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Modelling and Simulation of the SGT5 – 2000E Gas Turbine Model for Power Generation

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Abstract

In this paper, a steady-state flow modelling and simulation of an SGT5 – 2000E gas turbine model was carried using Ebsilon Professional Software. The design and off-design conditions were considered according to the environmental ambient air temperature. The model results were validated using design and actual operating data by applying the error percentage analysis. The validated results obtained ranged from -3.13% to 0.88% for design condition, while -3.24% to 2.66% for off-design condition. In addition, the model validation results were found to be in good agreement with the actual installed operating data. Thus, the model data obtained can be utilized for further studies such as energy, exergy and environmental analyses.

1. Introduction

The relevance of energy resources is important in almost all parts of the economy because of their necessity. The direct increase in energy and power consumption is a result of the increase in urbanization, population, technological development and number of industries [1,2]. The majority of the power plants in the country are gas turbine power plants because of their low cost, short installation period and abundant availability of natural gas. The gas turbine power plants can operate as a simple open cycle system, a combined cycle system, or combined heat and power system in the power plant generation setup [3,4]. The applications of gas turbine shaft or propulsion power and as cogeneration for industrial, automotive and commercial purposes had become reliable and popular [5-8]. "All gas turbines operate on the thermodynamic cycle known as the Brayton cycle" [9]. Simple gas turbine major components are compressors, combustors, turbines, and generators. It operates continuously by taking in the fresh air from the atmosphere into the compressor. The air is then compressed to high pressure in the compressor and discharged into the combustors. Here the compressed air is mixed with fuel and burnt continuously after ignition to liberate its energy at constant pressure. Then the combustion products with high pressure and temperature flow into the turbine, the burnt gas thermal power is converted to mechanical shaft power in the turbine, which in turn drives the generator shaft. The mechanical shaft power is converted to electrical power by the generator. the flue gases are discharged into the atmosphere at constant pressure after expansion through the turbine to low temperature and pressures.

The performances of all gas turbines are affected by local or ambient air conditions such as temperature, relative humidity, etc [10,11]. The rated capacities of all combustion turbines are based on the International Standard Organization (ISO) conditions of an ambient air temperature of 15 ^{0}C, relative humidity of 60% and ambient pressure of 101.325 kPa at sea level. The installed SGT5 – 2000E Power Plant is located in Ihobvor community an area of an average ambient air temperature of 26 ^{0}C, relative humidity of 70%, and ambient pressure of 1.013bar [12]. Ambient air temperature has a clear impact on the gas turbine power plant because the air mass is inversely proportional to its ambient air temperature and proportionate to its power produced. So, it is important to study the effect of ambient air temperature on power output and other output parameters of gas turbine power plants as reported in Saravanamuttoo et al. [9]. The report also stressed that this is very important to the customer, at any specific condition, the performance of the power plant must be guaranteed readily by the manufacturer of the gas turbine. Simon et al. [13] explained in their report that actual outcomes may differ depending on local site conditions. This makes it necessary to have design and guarantee performance for gas turbine power plants when the site ambient air temperature differs from the design ambient air temperature. Most manufacturers of gas turbine power plant models provide scanty information about the power plant because they are proprietary to them [14,15]. General data provided by them are usually compressor air inlet temperature, exhaust mass flow rate and temperature, compressor pressure ratio, power out, heat rate and thermal efficiency at ISO condition. In order to determine the other parameters such as mass flow rates of air and fuel, compressor discharge temperature and pressure, turbine inlet pressure and temperature. Many researchers used the thermodynamic model equations, which are sometimes passed through cumbersome iteration. This led to the use of modelling software, which makes the iteration process simple. Numerical modelling and simulations are considered as the initial move towards the underlying idea approval. The gas turbine power plant to be investigated requires a detailed model's execution to accurately assess the performances of power plants in ambient temperature variation and part load function [16,17]. A decent number of modelling tools have been utilized for power plant simulations and modelling analysis. From various programming software accessible, for example, APROS, Aspen, Autodynamics, HYSYS, gProms, SIMODIS, PowerSim, MMS, and ProTRAX; the Ebsilon Professional package has been chosen. This product package considers far-reaching power plant process cycle usage for steady-state and semi-dynamic simulations and plant parameters advancement measures. Albeit, physical equations depicting all parts in Ebsilon Professional programming condition are valid for steady-state estimation thus, it is conceivable to disregard the dynamic impacts by carrying out the arrangement of reenactments on a little timescale [18]. It is acknowledged by utilizing a blend of "time Series" and "ebsScript" include at the programming level to make and simulate such semi-dynamic systems [19].

EBSILON Professional 14 and ASPEN Softwares are alternative simulating programming software. The first is intended for chemical processes simulation, while the last is ideal to balance mass and energy in processes of power plants. For power plant analysis which involves energy and mass balance as in the thermodynamic and environmental analysis of gas turbine power plant, EBSILON Professional 14 is preferable. EBSILON Professional is one of the utmost utilized energy and mass balance computation programming in the European Countries that speak the German language [20]. It shows high intermingling solidness, high computation speed, and is adjusted to Microsoft environments. EBSILON has all the highlights needed for this investigation and speaks to the ideal apparatus for the study of gas turbines [21]. This part quickly sums up the principal highlights of this product

OEM-GTLib, created in participation with and dispersed by VTU Energy GmbH, contains countless gas turbine models. These are changed under genuine performance conduct and depend on legitimate producer information. Accordingly, one can choose the ideal gas turbine for one's capacity plant to study. The library contains gas turbines from Alstom, Siemens, Rolls Royce, General Electric, Solar Turbines, Centrax, and Hitachi [22].

EBSILON software has been used for various research works in power plant analysis by Miguez Da Rocha [23] on the study of retrofitted solar-powered Combined Cycle Power Plant; Jaszczur and Dudek [24] in the thermodynamics investigation of a gas turbine combined cycle incorporation with a high-temperature nuclear system; Garcia Sanchez – Cervera [25]; Wojcik and Wang [26] in steam power plant optimization of energy for an effective process of a post-combustion of CO_2 capture plant; and Wallentinen [27] in the analysis of a concentrated solar power gas turbine with thermal storage. Zyrkowski and Zymelka [28] analyzed the behaviour of 2 225MW coal units by utilizing EBSILON Professional Software. The model results were validated using test results from the power plant. Matjanov [29] also used the software in modelling and analysis of the Tashkent CHP enhancement with an air inlet cooling system. Design data were used to validate their models in their various researches without considering actual operating varied ambient air temperature. There are many studies carried out using ebsilon software, but these few were mentioned to ascertain that Ebsilon software has been previously used for research work. It has been employed as an accurate tool, precise and powerful tool which support the design, modification, or retrofitting of power plant for the past twenty years of successful applications and further development [27,30,31].

The SGT5 -2000E gas turbine model has been installed in Ihobvor community, Benin City, Edo State, Nigeria. The ISO condition parameters provided did not have detailed data that can be used for further studies. So, to determine these important parameters such as mass flow rates of air and fuel, compressor discharge temperature and pressure, turbine inlet temperature, thus, this study presents the modelling and simulation of the SGT5 -2000E gas turbine using EBSILON Professional. The focus of the study is to obtain these important data which will be validated by actual operating parameters from the installed power plant.

2. Methodology

The methodology for this study was based on the gas turbine model description, the thermodynamic equations, modelling and simulation of the SGT5 – 2000E model, validation of the model developed and simulation analysis.

2.1 Thermodynamic Operational Principle of the SGT5 – 2000E Model Power Plant

The installed SGT5 -2000E gas model Power Plant works on the Brayton cycle. The schematic and temperature–specific entropy (T – s) diagrams of the installed SGT5 -2000E Power Plant are illustrated in Figures 1 and 2 respectively. For the initial starting of the power plant, the air compressor (axial flow type) is initially driven by an electric motor until 60% of the turbine shaft speed is achieved. The maximum net thermal efficiency of the Azura Edo Power plant operating as the open simple cycle will always be less than 40% because 60% of the turbine shaft power is used to drive the air compressor. At the point when the turbine starting structure is activated, the air intake plenum taking fresh air from the surrounding air, filtered it at state 1 and is compressed in the 16-stage air compressor. At the 11^{th} stage, the air extraction valve is opened and the variable IGVs are closed during start-up for pulsation protection. The extraction bleeding valve closes automatically at 95% speed of the turbine shaft, which allows the compressed air to enter into the silo combustor annular space at state 2, then enters liners of the combustor. For appropriate combustion, the compressed air flows into the burning zone through the metering hole in each of the combustor liners.

