

OPTIMIZATION OF THE SYSTEM PERFORMANCE OF A GAS-TURBINE PLANT

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Abstract

This paper presents exergy based analysis and power turbine inlet temperature drop techniques for the optimization of the plant exergetic efficiency. Actual field investigation of the individual component performance was achieved using data from plant operational log sheets. Pertinent parameters which could not be measured directly were obtained using appropriate existing equations. The parameters investigated were the thermal efficiency, work ratio, thermal discharge index, net plant heat rate, and the exergetic efficiency. The exergetic efficiency and exergy destruction in the plant were largely affected by variation in the power turbine inlet temperature. For a 15percent reduction in power turbine inlet temperature an increase of 83.71percent in exergetic efficiency was obtained.

Keywords: Optimization, Energy, Gas-turbine, Efficiency

1.0 INTRODUCTION

The gas turbine is a very satisfactory means of producing mechanical power. Air is drawn in through the air intake system into the compressor from the atmosphere. The compressed air is mixed with gas and burnt in the combustion chamber. The products of this combustion are allowed to expand and the mechanical work produced drives the turbine which in turn drives the compressor(Lebele-Alawa,2009). The energy produced is a crucial prerequisite for economic and social development. Energy is used to process raw materials and convert them to more useful products. Energy availability is also critical for ensuring the services in the areas of transportation, water and healthcare delivery, operations of tools and gadgets in institutions and homes and telecommunication amongst others thereby improving our standard of living. There is an increase demand for energy all over the world, while there is also a corresponding decline in the energy resources. With the limiting energy resources and large energy demands, it becomes necessary to understand the mechanisms which degrade the quality of energy (ability to do work) and energy resources and to develop a systematic approach to optimizing the system performance.

Ogbonnaya(2011) studied gas-turbine performance optimization using compressor online water washing technique. The online washing method was done on weekly intervals for a period of 15weeks. De-mineralized water was injected through a configuration of small nozzles in the air flow before the first compressor stage. The results showed that the plant operating pressure increased from 7.0 to 8.8bar. This yielded a compressor efficiency of 82percent and an overall operational efficiency of 46.25percent. Marrone et al(2009) worked on optimizing the geometry of the cooling system for the first stage stator nozzles by means of a genetic algorithm. This was interfaced with a thermo-fluid dynamic code which computed the Nusselt number. Cooling of the gas turbine inlet air has also been adopted as an optimization technique. Review of the techniques adapted to cool gas-turbine inlet air revealed that the efficiency of evaporative cooler largely depends on the moisture present in the air (Ibrahim et al,2011).

The need for power utilization has given rise to increased susceptibility to failure as a result of improper integration of the synthesis and optimization of the components of a system. Therefore, a critical need exists for improved analysis and optimization of system performance. Exergy based analysis which can relate every system component to overall system requirements in a framework, is reasonably, a mature and holistic approach for the analysis and optimization of power plants. This is the focus of the present work. Exergy is a thermodynamic property which enhances our understanding of thermal and chemical processes by providing information on the loss mechanism present in a process. Exergy allows one to examine any complex process in relation to theoretical most efficient manner by which the process could be carried out with respect to a given reference environment. Exergy analysis allows one quantify the loss of efficiency in a process due to loss of quality of energy.

Exergy analysis can thus, point in the direction of how a process can be improved and to what extent within the process. The present work was carried out on an operational gas-turbine plant at Kolo Creek in southern Nigeria. The power plant has a rated capacity of 20mW and operates on Brayton cycle.

2. Materials and methods

The research methodology involve collection of data from actual plant operation log sheets from January 2006 to December 2009. Actual field investigation of the individual component performance was also done. Parameters which could not be directly measured were derived using appropriate existing equations.

Parameters considered during the data collection were the pressures, temperatures and mass flow rates at various state points as indicated in figure1.

