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Quantitative Modeling of Aero Derivative Gas Turbine – In alliance with the Nigerian Industrialization Project

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Abstract – Reference technology is LM2500 plus G5 for power generation. The study is centered on modeling. Applied thermodynamics principle for the modeling of aero derivative gas turbine is Brayton cycle. International Standard Organization (ISO) was referenced. Some of the LM2500 plus G5 parameters were referenced, while gas/steam table was referenced for others as detailed in the parameter analysis sections. Upgrade was done by introducing intercooler into the turbine as auxiliary accessory to reduce inlet air temperature to 15^{0}C and additional blade stage to the compressor. Modeling was done in three fronts of 15^{0}C inlet air temperature and additional blade stage introduction into the axial compressor, applied the turbine output power loss principle due to increase in ambient air inlet temperature of 30^{0}C (indicative of the Nigerian actual environmental condition) and finally, economic gains of the upgrades. Modeled turbine at 30 C ambient air inlet temperature by applying the rule of thumb (17 blade stage) resulted to 30 mw. Modeled turbine at 15^{0}C ambient air inlet temperature with Intercooler in place (17 blade stage) generated an output power of 35.5 mw. Modeled turbine at 15^{0}C ambient air inlet temperature with Intercooler and an additional Compressor stage (18 blade stage) generated 48 mw. Economic gains of modeling intercooler into the aero derivative gas turbine per hour = 5,323mw = NGN 638,760,000 per hour of power generation at NGN 120,000,000 per mw. Economic Gains of modeling an additional compressor blade stage into the ADGT = 12,764 MW = NGN 1,531,680,000 per hour of power generation

Key words– Modeling, turbine, ambient. blade, intercooler, generation, compressor

I. INTRODUCTION

The technology of interest is LM2500+G5 (land and marine) aero derivative gas turbine. This is used for power generation in the power sector, power generation and gas compression in oil and gas. It serves propulsion and power generation purposes for marine sector of the economy. Research focus is modeling of aero derivative gas turbine, a path way to technology transfer to the Nigerian industrial sector.

Other major areas to be x-rayed in the future are the aero derivative gas turbine technology transfer, marketability, sustainability, research and development premised on Nigeria technological need. Every nation in the world today is determined more than ever before to harness her industrial potentials and capabilities. The basic input of industrialization are research, development, technology transfer, and sustainability. For the aforementioned to thrive, Government must create enabling environment by supporting and pioneering industrialization effort.

II. LITERATURE REVIEW

Neeraj & Bansode (2016) presented that during turbine engine operation, the inlet air ambient temperature is very significant. The equipment is designed to perform maximally at ambient air temperature of 15^0 C. Changes in ambient air temperature affect power output of the turbine directly. Conducted experiment indicated the existence of a relationship between the axial compressor inlet air temperature and the eventual power output of the turbine. It therefore means that the lower the ambient air temperature, the higher the power output. The higher the ambient air inlet temperature, the lower the power output. Which means the inlet air ambient temperature is inversely proportional to the engine power output. By implication, it means that every 1^0 C increase in ambient temperature of the inlet air into the turbine axial compressor results to approximately 1 % decrease in power output. It is critical to maintain inlet air temperature as low as the designed temperature and according to ISO standard to achieve a stable and maximum power output of the turbine.

Wadia *et al* (2002) presented that the GE LM2500 Plus series aero derivative gas turbine is an upgrade of LM2500 gas turbine. Zero stage was introduced into the LM2500 plus series gas turbine. Pressure ratio improved to 23.3:1. The 17-stage axial compressor engine efficiency is well above 91% polytropic. The engine is used for power generation or gas compression in the power generation and or oil and gas industry.

Shuwei et al (2018) established that mathematical models are important gate ways in modeling aero derivative engines. Mathematical models' applicability is dominant in research, design, and development and monitoring during turbine engine operation. Due to the complexities of aero engine modeling, intelligent algorithms such as neural network, support vector machines are employed in mapping inputs and outputs. However, the exact derivative-based modeling method of aero engine, transforming the problem of derivative calculation to problem of differential calculation with a chain derivation method. This resulted to time improvement ranging 54% to 55% in steady and transient states of the component level model.