Fuel is given to the streamlines, each ending at a fuel nozzle. The fuel is introduced into the burning chamber by the nozzles at a steady rate depend on the load and speed required by the gas turbine.

The presented fuel blends in with the compressed air and is exploded utilizing either of the sparkle plugs. At the moment when the fuel blend is lighted in one burning chamber; fire is spread through cross-interfacing fire cylinders to all other ignition chambers. At the point when the turbine rotor approximates the working rate, the pressure of the combustion causes the spark plugs to withdraw subsequently eliminating their electrode from the hot zone. It is planned for appropriate dilution and cooling. At state 3, the hot gas flows into a 4-stage turbine after expanding into different progress pieces joined to the aft end of the chamber liners from the combustor. In every turbine nozzle, the hot gas kinetic energy is raised as the drop in pressure in each after row of rotating blades. Part of the burnt gas kinetic energy is transferred into useful work on the rotor of the turbine. The flue gases, after flowing through the third stage of blades, are coordinated into the fumes hood and diffuser, which contains a progression of going vanes to divert the gases from axial flow to radial flow directions with the least fumes hood losses. At state 4 the gases flow into the fumes plenum and are released with the air through the fumes stack. The turbine uses part of the power generated to drive the air compressor as earlier mentioned and the rest is obtainable for valuable work at the gas turbine output flange, which is coupled to a 3000-rpm generator. The generator will convert the mechanical power to electrical power, which is supplied to the national grid.

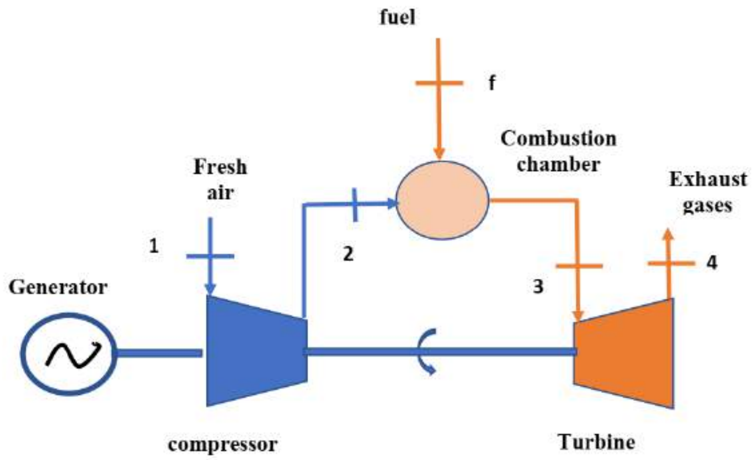

Figure 1: A Typical Schematic Diagram of SGT5 – 2000E Model Power Plant

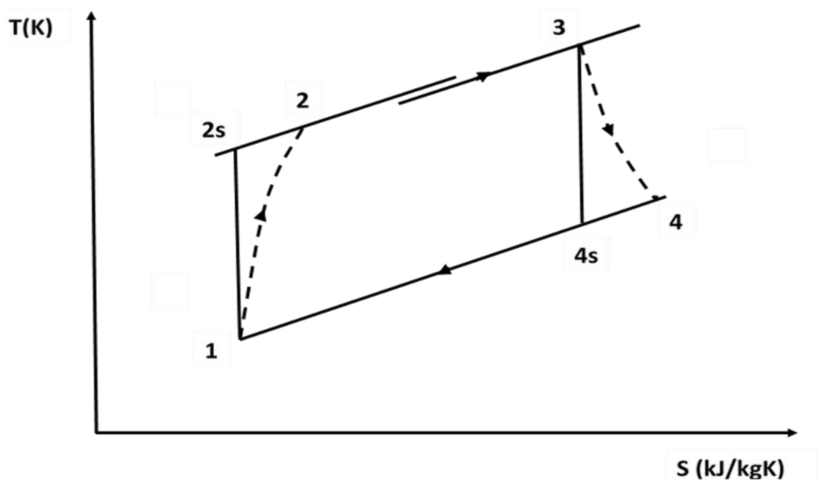

Figure 2: A Typical T- s Diagram of the Installed SGT5 -2000E Power Plant Cycle

2.2 Thermodynamic Model Equations

The compressor discharged temperature is calculated using Equation (1) and exhaust gas temperature from the turbine was determined by Equation (2) as provided in Cengel and Boles [32]and Ehyaei et al. [33]. The pressure ratio was evaluated using Equation (3).

$$T_2 = T_1\left[\left(\frac{r_{pc}^{\frac{\gamma-1}{\gamma}}-1}{\eta_c}\right)+1\right] \tag{1}$$

$$T_4 = T_3\left[1-\eta_T\left(1-\frac{1}{r_{p_t}^{\frac{\gamma_g-1}{\gamma_g}}}\right)\right] \tag{2}$$

$$r_{p_c} = \frac{p_2}{p_1} \quad \text{and} \quad r_{pt} = \frac{p_3}{p_4} \tag{3}$$

Equations (4) to (13) was used to determine the various performance of the gas turbine model were obtained from Cengel and Boles [32]; Rogers and Mayhew [34]; Eastop and McConkey [35]; Egware and Obanor [36]; Egware et al. [37].

The work done by compressor and turbine was evaluated using Equations (4) and (5) respectively.

$$W_C = \dot{m}_a c_{pa}\left(T_2 - T_1\right) \tag{4}$$

$$W_T = \dot{m}_g c_{pg}\left(T_3 - T_4\right) \tag{5}$$

The thermal power of the power plant was determined by applying Equations (6) and (7).

$$P_{themal} = W_T - W_C \tag{6}$$

$$P_{thermal} = \dot{m}_g * c_{pg}\left(T_3 - T_4\right) - \dot{m}_a c_{pa}\left(T_2 - T_1\right) \tag{7}$$

The heat supply by the fuel, gross thermal efficiency, and the heat from flue gas of the gas turbine was determined using Equations (8) to (10) respectively.

Heat supply, $HS = \dot{m}_f LHV$ (8)

Gross thermal efficiency, $\eta_{thgr} = \frac{P_{net}}{\dot{m}_f LHV}$ (9)

Flue gas losses

Flue Gas Heat, $Q_{flue} = \dot{m}_g c_{pg} \left(T_4 - T_1\right)$ (10)

The electrical power generated, P_{net} is expressed in Equation (11)

$P_{net} = P_{thermal} - P_{oss}$ (11)

where P_{loss} is the total losses for mechanical, generator and auxiliary losses

The net thermal efficiency (η_{net}) was determined as expressed in Equation (12)

$$\eta_{net} = \frac{P_{net}}{\dot{m}_f * LHV} \tag{12}$$

The heat rate (HR) of the gas turbine power plant was computed using Equation (13)

$$HR = \frac{3600}{\eta_{net}} \tag{13}$$

2.3 Modelling and Simulation of the SGT5 – 2000E Gas Turbine Power Plant

The model of the installed turbine power plant is the siemens SGT5 – 2000E. The nominal conditions for the Model ISO parameters are presented in Table 1. The power plant model structure executed in EBSILON Professional programming is shown in Figure 3. The principal model data are recorded in Table 1.

Table 1: Nominal conditions for ISO SGT5 - 2000E Gas Turbine Power Plant

	ISO Design condition	
S/N	Parameters	Design values
1	Power (MW)	166
2	Heat Rate(kJ/kWh)	10375
3	Thermal Efficiency (%)	34.7
4	Turbine Exhaust Temperature (^{0}C)	541
5	Exhaust mass flow rate (kg/s)	525
6	Pressure Ratio (r_p)	12
7	Ambient air temperature (^{0}C)	15
8	Ambient air pressure (bar)	1.013
9	Lower Heating Value, LHV (MJ/kg)	45.011

The assumptions made in the modelling the Power plant are as follows:
(i) The simulations are performed at a steady state.
(ii) Neglecting the transient impact caused by start-up and shut down during operation.
(iii) The pressure drops in Ebsilon was considered for each component nominal pressure drops.