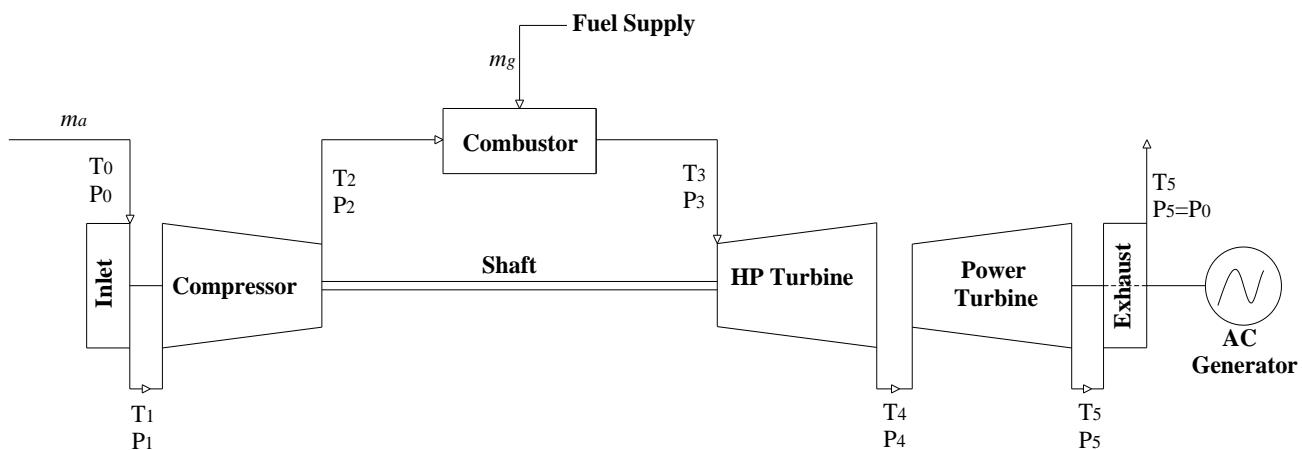


Fig 1: Simplified arrangement of plant component

In the analysis and treatment of the data, mean values of daily parameters were computed using statistical methods. This was followed by monthly average and the overall average for the period of the research. The actual performance of the power plant over the period of its installations was determined from these average parameters. From the measurements, more parameters such as pressure ratios, thermal efficiency, isentropic and actual, mechanical efficiency, specific fuel consumptions, heat rates, thermal discharge index and work ratio were obtained.

3.0 RESULTS AND DISCUSSIONS

Table1 shows the overall average of working parameters of Kolo Creek Gas Turbine Power Plant, extracted from control room log sheets for the period under investigation. The parameters for calculation in the analysis were based on table1.

Table1: Summary of average working parameters of Kolo Creek Gas Turbine Power plant.

Component	Parameter	Unit	Actual Value from Log Sheets	Calculated Value
Compressor	Inlet temperature, T_1 Outlet temperature, T_2 Inlet pressure, P_1 Outlet pressure, P_2 Mass flow rate, m_a Compressor work, W_c	K K Bar Bar kg/s KJ/kg	303.50 - 1.013 5.91 87.47 -	- 552.07 - - - 249.81
Combustion Chamber	Inlet temperature, T_2 Outlet temperature, T_3 Inlet pressure, P_2 Outlet pressure, P_3 Mass flow rate, m_g	K K Bar Bar kg/s	- - 5.91 - 87.47	5.91 1129.40 - 5.73 -
HP Turbine	Inlet temperature, T_3 Outlet temperature, T_4 Inlet pressure, P_3 Outlet pressure, P_4 Mass flow rate, m_g	K K Bar Bar kg/s	- 887.83 - - 87.47	1129.40 - 5.73 1.93 -
Power Turbine	Inlet temperature, T_4 Outlet temperature, T_5 Inlet pressure, P_4 Outlet pressure, $P_5=P_{atm}$ Mass flow rate, m_g Power output, P	K K Bar Bar kg/s MW	887.83 - - 1.013 87.47 10.45	- 768.36 1.93 - - -
Exhaust	Temperature of effluent gases, T_5 Pressure of effluent gases, P_5 Mass flow rate	K Bar kg/s	- 1.013 87.47	768.36 - - -

3.1 Performance Criteria

The parameters were computed from standard thermodynamic equations from Eastop and McConkey,2001, Nag,2001 and Rogers and Mayhew, 2009.

