The Nigerian Institution of Mechanical Engineers
(A division of the Nigerian Society of Engineers)
Minna Chapter

A case study of Performance Analysis of Gas Turbine Power Plant in Pacific Energy Company Limited, Olorunsogo Papalanto, Nigeria

[1]Oyebamiji, M.O., [2]Nasir A.A, [3]Godfrey, M.

[1]Department of Mechanical Engineering, Federal University of Technology, Minna, Nigeria, oyebamijimuideenoladele@gmail.com a.nasir@futminna.edu.ng

[2]Quality/Environmental Management System, Sonates Resources Investment Ltd, Abuja, Nigeria, godfreymnet@gmail.com

Corresponding author: **[1]Oyebamiji, M.O.** oyebamijimuideenoladele@gmail.com, *+2348036322777*

Abstract

.
There is need to improve on the thermal power plants that is gaining momentum within the electricity generation industry to support hydro power plant. The failure of hydropower plant meeting up energy generation in Nigeria is becoming major task. This paper present issues of efficiency to boost availability of gas turbine power plant in Pacific Energy Company Limited, Olorunsogo Papalanto, Nigeria. The paper carried out thermal efficiency of two turbines of out eight turbines with 42 MW each at power rating of 0.8. The thermal efficiency of the power plant and network output and effect of ambient temperature were determined. The parameters such as ambient temperature, T_1 (^oC), ambient pressure, P_1 (bar), compressor pressure, P_2 (bar) and compressor discharge temperature, T_2 (^oC) and turbine exit temperature, T_4 (^oC) for a period of a year 2020 per hour and average of data were used in the analysis. Thermal efficiencies of turbines (η) include at ambient temperature of 33 ^oC, (η) for turbine 1 was 7.4% and turbine 2 as 9.3%. The thermal efficiency was high in rainy season due to low ambient temperature. Gas turbine power plant 1 has 45.3 kJ/kg-hour of network output and turbine 2 has network output of 59.7 kJ/kg-hour. In conclusion, thermal efficiency increases with increase network output and decrease with ambient temperature of gas turbine.

1.0 Introduction

Meeting growing demand of electricity efficiently requires improvement in the usage patterns, cost of production of energy at optimum environmental conditions and the use of modern technology in generating electricity. The need for supplying efficient electricity through thermal system is increasing due to deteriorating nature of hydropower system in Nigeria especially during the dry season (Abam *et al.*, 2011). Ensuring efficiency and availability of electricity in Nigeria requires both hydropower system and thermal power system for growth and development. Nigeria powers faces fluctuating of rain by nature and this affect hydroelectricity generation of energy especially during the dry season (Abam *et al.*, 2012; Ibrahim *et al*, 2010). Hence, there is need to improve on the thermal power plants that is gaining momentum within the electricity generation industry. To provide adequate power to ensure that Nigeria is among the industrialized nations, the energy generation of the country has been looked into with a view to improving the efficiency, availability and reliability of electricity in the country through thermal power plants. Looking into this, the energy generation has to be followed by reducing the environmental impact of the thermal power plants.

Therefore, to achieve improve energy generation, this paper analyses performance of gas turbine and this affects its availability, reliability and improved power supply in the country. Performance Gas turbine power plant is one of the leading modern options of generating electricity in thermal power system, need to be evaluated based on performance. This type analysis will support continuous improvement on maintenance and sustainability of electricity power generation (Saif and Tariq, 2014; Xiaojun *et al.*, 2010; ASME, 2005). It is because thermal power plant is fast in start-up time and controllable and react quickly to environmental fluctuations and this affect electricity supply. One sector in which gas turbine have found wide and increasing application for power generation is the oil and gas industries, and this increased popularity has raised the need to develop several methods optimize performance under any constraints environmental condition (Basrawi *et al.*, 2011; Rahman *et al.*, 2011a; Rahman *et al.*, 2011b).

Pacific Energy Company Limited, Olorunsogo Papalanto, Nigeria thermal power plant uses natural gas, which is readily available in Nigeria as the working fluid that powers the gas turbines but the environmental conditions such as humidity, ambient temperature of air, air density and pressure ratio keep changing. All these have great effect on the power output of the plant. This paper carried out performance analysis of gas turbine and Pacific Energy Company Limited, Olorunsogo Papalanto, Nigeria served as case study. The performance analysis of two (2) gas turbine power plants out of total eight (8) that belong to Pacific Energy Company Limited, Olorunsogo Papalanto, Nigeria is considered in this paper.