The SGT5 – 2000E Turbine model comprises of an air compressor, a combustion chamber, a gas turbine, and a generator, which was modelled by selecting and connecting the components mentioned together. The ISO conditions of 1.013bar pressure and 15^{0}C temperature for ambient air were used to establish the design performance outcomes of SGT5 – 2000E Turbine. The values of ambient air temperature, pressure, pressure ratio, exhaust mass flow rate and temperature from Table 1 were used to perform the modelling of the ISO SGT5 – 2000E Turbine. The "genera input value" was utilized to set the nominal mass flow rate and temperature of the flue gas, whereas the pressure ratio was set in the gas turbine. The isentropic efficiencies of the compressor and turbine used are 90% and 85% respectively. The mechanical and generator efficiency used is 99% and 98.5% respectively. The composition and LHV of the fuel when inputted accordingly. After inputting all data and performing the model simulation, the mass flow rates of air and fuel are attained for the nominal conditions in the gas turbine. This formed the root profile for further simulations of the model.

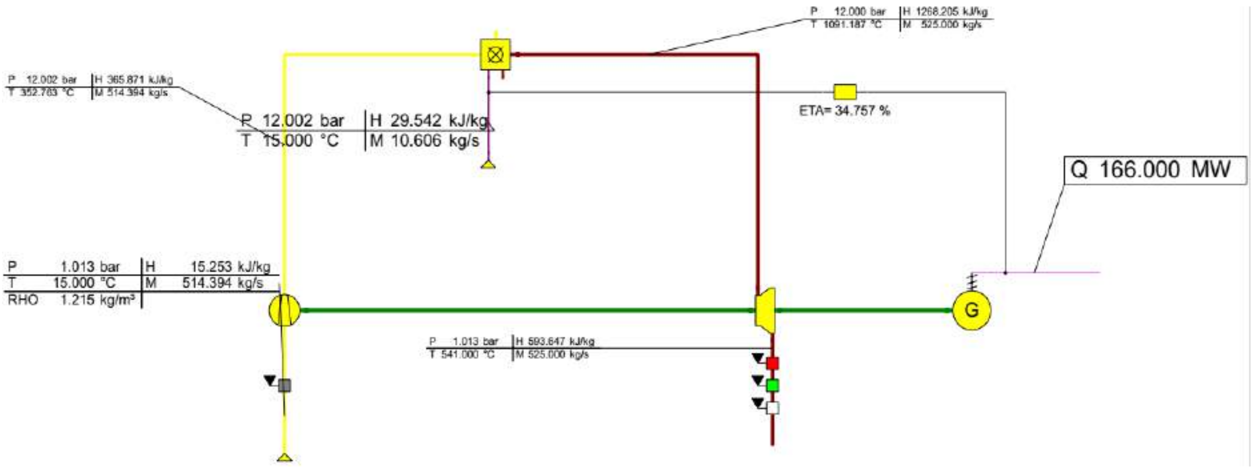

Figure 3: Topology of ISO SGT5 – 2000E based on the Fixed Exhaust Data

Then, the values of air and fuel mass flow rates gotten will be used to carry out the simulation again without the exhaust temperature and mass flow rate. When the initial results for exit conditions are obtained, the nominal condition for the gas turbine will be achieved. This explanation is illustrated in Figure 4. This is done to enable the mass flow rate of air and mass flow rate of fuel that enters the compressor and combustion chamber respectively to determine the gas turbine performance under partload. This will make it possible to input the new parameters of air for off-design conditions using the "boundary value input".

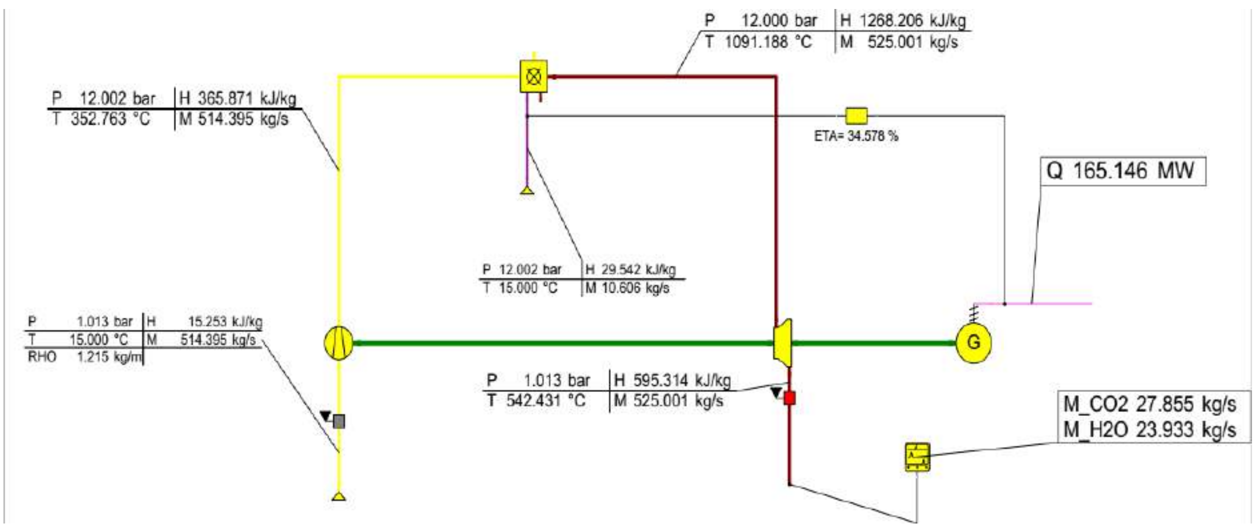

Figure 4: Topology of ISO SGT5 – 2000E

2.4 The SGT5 – 2000E Gas Turbine in Off-design Modelling

When the gas turbine is operating in the design condition is also referred to as nominal condition or ISO condition as the case may be or 100% off-design condition. When the power plant operates at any condition different from nominal specifications is known as off-design condition performance. The off-design conditions can be experienced when there are changes in ambient air conditions such as temperature and humidity, or when the power output generated by the turbine is less or more than the design power output. The Off-design model will be carrying out to change air ambient temperature and when the loads' performance is different from nominal.

The variation analysis of power output and thermal efficiency with ambient air temperature was modelled using the EBSILON Professional Software for the Power Plant. Figure 5 represents the simulated outcome of the installed SGT5 – 2000E Power plant at 26 ^{0}C ambient air temperature.

