Gas Turbines waste heat/power recovery in tropical climate zones: Case study

استعادة الحرارة / القدرة المبددة فى التربينات الغازية التى تعمل فى المناطق الحارة : دراسة حالة

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الملخص

تم تطبيق هذا البحث على محطة التوليد الكهربية الكائنة بالمملكة العربية السعودية (محطة الجوف) – تتكون من 7 تربينات غازية بطاقة 25 ميجا وات للوحدة الواحدة والتى تعمل منذ 1998 - بالاضافة الى وحدتين بطاقة 82 ميجا وات للوحدة الواحدة مزودة بمنظومة تبريد الهواء الداخل الى الضاغط – تعملان منذ 2007

يهدف هذا البحث الى بيان عملى عن كيفية وخطوات اجراء قياسات عملية على محطات التوليد ذات التربينات الغازية فى مواقع التشغيل – بغرض قياس وتقييم الأداء لها ومقارنته بالتصميم

لقد اشتملت النتائج على قيم الأداء لمحطات التربينات الغازية ممثلة فى القدرة الكهربية – ومعدل الحرارة المضافة بالكيلو جول لكل كيلووات ساعة عند الأحمال المختلفة للمحطة . أيضا تم مقارنة بيانات الأداء الفعلية بنظيراتها التصميمية والتى أوضحت تباعدا ملحوظا بينهما بسبب ارتفاع حرارة الجو عن التصميم . أيضا أوصت الدراسة بأفضلية تشغيل المحطة عند أحمال مرتفعة قريبة من 100 % لأسباب اقتصادية كلما أمكن ذلك - أخيرا أوصت الدراسة بضرورة اضافة منظومة التبريد الى للهواء الجوى باستخدام عادم التربينة وذلك لتلاشى فقد القدرة وكذلك استرجاع حرارة العادم – عندئذ يمكن تلاشى التباعد الملحوظ بين الأداء الفعلى للتربينات الغازية والأداء التصميمى فى المناطق الحارة.

Abstract

The present study is applied on the gas turbines power plant that is currently under operation in northern kingdom of Saudi Arabia (KSA) since 1998. The plant consists of 7 gas turbine of 25MW each without inlet air cooling. More 2 gas turbine of 82MW each with inlet air cooling were added in_Dec 2007._In this area, during summer season, the ambient temperature may reach 50 °C, or higher in July and August. The objective is to present field results of this gas turbines power plant operation in order to measure and evaluate the performance. The results showed that, operating the plant at higher loads near to 100% is recommended for economic considerations, heat /power recovery by cooling the inlet air is advised, to overcome the problem of electricity black-out. Finally the results showed that disagreement between design and measured performance values is mainly due to higher ambient temperatures.

Key words

Gas turbines, power plant, Case study

1-Introduction

The gas-turbine (GT) operates on the principle of the Brayton cycle, where the mixture of compressed air and fuel (natural gas or fuel oil) is burned at constant pressure. The resultant hot gas is used to expand through a turbine to get work. More than 50 percent of the turbine power is consumed by the air compressor [1]. The simple cycle gas turbine is the most common type with efficiencies ranging from 30 to 40 percent (8968-11606 kJ/kWh) [1-9].

The GT plant performance is indicated by two Parameters, Power output ($\dot{W}$) and Heat rate (HR in kJ/kWh [1-9]. At present, GT efficiencies are in the range of 45-50% (a heat rate of 8000KJ/KWh to 7199KJ/KWh). Using a new metallurgy,

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advanced thermal barrier coating and advanced cooling system for blades, the fired inlet temperatures can be high as 1427C and PRs of 40:1[1-5, 31].

Site conditions especially ambient temperature has a strong influence on the gas turbine performance. However, in tropical climate zones, about up to 20% loss in power is possible due to the high ambient temperature. So, some researches [10-22] were directed to GT power augmentation by cooling the inlet air. The power output gain of 1% is achievable for every 1.5-2.0 °C drop in the inlet air temperature using water chillers as reported by researchers and manufacturers [1-3, 10-22].

Other researches [23-30] were focused on environmental issues like NOx reduction using bypass air and catalyst systems. The new GTs also utilize low NOx combustors to reduces the NOx emissions, where The use of NG fuel and the use of the new dry low NOx combustors have reduces NOx levels below 10 PPm. New research in combustors such as catalytic combustion have great promise and values of 2 ppm can be achieved in the future. Also for future plants CO2 capture and storage system are included.

Finally, integration between renewable energy (solar energy) and GTs were studied for future applications [32-36].

1.1-Main components of the present GT

The present GT is identified as single shaft, simple open cycle mechanical power generator consisting of **[2]:**

1. Air filtration system: to avoid compressor fouling, erosion and corrosion that may cause by dirty and sandy air.

2. Eighteen stage axial compressor

3. A combustion chamber equipped with eight combustors arranged in circular array around the GT axis

4. A three stage reaction type turbine.

The complete package of gas turbine can be divided into four major assemblies:

a. Inlet section assembly

b. Compressor/ combustor section assembly

c. Turbine section assembly

d. Exhaust section assembly.

1.2-Gas path description:

Atmospheric air is drawn through the inlet manifold and filtration system into the compressor where it is pressurized up to about 11 atmospheres and forced into the combustor chamber and baskets in a steady flow. Fuel is injected into the combustors burners, raising the temperature of the mixture of air and combustion products. Then the compressed and heated gas flows through the turbine, dropping in temperature and pressure as the heat energy is converted into mechanical work. A portion of the power produced by the turbine is used for driving the compressor and the balance of power is used to drive the generator. The gas is then discharged through the exhaust diffuser and axial exhaust manifold into the exhaust stack.