2.0 Performance of Gas Turbine Power Plant

Gas turbines are widely used for electricity generation in most countries of the world. They can be started and stopped easily allowing them to be brought into service as required to meet energy demand at peak conditions (Jaber *et al.*, 2007). Because of natural gas availability and at low prices compared to distillate fuels, many countries of the world, example Nigeria utilizes large conventional gas turbines as based load units (Abam *et al.*, 2012). The average efficiencies of gas turbine plants in the Nigerian energy utility sector over the past decade was in the range 27-30% (Abam *et al.*, 2011). The low efficiencies of the gas turbine plants are tired to many factors which include: operation mode, poor maintenance procedures, age of plant, discrepancies in operating data, high ambient temperature and relative humidity. Power output and efficiency of a gas turbine plant depends largely on the condition of the compressor inlet air temperature (Cortes and William, 2003). The performance output during hot conditions is less compared to the performance at high air temperature and humid environment, so cooling the inlet air temperature to gas turbine, increases the air density, which enhances the mass flow rate of air and gives better power output (Jaber *et al.*, 2007).

Overall efficiency of the gas turbine cycle depends primarily upon the pressure ratio of the compressor. It is important to realize that in the gas turbine the processes of compression, combustion and expansion do not occur in a single component as they occurred in a reciprocating engine. It is well known that the performance can be qualified with respect to its efficiency, power output, specific fuel consumption as well as work ratio. There are several parameters that affect its performance including the compressor compression ratio, combustion inlet temperature and turbine inlet temperature (TIT) (Mahmood and Mahdi, 2009, Rahman *et al.*, 2010).

Taniquchi and Miyamae (2000) carried out the study on the effects of ambient temperature, ambient pressure as well as the temperature of exhaust gases on performance of gas turbine. It was discovered that there is an obvious drop in the power output as the ambient air temperature increases, if an increase of intake air ambient temperature from ISO condition 15 to 30°C which is 10% decrease in the net power output. This is particularly relevant in tropical climates where the temperature varies 25 to 35°C throughout the year (Boonnasa *et al.*, 2006). Figure 1 illustrates gas turbine power plants with four components that include the following major components: compressor, combustion chamber (CC), turbine and generator.

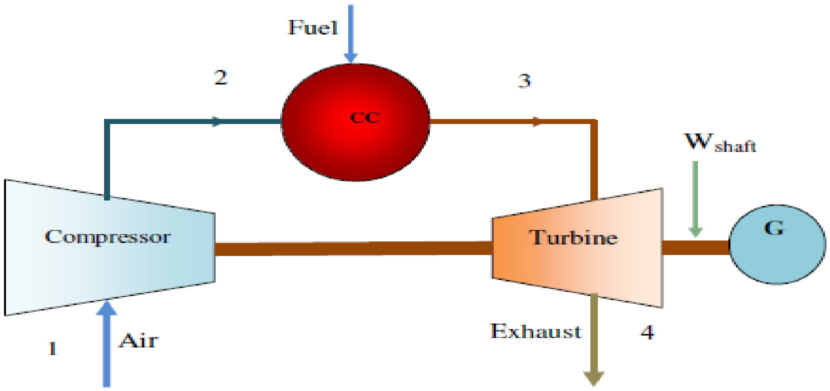

Figure 1: Gas Turbine Power Plants

The fresh atmospheric air is drawn into the circuit continuously and energy is added by the combustion of the fuel in the working fluid itself. The products of combustion are expanded through the turbine which produces the work and finally discharges to the atmosphere. Several gas turbines are being widely used for power generation in several countries all over the world. Many of these countries have a wide range of climatic conditions, which impact the performance of gas turbines (Amir and Ali, 2006). The Gas Turbine power plant works on a Joule-Brayton cycle Zhang *et al.*, 2009). The gas turbine is a complex machine, and its performance and reliability are governed by many standards. The reliability of the turbines depends on the mechanical codes that govern the design of gas turbines (ASME, 2005). The mechanical standards and codes have been written by both ASME and the American Petroleum Institute (API) amongst others. The major variables that affect the gas turbines are types of application, plant location and site configuration, plant size and efficiency, types of fuel, enclosures and plant operation mode (Saif and Tariq, 2014). The performance of a gas turbine power plant is sensible to the ambient condition. As the ambient air temperature arises, less air can be compressed by the compressor since the withdrawing capacity of compressor is given, and so the gas turbine output is reduced at a given turbine entry temperature. Additionally, the compression work increases because the limited volume of the air increases in proportionality to the intake air temperature (Xiaojun *et al.*, 2010).