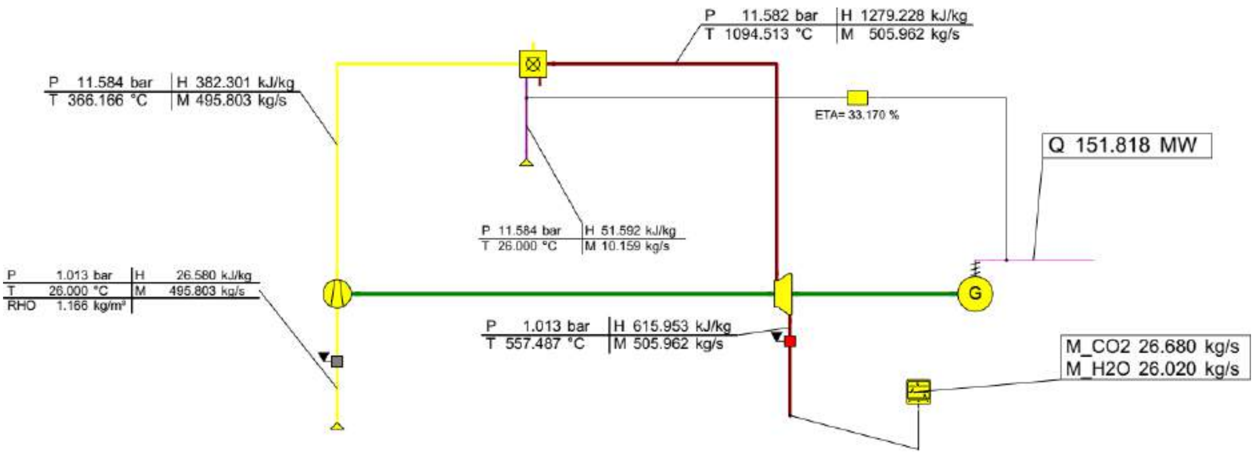

Figure 5: Topology of the Installed SGT5 – 2000E

2.4.1 Variation of Ambient Air Temperature

The power plants are affected by the change in ambient conditions. The variations of thermodynamic performance that occurred are due to the change in the air properties entering the air compressor like humidity and temperature. For this research, the variation of humidity (φ) when the temperature changes will be considered according to the installed SGT5 – 2000E Plant in Ihobvor community weather information which can be obtained from Equation (16). This equation was determined by using the data curve fit from data obtained in Azura EIA [38]. Additionally, the change in temperature affects the density (ρ) of air inversely, and the density of air is proportional to its mass flow rate. Accordingly, the mass airflow rate into the compressor with being reduced at high ambient air temperature since the compressor volume flow rate is fixed. This implies that gas turbines that operate in hotter weather conditions generate low power output because of the low mass flow of air involved. Consequently, it is important to study the effect of ambient air temperature on the gas turbine outcome parameters. The various ambient air density and mass flow rates is calculated using Equations (14) and (16) respectively for different ambient air temperatures.

$$\rho_i = \rho_N \frac{T_N}{T_i} \tag{14}$$

$$\dot{m}_i = \dot{m}_N \frac{\rho_i}{\rho_N} \tag{15}$$

$$\phi = -0.0408t_1^3 + 3.1218t_1^2 - 80.238t_1 + 774.44 \tag{16}$$

where T_N, ρ_N, and $\dot{m}_N$ are ambient air absolute temperature, density and air mass flow rate at nominal or design condition respectively; T_i, ρ_i and $\dot{m}_i$ are ambient air absolute temperature, density and air mass flow rate at off-design conditions respectively; t_i and ϕ are ambient air temperature and relative humidity respectively.

The mass flow rate of fuel was computed by EBSILON software according to the air ratio entered. In Ebsilon, the parameter that helps to control the maximum temperature obtained from the gas turbine cycle is represented by "ALAM" and is known as air ratio. For a given amount of fuel, the ratio of the actual mass of air to the stoichiometric mass of air is defined as the air ratio. When the air ratio is changed, the combustion chamber will take in less or more fuel. The turbine inlet temperature of the cycle will also change. If the air ratio is increased more air will be accepted about close to the stoichiometric air, which will result in the reduction of turbine inlet and exhaust temperatures. However, the gas turbine inlet and exhaust temperatures will be increased when the air ratio is reduced.

2.4.2 Variation of Partload

When an amount of power delivered is lower or higher than the nominal power, which is enough for satisfying the demand, is term as partload operation by the gas turbine. This happens when the load demand is different from the nominal load. A controller is introduced to take care of the load adjusting accordingly as needed. The off-design model for the partload variation is shown in Figure 6. Once the correct load is inputted in the controller the air ratio is adjusted accordingly. The amount of air and fuel flow rates needed to generate the corresponding power is then calculated. Partload of 40 – 110% of the nominal load condition was considered for this analysis. GT partload is controlled by using the "GT load level controller" to regulate the exhaust mass flow rate in the GT combustor and the range 40-110%.was used for the load variation analysis.

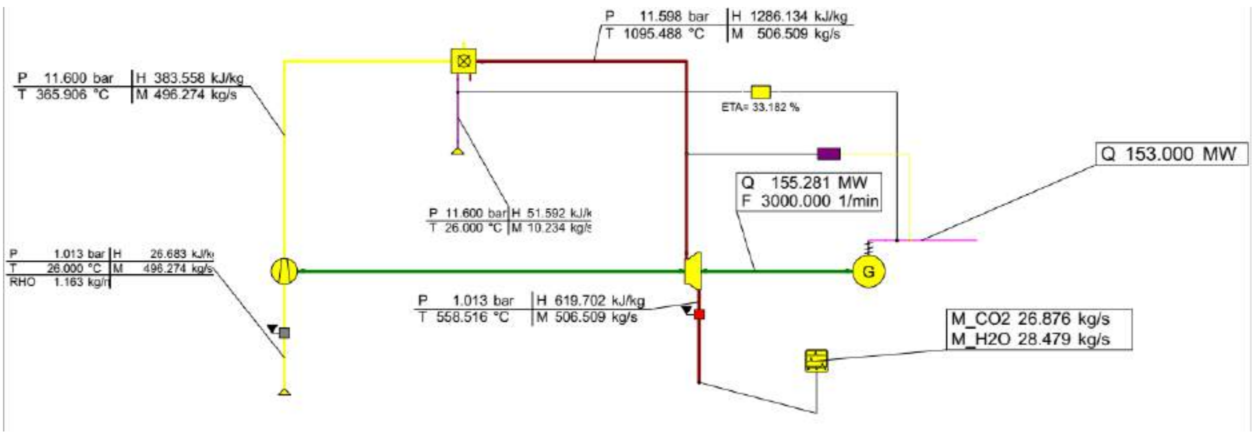

Figure 6: Topology of the Installed SGT5 – 2000E for Part Load Analysis

2.5 Variation of Efficiency in Components

The pressure ratio and power output of gas turbine changes during off-design operations. The compressors and turbine's isentropic efficiencies are also affected. The isentropic efficiencies of the compressor and turbine vary directly to their mass flow rates. Normally, the designed or defaults isentropic efficiencies are used for design conditions simulation, while for off-design simulation the isentropic efficiencies will now vary according to the actual mass flow rate. Tables 2 and 3 show the correction curves for the compressor and turbine to take care of the variation between isentropic efficiencies and mass flow rates. The isentropic efficiency values are denoted by ETAI as shown in Tables 2 and 3.

Table 2: Variation of Compressor Isentropic Efficiency with Mass flow rate of Air

$\dot{m}/\dot{m}_N$	$ETAI/ETAI_N$
0	0
0.4	0.9
1	1
1.2	1.1

Table 3: Variation of Turbine Isentropic Efficiency with Mass flow rate of Exhaust Gas

$\dot{m}/\dot{m}_N$	$ETAI/ETAI_N$
0	0.85
0.4	0.9
0.7	0.95
1	1
1.2	1.1

2.6 Validation of the Model

The model built is used for monitoring existing gas turbine power plant performance, the values obtained was compared with the ISO and installed SGT5 – 2000E data. The Power generated, exhaust mass flow rate, turbine exhaust temperature, and heat rate was used for ISO condition validation. The operating data from the operating SGT5 -2000E was used for the line operating validation for the model. Equation (17) can be utilized to compute the percentage error in the model [23,27,29, 39-41].

$$\%\text{ModelError} = \frac{(\text{Actual data - Model data})*100\%}{\text{Actual data}} \tag{17}$$

The actual operating data obtained from the installed SGT5 -2000E power plant are presented in Table 4.

Table 4: Actual Performance Data for GT11

S/N	P_{net} (MW)	t_1 (^{0}C)	t_2 (^{0}C)	p_1 (bar)	p_2 (bar)	t_3 (^{0}C)	t_4 (^{0}C)	$\dot{m}_a$ (kg/s)	$\dot{m}_f$ (kg/s)	$\dot{m}_g$ (kg/s)
1	159.41	21	363.6	1.0091	11.06	1103.0	534.0	504.01	10.74	514.75
2	157.86	22	365.8	1.0093	11.04	1103.5	535.1	502.33	10.78	513.11
3	156.16	23	368.0	1.0094	11.03	1104.0	536.3	500.65	10.82	511.46
4	154.46	24	370.2	1.0096	11.01	1104.5	537.5	498.97	10.85	509.82
5	152.82	25	372.5	1.0097	11.00	1105.0	538.6	497.29	10.89	508.18
6	149.86	27	376.9	1.0100	10.97	1106.0	541.0	493.93	10.96	504.89
7	147.14	29	381.3	1.0103	10.94	1107.0	543.3	490.57	11.03	501.60
8	145.49	30	383.5	1.0105	10.92	1107.5	544.5	488.89	11.07	499.95
9	144.04	31	385.8	1.0106	10.91	1108.0	545.7	487.20	11.10	498.31
10	142.60	32	388.0	1.0108	10.89	1108.5	546.9	485.52	11.14	496.66
11	141.27	33	390.0	1.0109	10.88	1109.0	548.0	483.84	11.18	495.02
12	139.80	35	392.0	1.0112	10.85	1110.0	550.0	480.48	11.25	491.73

3. Results and Discussion

In this research work, EBSILON Professional 11.4 was used for the modeling and simulation of SGT5 – 2000E. The validation and analysis results of SGT5 -2000E for design and off-design conditions are presented in the section.

