Optimisation of Gas Turbine Power Output

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Abstract— Currently, the usefulness of gas turbines in the power generation business cannot be undermined. But to get the most value from their use, they should be operated in a way to maximise power output, minimise fuel consumption with increased thermal efficiency and minimal negative impact on the environment. This research aims at improving the power output of GT 20, which will also help to improve its performance; and will be significant at providing more power to Nigeria's national grid from the unit for increased production activities. Experimental plants GT 20 and GGTPP were compared with a model plant used to simulate their behaviour. The results of the study show that GT 20 maximum power output from its operation was 92MW with efficiency of 0.34 at a pressure ratio of 10.27. Comparison made between GT 20 and the two other power plants - GGTPP and the model indicate that the low performance of GT 20 was due to unfavourable operating conditions of the unit, but the unit is a much older plant. A mix of optimisation variables of mass flow, compressor inlet temperature, turbine inlet temperature, and pressure ratio were applied to the model and used to optimise GT 20 and its power output. The plant has been optimised at a power output of 100.07 MW and an efficiency of 0.35, with improvements in its performance measures by 5%, 2.94%, and 8.77% in specific fuel consumption, overall efficiency and power output respectively.

Keywords— Gas Turbine, Simulation, Optimisation, Power Output, Performance.

INTRODUCTION

Adequate and sustained power supply is a frame that supports the economic growth and development of any nation. Gas turbines are internal combustion engines that operate in a Brayton cycle and used to generate large amount of electricity for industrial as well as for public use. The abundance of natural gas in Nigeria clearly favours the adoption of gas turbines for the provision of power for domestic and industrial purposes.

Power plant performance is critical and not to be undermined if optimal utilisation of the plant as well as environmental sustainability is to be promoted.

Gas turbine power plants performance is influenced by the prevailing operating conditions. The power output for both plants increases with increasing pressure ratio (Ayadju and Ighodalo, 2016), although there is a limit to this. It is proper to understand how these influences affect performance.

The parameters related to system elements like compressor, combustor and the turbines, etc. are matched for their peak performances to achieve their maximum efficiency in the integrated system (Yadav, Khan, and Grover, 2010). Key parameters for performance measurement are specific fuel consumption and power output. The performance of a gas turbine power plant is commonly presented in function of power output and specific fuel consumption (dos Santos, Andrade, and Zaparoli, 2012). Therefore, it is pertinent to optimise the power output to ensure a most beneficial utilisation of the plant. This can be achieved by applying a model plant to the study of the variables that affect performance so as to be able to make the necessary adjustments.

The gas turbine (GT) performance is affected by component efficiencies and turbine working temperature (Rahman, Ibrahim and Abdalla, 2011). The increase in ambient temperature causes decreased thermal efficiency, but the increase in turbine inlet temperature increase thermal efficiency (Rahman, Ibrahim, Taib, Noor, and Bakar, 2010). It is important to mention that over the year's researchers have carried out studies with the intent of improving gas turbine power plant performance. New and advanced materials are to be developed for matching better performance at high firing temperatures (Yadav, Khan, and Grover, 2010).

THEORETICAL ANALYSIS

2.1 Major Mechanical Components of Gas Turbine Power Plants

The schematic diagram of a gas turbine power plant with its major mechanical components is shown in figure 1, 1-2 is compressor, 2-3 is combustion chamber, and 3-4 is turbine (expander). Usually, in gas turbines shafts are used to connect expanders to compressors to drive them.

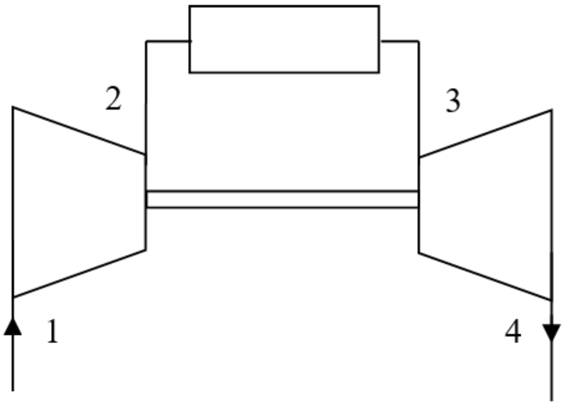

Fig. 1: Schematic Diagram of Gas Turbine Power Plant

2.1.1 Compressor:

The compressor is used to compress ambient air from the environment to a higher pressure and temperature. The governing equations for the isentropic process in the compressor are given in equations (1) and (2) in respect of the outlet temperature and compressor work