Gas turbines are constant volume machines; at a given shaft speed, they always move the same volume of air. In gas turbines, since the combustion air is taken directly from the environment, their performance is strongly affected by weather conditions (Mahmoudi *et al.*, 2009). There is a loss of 3.36 % in thermal efficiency at pressure ratio 9 as ambient temperature increases from 283 K to 313 K, this loss increases up to 3.89 % at pressure ratio 21, which reveals that, as pressure ratio increases percentage loss in thermal efficiency increases on increasing the ambient temperature (Basrawi *et al.*, 2011).

The thermal performance analysis reveals that, the ambient temperature and compression ratios are strongly influence the performance of gas turbine power plant. Saif and Tariq (2014) stated that that the performance of gas turbine can be qualified with respect to its efficiency, power output, specific fuel consumption as well as work ratio. There are several parameters that affect its performance including the compressor compression ratio, combustion inlet temperature and turbine inlet temperature (TIT) (Abam *et al.*, 2018; Al-Sayed, 2008; Boonnasa *et al.*, 2006; Mahmood and Mahdi, 2009).

Simple gas turbine cycle calculations with realistic parameters are made and confirm that increasing the turbine inlet temperature no longer means an increase in cycle efficiency, but increases the work done (Aram and Mohd, 2013). Further, the performance improvement of the gas turbine is dependent on the maximum temperature tolerance of the first stage blades and is also reliant on inter stage cooling at the compression stage (Carniere *et al.*, 2006). Several methods and technologies are available to augment this power loss but this entails additional plant and equipment installation as well as additional operational requirements (Ibrahim *et al.*, 2010;

Horlock, 2003). Many of these methods such as use of air cooler regenerative steam injection effusive blade cooling techniques (Saif and Tariq 2014; Ablay, 2013). Some use desiccant-based evaporative cooling absorption chillers (Kang and Ahn 2017). The effect of relative humidity on the gas turbine power plant addresses issues of the air cooling and enhances compressor efficiency (Cao and Dai, 2017).

3.0 Materials and Methods

Daily turbine operating data of two units of gas turbine were collected within a single calendar year 2020. The obtainable data from Pacific Energy Company Limited, Olorunsogo Papalanto, Nigeria include the following:

i. ambient temperature, T_1 (^oC)
ii. ambient pressure, P_1 (mmH_2O)
iii. compressor pressure, P_2 (bar)
iv. compressor discharge temperature, T_2 (^oC)
v. turbine exit temperature, T_4 (^oC)

These data are obtained from operation data sheet of **42 MW Hitachi turbine** plant with power rating factor of 0.8. The parameter influence gas turbine performance is such as compression ratio, turbine inlet temperature, air to fuel ratio, and ambient temperature (Al-Sayed, 2008). The paper noted turbine efficiency as (η_t). Figure 2 represents T-S diagram of open gas turbine cycle process.

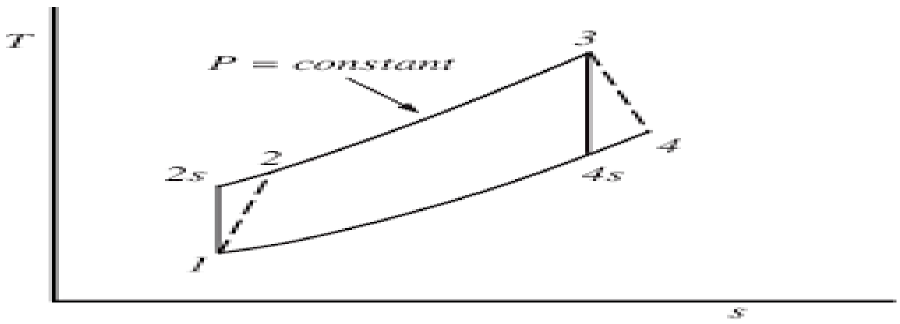

Figure 2: T-S diagram of Gas Turbine

Where:

Line 1-2 is irreversible adiabatic compression

Line 1-2s is ideal isentropic compression

Line 2-3 is constant pressure heat input by combustion chamber

Line 3-4 is irreversible adiabatic expansion

Line 3-4s is isentropic expansion

By assuming the change in kinetic energy between the various points in the cycle is negligible, the following heat flow expression is valid (Rajput, 2013):

$$W_i = c_p(T_2 - T_1) \tag{1}$$

$$H_i = c_p(T_3 - T_2) \tag{2}$$

$$W_o = c_p(T_3 - T_4) \tag{3}$$

Network output, $W_n = c_p(T_3 - T_4) - c_p(T_2 - T_1)$ (4)

Thermal efficiency, $\eta = W_n/H_i$ (5)

Turbine isentropic efficiency, $\eta_t = c_p(T_3 - T_4)/c_p(T_3 - T_4 s)$

$$\eta_t = T_3 - T_4/(T_3 - T_4 s) \qquad (6)$$

Where:

W_i = work input by compressor in kJ/kg

c_p = specific heat at constant pressure and it is assumed to be 1.005 kJ/kgK

T_2 = irreversible adiabatic compressor temperature in Kelvin (K)

T_1 = ambient temperature in Kelvin (K)

H_i = heat supplied by combustion chamber in kJ/kg

T_3 = temperature in which heat is supplied by combustion chamber at constant pressure in Kelvin (K)

W_o = work output by turbine in kJ/kg

T_4 = irreversible adiabatic expansion temperature in Kelvin (K)

T_{4s} = isentropic expansion in turbine in Kelvin (K)

4.0 Results and Discussions

4.1 Performance Analysis

Table 1 is the average data obtained per hour in year 2020 from turbine 1

Table 1: Turbine 1 Average hourly data in Year 2020

Parameter	**Quantity**	
	Measured	**Conversion**
Time	1 hour	
Ambient temperature, T_1	30.8 ^oC	303.95 K
Compressor discharge temperature, T_2	371.5	644.65 K
Turbine exit temperature, T_4	546.5	819.65 K
Ambient pressure, P_1		1 bar
Compressor pressure, P2		9.8 bar

Thus: $P_3 = P_2 = 9.8\, bar$, $P_4 = P_1 = 1\, bar$ and air-fuel constant, $\gamma = 1.4$

Isentropic process in compressor stage: pressure ratio, $\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} = \frac{T_{2s}}{T_1} = \left(\frac{9.8}{1}\right)^{\frac{0.4}{1.4}} = \frac{T_{2s}}{303.95}$

$(9.8)^{0.2857} = \frac{T_{2s}}{303.95} \rightarrow T_{2s} = 1.9 \times 303.95 = 577.5\ \text{K}$

Isentropic efficiency of the compressor, $\eta_c = \frac{T_{2s} - T_1}{T_2 - T_1} = \frac{577.5 - 303.95}{644.65 - 303.95} = \frac{273.55}{340.7} = 0.8$

$W_i = c_p(T_2 - T_1) = 1.005(644.65 - 303.95) = 342.4\, kJ/kgK$

Temperature, T_3, at combustion chamber is related as Rahman *et al.*, (2011b):

$$T_4 = T_3(1 - \eta_t R_g) \tag{7}$$

$$R_g = 1 - \frac{1}{P_r^{\left(\frac{\gamma_g - 1}{\gamma_g}\right)}} \tag{8}$$

Let, $\gamma_g = 1.3, P_r = \left(\frac{P_2}{P_1}\right) = 9.8$, isentropic efficiency of turbine (η_t) is 0.8 due to power rating value of 0.8 and $R_g = 1 - \frac{1}{9.8^{\left(\frac{0.3}{1.33}\right)}} = 1 - \frac{1}{9.8^{(0.226)}} = 1 - \frac{1}{1.68} = 1 - 0.6 = 0.4$

$819.65 = T_3(1 - 0.8 \times 0.4) = T_3(0.68) \quad \rightarrow T_3 = \frac{819.65}{(0.68)} = 1205.4\, K$

If, $\eta_t = \frac{T_3 - T_4}{T_3 - T_{4s}} = \frac{1205.4 - 819.65}{1205.4 - T_{4s}} = \frac{385.75}{1205.4 - T_{4s}} = 0.8$