3.1 Model Validation

The validation results of the SGT5 – 2000E model for both ISO and Azura Edo guarantee conditions developed in Ebsilon are presented in Table 5. Equation (17) was used to compute the percentage error or deviation between the design data and model data. This was carried out to ensure that the model is consistent with the ISO design data, which form the root profile of the model that all other conditions simulation was done.

Table 5: Results of Model validation for ISO Design Data

S/N	Parameters	Design	Model	Diff	%Error
1	Power (MW)	166	165.128	0.872	0.525301
2	Heat Rate(kJ/kWh)	10375	10411.55	-36.5452	-0.35224
3	Thermal Efficiency (%)	34.7	34.577	0.123	0.354467
4	Turbine Exhaust Temperature (^{0}C)	541	542.389	-1.389	-0.25675
5	Exhaust mass flow rate (kg/s)	525	525	0	0

As presented in Table 5, the error between the ISO and model data is 0.525%, -0.352%, 0,354%, -0.257% and 0% for net Power, Heat rate, net thermal efficiency, turbine exhaust temperature and turbine exhaust mass flow rate respectively. The results obtained from the model validation revealed that the model data are found to be a good agreement with the ISO. These validation results are also in the same range as the works of Miguez Da Rocha [23], Wallentinen [27]; Matjanov [29] and Jingzhi [42] that employed the use of EBSILON Professional. Since the results of the model are consistent with the ISO, the off-design conditions results is presented and compared with operating parameters from the Installed SGT5 -2000E Power Plant. This will show how the model mimics the actual operation of the power plant.

3.2 Ambient Temperature Variation

The ambient temperature ranging from 15 ^{0}C to 35 ^{0}C was utilized in the EBSILON Model developed to evaluate the power plant performance. The variation of density and the air mass flow rate for various ambient temperatures were determined using Equations (14) and (15) and the outcome is presented in Table 6. The relative humidity values obtained using Equation (16) are illustrated in Table 7.

Table 6: Model Results for Density and Air Mass Flow rate for various ambient air temperatures

t (^{0}C)	T (K)	Density (kg/m^3)	$\dot{m}_a$ (kg/s)
15	288.15	1.215	514.395
26	299.15	1.170323	495.4803
21	294.15	1.190217	503.9025
22	295.15	1.186184	502.1952
23	296.15	1.182179	500.4995
24	297.15	1.1782	498.8151
25	298.15	1.174249	497.1421

26	299.15	1.170323	495.4803
27	300.15	1.166424	493.8295
28	301.15	1.162551	492.1897
29	302.15	1.158703	490.5607
30	303.15	1.154881	488.9425
31	304.15	1.151084	487.3349
32	305.15	1.147312	485.7379
33	306.15	1.143564	484.1513
34	307.15	1.139841	482.575
35	308.15	1.136142	481.009

Table 7: Temperature and Relative Humidity relationship for Azura Edo Power Plant Location

S/N	t (^{0}C)	RH (%)
1	20	92
2	21	88.307
3	22	85.7168
4	23	83.9846
5	24	82.8656
6	25	82.115
7	26	81.488
8	27	80.7398
9	28	79.6256
10	29	77.9006
11	30	75.32
12	31	71.639
13	32	66.6128
14	33	59.9966
15	34	51.5456
16	35	41.015

The various values of relative humidity and air mass flow rates in Tables 6 and 7 were used in off-design modelling for ambient air temperature as described in Section 2.4.1 and the model performance results are showed in Table 8. The turbine inlet and exhaust temperature evolution with ambient air for off-design conditions are shown in Figure 7.

Table 8: Variation of Parameters in Ambient air temperature Off – Design

t_1 (^{0}C)	t_2 (^{0}C)	t_3 (^{0}C)	t_4 (^{0}C)	p_2 (bar)	p_3 (bar)	$\dot{m}_a$ (kg/s)	$\dot{m}_f$ (kg/s)	$\dot{m}_g$(kg/s)	P_{net} (MW)	η_{thnet} (%)
15	363.8	1091.2	542.4	12.00	12.00	514.40	10.61	525.00	165.15	34.58
21	360.1	1093.6	551.2	11.77	11.77	503.90	10.36	514.26	157.72	33.80
22	361.3	1093.9	552.5	11.73	11.73	503.20	10.32	513.51	156.50	33.67
23	362.5	1094.1	553.8	11.69	11.69	500.50	10.27	510.77	155.28	33.54
24	363.7	1094.2	555.1	11.65	11.65	498.82	10.24	509.05	154.07	33.41
25	364.8	1094.3	556.4	11.61	11.61	497.19	10.19	507.39	152.85	33.28
26	366.2	1094.5	557.5	11.58	11.58	495.80	10.16	505.96	151.82	33.17
27	367.2	1094.4	559.0	11.54	11.54	493.83	10.11	503.94	150.44	33.03

28	368.4	1094.4	557.9	11.50	11.50	492.19	10.07	502.26	147.28	32.90
29	369.6	1094.2	561.0	11.46	11.46	490.56	10.03	500.59	148.02	32.77
30	370.8	1094.1	562.1	11.42	11.42	488.94	9.99	498.93	146.84	32.64
31	371.9	1093.9	563.9	11.38	11.38	487.36	9.94	497.30	145.64	32.51
32	373.1	1093.6	564.1	11.35	11.34	485.94	9.90	495.84	144.43	32.38
33	374.2	1093.3	565.0	11.37	11.31	484.15	9.86	494.01	143.25	32.24
34	375.3	1092.8	565.9	11.27	11.27	482.58	9.81	492.39	142.02	32.11
35	376.4	1092.3	567.0	11.23	11.23	481.009	9.77	490.78	140.82	31.98

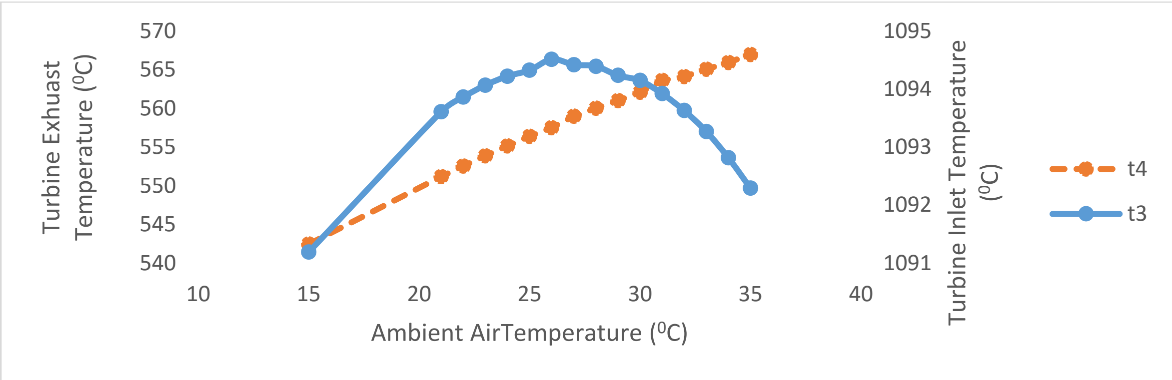

Figure 7: Model Variation of ambient air temperature with Turbine Inlet and Exhaust temperatures

As the ambient air temperature increases and the air mass flow rate reduces, the pressure ratio within the turbine is decreased as observed in Table 8. Also, as a result of the gas turbine allowing capacity and the decrease in air and fuel mass flow rates, the pressure, p_3 at the turbine inlet is lowered due to the increase in ambient air temperature. This results in a lesser pressure ratio in the gas turbine section, and the net power output is also reduced. It was also observed in Table 8 that as the ambient air temperature increased from 15 ^{0}C to 35 ^{0}C, the net power generated decreases from 165.146 to 140.816MW, the net efficiency from 34.578 to 31.577%.