To assure good starting characteristics, two bleeds are fitted at the 6^{th} and 11^{th} stage of the compressor. They remain open during the starting cycle up to 90% of the design speed.

1.3-Inlet guide vanes:

The first stage of the compressor vanes is a flow distributer conveying the air towards the first stage of rotating blades according to a suitable angle. A variable setting mechanism permits to select the proper incidence of the air entering the first rotor. This arrangement improves starting acceleration and prevents compressor surge or stall.

1.4-Thrust bearing:

It is housed in the inlet casing. it is capable of transmitting the full designed thrust load in either direction. Thrust bearing shoes on both sides of the shaft thrust ring are equipped with thermocouples for monitoring bearing temperature by the control system. Continuous oil flow is admitted to the bearings for lubrication purpose.

1.5-Combustion system

The combustion system is contained in a carbon steel casing which is part of the compressor-combustor shell and provides housing for the following components:
Combustor baskets, fuel nozzles, spark plug igniters, flame detectors, and cross flame tubes.

1.6-Turbo-set configuration:

Figure (1) shows the block diagram of the turbo-set configuration. In this diagram the main two auxiliaries groups are: the starting package and the reduction gear unit.

1.7-starting package

A gas turbine, like other internal combustion engines, cannot produce torque at zero speed. Therefore a starting device must be used to crank the turbine for startup.

The main components of the starting package are: the driving motor, the torque converter, the electro-brake, the turning gear with relevant clutch, and the double toothed coupling. The turbo-set starting diagram is given in Fig. (2).

1.8-Lubricating oil system:

The basic requirement of all rotating equipment is a reliable lubricating system. Such a system must supply cooled, cleaned and pressurized oil to the bearings. The GT lube oil system consists of: oil reservoir, pumps to pressurize the oil, heat exchanger to remove the heat absorbed by the oil, a filtering system to remove contaminants, a system of valves to regulate at the proper pressure, oil vapor extractor, and oil heaters. A block diagram for the oil system is shown in Fig. (3).

1.9-Air system

The air system is divided into 5 portions:

-***Instruments air, Fig. (4):*** Used for operating the pneumatic control system (valves and actuators). This is during the startup and shutdown.

-***Atomizing air, Fig. (5):*** it is needed during running on fuel oil only. The requested supply is taken from a bottle filled by pressurized air from reciprocating compressor.

-***Compressor bleed**, **Fig. (6)**:* is used to avoid compressor surge during starting phase of the unit. For the same reason during shutdown. The air is discharged from the axial compressor to the exhaust of the unit through two bleed valves. During the starting phase the bleeds remain open up to 90% of rated speed. The source is from 6^{th} and 11^{th} axial compressor stage.

-***Sweep air, Fig. (7)*** Is used to clean the liquid injectors for removing the liquid fuel from flow-divider and injectors themselves. It operates only at the end of change-over from liquid fuel to gas fuel.

-***Cooling system:*** The turbine is provided of separate cooling paths for the rotor and stationary parts. Compressor discharge air, bled at the combustor casing, is used for both the rotor and the stator cooling after having been filtered and cooled into an air-to-air cooler externally to the machine. As shown in Fig. (8).

The rotor cooling flow enters the machine at the torque tube section. The first rotor blades has been constructed with air passage to allow cooling air-flow through the blade before it is discharged in the gas stream at the tip of the blades.

The stator cooling air serves many functions. All the cooling air is supplied to the blade ring cavity by a single air box. A percentage of the air enters the first row vanes through wall span wise holes and is discharged into the gas stream through the trailing edge slots, Fig.(9). The air cooling system has been designed to ensure long life and reliability of the working parts. A continuous monitoring system for temperature supervision was supplied.

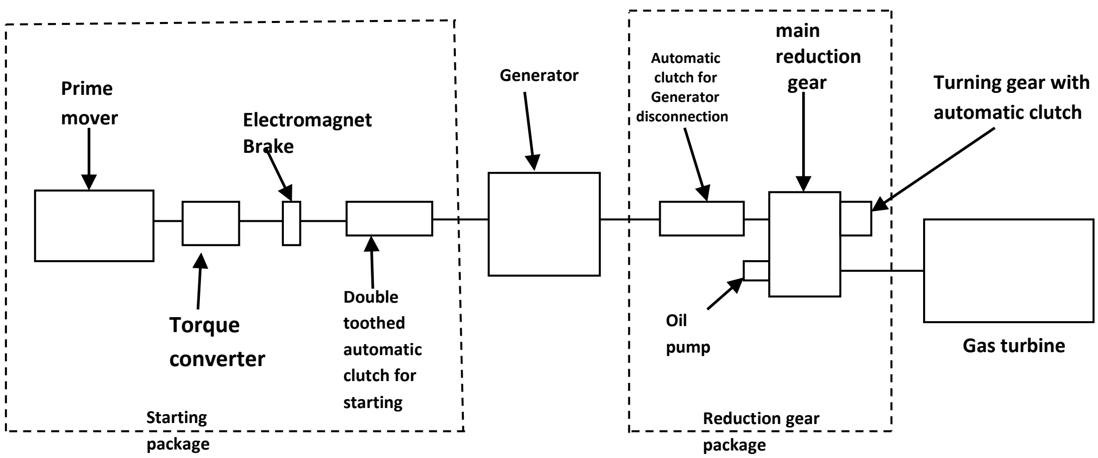


Fig.(1):block diagram for turbo-set configuration[2-5]