David et al (2023) presented that the LM2500 plus G5 aeroderivative gas turbine is the 5th generation of the its model with a higher-level power ouput. Its thermal efficiency is above 39%. It is applied for both land and marine operation. The LM2500 series has undergone series of improvement resulting to LM2500 plus G5 with higher availability and reliability. The LM2500 plus G5 free power turbine is six - turbine. Lower fuel consumption and more megawatt. LM2500 plus G5 has efficiency rating of 90% high-pressure turbine, 92% free power turbine and 90% for combustion.

Badeer (2020) presented that the land and marine gas turbine have power output range is between 30mw and 47 mw. The operational advantage aero derivative gas turbine visible in its unique economic advantage to the end user. Major components are the gas generator, combustor, high-pressure turbine and free power turbine). The gas generator consists of the axial compressor, combustor and high-pressure turbine, aerodynamically coupled to the high-pressure power turbine. The power turbine is a six stage – low speed (3600 rpm)

III. MODELING OF AERO DERIVATIVE GAS TURBINE

The thermodynamics principle applied in this thesis for the modeling of aero derivative gas turbine is the Brayton cycle. Reference technology is LM2500 Plus G5 aero derivative gas turbine. While, reference standard is International Standard Organization

3.1 DATA COLLECTION AND SOURCES

Aero derivative gas turbines for power generation and gas compression are designed according to customer requirement of power output and operating conditions. Data were determined and collated based on International Standard Organization (ISO) and site-specific requirement for the design of a low speed 40 MW aero derivative gas turbine required for power generation in Esan central local government of Edo state, Nigeria. Importantly, the intended operating conditions (atmospheric pressure, air ambient temperature and fluid flow rate) and referenced operating parameters (Axial compressor exits pressure, mass flow rate of air, specific heat capacity of air at a constant pressure, specific heat capacity of air at a constant volume, specific heat, mass flow rate of air, specific heat capacity of gas at a constant pressure, compressor efficiency, lower calorific value, Free Power Turbine efficiency, High Pressure Turbine efficiency and combustion efficiency) were used in determining preliminary design, upgrade, modeling, manufacturing, installation, operating and maintenance data. Additionally, aero derivative gas turbine technology transfer data (technical information, strategic processes, procedures, labor, skill, scientific and knowledge of the transaction were obtained from published industry research and aero Field Engineers and specialist interview.

3.2 METHOD OF ANALYSIS

Calculations, numerical analysis, modeling and simulations were done with Microsoft word, Excel and MATLAB - Simulink software. The impute data of power output requirement, site specific pressure, ambient temperature, fluid flow rate, performance requirement, efficiency, thermodynamic cycle calculations were used to achieve the aforementioned aero derivative gas turbine modeling.

3.3.1. Major Components of Aero derivative Gas Turbine Modeling at 15^0 C ambient air inlet temperature with Intercooler in place

3.3.1.1 Modeling parameters and assumptions

The LM2500 plus G5 aero derivative gas turbine mass flow rate of air used in this thesis is 84.5 kg/s. This shall be the baseline reference. The engine is able to accommodate varying flow to deliver on specification (Wadia *et al*, 2002)

The efficiency adopted is that of LM2500 series aero derivative gas turbine major components. 92% for axial compressor (Wadia et al, 2002), 90% for high-pressure turbine, 92% free power turbine and 90% for combustion (David *et al*, 2023). While pressure ratio for the referenced aero derivative gas turbine technology LM2500 plus G5 is 25:1 bar

Axial compressor inlet temperature T_1=288.15 K (15^0 C) and pressure P1=1.01325 bar at ISO standard were referenced as base line (Neeraj & Bansode 2016). Then specific heat capacity of air at a constant volume = Cv_air = 0.718 kJ/kg.K, specific heat capacity of air at a constant pressure at T1 = Cp_air=1.006 kJ/kg.K, specific heat capacity of air at a constant pressure at T2 = Cp_air2=1.12 kJ/kg.K, specific heat capacity of gas at a constant pressure Cp_gas=1.15 and specific heat ratio = Y =1.4 are from look up tables (Basil, 2001). While mass flow rate of fuel = m_fuel=1.83 kg/s was determined from the relationship among Lower calorific value for natural gas of methane rich density of 0.717 = LCV= (36000kj/nm^3) = 50209 kj/kg, power output of reference technology of 36.8 mw and equipment efficiency of 40% (GEA34108A, 2024)