As observed in Figure 7, as the ambient air temperature increases, the turbine inlet temperature increases to pick value at 26 ^{0}C of the ambient air temperature and start decreasing until it gets to 35 ^{0}C. This shows that the maximum turbine inlet temperature of 1094.513 ^{0}C for the model was obtained at 26 ^{0}C. The turbine exhaust temperature increases as the ambient temperature rise as shown in Figure 7, this is due to the decrease in pressure ratio.

The model results have shown that the gas turbine generates more power at lower ambient temperature conditions, this is because of the large mass of gas that was expanded in the turbine and at a higher pressure ratio. Consequently, this makes the gas turbine generate more power at low ambient air temperature than the high ambient air temperature.

3.3 Partload Variation

The various values obtained from the 40% to 110% partload off-design condition that was discussed in Section 2.4.2 are shown in Table 9. Figure 8 represents the mass flow rate and temperature of the gas turbine exhaust variation when the nominal power output varied from 40% to 110% for the gas turbine performance.

Table 9: Model Results for Partload Variation off – Design

S/N	Part Load (%)	$\dot{m}_a$ (kg/s)	$\dot{m}_f$ (kg/s)	$\dot{m}_g$ (kg/s)	λ	t_4 (^{0} C)	t_3 (^{0} C)	p_3 (bar)	η_{th} (%)	P_{net} (MW)	p_2 (bar)	r_{pc}	r_{pt}
1	110	495.4	11.1	506.4	2.86	593.68	1152.13	11.83	33.736	168.3	11.84	11.68	11.68
2	100	496.3	10.2	506.5	3.1	558.52	1095.49	11.60	33.182	153	11.60	11.45	11.45
3	90	494.8	9.4	504.3	3.36	526.38	1041.90	11.32	32.462	137.7	11.32	11.17	11.17
4	80	496.2	8.6	504.8	3.7	489.37	981.76	11.07	31.689	122.4	11.07	10.93	10.93
5	70	496.3	7.7	504.1	4.1	453.39	922.58	10.80	30.719	107.1	10.80	10.66	10.66
6	60	496.6	6.9	503.5	4.6	416.42	861.72	10.51	29.521	91.8	10.51	10.38	10.37
7	50	496.2	6.1	502.3	5.23	379.30	799.77	10.20	27.251	76.5	10.20	10.07	10.06
8	40	495.2	5.2	500.5	6.06	341.41	736.01	9.86	26.1	61.2	9.86	9.73	9.73

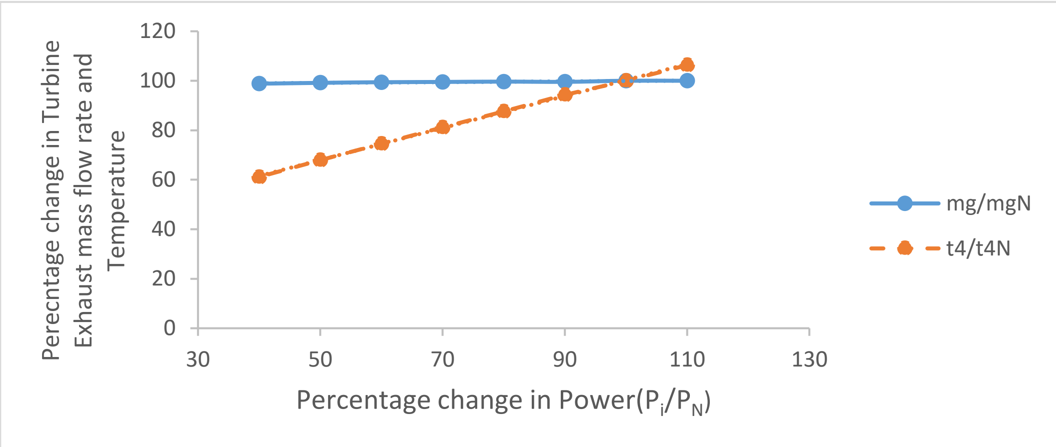

Figure 8: Variation of the percentage change in exhaust mass flow rate and turbine exhaust temperature

In Table 9, it was observed that as the partload reduces, mass flow rates of fuel and exhaust, the turbine inlet and exhaust temperatures, net efficiency, pressure ratio, and mass of carbon dioxide emitted decrease. While the AR increases as the partload decreases. As shown in Figure 8, the percentage of exhaust mass flow rate with the nominal value was almost constant with a change in partload. This shows that partload variation has little effect on the exhaust mass flow rate of the gas turbine power plant. The percentage change in the turbine exhaust temperature increased as the partload increases. This means the higher the partload, the higher the exhaust temperature and this will be favourable when utilizing the exhaust temperature for further processes like HRSG at high partload.

3.4 Comparison of Model Data with Actual Operating Data

As mentioned, in Section 2.6, the model validation was done in two ways. The first was presented in Section 3.1, which then other conditions data was generated was the model analysis for changing in ambient air temperature. The second is presented in this section. For the validation of the model results with operating data from the Azura Edo Power plant. The compressor exit, turbine inlet and exhaust temperatures, air, fuel, and exhaust mass flow rates, and net power out from the Installed SGT5 -2000E model power plant at various states were compared with model data. The variations of the operating data with model data with ambient air temperature for the various GT units are presented in Figures 9 to 15. The summary of the resulted errors between model data and operating data for various parameters and GT units is presented in Table 10.

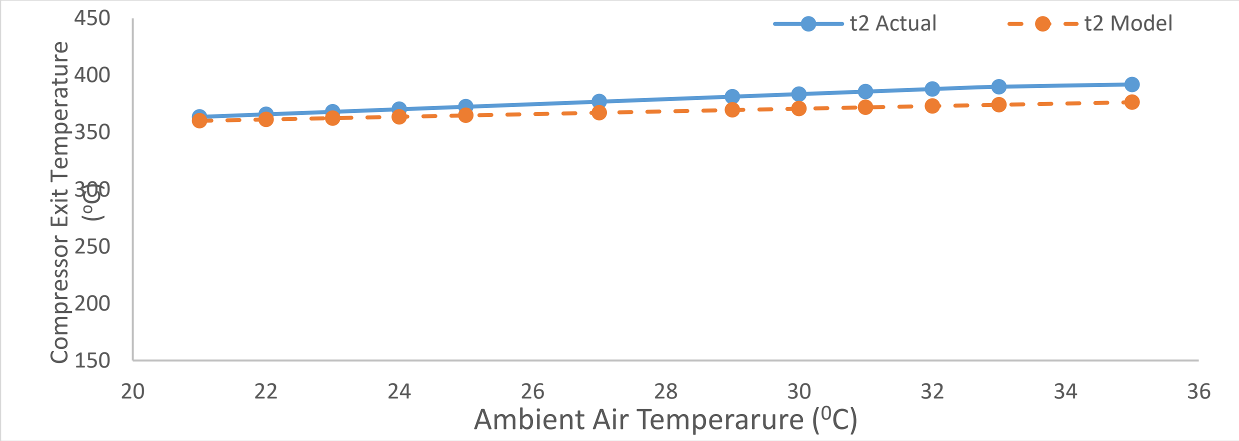

Figure 9: Model Validation for variation in compressor exit temperature with ambient air temperature for the GT