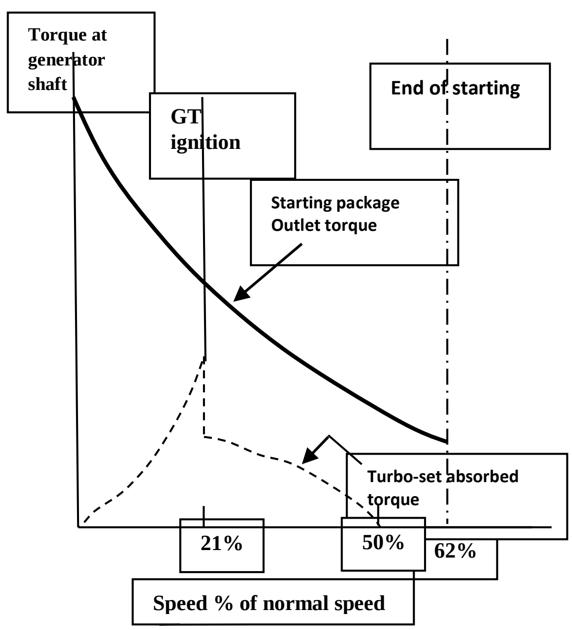
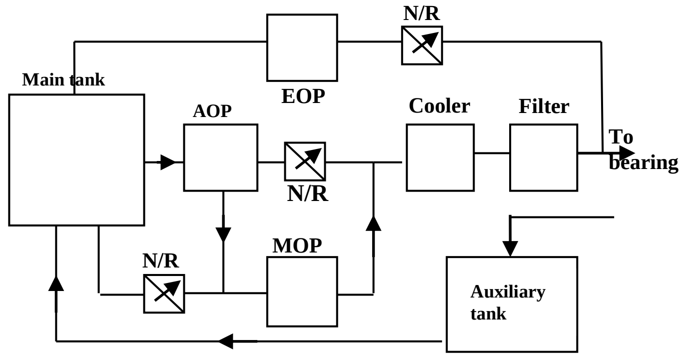


Figure (2): Turbo-set starting diagram [2-5].



AOP: Auxiliary oil pump, driven by A.C motor EOP: Emergency oil pump, driven by D.C. motor MOP: Main oil pump, driven mechanically by the unit N/R: non return (check) valve
Fig.(3): oil system block diagram[2-5]

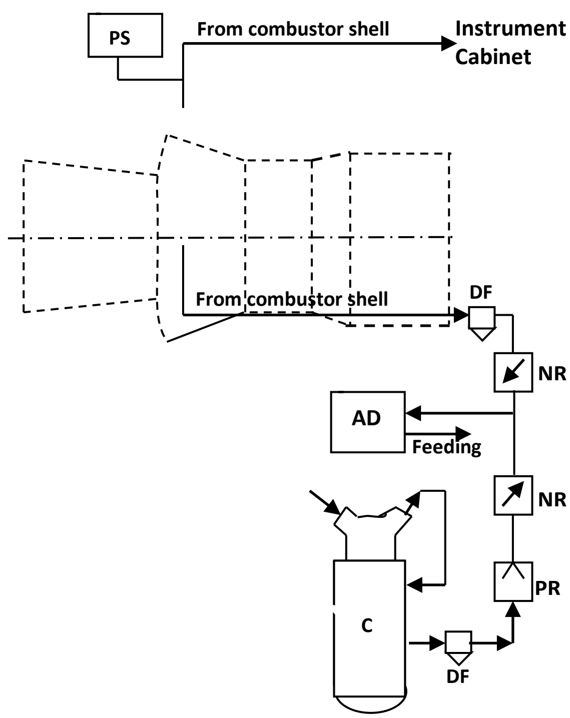


Fig.(4): Instruments air
C: service air compressor, DF: Drain filter, PR: pressure reducer, NR: check valve, AD: Air drier.

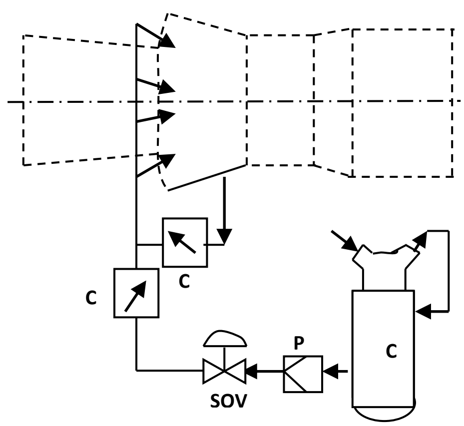


Fig.(5): Atomizing air system[2-5]
C: Atomizing air compressor, PR: pressure reducer, SOV: shut-off valve, CV: check valve (no return valve)