The flow rates shall be determined from thermodynamic energy equations as follow

$$Q_{in} = P_{out\ ref\ tech} / \eta_{Turbine\ thermal} \tag{3.1}$$

I applied equation 3.1 to determine quantity of heat Q_{in} supplied to the turbine to generate output power $P_{out\ ref\ tech}$ at a reference turbine thermal efficiency $\eta_{Turbine\ thermal}$

$$Q_{in} = P_{out\ ref\ tech} / \eta_{Turbine\ thermal}$$

$$Q_{in} = 36800/0.40$$

$$= 92000 \text{ kj/s}$$

$$\dot{m}_{fuel} = Q_{in}/LCV \tag{3.2}$$

To determine the new fuel mass flow rate $\dot{m}_{fuel}$ equation 3.2 was applied as indicated below

$$\dot{m}_{fuel} = 92000/50209$$

Mass flow rate of gas shall be determined by applying equation 3.3 below

$$\dot{m}_{gas} = (\dot{m}_{air} + \dot{m}_{fuel}) \tag{3.3}$$

I applying equation 3.3, I determined the gas mass flow rate as detailed below.

$$\dot{m}_{gas} = 1.83 + 84.5$$

$$= 86.33 \text{ kg/s}$$

3.3.1.2. Compressor Exit Temperature T_2

Applying equation 3.5, I derived exit temperature for the axial compressor as indicated below.

Where η_s is compressor efficiency (92%), P_1 and P_2 are pressures – in and out (1.01325 bar and 25 bar for a 17-stage axial compressor, T_1 and T_2 are axial compressor inlet and outlet temperatures.

The atmospheric air inlet temperature T_1 is 288.5 k at ISO condition, T_{2s} is isentropic temperature and γ is the ratio of specific heat for air at constant pressure and volume (CP/CV = 1.005/0.718 = 1.4).

$$T_{2s} = T_1 \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} \tag{3.4}$$

$$T_{2s} = 288.15 \left(\frac{25}{1.01325}\right)^{\frac{1.4-1}{1.4}}$$

= 741 K – Isentropic

Applying equation 3.6, I derived actual exit temperature for the axial compressor as indicated below.

$$T_2 = T_1 + \left(\frac{T_{2S} - T_1}{\eta_c}\right) \tag{3.5}$$

$$T_2 = 288.15 + \left(\frac{741 - 288.15}{0.92}\right)$$

= 781 K

3.3.1.3 Compressor Work

Applying equation 3.6, I derived the compressor work as detailed below

$$w_c = \dot{m}_{air}\, C_{P.air}\,(T_2 - T_1) * \eta_{compr.} \tag{3.6}$$

$$w_c = 84.5\, X\, 1.006\,(781 - 288.15) * 0.92$$

= 38,577 KW

= 38.5 MW

3.3.1.4. High Pressure Turbine Inlet Temperature (T_3)

To determine turbine inlet temperature, I applied energy balance equation across the combustor

$$T_3 = \frac{\dot{m}_{air}\ X\ C_{P.air}\ X\ T_2 + \dot{m}_{fuel}\ X\ LHV\ X \eta_{cb}}{(\dot{m}_{air} + \dot{m}_{fuel}) X C_{p.gas}} \tag{3.7}$$

Applying equation 3.7, where mass flow rate of air is 84.5 kg/s, mass flow rate fuel is 1.83 kg/s, lower calorific value of fuel is 50209 kj/kg, axial compressor exit temperature T_2 is 781 K, combustion efficiency η_{cb} is 0.95, specific heat of air at constant pressure is 1.006 Kj/kg.k, specific heat of gas at constant pressure is 1.15 Kj/kg.k

$$T_3 = \frac{84.5X\ 1.12X\ 781 + 1.83\ X\ 50209\ X 0.95}{(84.5 + 1.7) X 1.15}$$

T_3 = 2149 K

3.3.1.5 High Pressure Turbine Outlet Temperature (T_4)

The turbine outlet temperature T_4 is the inlet temperature of the free power turbine and was determined using the following equation 3.8