3.1.1 Thermal Efficiency, η_{th}

$$\eta_{th} = \frac{\text{Net Work output}}{\text{Heat Supplied}} = \frac{C_{pg}(T_4 - T_5)}{C_{pg}(T_3 - T_2)} \quad (1)$$

Substituting values from table1 gives the required results

$$\eta_{th} = 20.73\%$$

3.1.2 Work Ratio

$$\text{Work Ratio} = \frac{\text{Net Work output}}{\text{Gross Work output}} \quad (2)$$

Gross work = sum of work done by compressor and power turbine.

Work done by compressor is given by:

$$C_{pa}(T_2 - T_1) \quad (3)$$

Work output from L.P. Turbine is,

$$C_{pg}(T_4 - T_5) \quad (4)$$

Substituting values from table1, the gross work output is obtained.

∴ Work ratio = 0.36

3.1.3 Thermal Discharge Index (TDI)

This is the number of thermal energy units discharged to the environment for every unit of electrical energy generated by a power plant. According to El Wakill (1985), Thermal Discharge Index is strongly dependent on the thermal efficiency of the plant, the lower it is, the more efficient the power plant; thus a low value of Thermal Discharge Index is desirable.

$$TDI = \frac{\text{Thermal Discharge to the Environment (MW)}}{\text{Electrical Output (MW)}}$$

$$TDI = \frac{P_{th}(1 - \eta_{th})}{P_{th}\eta_{th}} = \frac{1}{\eta_{th}} - 1 \quad (\text{El Wakil; 1985})$$

Where P_{th} is the thermal exergy input and η_{th} , the thermal efficiency.

$$\text{Hence, } TDI = \frac{1}{0.2072} - 1 = 3.82$$

3.1.4 Net Plant Heat Rate (NPHR)

This is an important measure of plant performance; it is defined as the amount of fuel energy input required to generate one KWh and deliver it to the transmission lines. According to Nag (2005), NPHR is given by;

$$NPHR = \frac{3600}{\eta_{th}}$$

$$\text{So that } NPHR = \frac{3600}{20.73 \times 10} = 17.37 \text{ MJ/KW}$$

3.1.5 Exergetic Efficiency

$$\eta_{e,eff} = \frac{E_{X,DESIRED}}{E_{X,FUEL}} = \frac{9.66}{59.16} = 16.33\%$$

3.1.6 Effects of Variation of Power Turbine Inlet Temperature

Assuming a 15% reduction in power turbine inlet temperature (T_4), with all other parameters remaining constant, a considerable reduction in the exhaust temperature (T_5) will be achieved.

For a 15% reduction in T_4 ;

Let $T_{4,NEW} = T_4'$ and $T_{5,NEW} = T_5'$

$$\text{Then } T_4' = 887.83 - (0.15 \times 887.83) = 754.66 \text{ K}$$

$$\text{From equation 4.6, } T_{5S} = \frac{T_4}{\left(\frac{P_4}{P_5}\right)^{\frac{1}{\gamma-1}}}$$

$$\therefore T_{5S} = \frac{754.66}{\left(\frac{1.93}{1.013}\right)^{\frac{1.4-1}{1.4}}} = 627.72\text{K}$$

Also from equation 4.7, $T_5' = T_4 - \eta_{PT}(T_4 - T_{5S})$

$$\therefore T_5' = 754.66 - 0.80(754.66 - 627.72) = 653.11\text{K}$$

With values of $T_4' = 754.66\text{K}$, $T_5' = 653.11\text{K}$, $m_g = 87.83\text{kg/s}$, $C_{pg} = 1.15\text{KJ/kg}$

$$W_{PT,IDEAL} = m_g C_{pg} (T_4' - T_5') = 87.83 \times 1.15 (754.66 - 653.11) = 10.26\text{MW}$$