$385.75 = 0.8(644.65 - T_{4s}) \rightarrow T_{4s} = 1205.4 - \frac{385.75}{0.8} = 723.2\, K$

$H_i = c_p(T_3 - T_2) = 1.005\ (1205.4 - 644.65) = 563.6$ kJ/kg

$W_o = c_p(T_3 - T_4) = 1.005\ (1205.4 - 819.65) = 387.7$ kJ/kg

Network output, $W_n = W_o - W_i = 387.7 - 342.4 = 45.3$ kJ/kg

Thermal efficiency, $\eta = \frac{W_n}{H_i} = \frac{45.3}{563.6} = 0.08$

Table 2 is the average data obtained per hour in year 2020 from turbine 2

Table 2: Turbine 2 Average hourly data in Year 2020

Parameter	**Quantity**	
	Measured	**Conversion**
Time	1 hour	
Ambient temperature, T_1	31 ^{o}C	304.15 K
Compressor discharge temperature, T_2	365.5	638.65 K
Turbine exit temperature, T_4	538.5	811.45 K
Ambient pressure, P_1	1 bar	
Compressor pressure, P2	9.7 bar	

The analysis is the same as for turbine 1: $P_3 = P_2 = 9.7\ bar$, $P_4 = P_1 = 0.0108\ bar$ and air-fuel constant, $\gamma = 1.4$. Isentropic process in compressor stage: pressure ratio:

$$\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} = \frac{T_{2s}}{T_1} = \left(\frac{9.7}{1}\right)^{\frac{0.4}{1.4}} = \frac{T_{2s}}{304.15} \rightarrow (9.7)^{0.2857} = \frac{T_{2s}}{304.15} \rightarrow T_{2s} = 1.9 \times 304.15 = 577.9\text{ K}$$

Isentropic efficiency of the compressor, $\eta_c = \frac{T_{2s}-T_1}{T_2-T_1} = \frac{577.9-304.15}{638.65-304.15} = \frac{273.75}{334.5} = 0.82$

$$W_i = c_p(T_2 - T_1) = 1.005(638.65 - 304.15) = 336.2\ kJ/kg$$

T_3, at combustion chamber: $T_4 = T_3(1 - \eta_t R_g)$; $R_g = 1 - \frac{1}{P_r^{\left(\frac{\gamma g-1}{\gamma g}\right)}}$; $\gamma_g = .3, P_r = \left(\frac{P_2}{P_1}\right) = 9.7$, isentropic efficiency of turbine (η_t) is 0.8 due to power rating value of 0.8 and $R_g = 1 - \frac{1}{9.7^{\left(\frac{0.3}{1.33}\right)}} = 1 - \frac{1}{9.7^{(0.226)}} = 1 - \frac{1}{1.67} = 1 - 0.6 = 0.4$

$$811.45 = T_3(1 - 0.8 \times 0.4) = T_3(0.68) \quad \rightarrow T_3 = \frac{811.45}{(0.68)} = 1193.3\ K$$

If, $\eta_t = \frac{T_3 - T_4}{T_3 - T_{4s}} = \frac{1193.3 - 811.45}{1193.3 - T_{4s}} = \frac{381.45}{1193.3 - T_{4s}} = 0.8$

$381.45 = 0.8(1193.3 - T_{4s}) \rightarrow T_{4s} = 1193.3 - \frac{381.45}{0.8} = 723.2\ K$

$H_i = c_p(T_3 - T_2) = 1.005\ (1205.4 - 638.65) = 569.6$ kJ/kg

$W_o = c_p(T_3 - T_4) = 1.005\ (1205.4 - 811.45\ ) = 395.9$ kJ/kg

Network output, $W_n = W_o - W_i = 395.9 - 336.2 = 59.7\ \mathrm{kJ/kg}$

Thermal efficiency, $\eta = \frac{W_n}{H_i} = \frac{59.7}{569.6} = 0.1$

4.2 Effect of Ambient Temperature on the Thermal Efficiency

The analysis of inlet ambient temperature on gas turbine thermal efficiency is presented base on seasonal changes in Nigeria. The ambient temperature in wet season is averagely varied between 23 °C to 29 °C while during dry season changes from 26 °C to 33 °C. The thermal efficiency and power output decreases linearly with increase of ambient temperature. This paper tabulated thermal efficiency based on obtainable ambient temperature using linear interpolation in Table 3.