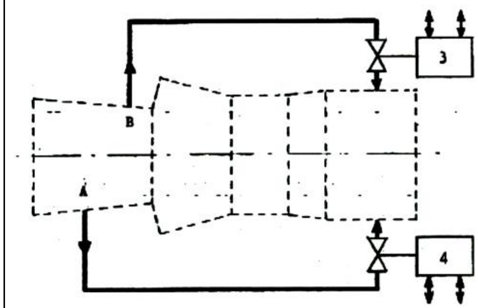
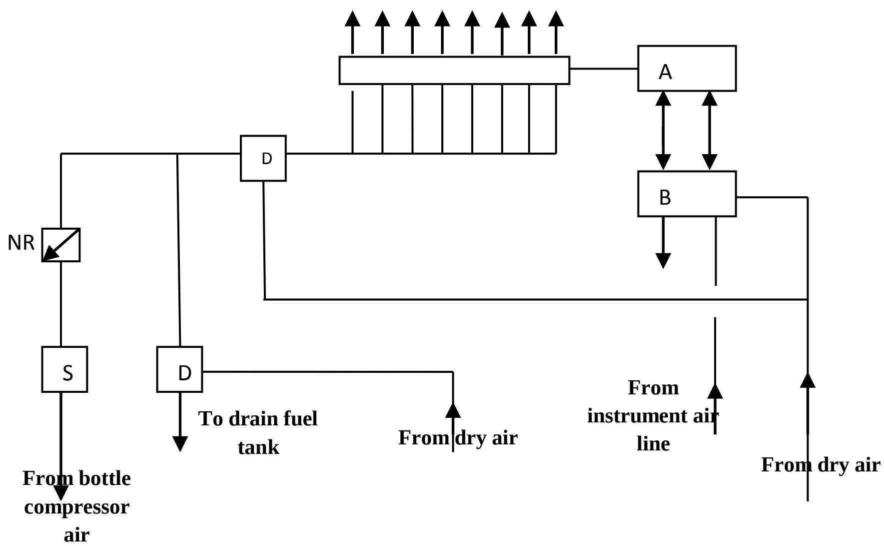


Fig.(6): compressor bleed line[2-5]
A: Bleed from the 6th stage
B: Bleed from the 11th stage 3&4: bleed valves



A: multiple drain collector valve *D: Drain Valve*	*B: Four ways pneumatic valve* *D: Drain Valve*	*NR: check valve* *S: Solenoid valve*

Fig.(7): Sweep Air diagram[2-5]

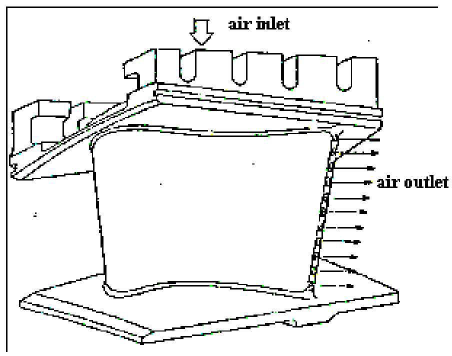


Fig. (9): air inlet and air outlet for blade cooling [2-5].

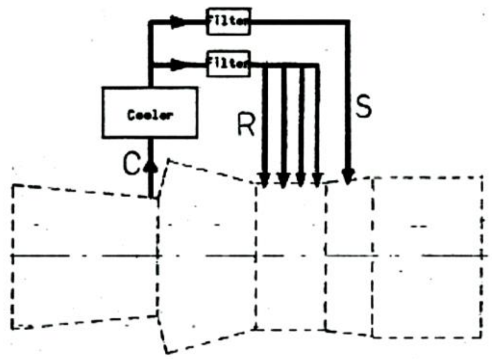


Fig.(8): Turbine air cooling
C:compressor delivery
S:Stator air cooling
R: Rotor air cooling

1.10-Testing objectives:

The present test objective is to determine the power output and the heat rate of a single gas turbine generating plant compared to the design ones at different loads.

2-Testing steps

2.1-Test preparation

Before the tests are carried out, it is necessary to perform the following actions before testing:

1. cleaning of gas turbine: compressor has to be washed wet and/or dry cleaning, using agent approved by the turbine manufacturer.
2. Check and clean inlet air filters, and check that the inlet duct, elbows, expansion joints, silencers, are in order and not obstructed.
3. Take-off the injectors and clean them carefully and thoroughly.
4. Check the instruments conditions and update its calibration before the tests.

During the test the main operating parameters of the gas turbine are given below:

a. Barometric pressure: it is measured by a precision mercury barometer. The reading accuracy is 0.25 mmHg.

b. Compressor inlet temperature: it is measured by three precision mercury-in-glass thermometers scaled 1/10 ^{0}C, placed in the air flow near the compressor inlet filters.

c. Power output: it is measured by using standard calibrated three phase KWh meter. This meter is connected to the transformer (current and voltage).

d. Turbine speed: it is measured using electronic taco-meter connected to the turbine control system.

e. Combustor shell pressure: it is measured by transducer supplied with the turbine control system.

f. Exhaust manifold temperature: it is measured by thermocouples which are mounted and connected to the control system.

g. Fuel temperature: it is measured by a thermometer immersed in the flowing fluid,

h. Distillate oil fuel flow: it is measured by a turbine type flow meter

i. Fuel analysis: it is done by the local laboratories.

j. Auxiliaries power absorption is measured by means of a KWh meter.

Symbols and units are summarized in table (1).

2.2-Test procedure:

To achieve the objective of the present study, operating parameters are recorded at part loads (50%, 75%), maximum continuous load or base load (100%), and peak load (120%), in the following sequence:

a. after calibration of instruments and preparation

b. the unit will be started and brought directly to base load exhaust temperature control

c. After the unit has reached base load it will be allowed to stabilize for at least 3 hrs.

d. all reading will be taken and recorded. Three sets of data will be obtained. The duration of each set of measurements will be 20 minutes (5:20 according to ISO standard). A sample of test data sheet that was recorded during the test is summarized in tables (2 and 3).