$$W_{PT,ACTUAL} = \eta_{PT} W_{PT,IDEAL} = 0.80 \times 10.26 = 8.21\text{MW}$$

$$I_{PT,MC} = 8.21 - 10.26 = -2.05\text{MW}$$

$$I_{X,PT}^0 = m_g T_0 \left[C_{p,g} \ln \left(\frac{T_5'}{T_4'} \right) - R_g \ln \left(\frac{P_5}{P_4} \right) \right] = 0.535\text{MW}$$

$$E_{X,LOSS} = I_{PT,MC} + I_{X,PT} = 1.52\text{MW}$$

$$E_{X,OUT} = m_g \left[C_{p,g} (T_5' - T_0) - T_0 \left\{ C_{p,g} \ln \left(\frac{T_5'}{T_0} \right) - R_g \ln \left(\frac{P_5}{P_0} \right) \right\} \right] = 11.82\text{MW}$$

From exergy balance equation for the power turbine,

$$E_{X,IN} = E_{X,OUT} + E_{X,PT} + E_{X,LOSS}$$

$$E_{X,PT} = 31.1 - 11.82 - 1.52 = 17.76\text{MW}$$

Were $E_{X,IN} = 31.1\text{MW}$

$$\eta_{e,PT} = \frac{11.82 + 17.76}{31.1} = 95.1\%$$

$$\eta_{e,OVERALL} = \frac{E_{X,DESIRED}}{E_{X,FUEL}} = \frac{17.76}{59.16} = 30\%$$

$$\text{Increase in exergetic efficiency} = \frac{30.0 - 16.33}{16.33} = 83.71\%$$

Exergy loss in exhaust due to 15% reduction in power turbine entry temperature,

$$E_{X,LOSS} = m_g C_{pg} \left\{ (T_5' - T_0) - T_0 \ln \left(\frac{T_5'}{T_0} \right) \right\} = 11.82\text{MW}$$

Table 2: Summary of results for percentage reduction in Power Turbine entry temperature

PTET % Reduction	Power Turbine Efficiency (%)	Overall Exergetic Efficiency (%)
0	90.30	16.33
10	92.99	25.30
15	95.10	30.0
20	95.37	33.62
25	95.97	37.0
30	96.17	40.20

Reducing the power turbine inlet temperature with all other parameters remaining constant brings about reduction in the temperature of effluent gases; also, there is considerable increase in the power turbine efficiency and overall exergetic efficiency.

4. CONCLUSIONS

1. The effect of power turbine inlet temperature on the plant exergetic efficiency and exergy destruction has been analyzed. It was confirmed that the exergetic efficiency and exergy destruction in the plant were largely affected by variation in the power turbine inlet temperature. For a 15% reduction in power turbine inlet temperature, an increase of 83.71% in exergetic efficiency was realized.
2. It was discovered that as the system operates close to its maximum rated capacity (20MW), the efficiency and hence performing parameters are improved.

$$\text{Thermal efficiency, } \eta_{th} = \frac{\text{Net Work Output}}{\text{Heat Supplied}} = \frac{C_{Pg}(T_4 - T_5)}{C_{Pg}(T_3 - T_2)}$$

$$\eta_{th} = \frac{1.15(887.83 - 768.36)}{1.15(1129.40 - 552.07)} = 0.207$$

$$\therefore \eta_{th} = 20.694$$

$$\begin{aligned} \text{Work Ratio} & \frac{\text{Net WorkOutput}}{\text{Gross WorkOutput}} = \frac{C_{Pg}(T_4 - T_5)}{C_{Pa}(T_2 - T_1) + C_{Pa}(T_4 - T_5)} \\ & = \frac{1.15(887.83 - 768.36)}{1.005(552.07 - 303.50) + 1.15(887.83 - 768.36)} = 0.355 \end{aligned}$$

$$\text{Thermal discharge index (TDI)} = \frac{1}{\eta_{th}} - 1 = \frac{1}{0.207} - 1 = 4.831$$

$$\text{NSHR} = \frac{3600}{\eta_{th}} = 17.40 \text{ MJ/KWH}$$

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