$$T_{2i} = T_1(P_2/P_1)^{(\gamma_a-1)/\gamma_a}$$
(1)

$$W_{ci} = C_{pa}(T_{2i} - T_1) \qquad (2)$$

The actual compressor work is related as

$$W_c = W_{ci}/\eta_c$$
(3)

where, T_1 is ambient temperature of air at compressor inlet, P_1 is ambient pressure, P_2 *is* pressure to which air is compressed, $P_2/P_1 = r_p$ is pressure ratio, η_c *is* Isentropic efficiency of compressor, T_{2i} *is* isentropic compressor outlet temperature, W_c is actual compressor work, W_{ci} is isentropic compressor work, η_c is compressor isentropic efficiency; $\gamma_a = 1.4$ is ratio of specific heats for air for the compression process, and $C_{pa} = 1.005 \text{kJ/kg}^o\text{C}$ is specific heat at constant pressure for air. The International Standard Organization (ISO) condition includes $T_1 = 15^oC$, $P_1 = 1.01325$ bar.

2.1.2 Combustion Chamber:

A mixture of compressed air and natural gas is ignited and burned at constant pressure in the combustion chamber. This process is modelled by the equation (4).

$$Q_{in} = m_a C_{pg}\,(T_3 - T_2)$$
(4)

where, Q_{in} is energy rate in combustion chamber, m_a is the mass flow of air, T_2 and T_3 are actual compressor outlet temperature and turbine inlet temperature respectively. The value of the specific heat at constant pressure for flue gas, $C_{pg} = 1.15$ kJ/kg°C..

2.1.3 Turbine:

The turbine is used to expand the flue gas through it, extract power and exit the exhaust gases. The isentropic process is modelled by equation (5)

$$T_{4i} = T_3/(P_2/P_1)^{(\gamma_g-1)/\gamma_g}$$
(5)

where $\gamma_g = 1.33$ is the ratio of specific heats for flue gas. The actual turbine work is

$$W_t = C_{pg}\,(T_3 - T_4)$$
(6)

The actual turbine work is also related as

$$W_t = \eta_t W_{ti}$$
(7)

where, T_4 is Temperature at turbine outlet, and , W_t is actual turbine work, W_{ti} is isentropic turbine work, and η_t is turbine isentropic efficiency.

The ideal thermal efficiency is given as

$$\eta_{th} = 1 - r_p^{1/(\gamma-1)/\gamma}$$
(8)

dos Santos, Andrade, and Zaparoli (2012) supplied the natural gas mass flow rate m_f, and thermal efficiency of gas turbine η as

$$m_f = (Q_{in}/FHV)/\,\eta_{combustor}$$
(9)

and

$$\eta = 3600/SFC\,.FHV$$
(10)

respectively, where, $\eta_{combustor}$ is combustion chamber efficiency, SFC and FHV are specific fuel consumption and fuel heating value respectively.

The overall efficiency, η_o for the gas turbine is

$$\eta_o = P_{net}/Q_{in}$$
(11)

where, P_{net} = net power output, and up and bottom parameter units being in mega Watt.

METHODOLOGY

Operational data of two different real experimental plants, GT 20 an old plant, and GGTTP a much newer one were collected and used for studying to determine their condition. The data include temperature, pressure, mass flow and power output, thereafter, the behaviour of the plants were modelled.

Comparison was made between the model and experimental data. A mix of optimisation variables of mass flow, compressor inlet temperature, turbine inlet temperature, and pressure ratio were applied to the program function and used to optimise the experimental plant GT 20 and its power output.

An optimisation method, was used to realise the efficiency improvement of all investigated compressors (Marchukou, Egorov, Popov, Baturin, Goriachkin, Novikova and Kolmakova, 2017).