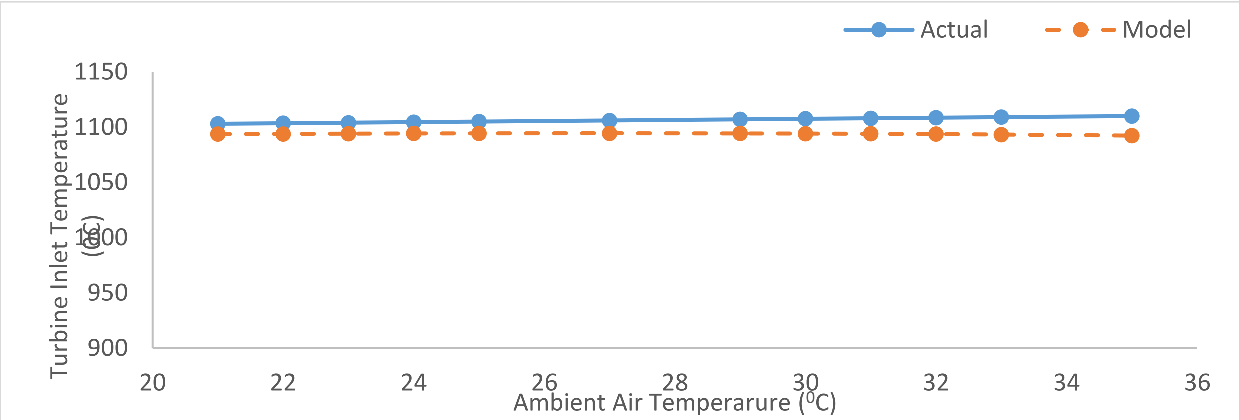

Figure 10: Model Validation for variation in turbine inlet temperature with ambient air temperature

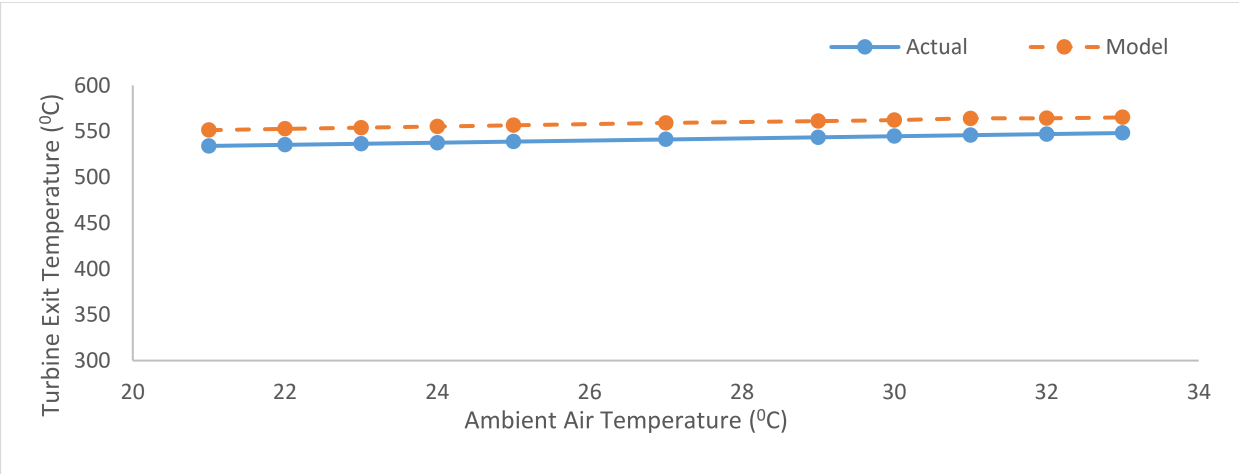

Figure 11: Model Validation for variation in turbine Exhaust temperature with ambient air temperature

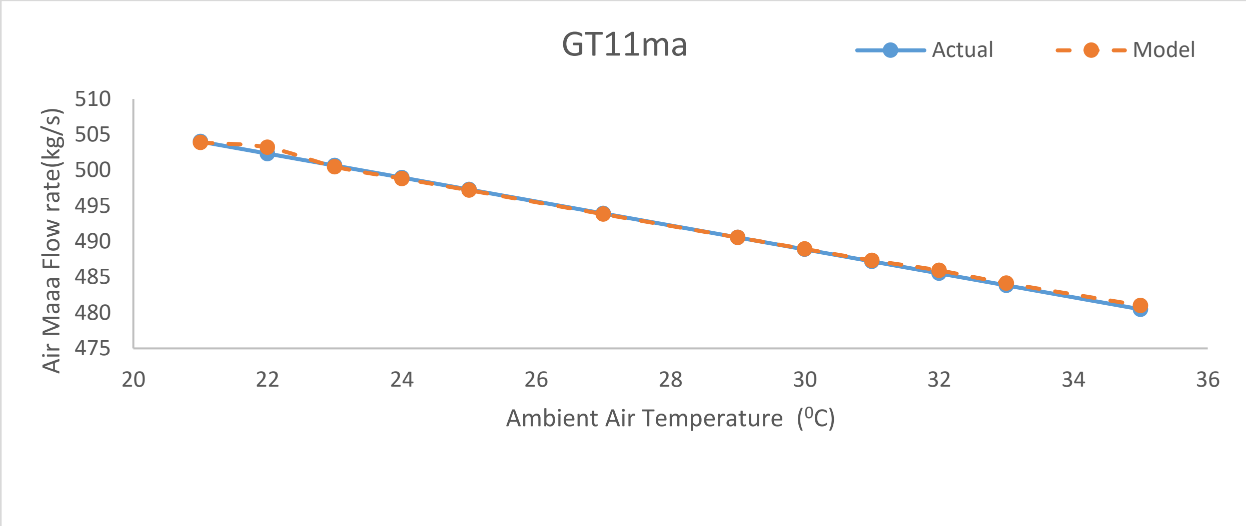

Figure 12: Model Validation for variation in air mass flow rate with ambient air temperature

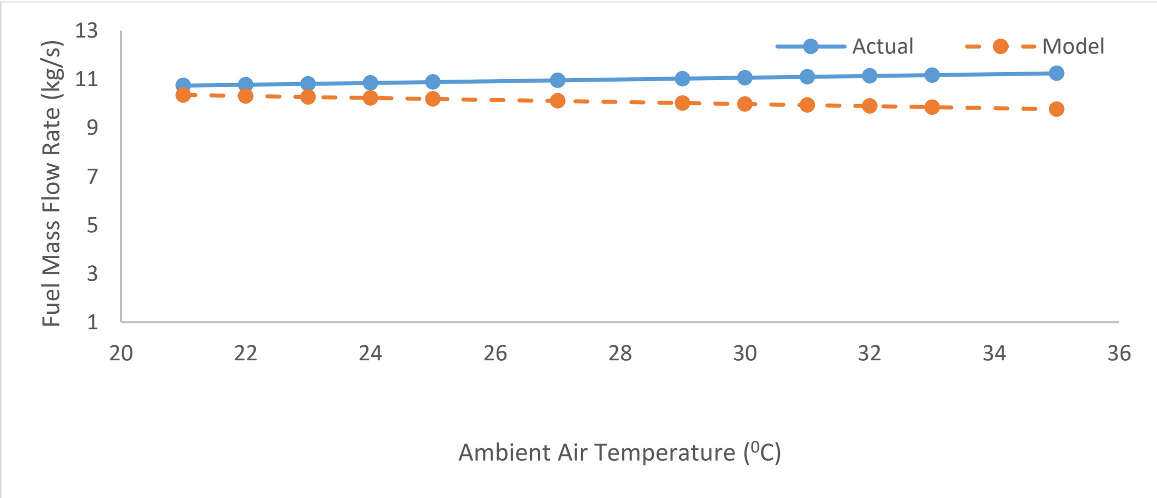

Figure 13: Model Validation for variation in fuel mass flow rate with ambient air temperature