Table (1): symbols and units.

Symbol	parameter	unit
P_{amb}	Ambient pressure	mmHg
T_{amb}	Ambient temperature	^oC
F_{CST}	Fuel counter at start test	Liter
F_{CET}	Fuel counter at end test	Liter
G_{fuel}	Fuel flow	L/hr
T_{fuel}	Fuel temperature	^oC
ρ_{fuel}	Fuel flow density at flow temperature	Kg/Liter
LHV	Lower heating value	KJ/Kg
n	Revolution per minute	rpm
Time	time	S
W_{Gros}	Gross power at generator flange	KW
C_{aux}	Current for auxiliary consumption	A
V_{aux}	Voltage for auxiliary consumption	V
W_{Aux}	Auxilary power	KW
W_{Net}	Net power	KW
HR_{Net}	Net heat rate	KJ/KWhr
W_{Corr}	Corrected net power to design conditions	KW

Table (2): Average data recorded during the test with distillate fuel oil.

	Base load 100%			75% base Load			50% base load			Peak Load		
Parameter	Run1	Run2	Run3	Run4	Run5	Run6	Run7	Run8	Run9	Run10	Run11	Run12
P_{amb}	701.7	701.7	701.7	702.6	702	701.4	701	701	701	701	701	701
T_{amb}	34.38	34.86	35.6	35.76	36.1	38.96	40.2	39.9	39.54	40.12	40.88	41.04
F_{CST}	31500	36700	41900	50600	54800	89700	17400	20800	23800	99200	4200	9000
F_{CET}	35145	40331	45515	53701	57904	92810	19880	23284	26285	102949	7930	12730
G_{fuel} (L/hr)	10935	10893	10845	9303	9312	9330	7440	7452	7455	11247	11190	11190
T_{fuel}	48.8	49.8	49.5	50.2	51	52.6	52.8	53	53	52.6	52.6	52.8
ρ_{fuel}	0.8159	0.8153	0.8155	0.8150	0.8145	0.8134	0.8133	0.8132	0.8132	0.8134	0.8134	0.8133
LHV	42578 KJ/kg											
N	37.72	37.48	37.25	29.56	29.555	29.53	19.81	19.82	19.82	38.95	38.67	38.66
Time(s)	1200											
C_{aux} (A)	500	505	520	510	530	520	500	480	510	525	540	520
V_{aux} (V)	380	380	379	379	378	382	380	385	385	380	380	380

Table (3): Correction factors and corrected values @ standard conditions

	Run 1	Run 2	Run 3	Run 4	Run 5	Run 6	Run 7	Run 8	Run 9	Run 10	Run 11	Run 12
Wnet	28818.56	28630.62	28445.56	22518.86	22504.62	22487.84	15002.27	15017.64	15000.64	29753.43	29529.04	29532.51
HRnet KJ/KWh	13181.62	13207.46	13238.06	14335.72	14349.86	14368.89	17173.23	17181.21	17207.61	13091.49	13124.12	13120.96
λ1	1.1238	1.1199	1.1139	1.1126	1.1099	1.087	1.0771	1.0795	1.0823	1.0777	1.0717	1.0704
λ2	1.0039	1.0039	1.0039	1.0052	1.0043	1.0035	1.0029	1.0029	1.0029	1.0029	1.0029	1.0029
β	0.9578	0.959	0.9608	0.9612	0.9621	0.9695	0.9727	0.919	0.971	0.9725	0.9746	0.9725
W corr	32512.6	32188.48	31809.09	25184.77	25085.29	24529.84	16205.81	16258.56	16282.27	32158.26	31738.04	31703.28
HR corrected	12625.35	12665.96	12719.12	13779.49	13806	13930.64	16704.4	15789.53	16708.59	12731.47	12790.77	12760.14

3-Data reduction and Calculations [1-5]:

To study the plant performance at site operating conditions (heat rate and output power), the operating data are measured and recorded at different loads. Then, the performance can be calculated as follows:

I-Computation of power output, and heat rate

-power output is calculated as follows:

$$W_I = \frac{n*3600}{t} \cdot \frac{1}{K_C} \cdot K_A \cdot K_V$$

$$W_n = W_I - W_{aux}$$

Where:

W_I: gross power output at generator terminals, KW

W_n : net power output at generator terminals, KW

W_{aux} : auxiliaries power consumption, KW

n : revolution number counted on the KWh-meter.

t : time during which the KWh-meter reading has been made, S

K_C : KWh-meter constant, revs/KWh

K_A : ratio of current transformers

K_V : ratio of voltage transformers

-Heat Rate is calculated as follows:

$$HR = \frac{V.3600}{t} . \rho . LHV . \frac{1}{W_n}$$

Where:

HR: heat rate, KJ/KWh

V: fuel volume read on the flow-meter during the time t

t: time during which the measurement of fuel consumption has been carried out, Seconds .

ρ : fuel density at the temperature of fuel during test, Kg/L.

LHV: lower heating value of the fuel oil, KJ/kg

W_n : net power output at generator terminals, KW

II-Computation of the power output and heat rate corrected to reference conditions

A-corrected net power output, W_{Corr}

$$W_{Corr} = \frac{W_n}{\lambda}$$

Where $\lambda = \lambda1.\lambda2$

λ1: factor taking into account the compressor inlet temperature λ2: factor taking into account the atmospheric pressure

B-corrected heat rate, HR_{corr}

$$HR_{corr} = \frac{HR}{\beta}$$

Where β: factor taking into account the compressor inlet temperature.