RESULTS AND DISCUSSION

Table I: Comparison of Experimental and Simulation Plants
(Simulation at P_1 = 1.013 bar, T_1 = 288K, T_3 = 1500K)

Power Plant/Output	Power (MW)	Mass Flow (kg/s)	Efficiency, ηo	Pressure Ratio, rp	SFC (kg/kW.hr)
GT 20 at Maximum Power Output	92	340.18	0.34	10.27	0.20
GGTPP at Maximum Power Output	145	442.43	0.36	11.5	0.19
Actual Cycle Simulation	180.42	500	0.36	10.86	0.19
Actual Cycle Simulation at Maximum Power Output	181.42	500	0.38	13.3	0.18

Table II: Optimisation of GT 20
(Constraints applied to the plant: 300K < = T_1 < = 304K; 1340K < = T_3 < = 1400K; r_p = 10.86)

Mass flow (kg/s)	T1 (K)	T3 (K)	SFC (kg/kW.hr)	ηo	P (MW)
300	301	1340	0.20	0.34	81.56
300	301	1380	0.19	0.34	87.10
320	301	1380	0.19	0.34	92.91
288	304	1400	0.19	0.34	85.29
288	304	1400	0.19	0.35	86.62
300	300	1400	0.19	0.35	90.23
337	303	1392	0.19	0.34	98.94
320	300	1400	0.19	0.35	96.24
334	301	1400	0.19	0.35	100.07
330	302	1400	0.19	0.35	98.49
335	302	1400	0.19	0.35	99.98
340	302	1380	0.19	0.34	98.33
331	302	1400	0.19	0.35	98.79
331	301	1400	0.19	0.35	99.17
332	301	1400	0.19	0.35	99.47
333	301	1400	0.19	0.35	99.77

The data analysis indicates that GT 20 performance was lowest when compared with GGTPP and the simulation (Table 1). A cause to this low performance could arise from the prevailing operating conditions on the one hand and age among other factors on the other hand. Its maximum derived power output and efficiency were 92MW and 0.34 respectively at a mass flow of 340.18kg/s when compared with those of GGTPP which stood at 145MW and 0.36 at mass flow of 442.43kg/s. This low efficiency of GT 20 can be attributed to its low pressure ratio of 10.27, and consequently it incurred the highest specific fuel consumption of 0.2kg/kW.hr among the plants. Actual cycle simulation has shown that power output can be increased by raising mass flow just as efficiency can be increased by raising pressure ratio. The values show that at 500kg/s the model outputted a power of 180.42MW and efficiency of 0.36 at pressure ratio of 10.86, while a maximum power output of 181.42MW and efficiency of 0.38 were achieved at pressure ratio of 13.3. A mix of optimisation variables with compressor inlet temperature constrained to between 300K and 304K, turbine inlet temperature in the limits of 1340K to 1400K presented in Table 2 indicate that GT 20 was optimised to an efficiency of 0.35 and with power output of 100.07MW when the compressor inlet temperature, turbine inlet temperature and mass flow were set to 301K, 1400K and 334kg/s respectively. By this, GT 20 improvements in its performance measures were 5%, 2.94%, and 8.77% in specific fuel consumption, overall efficiency and power output respectively when compared to its existing state. More electrical energy will be available from use from GT 20, better plant utilisation as well as less impact on the environment from harmful emissions due to operation of the plant.

CONCLUSION

It was seen that unfavourable operating conditions caused the low performance of the GT 20. Its maximum power output and efficiency were 92MW and 0.34 respectively at a pressure ratio of 10.27 and mass flow of 340.18kg/s. But GGTPP which was operated at a higher pressure ratio of 11.5 and mass flow of 442.43kg/s had a better performance in terms of power output and efficiency with values of 145MW and 0.36 accordingly. This observation was further strengthened with the model plant power output of 181.42MW and efficiency of 0.38 when its mass flow and pressure ratio were set at 500kg/s and 13,3 respectively. The mix of optimisation variables of mass flow, compressor inlet temperature, turbine inlet temperature, and pressure ratio applied to the model was used to enhance the performance of GT 20. The plant has been optimised at a power output of 100.07 MW and efficiency of 0.35, with improvements in its performance measures by 5%, 2.94%, and 8.77% in specific fuel consumption, overall efficiency and power output respectively.

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