Table 3: Effect of Ambient Temperature on the Thermal Efficiency

Temperature, °C	**Thermal Efficiency**	
	Turbine 1	Turbine 2
23	0.1071	0.1348
24	0.1027	0.1292
25	0.0986	0.1240
26	0.0948	0.1192
27	0.0913	0.1148
28	0.0880	0.1107
29	0.0850	0.1069
30	0.0821	0.1033
31	0.0795	0.1000
32	0.0770	0.0969
33	0.0747	0.0939

Figure 1: Thermal efficiency of the Plant

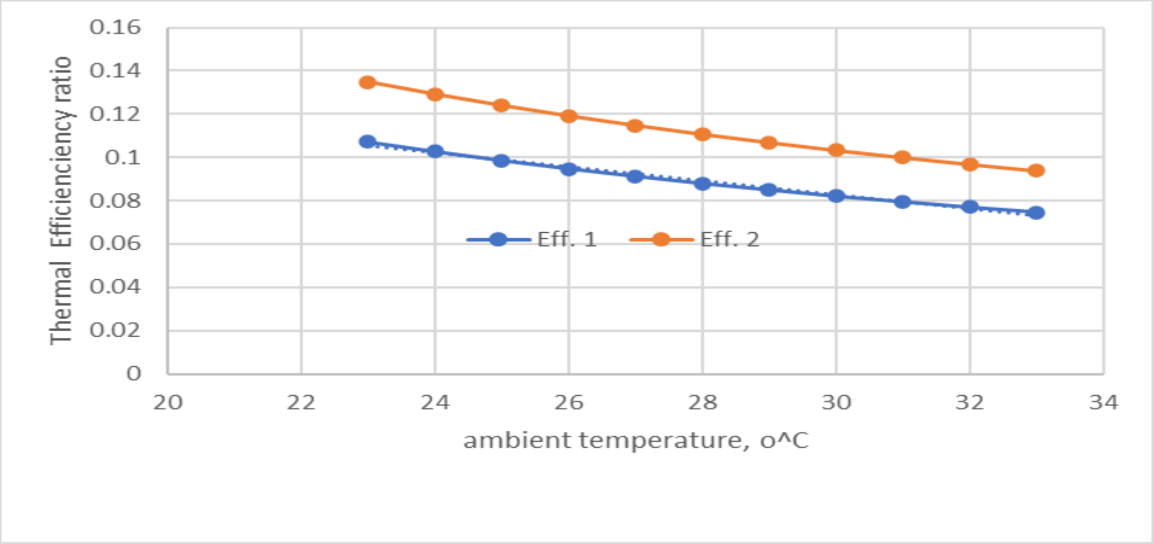

Figure 4.2 also illustrates the network output per seconds for each turbine.