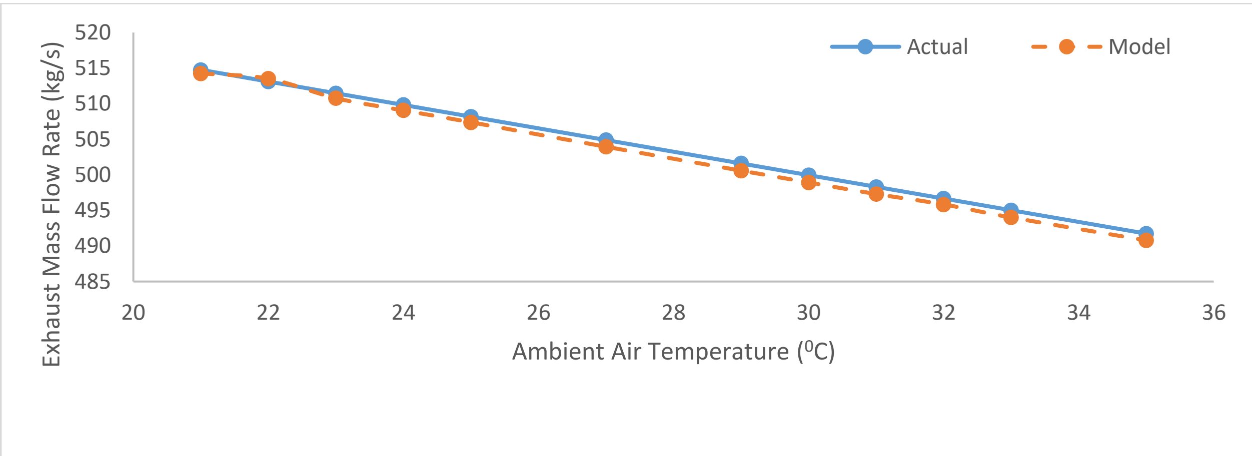

Figure 14: Model Validation for variation in exhaust mass flow rate with ambient air temperature

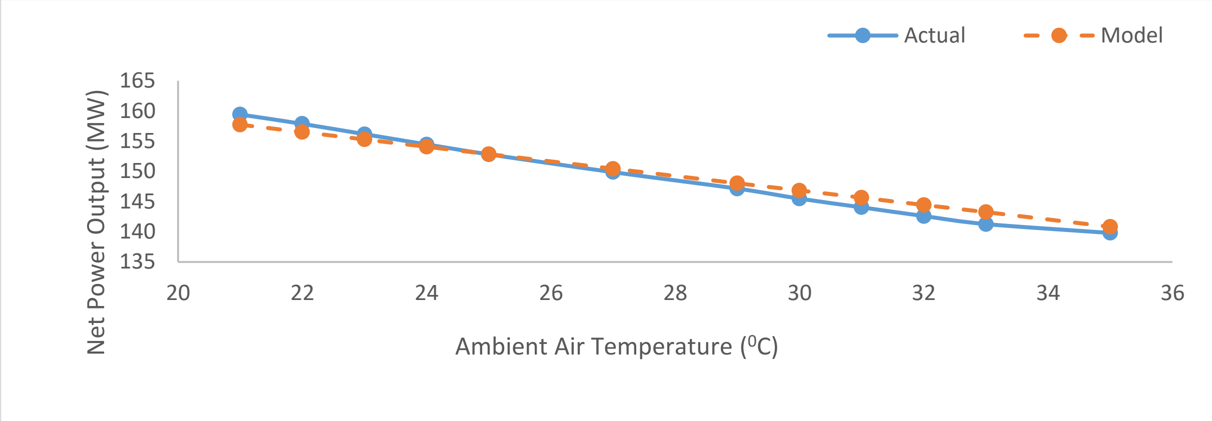

Figure 15: Model Validation for variation in net Power out with ambient air temperature

Table 10: Model Validation Results Using Operating Data

S/N	t_2 (%)	t_3(%)	t_4(%)	m_a(%)	m_f(%)	m_g(%)	P_{net}(%)
1	0.96	0.85	-3.22	0.02	3.58	0.10	1.06
2	1.23	0.87	-3.25	-0.17	4.29	-0.08	0.86
3	1.50	0.90	-3.27	0.03	5.03	0.14	0.56
4	1.78	0.93	-3.29	0.03	5.69	0.15	0.26
5	2.05	0.97	-3.30	0.02	6.37	0.16	-0.02
6	2.57	1.05	-3.33	0.02	7.75	0.19	-0.39
7	3.08	1.15	-3.25	0.00	9.11	0.20	-0.60
8	3.33	1.21	-3.23	-0.01	9.78	0.21	-0.93
9	3.59	1.27	-3.34	-0.03	10.46	0.20	-1.11
10	3.84	1.34	-3.15	-0.09	11.13	0.17	-1.29
11	4.05	1.42	-3.11	-0.06	11.80	0.20	-1.40
12	3.98	1.60	-3.09	-0.11	13.14	0.19	-0.73
AVG	**2.66**	**1.13**	**-3.24**	**-0.03**	**8.18**	**0.15**	**-0.31**

In Figures 9 to 15, it was observed that the model values for compressor exit, turbine inlet and exhaust temperatures were close to the actual operating data, which both results increased linearly as ambient air temperature increases for the turbine unit. The slightly different notice was as a result of differences in pressure ratio observed for the model and actual data as shown in Tables 4 and 8. The model and operating values for air and exhaust flow rates decrease with an increase in ambient air temperature and slight differences between were observed as shown in Figures 12 and 14 in the gas turbine power plant. The comparison between the model and operating values for fuel flow rate as shown in Figure 13 revealed that model data decreases slightly while the operating data increases as ambient air temperature increases. The high mass flow rate of fuel in the operating data could be attributed to the fact that the power plant needed more fuel to meet up power generated at a low-pressure ratio value compared to the corresponding model value with a higher-pressure ratio. As shown in Figure 15, both the model and operating data for net power output decrease as ambient air temperature increases. It was observed that much difference was not noticed between the model data and operating data. This also shows that the mass flow rate of fuel plays an important role in power generation, which resulted from the increased fuel mass flow rate to reduce the effect of the low-pressure ratio observed in the operating data.

Table 10 reports the validation results showing that the average error between the model and operating values for compressor exit, turbine inlet and exhaust temperatures, air, fuel and exhaust flow rates and net power is 2.66, 1.13, -3.24, -0.03, 8.18, 0.15 and -0.31 respectively for the GT unit. The moderate deviations observed in turbine inlet and exhaust temperatures, mass flow rates of fuel, and net power generated are indications of the difference in pressure ratios between the model and actual data. Also, it indicates some additional uncertainty in the modelling of the SGT5 – 2000E gas turbine on the same principle. The validation results are in good agreement with the results obtained from Miguez Da Rocha [23] and Wallentinen [27] that used the same software for their study. Therefore, the data obtained from the model validation show that the developed model is adequate and consistent. Thus, its results can be used for further studies in the thermodynamic evaluation of the SGT5 – 2000E gas turbine model power plant.

4. Conclusion

The modelling and simulation of the SGT5 – 2000E gas turbine model were carried out utilizing EBSILON professional software. The modelling was done for design and off-design conditions to provide parameters that were normally not given by manufacturers of gas turbine models. The results of model validation obtained were in good agreement with actual operating data. Also, the data obtained will be useful in thermodynamic and environmental analyses of the installed gas turbine model.

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Nomenclature

ASHRAE	American Society of Heating, Refrigeration and Air Conditioning Engineers
C	Compressor
CC	Combustion chamber
CIT	Compressor Inlet Temperature (^{0}C, K)
c_p	Specific heat capacity at constant pressure (kJ/kg K)
GE	General Electric
GT	Gas turbine unit
HR	Heat rate (kJ/kWh)
IPP	Independent Power Producer
ISO	International Standard Organization
LHV	Lower heating value (kJ/kg)
$\dot{m}$	Mass flow rate (kg/s)
NBET	Nigerian Bulk Electricity Trading PLC
P	Pressure (bar)
P	Power (MW)
Q	Heat supply/removal rate (MW)
r_p	Compressor Pressure ratio
T	Absolute temperature (K)
t	Temperature (^{0}C)
TIT	Turbine Inlet Temperature (^{0}C, K)
W	Work done (MW)
W.R	Work ratio

Greek Symbols

ϕ	Relative Humidity
η_c	Compressor isentropic efficiency
η_{cc}	Combustion efficiency
η_{gen}	Generator efficiency

η_{net}	Net Thermal Efficiency
η_{o}	Overall Thermal Efficiency
η_{oc}	overall combined cycle efficiency
η_{T}	Turbine isentropic efficiency
$\eta_{thermal}$	Thermal efficiency
η_{thgr}	Gross thermal efficiency
ϒ	Specific heat capacities ratio
Ρ	Density (kg/m^3)

Subscripts

a	Air
AC	Air compressor
Amb	Ambient
CC	Combustion chamber
f	Fuel
G	Exhaust gas
GT	Gas turbine
T	Turbine

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