$T_4 = T_3 - \frac{W_c}{\dot{m}_g X C_{p.gas}}$ (3.8)

$$T_4 = 2149 - \frac{38577}{86X1.15}$$

= 1761 K

3.3.1.6. High-pressure turbine work

$w_t = (\dot{m}_{air} + \dot{m}_{fuel})\, C_{P.gas}\,(T_{in} - T_{out}) * \eta_{HPT.}$ 3.9)

Applying equation 3.9 to derive turbine work as seen below, knowing that high-pressure turbine work W_{HPT} is equal to compressor work w_c

$$W_{HPT} = \dot{m}_g\, C_{P.gas}\,(T_3 - T_4) * \eta_{HPT.}$$

$w_{HPT} = 86X1.15(2149 - 1760) * 0.90$

39,719 KW

39.7 MW

3.3.1.7. Free Power Turbine Output Temperature

The free power turbine output temperature T_5 was determined via thermodynamic energy balance equations as in indicated below

$T_5 = T_4 - \frac{W_c}{C_{p.gas.}\, X\, \dot{m}_g}$ (3.10)

Applying equation 3.10, we shall determine the free power turbine exit temperature $T_{5.}$

$$T_5 = 1761 - \frac{38577}{1.15\, X86}$$

= 1372 K

3.3.1.8. Free Power Turbine Power Output

$P_{FPT} = \dot{m}_{gas}.\, c_{p.gas}\,(T_4 - T_5) * \eta_{FPT.}$

(3.11)

Applying equation 3.1 to derive free power turbine output

$$P_{FPT} = 86\,\mathrm{X}\,1.15(1761 - 1372) * 94$$

= 35,491 KW

= 35.5 MW

3.3.2. Major Components of Aero derivative Gas Turbine Modeling at 15^0 C ambient air inlet temperature with Intercooler and an additional Compressor stage

The upgrade modeling was done in two fronts of compressor inlet atmospheric air temperature reduction to improve power output and the introduction of additional blade stage (from 17 stage compressor to 18 stage compressor) to improve power output.

The intercooler with cooling range of 30º to 15º C was modeled into the aero derivative gas turbine engine with reference to the fact that there exists a relationship between turbine compressor inlet temperature and turbine power output. A higher ambient temperature results in decrease in air density and reduced mass flow rate. The turbine power output is inversely proportional to the compressor inlet temperature – $T_{1.}$ The relationship is driven by changes in air density ρ and mass flow rate ṁ.

To achieve the target of improved compression ratio of the turbine engine axial compressor and therefore improved power output from the Free Power Turbine, an additional stage was modeled into the engine compressor. The effect of additional stage on the outlet pressure of the compressor P_2 and temperature T_2 was modeled into the axial compressor, using pressure increase and power increase models respectively.

The additional stage (S+1), shall be determined by referencing the LM2500 plus G5 discharge pressure b and number of stages 17 and applying the principle of stage stacking we use the n-th root of the compressor discharge pressure

$PR_{stage\ S+1} = \sqrt[a]{b}$ (3.12)

Applying equation 3.1, I determined the additional stage as follow

$PR_{stage\ S+1} = \sqrt[17]{25}$

= 1.2

Where a is the number of compressor blade stages = 17, b is compressor discharge pressure.

The pressure increase model is premised on the fact that for an additional stage (S+1), the product of the original pressure ratio and the additional stage pressure ratio (PR_{STAGE}) of the new stage is the new overall pressure ratio.

Therefore, new pressure ration shall be derived from the model below.

$\mathrm{PR_{new}} = \mathrm{PR_{existing}}\, X\, \mathrm{PR_{stage\ S+1}}$ (3.13)

Applying equation 3.13 to determine new pressure ratio - $\mathrm{PR_{new}}$, where existing pressure ratio $\mathrm{PR_{existing}}$ is 25.4:1 and stage pressure ratio is 1.2

$\mathrm{PR_{new}}$ = 25.4 X 1.2 = 30 bar

The compressor P_{2new} (new compressor exit pressure) shall be derived from

$\mathrm{P_{2new}} = \mathrm{P_{inlet}}\, X\, \mathrm{PR_{new}}$ (3.14)

$\mathrm{P_{2new}}$ = 1.01325 X 30 = 30.6 bar

Free power turbine power increases with an increase in mass flow rate ṁ and pressure ratio. This increase is however indirectly proportional to enthalpy in the free power turbine. The simplified power increase model shall be applicable as indicated below

Power increase in percentage is indicated model below

$$\Delta P_{\%} = \left[\left(\frac{PR_{new}-PR_{baseline}}{PR_{baseline}}\right) X \left(\frac{100}{1}\right)\right] \quad (3.15)$$

$$=\left[\left(\frac{30-25}{25}\right) X \left(\frac{100}{1}\right)\right]$$

= 20%

Compressor exit pressure increase from 25 bar to 30 bar indicating a 20% increase. Therefore, new ratio shall be as follow:

Old exit pressure P_{OLD} is 25 bar, new exit pressure P_{NEW} is 30 bars.