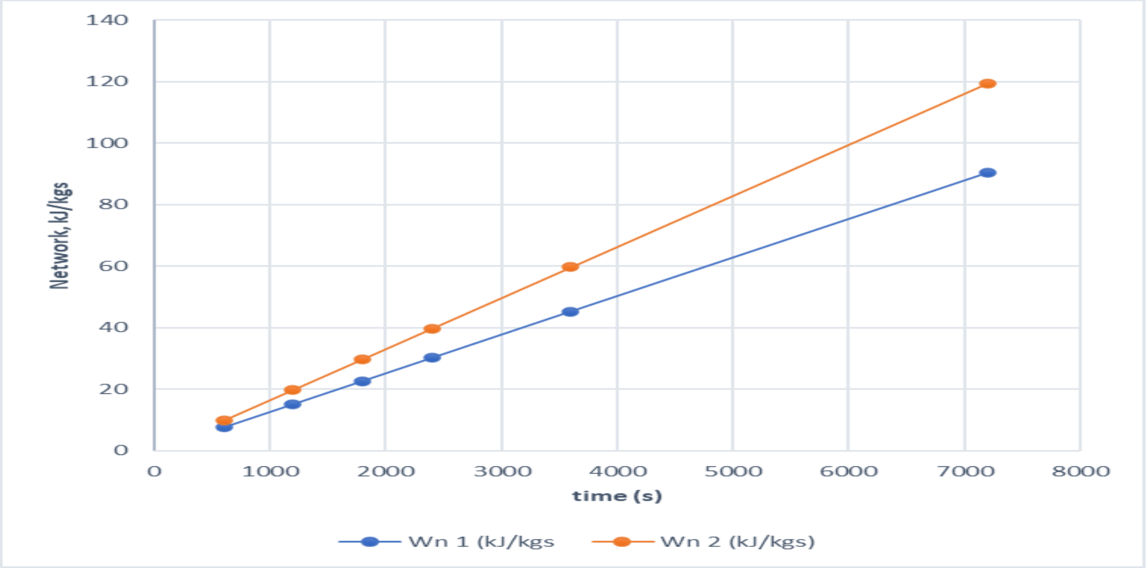

Figure 2: Network output of the Plant

4.3 Discussion

The ambient temperature varied between 23 ^{o}C to 33 ^{o}C per seasonal changes and this leads to significant effect on thermal efficiency in Table 3 and Figure 1. At inlet temperature of 23 ^{o}C the thermal efficiencies of turbines(η_t) ratio include as turbine 1 as 0.1071 (10.71 %) and turbine 2 as 0.1348 (13.48 %) ratio while at 33 ^{o}C, (η_t) for turbine 1 was 0.0747 (7.4%) and turbine 2 as 0.0939 (9.3%). The thermal efficiency incremental changes from rainy season to dry season in turbine 1 was 0.026 while turbine 2 was 0.0323 in turbine. In rainy season when the ambient temperature was low thermal efficiency ratio is high and it is opposite in dry season. The thermal efficiency ratio due to incremental changes in turbine 2, from 0.1348 to 0.0939 was 0.0409 as temperature changes from 23 to 33 ^{o}C. Also, turbine 1 has 0.0324 as efficiency ratio changes within the temperature observed. Therefore, turbine 2 has 26.2% better thermal efficiency ratio when compared with turbine 1.

Figure 2 also illustrates network output of the gas turbine power plant in which 45.3 kJ/kg-hour of network output was experienced in turbine 1 while 59.7 kJ/kg-hour of network output will be gained in turbine 2. It also means, turbine 2 has 31.8% of kJ/kg-hour of turbine 1. Compression compressor (pressure) ratio of turbine 1 is higher at 9.8:1 while the same ratio of 9.7:1 was obtained in turbine 2. According to Rahman *et al.*, (2011b) such deviation of thermal efficiency at lower compression ratio is not significant while the variation at higher compression ratio is vital for thermal efficiency. Low ambient temperature leads to a higher air density and a low compressor work gives a higher gas turbine network output which in turn provides a higher thermal efficiency. However, the outcome of the paper agreed by Rahman *et al.*, (2011b); Saif, and Tariq (2014) that thermal efficiency increases with lower ambient turbine inlet temperature.

5.0 Conclusion

This paper concluded that both thermal efficiency and network output increases with decrease in ambient temperature of gas turbine. The thermal efficiency of turbine 1 is affected by ambient temperature as a result of air density changes and compressor work. Therefore, as a result of this, turbine 2 has better thermal efficiency and network output.

REFERENCES

Abam, F.I., Ugot, I.U. and Igbong, D.I. (2011). Thermodynamic assessment of grid-based gas turbine power plants in Nigeria. Journal of Emerg. Trends Eng. Applied Sci., 2(1), 1026-1033.

Ablay, G.A.(2013). modeling and control approach to advanced nuclear power plants with gas turbines. *Energy Conversion and Management.* 76: 899-909

Amir, A.Z. and Ali, H.G., (2006). Performance improvement of a gas turbine cycle by using a desiccantbased evaporative cooling system, Energy 31 (1), 2652–2664.

ASME (2005): American Society of Mechanical Engineers, Performance Test Code on Test Uncertainty: Instruments and Apparatus PTC 10, 19.1, 22, 46. ASME, New York, 2005.

Abam, F.I., Ugot, I.U and Igbong, D.I. (2012). Performance Analysis and Components Irreversibility of A (25 MW) Gas Turbine Power Plant Modeled with a Spray Cooler, American J. of Engineering and Applied Sciences 5 (1): 35-41, 2012.

Al-Sayed, A.F. (2008). Aircraft Propulsion and Gas Turbine Engines. Taylor and Francis, ISBN 978-0-8493-9196-5.

Aram, M.A. and Mohd. T. (2013). Thermal analysis of a gas turbine power plant to improve performance efficiency, 6, (2013), 43-54.

