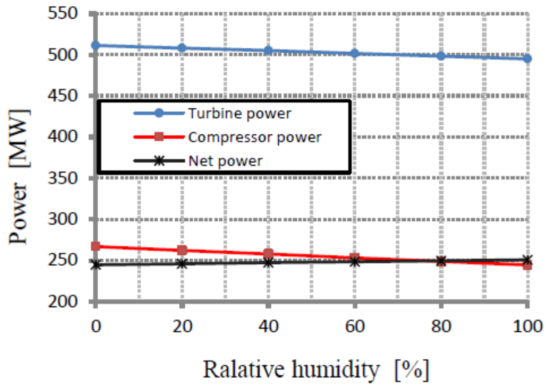


Figure 13: Effect of relative humidity on power (AAT=40 ºC).

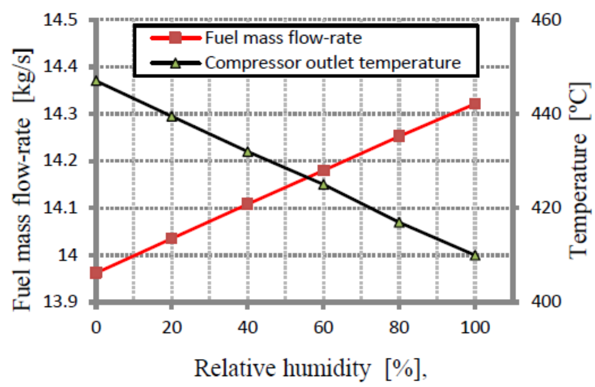


Figure 14: Effect of relative humidity on fuel flow-rate and compressor outlet temperature (AAT=40 ºC).

EFFECT OF PART LOADING

When less than full power (at lower load demand) is required from GT power plant the plant modifies its operation at off-design point or at part-load conditions. At part-loads, both the power output and thermal efficiency decrease. The GT thermal efficiency can be reduced to about 50% of its original rated value when operating at 25% part-load [1,3,14]. This is due the fact that at part-load operation, the turbine and compressor powers decrease with decreasing air mass flow. The thermal efficiency decreases because the decrease in compressor power is less than the decrease in turbine power. This in turn will tend to reduce the net power output of the GT cycle, and becomes less than the design value (Figure 15).

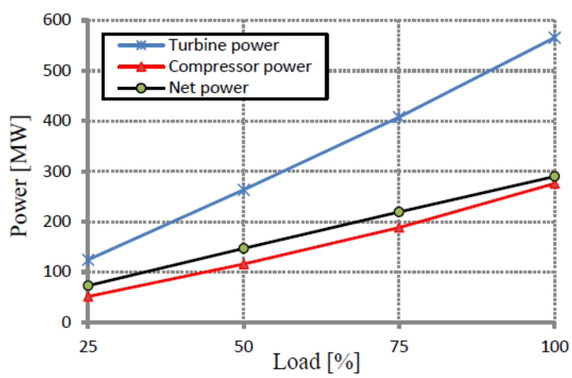


Figure 15: Effect of part-loading conditions on power (AAT=40 ºC and =60%).

The analysis of part load operation was based on varying the air flow-rate at compressor inlet by IGVs control and fuel flow-rate control in the CC, at constant TIT (equals 1242 °C). The effects of AAT on power output and the thermal efficiency of the GT unit, when part-loads are considered, are shown in Figures (16 and 17). The increase in AAT decreases net power output and thermal efficiency at all part-load values. It can be shown that; the power output is directly proportional with the load required. At ISO conditions and 25% part-load, the thermal efficiency decreases from 39.46% to 26.54% (33% of the design value).

Also, the effects of part-load operation on the performance of GT power plant were studied at different relative humidity and fixed ambient and turbine inlet temperatures (T_1

= 40 ºC and TIT=1242 °C). Figs. 18 and 19 illustrate the variation in the power output and thermal efficiency in relation to the part load operation at different RHs. The increase in RH increases the power output and mostly no effect is noticed on the thermal efficiency at all part-loads.

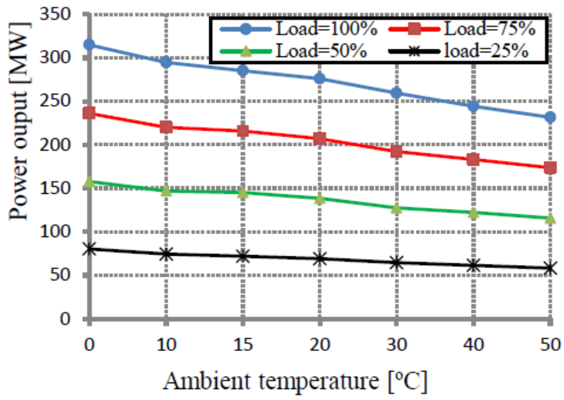


Figure 16: Effect of AAT on power output at part-loading conditions (ϕ=60%).