Ratio of new to old exit pressure shall be = 30/25 = 1.2

New flow shall be determined as follow

Since referenced old flow is 84.5 kg/s, new flow relationship between pressure and air mass flowrate shall be approximated using continuity equation 3.16 as follows

$$m_{new} = m_{old} \text{ X } PR_{stage\ S+1} \quad (3.16)$$

$$m_{new} = 84.5 \text{ X } 1.2 = 101.4 \text{ kg/s}$$

3.3.2.1. Compressor Exit Temperature T_2

Applying equation 3.2, I derived exit temperature for the axial compressor as indicated below.

Where η_s is compressor efficiency (92%),P_1 *and* P_2 are pressures – in and out (1.01325 bar and 30.4 bar for an upgraded 18-stage axial compressor, T_1 and T_2 are axial compressor inlet and outlet temperatures. The atmospheric air inlet temperature T_1 is 288.15 at ISO condition, achieved by intercooler modeling into the aero derivative gas turbine, T_{2s} is isentropic efficiency and γ is the ratio of specific heat for air at both constant pressure and volume (CP/CV = 1.005/0.718 = 1.4).

Applying equation 3.4, I derived the axial compressor isentropic exit temperature as indicated below

$$T_{2s} = T_1 \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}}$$

$$T_{2s} = 288.15 \left(\frac{30}{1.01325}\right)^{\frac{1.4-1}{1.4}}$$

= 802 K – Isentropic

Applying equation 3.5, I derived actual exit temperature for the axial compressor as indicated below.

$$T_2 = T_1 + \left(\frac{T_{2S} - T_1}{\eta_c}\right)$$

$$T_2 = 288.15 + \left(\frac{802 - 288.15}{0.92}\right)$$

= 847 K

3.3.2.2 Compressor Work

Applying equation 3.6, I derived the compressor work as detailed below

$$w_c = \dot{m}_{air}\, C_{P.air}\, (T_2 - T_1) * \eta_{compr.}$$

$$w_c = 101.4 \, X \, 1.006\, (847 - 288.15) * 0.92$$

= 52,451 KW

= 52 MW

3.3.2.3. High Pressure Turbine Inlet Temperature (T_3)

To determine turbine inlet temperature, I applied energy balance equation 3.7 across the combustor

$$T_{turbine\ in} = \frac{\dot{m}_{air}\ X\ C_{P.air}\ X\ T_2\ + \dot{m}_{fuel}\ X\ LHV\ X \eta_{cb}}{(\dot{m}_{air} + \dot{m}_{fuel}) X C_{p.gas}}$$

Applying equation 3.8, where mass flow rate of air is 101.4 kg/s, mass flow rate fuel is 1.7 kg/s, lower heating value of fuel is 50209 Kj/kg, axial compressor exit temperature T_2 is 847 K, combustion efficiency η_{cb} is 0.95, specific heat of air at constant pressure is 1.006 Kj/kg.k, specific heat of gas at constant pressure is 1.15 Kj/kg.k

$$T_3 = \frac{101.4\ X\ 1.12 X\ 847 + 1.83\ X\ 50209\ X 0.95}{(111.6 + 1.83) X 1.15}$$

T_3 = 2,045 K

3.3.2.4 Turbine Outlet Temperature (T_4)

The turbine outlet temperature T_4 is the inlet temperature of the free power turbine and was determined using the following equation 3.8

$$T_4 = T_3 - \frac{W_c}{\dot{m}_g X C_{p.gas}}$$

$$T_4 = 1738 - \frac{43079}{86 X 1.15}$$

= 1,603 K

3.3.2.5. High-pressure turbine work

$$w_t = \left(\dot{m}_{air} + \dot{m}_{fuel}\right) C_{P.gas}\, (T_{in} - T_{out}) * \eta_{HPT.}$$