4-Results and discussion

As discussed above, to obtain and investigate the performance of the existing GT plant, the following steps are taken:

Step-1: recording all measured parameters in the test data sheet and calculating the average value for each parameter as given in table (2). The load variation steps are 50%,75%, 100%, 125% as shown in table(2)

Step-2: calculating the net power and net heat rate at different loads. Then making correction as summarized in table (3)

4.1-Variation of plant performance at different loads (net power and heat rate)

Variation of both of Power output and HR are plotted in Fig.(10) against load variation compared with design (ISO) conditions. It is clear that, the higher operating load the most economic operation is. As example operating the plant at 100% load instead of 50%, can decrease the HR from 13930 down to 12625 and from 17173 to 13181 KJ/KWh at ISO and actual conditions respectively.

4.2-Comparison of actual power output and heat rate with the design ones.

In order to compare the actual performance results with the design ones, The plant performance at both actual and design conditions are plotted in Fig.(10). It is noticed that the HR actual results are higher than design ones (as increase percentage of 4.4%, 4.03%, 2.8% at load of 100%, 75% , 50% respectively).

In general, a disagreement between design and measured performance values is due to high ambient temperatures.

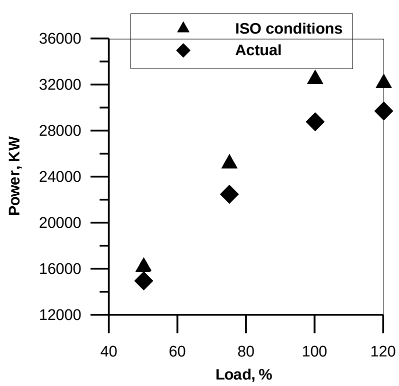


Fig. (10a): Variation of design and actual Power output at different loads.

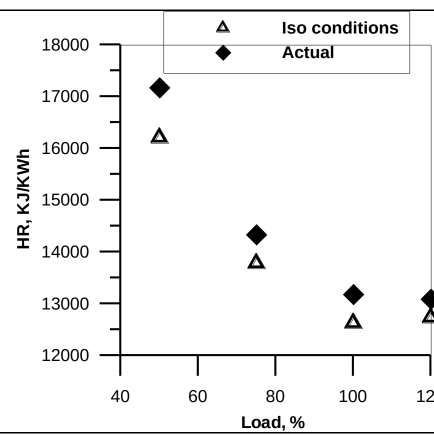


Fig. (10b): Variation of design and actual HR at different loads.

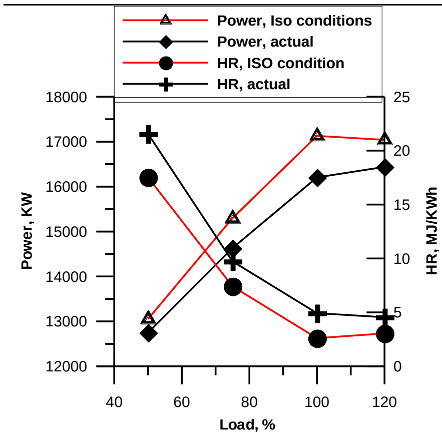


Fig. (11): comparison between actual and design performance results at different loads.

5- Recommendation for adding absorption chiller as a heat/power recovery system

As depicted from Figures(10 and 11)The increase in air temperature at inlet of GT compressor, causes a significant reduction in output power as shown in table(7).

Currently, there are two basic systems available for compressor inlet cooling: Evaporative cooler and absorption chiller systems when the ambient temperature is extremely high with low relative humidity the chiller is the more suitable cooling solution [1-3].

Table. (4): influence of compressor inlet temperature on power output (base rating) [1-3].

T1 C	Power change factor
10	1.03
15	1
20	0.965
30	0.9
40	0.8375
50	0.775

From table (4), for decrease in inlet air temperature from 50 C to 15 C, the net output power increases around 22%. Consequently, 22% increase in power means adding of 0.22x7x25=38.5 MW of power output to overcome the blackout problem during peak load.

6-Conclusion

The performance of Al-Jouf power plant is studied and compared with the design at 50, 75%, 100, 120% loads. It is concluded that:

1. For the single GT cycle the actual HR is between 17173.23, 14335.72, 13181.62 KJ/KW.hr at loads of 50%, 75%, 100% respectively. While the corresponding designs ones are 16704.4, 13779.49, 12625.35 at loads of 50%, 75%, 100% respectively. So, operating the plant at higher loads near to 100% is recommended.

2. As a simple solution, heat/power recovery by cooling the inlet air, is recommended for economic considerations and to overcome the problem of electricity black-out.

3. As noticed, the flue gases is exhausted at higher temperatures (about 410C), so heat recovery system is recommended.

4. 4-New technologies for low Gas emissions should be considered in future projects (NOx , CO and CO2 capture and storage system).