Boonnasa S, Namprakai P, Muangnapoh T (2006). Performance improvement of the combined cycle power plant by intake air cooling using an absorption chiller. Energy, 31(12): 2036-2046.

Basrawi, F., Yamada, T., Nakanishi, K. and Naing, S, Effect of ambient temperature on the performance of micro gas turbine with cogeneration system in cold region. Applied Thermal Engineering. 31: (2011), 1058-1067.

Carniere, H., Willocx, A., Dick, E., DePaepe, M., (2006). Raising cycle efficiency by inter cooling in air cooled gas turbine, Applied Thermal Engineering 26 (16) (2006) 1780-1787.

Cao, Y., Dai, Y. (2017). Comparative analysis on off-design performance of a gas turbine and ORC combined cycle under different operation approaches. Energy Conversion and Management. 2017;135:84-100

Cengel, Y.A. and M.A. Boles, 2010. Thermodynamics: An Engineering Approach. 7th Edn., McGraw-Hill, New York, ISBN-10: 9780073529325 pp: 1024.

Cortes, C.R. and D.F. Williams, 2003. Gas turbine inlet air cooling techniques: An overview of current technologies. Power-Gen International, Las Vegas, Neva, USA

Himmelblau, D.M. and J.B. Riggs, 2012. Basic Principles and Calculations in Chemical Engineering. 8th Edn., Prentice Hall London, New Jersi, ISBN: 9780132346603, pp: 752.

Horlock, J.H (2003). Advance Gas Turbine Cycles. Elsevier Sci. Ltd., UK. Ibrahim TK, Rahman MM, Alla AN (2010). Study on the effective parameter of gas turbine model with intercooled compression process. Sci. Res. Essays, 5(23): 3760-3770.

Ibrahim, T.K. Rahman, M.M and Abdalla, A.N. (2010). Study on the effective parameter of gas turbine model with intercooled compression processes (2010): Academic Journals - Scientific Research and Essays. ISSN: 1992-2248 5 (Suppl. 23) (2010). ISSN: 1992-2248 3760-377

Jaber, Q.M., J.O. Jaber and M.A. Khawaldah, 2007. Assessment of power augmentation from gas turbine power plants using different inlet air cooling systems. J. Mech. Ind. Eng., 1: 7-15.

Johnke, T. and Mast, M. (2002). Gas Turbine Power Boosters to enhance power output. Siemens Power for generation, Siemens Power Journal

Kang, S and Ahn, K. (2017). Dynamic modelling of solid oxide fuel cell and engine hybrid system for distributed power generation. Applied Energy. 2017;195: 1086-1099

Kumara, N.R., K.R. Krishnab and A.V.S.R. Rajuc, 2007. Performance improvement and exergy analysis of gas turbine power plant with alternative regenerator and intake air cooling. Energy Eng., 104: 36-53. DOI: 10.1080/01998590709509498

Mahmood, F.G and Mahdi, D.D. (2009). A new approach for enhancing performance of a gas turbine (Case study: Khangiran Refinery). Appl. Energy, 86: 2750–2759.

Mahmoudi, S.M, Zare, V, Ranjbar, F. and Farshi. (2009). Energy and exergy analysis of simple and regenerative gas turbines inlet air cooling using absorption refrigeration. J. Appl. Sci., 9(13) 2399-2407.

Rahman, M.M, Ibrahim TK, Kadirgama K, Mamat R, Bakar RA (2011a). Influence of operation conditions and ambient temperature onperformance of gas turbine power plant. Adv. Mater. Res., 189-193: 3007-3013.

Rahman, M.M., Ibrahim, T.K. and Abdalla, A.N. (2011b). Thermodynamic performance analysis of gas-turbine power-plant (2011). International Journal of the Physical Sciences, 6(14), 3539-3550.

Saif, M., and Tariq, M. (2014). Thermal analysis of a gas turbine power plant at uran, India, International journal of current research, 6, (2014) 5459-5463.

Xiaojun, S, Brian, A, Defu, C., and Jianmin, G., (2010). Performance enhancement of conventional combined cycle power plant by inlet air cooling, inter-cooling and LNG cold energy utilization. Appl. Ther. Eng., 30(1), 2003-2010.

Zhang, W., Lingen, O., and Fengrui, S., (2009). Power and efficiency optimization for combined Brayton and inverse Brayton cycles, Applied Thermal Engineering 29 (14-15) (2009) 2885-2894.