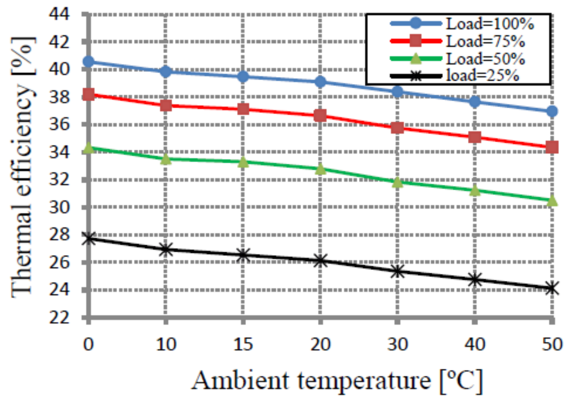


Figure 17: Effect of AAT on thermal efficiency at part-loading conditions (ϕ=60%).

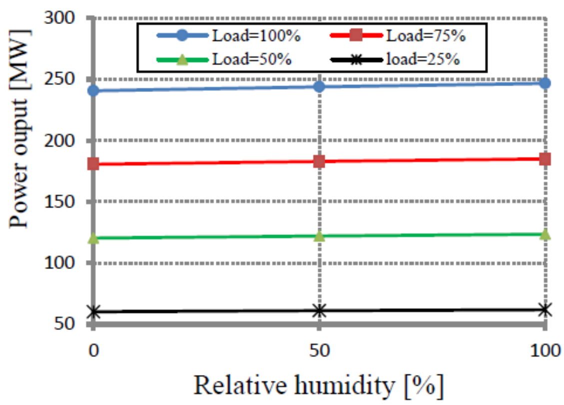


Figure 18: Effect of relative humidity on power output at part-loading conditions (AAT=40 ºC).

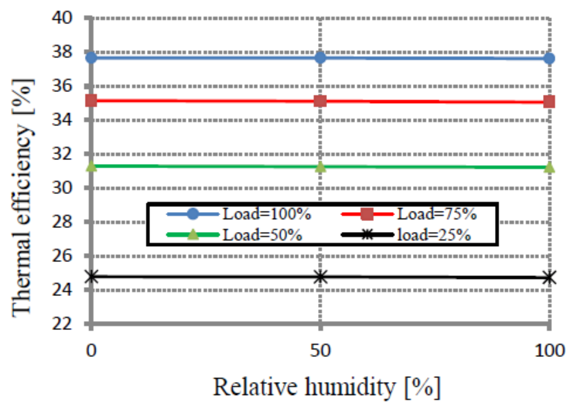


Figure 19: Effect of relative humidity on efficiency at part-loading conditions (AAT=40C).

THERMODYNAMIC MODEL VALIDATION

In order to confirm the validity of the computer program based on thermodynamic model, a comparison is made between the calculated results and actual GT data provided by manufacturer [17], as shown in Table (2). The agreement between manufacturer values and calculated ones is exceptionally good for all parameters.

Table 2: Comparison of manufacturer and calculated results at full-load and T_1= 27ºC, ϕ= 65%.

Parameter	Unit	Siemens	Model	Difference/%
Net gas turbine power output,	MW	261.3	264.32	-1.16
Net heat rate, HR	kJ/kWh	9292.	9330.2	-0.41
Exhaust flow rate, $\dot{m}_4$	kg/s	646.	647.	-0.16
Exhaust temperature, T_4	ºC	586.	583.	0.51

Figures (20, 21, 22 and 23) show the comparisons results between the predictions along with manufacturer [17] for the power output and thermal efficiency at different ambient conditions. All results are shown clearly in good agreement.

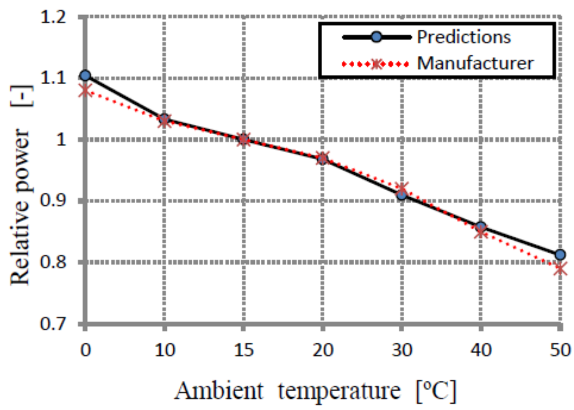


Figure 20: Comparison results between the predictions along with manufacturer for power output with effect AAT (ϕ= 60%)

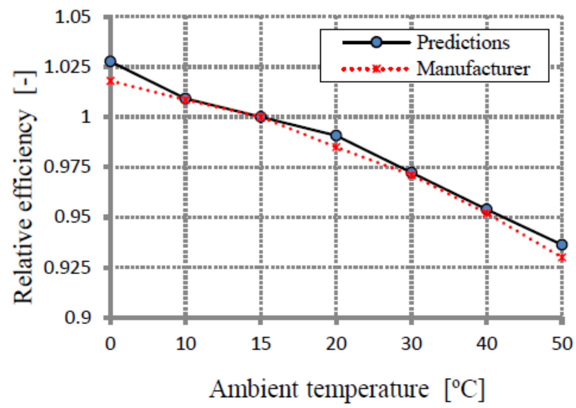


Figure 21: Comparison results between the predictions along with manufacturer for thermal efficiency with effect AAT (ϕ=60%).