Nomenclature

A_{aux}: Current for auxiliary consumption, A
F_{CET}: Fuel counter@ end of test, L
F_{CST} : Fuel counter@ start of test, L
G_{Fuel}: fuel flow, L/hr
HR: Heat rate, KJ/KWh.
HR_{Net}: net heat rate, KJ/KWh
LHV: Lower heating value of fuel, KJ/Kg
$\boldsymbol{m^o}$ **:** mass flow rate (kg/s)
n : revolutions, rpm
p : pressure (bar)
P_{amb}: ambient pressure, mmHg,
PR: pressure ratio
T_{amb}: Ambient temperature, C
T_{Fuel}: fuel temperature, C

TIT: Turbine inlet temperature (^{0}C)
V_{aux}: voltage for auxiliary consumption, V
W : power output (MWe)
W_{Aux}: auxiliary power, KW
W_{Corr}: corrected net power, KW
W_{Gross}: Gross power @ generator Flange, KW
W_{Net}: Net power, KW

Greek Symbols

ρ_{Fuel}: fuel flow density, Kg/L
η : efficiency

Subscripts

e: electricity
exp: expander
hp: high-pressure
lp: low-pressure
max: maximum

References

[1.] Meherwan, P,B, 2002"Gas turbine engineering handbook" the boyce consultancy, Fellow, American society of mechanical engineers and Institute of Diesel and gas turbine engineers, UK.

[2.] Fiat Avio , TG20 gas turbine technical and training manual.

[3.] General Electric- Model PG7111(EA) "Gas Turbine operation and maintenance manuals".

[4.] Frank J. Brooks " GE Gas Turbine Performance Characteristics" GE Power Systems, Schenectady, NY, GER-3567H.

[5.] J. H. HorlockF.R.Eng., F.R.S., 2003" Advanced Gas Turbine Cycles" Whittle Laboratory Cambridge, U.K.

[6.] M. M. Rahman, Thamir K. Ibrahim and Ahmed N. Abdalla, 2011"Thermodynamic performance analysis of gas-turbine power-plant" International Journal of the Physical Sciences Vol. 6(14), pp. 3539-3550, 18 July.

[7.] P. Centeno, I. Egido, C. Domingo, F. Fernández, L. Rouco , M. González, "Review of Gas Turbine Models for Power System Stability Studies" Universidad Pontificia Comillas,

[8.] KAWASAKI, 2003 "Gas turbines" Texas Technology Showcase, 17 - 19 March, 2003 Houston, TX

[9.] www.kawasaki.com/gtd

[10.] Y.S.H. Najjar, "Enhancement of Performance of the Gas Turbine Engines by Inlet Air Cooling and Cogeneration System". Applied Thermal Engineering, Vol. 16, 1996, 163-173.

[11.] M.M. Alhazmy, Y.S.H. Najjar, 2004 "Augmentation of Gas Turbine Performance Using Air Coolers". Applied Thermal Engineering, Vol. 24, 415-429.

[12.] Amell, F.J. Cadavid, 2002"Influence of the Relative Humidity on the Air Cooling Thermal Load in Gas Turbine Power Plant". Applied Thermal Engineering, Vol. 22, 1529-1533.

[13.] Mohanty, J. Paloso, 1995"Enhancing Gas Turbine Performance by Intake Air Cooling Using an Absorption Chiller". Heat Recovery Systems, Vol. 15, 41-50.

[14.] J.S. Andreppont, 2000 "Combustion Turbine Inlet Air Cooling, Benefits, Technology Options and Applications for District Energy". International District Energy Association (IDEA) 91st Annual Conference, Montreal.

[15.] M. Ameri, K. Montaser, 2002"The Study of Capacity Enhancement of a 37.5 MW Gas Turbine Using Chilled Water Thermal Energy Storage System". Proceedings of the 15th International Conference of Efficiency Costs, Optimization, Simulation and Environmental Impact of Energy Systems (ECOS, 2002), Berlin, Germany.

[16.] M. Ameri, H. Nabati, A. Keshtgar, 2004 "Gas Turbine Power Augmentation Using Fog Inlet Air-cooling System". Proceedings of ESDA04, 7th Biennial Conference on Engineering Systems Design and Analysis, Manchester, United Kingdom.

[17.] R.S. Johanson, 1998 "The Theory and Operation of Evaporative Coolers for Industrial Gas Turbine Installations". Gas Turbine and Aeroengine Congress and Exposition, June 5-9, Amsterdam, The Netherlands, Paper No. 88-GT-41.

[18.] I.S. Ondryas, D.A. Wilson, M. Kawamoto, G.L. Haub, 1991 "Options in Gas Turbine Power Augmentation Using Inlet Air Chilling". Engineering for Gas Turbine and Power, Transaction of the ASME, Vol. 113, 203- 211.

[19.] M. Mercer, 2002 "One-Stop Shop for Inlet Cooling Systems". Diesel and Gas Turbine Worldwide, June Issue, 10-13.

[20.] A.M. Bassily, 2001 "Effects of Evaporative Inlet and After cooling on the Recuperated Gas Turbine Cycle". Applied Thermal Engineering, Vol. 21, 1875- 1890.

[21.] D.A. Kolp, W.M. Flye, H.A. Guidotti, 1995 "Advantages of Air Conditioning and Supercharging an LM6000 Gas Turbine Inlet". Engineering for Gas Turbine and Power, Transactions of the ASME, Vol. 117, 1995, 513-527.

[22.] M.Jonsson, J. Yan, 2005" Humidified gas turbines—a review of proposed and implemented cycles" Energy 30 , 1013–1078.

[23.] Roger E. Anderson, Scott MacAdam, Fermin Viteri, 2008" ADAPTING GAS TURBINES TO ZERO EMISSION OXY-FUEL POWER PLANTS" Proceedings of ASME Turbo Expo 2008: Power for Land, Sea and Air GT 2008 June 9-13, Berlin, Germany.