Applying equation 3.9 to derive turbine work as seen below, knowing that high-pressure turbine work W_{hpt} is equal to compressor work w_c

$W_{HPT} = \dot{m}_g\, C_{P.gas}\, (T_3 - T_4) * \eta_{HPT.}$

$w_{HPT} = (101.4 + 1.12) X 1.15 (2232 - 1791) *$ 0.90

=47,206 KW

=47 MW

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3.3.2.6. Free Power Turbine Output Temperature
The free power turbine output temperature T_5 was determined via thermodynamic energy balance equations as in indicated below

$$T_5 = T_4 - \frac{W_c}{C_{p.gas.} X \dot{m}_g}$$

Applying equation 3.10, we shall determine the free power turbine exit temperature $T_{5.}$

$$T_5 = 1791 - \frac{43709}{1.15 X 86}$$

= 1161 K

3.3.2.7. Free Power Turbine Power Output

$$P_{FPT} = \dot{m}_{gas}.c_{p.gas} (T_4 - T_5)*\eta_{FPT.}$$

Applying equation 3.11 to derive free power turbine output

$P_{FPT} = (101.4 + 1.12)$ X 1.15(1791 − 1351)*0.94
= 48,255 KW
= 48 MW

3.3.3. Economic Gains of Modeling Intercooler and an additional Compressor stage into the Aero derivative Gas Turbine
The major reference gain is in power output which resulted from intercooler introduction and an additional compressor blade stage.
3.3.3.1. Economic Gains of Modeling Intercooler into the Aero derivative Gas Turbine
When operating outside ISO condition and without intercooler at ambient air inlet temperature of 30^0 C. I applied the rule of thumb, where every 1^0 C rise in atmospheric air temperature result to 1% loss of power turbine power. The percentage power drop shall be derived as follows using gas turbine ambient temperature correction models (Neeraj & Bansode, 2016)

Percentage drop Δ% = $\Delta T \ X \ \frac{1\%}{1^0 C}$ (3.16)

Δ%= 303.15-288.15 X $\frac{1}{1}$ = 15%

Power output at 303.15 k = $P_{T.303.15}$ = $P_{T.288.15} X (1 - \Delta\%)$ (3.17)

= 35,491 X (1 – 0.15)

= 30.167 MW

Power gain with intercooler in place shall be $P_{gain} = P_{T.288.15}$ -$P_{T.303.15}$ (3.18)

= 35,491 – 30,167

= 5,323 mw

Therefore, monetary value gain per installed turbine (with intercooler in place), when cost of 1 mw of electricity in Nigeria is one hundred and twenty million naira (120,000,000) shall be

= (5.323 X120, 000,000)

=NGN 638,760,000 per hour of power generation

3.3.3.2. Economic Gains of Modeling an additional Compressor Blade stage into the Aero derivative Gas Turbine
The free power turbine power output prior to additional stage was 35,491 MW, while the free power turbine power output post additional stage became 48,255 MW.
Therefore, power gain shall be $P_{gain} = P_{new}$ - P_{old}
= 48,255 – 35,491
= 12,764 MW
Monetary value gain per upgraded turbine (Additional compressor blade stage), when cost of 1 mw of electricity in Nigeria is one hundred and twenty million naira (120,000,000)
= (12,764 X120, 000,000)
= NGN 1,531,680,000 per hour of power generation

IV. CONCLUSION

Summarily, the turbine modeling at 30^0 C ambient air inlet temperature was done by applying rule of thumb for a 17-blade stage compressor and it generated 30 mw of power. Modeling at 15^0 C ambient air inlet temperature with Intercooler in place for a 17-blade stage compressor by apply energy balance thermodynamic equations, generated 35.5 mw. Modeling at 15 C ambient air inlet temperature with Intercooler and an additional Compressor blade stage (now 18 blade stage) generated 48 mw. The economic gains of modeling intercooler into the aero derivative gas turbine per hour was 5,323 mw. That is NGN 638,760,000 per hour of power generation at NGN 120,000,000 per mw. Economic Gains of modeling an additional compressor blade stage into the aeroderivative gas turbine compressor was 12,764 MW, implying NGN 1,531,680,000 per hour of power generation.

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