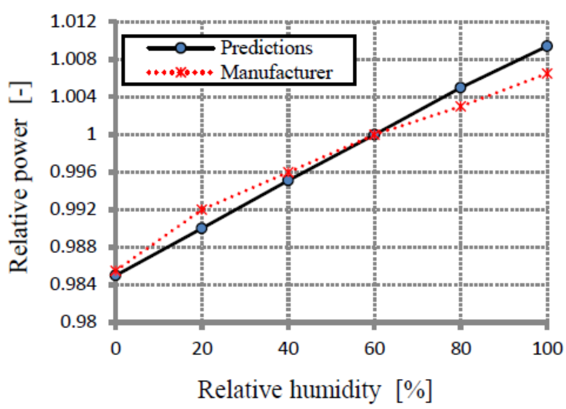


Figure 22: Comparison results between the predictions along with manufacturer for power output with effect relative humidity (AAT=40 °C).

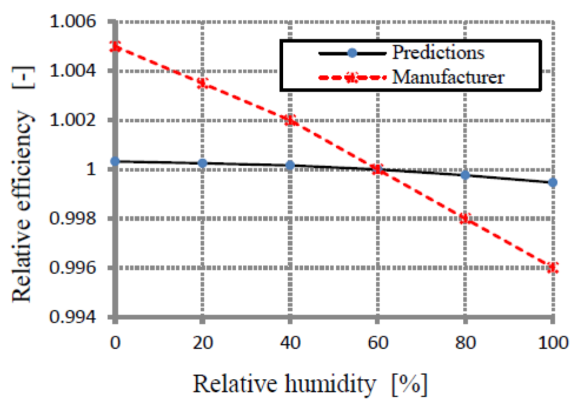


Figure 23: Comparison results between the predictions along with manufacturer for thermal efficiency with effect relative humidity (AAT=40 °C).

CONCLUSIONS

A flexible and general computer program based thermodynamic model has been developed for parametric study and performance evaluations of the GT unit. The model can evaluate the effects of various input and design parameters on the power output and thermal efficiency in a single-shaft constant speed GT engine. The performance evaluation of a 285 MW GT power plant under varying ambient conditions including part-load operation is made in the present work.

Based on the above analysis, the following conclusions can be drawn:

- It is evident from the present study that the thermodynamic performance parameters are having strong dependency on ambient and loading conditions.
- Ambient air temperature has a significant influence on the power output and thermal efficiency of the gas turbine power plants.

- The relative humidity has a negligible effect on both net power output and thermal efficiency at low ambient air temperatures. However, increases in net power output and decreases in thermal efficiency of the GT power plants at higher values of ambient air temperature were noted.
- The ambient temperature and relative humidity have a greater influence on power output than on thermal efficiency of the GT power plants.
- The part-load performance of the selected gas turbine plant is predicted. Under the 25% part-load condition, the thermal efficiency decrease to 33% of the design value.
- The verification of the developed thermodynamic model against relevant manufacturer data showed that it can be used to predict the effect of ambient conditions and part-load operation on the thermodynamic performance of the gas turbine with sufficient accuracy.
- Decreasing the compressor inlet temperature is a useful tool for increasing the net power generation capabilities of GT cycle power plants. Therefore, retrofitting of the existing GTs with inlet air cooling system gives a better system performance, and may prove to be an attractive investment opportunity to the Libyan government.

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NOMENCLATURE

C_p	specific heat (kJ/kg.ºC)
h	specific enthalpy (kJ/kg)
HR	heat rate (kJ/kWh)
LHV	lower heating value (kJ/kg)
$\dot{m}$	mass flow rate (kg/s)
MMW	molecular weight of humid-air, (kmol/kg)
p	pressure (bar)
P	power (kW)
Q	heat supply (kW)
R	gas constant, (kJ/kg. ºC)
SFC	specific fuel consumption (kg/kWh)
T	temperature (°C or K)
V	volume flow-rate (m^3/s)

Abbreviations

AC	air compressor
AFR	air to fuel ratio
AMF	dry-air mass fraction
AAT	ambient air temperature
CC	combustion chamber
EG	electrical generator
GT	gas turbine
IGV	inlet guide vanes
RH	relative humidity
TIT	turbine inlet temperature
VMF	vapor mass fraction

Greek Letters

ρ	density (kg/m^3)

γ	isentropic exponent (-)
η	efficiency (%)
ϕ	relative humidity (%)
ω	specific humidity (kg_v/kg_a)
Δp	pressure drop (bar)
π	pressure ratio (-)

Subscripts

a	air
h	humid-air
c	compressor
e	electrical energy
f	fuel
g	gas products
n	net
r	reference or design condition
t	turbine
v	vapor

1,2,3,4,5: cycle state points in Fig. 1.