[24.] C.Y. Liu, G. Chen, M. Assadi, X.S. Bai, 2011"characteristics of Oxy-fuel combustion in gas turbines" El-sevier, 2011

[25.] Hanne M. Kvamsdal1, Ola Maurstad1, Kristin Jordan, and Olav Bolland" BENCHMARKING OF GAS-TURBINE CYCLES WITH CO2 CAPTURE" 1SINTEF Energy Research, N-7465 Trondheim, Norway

[26.] Hanne M. Kvamsdal1, Olav Bolland, Ola Maurstad, and Kristin Jorda" A qualitative comparison of gas turbine cycles with CO2 capture" The Norwegian University of Science and Technology (NTNU), N-7491 Trondheim, Norway

[27.] Shinji, K. and Nobuhide, K., 2002" performance evaluation of gas turbine –fuel cell hybrid micro generation system" proceedings of ASME TURBO EXPO 2002, Amsterdam , The Netherlands.

[28.] A. Boudghene Stambouli and E. Traversa, 2002" Solid oxide fuel cells (SOFCs): a review of an environmentally clean and efficient source of energy" Renewable and Sustainable Energy Reviews 6 (2002) 433–455

[29.] M J Moore, 1997" No*x* emission control in gas turbines for combined cycle gas turbine plant" Proc Instn Mech Engrs Vol 211 Part A

[30.] [30] Christian Kaufmann, 2010 "INDUSTRIAL APPLICATIONS OF GAS TURBINES, Session #7 "Emission Reduction Case Study" Fall 2010 Course.

[31.] Mitsubishi Heavy Industries Technical Review (Mar. 2010) "Development of Key Technology for Ultra-high-temperature Gas Turbines" Vol. 47 No. 1

[32.] AQUA-CSP 2007: Trieb, F., Schillings, C., Viebahn, P., Paul, C., Altowaie, H., Sufian, T., Alnaser, W., Kabariti, M., Shahin, W., Bennouna, A., Nokraschy, H., Kern, J., Knies, G., El Bassam, N., Hasairi, I., Haddouche, A., Glade, H., Aliewi, A., "Concentrating Solar Power for Seawater Desalination" German Aerospace Center (DLR), Study for the German Ministry of Environment, Nature Conversation and Nuclear Safety, Stuttgart 2007, (www.dlr.de/tt/aqua-csp)

[33.] J.O. Jaber, S.D. Odeh, S.D. Probert, 2003"Integrated PV and gas turbine system for satisfying peak-demands". Applied Energy, Vol. 76, 2003, 305-319.

[34.] J.O. Jaber, O.O. Badran, N. Abu-Shikhah, 2004 "Sustainable energy and environmental impact: role of renewables as clean and secure source of energy for the 21st century in Jordan". Clean Technologies and Environment Policy, Vol. 6, 174-186.

[35.] J.O. Jaber, A. Al-Sarkhi, B. Akash, M. Mohsen, 2004"Medium-range planning economics of future electrical power generation". Energy Policy, Vol. 32, 357- 366.

[36.] Peter Schwarzbo¨zl, 2006 "Solar gas turbine systems: Design, cost and perspectives"Solar Energy 80 (2006) 1231–1240.

Appendix-A:
Example of calculations:

Run 1 with distillate fuel oil:

1-Gross power= $W_I = \frac{n*3600}{t} . \frac{1}{K_C} . K_A . K_V$

= (37.72*3600/1200)*(3000*120/1400)=29098.3 KW

2-W_{Aux}=$\sqrt{3}$ $*A_{aux}$ .V_{aux} . Cosϕ= $\sqrt{3}$ *500*380*0.85=279.7KW

where ,Cosϕ is assumed equal to 0.85

3-W_{Net} =Gross power-Aux. Power =29098.3-279.7=28818.6KW.

Net HR= $\frac{Gfuel.LHV}{NetPow}$ = 10935*0.8159*42578/ 28818.6 = 13181.6 KJ/KWh

Where, Gfuel(Kg/hr) = Gfue(L/hr)* δFuel (kg/L)= 10935* 0.8159 kg/hr

Note: λ1=1.1238 using an ambient temperature equal to 34.38C

λ2 = 1.0039 using an ambient pressure equal to 701.7 mmHg

β=0.9578 using an ambient temperature equal to 34.38 C

4-Power Corrected=$\frac{NetPower}{\lambda 1.\lambda 2}$ = $\frac{28818.6}{1.1238*1.0039}$ = 25544.3 KW

5-HR corrected= $\frac{HR}{\beta}$= 13181.6/ 0.9578= 13762.4 KJ/KWh

Table (A1): Sample of actual test data.

Fuel type	Distillate fuel oil
load	50%
Time at start run	17:26
Time at stop run	17:46
Test time	20 minutes
Active power at generator terminals	15.284MW
Current for auxiliary consumption	500 A
Voltage auxiliary	380V
Compressor inlet temperature	40.3,40.2, 40.1 C
Ambient pressure	701 mmHg.
Compressor inlet static loss	-176 mmH2O
Exhaust turbine static loss	-20 mmH2O
Combustor shell pressure	8.2 bar
Compressor discharge temperature	371.9, 371.3, 371.1 C
Cooler discharge temperature	152.7,152.2, 152.7 C
Exhaust gas temperature	410.2, 410.5, 410.4 C
GT speed rpm	3611, 3614, 3613 rpm
Liquid fuel temperature	52.8 C
Liquid fuel (Liters)	17,400, 18,800
time	1200 seconds