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Classification: *GJRE-A Classification (FOR): 091307, 091305*

COMPUTER SIMULATION OF A GAS TURBINE PERFORMANCE

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Computer Simulation of a Gas Turbine Performance

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I. Introduction

The growth in electricity demand being experienced in Nigeria has resulted in the need to build power plants that generate maximum power output at ambient temperature. Due to their installation time, low installation cost and availability of natural gas in the country, many states of the country are currently building gas turbine power plants to meet this demand. However, one disadvantage that penalizes the gas turbine power rating is the adverse effect of the ambient temperature on the gas turbine power output and efficiency. In order to utilize the high economic and energy saving potential of a gas turbine power plant in their simple and combined cycles, it is important to identify their optimal design parameters and determine the impact of the deviation of these parameters from the standard conditions, on the overall performance of the plant. Gas turbines designed to operate at maximum efficiency at standard ambient temperatures and relative humidity may tend to reduce in performance due to adaptation problems resulting from variation in weather conditions as they are installed at different locations. Numerous methods of analysis of gas turbine systems have been proposed amongst which is the exergy method. The exergy method is a performance analysis of a thermal system based on the second law of thermodynamics which extends beyond the limits of energy-based analysis since exergy is generally not conserved as energy but is destroyed in the system. The exergy method assists the engineer in identifying the source and magnitude of performance loss in a thermal system by measuring the irreversibilities that occur in different devices and sections of the system. Significant works in the field of simple and cogeneration cycle gas turbine power plants have been recorded. Ogaji (1997) utilized first law to develop a computer simulation model for investigating the performance of various gas turbine cycles. Pankaj (2003) verified the impact of high ambient air temperature on the performance of various gas turbine models utilizing performance data obtained for each model as a basis for comparison and proposed the Earth Tube Heat Exchanger (ETHE) technology as the most effective and economical inlet air cooling method. Somkiat and Pichai (2004) performed an exergy evaluation of a combined steam and gas turbine plant to quantify exhaust loss and its effect on the environment. Mohamad and Mofid (2005) performed an exergy analysis of a regenerative gas turbine cycle to identify sources of performance loss in the plant. Naser (2005) compared various modified Brayton cycles with a regenerative, two-isothermal heat addition Brayton cycle using second law analysis. Kamal and Zuhair (2006) investigated the technical and economical feasibility of using turbine inlet air cooling and its effect on the performance of gas turbines in Khartoum which is a high ambient temperature and dusty area and proposed wetted media evaporative cooling to be the most economically feasible option for improving the performance of gas turbine plants in the area. Tamer (2006) determined the optimum design parameters of a Brayton- Heat Recovery Steam Generator (HRSG) cycle at maximum exergy and their effects on the exergetic efficiency. Sanjay et al (2009) utilized exergy analysis principles and a computer code to simulate the performance of a Brayton – diesel cycle. Ashok et al (2010) combined the first and second law analysis to develop a design methodology for parametric study and thermodynamic performance evaluation of a closed Brayton cycle with Heat Recovery Steam Generator (HRSG).

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In this paper, an exergy analysis was performed for a 33-MW gas turbine plant, which is an existing plant located in Port Harcourt, Nigeria. Mass and energy conservation laws were applied to each component and quantitative exergy balance of each component and the overall plant was also delivered. Based on the model equations developed, a computer program is written which serves as an efficient tool for quantifying the exergy flow rate at each state point in the cycle, evaluating the efficiencies and irreversibilities in each component and for the overall plant, and simulating the performance of the plant and its components when the ambient and turbine inlet temperatures are varied.

II. PROBLEM FORMULATION AND SOLUTION METHOD

The schematic of a GE-MS6001, 33-MW single shaft gas turbine system which operates on Brayton cycle is given in Figure 1 and shows the main work and exergy flows and the state points which were accounted for in this analysis. The plant consists of an axial flow air-compressor (AC), a combustion chamber (CC), and a gas turbine (GT). Figure 2 is the T-s diagram showing the losses due to inefficiencies of the components of the actual open cycle gas turbine plant.

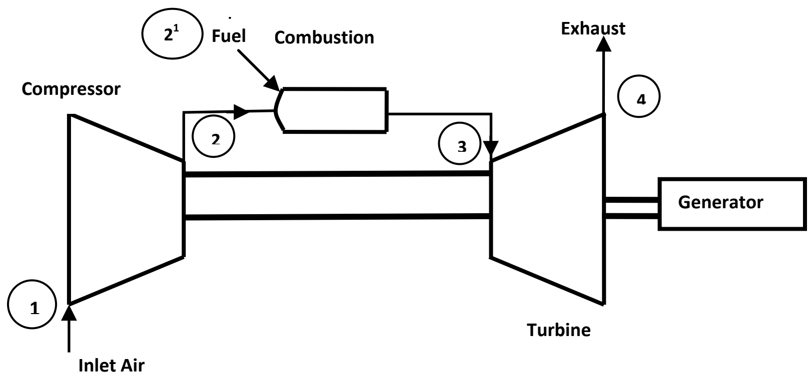

Figure1- the open-loop gas-turbine power plant

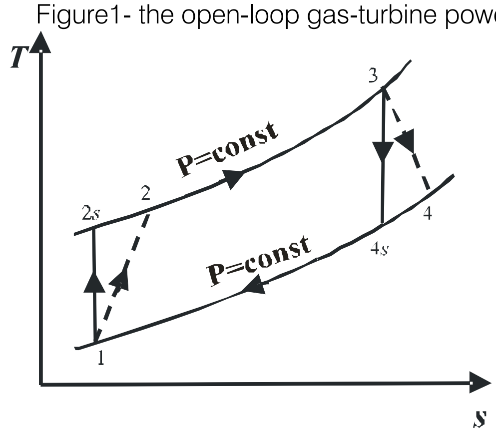

Figure2- T-s diagram of the actual open cycle gas turbine plant

The thermodynamic analysis of the gas turbine plant has been done by treating each component of the system as a control volume at steady state. This implies that the components experience no changes in their mass, energy, entropy, volume and exergy content. Hence, the amount of exergy entering the system in all forms (heat, work, mass transfer) must be equal to the amount of exergy leaving the system plus the exergy destroyed. A general exergy-balance equation, applicable to any component of a thermal system may be formulated by utilizing the first and second laws of thermodynamics (Mahamad and Mofid, 2005). The thermo-mechanical exergy stream may be decomposed into its thermal and mechanical components so that the balance in rate form gives

$$\dot{E}_i^m - \dot{E}_e^m = \left(\dot{E}_i^T - \dot{E}_e^T\right) - \left(\dot{E}_i^P - \dot{E}_e^P\right) \qquad (1)$$

where the subscripts i and e represent inlet and exit states; $\dot{E}^m$ is the exergy of the flow stream, $\dot{E}^T$ is the thermal component of the exergy stream; $\dot{E}^P$ is mechanical component of the exergy stream; the term on the left –hand side of the equation represent the change in exergy of the flow stream, the first and second terms on the right-hand side of the equation represent the changes in the thermal and mechanical components of the exergy stream, respectively.

The thermal and mechanical components of the exergy stream, assuming an ideal gas with constant specific heat, may be written, respectively, as

$$\dot{E}^{T} = \dot{m}c_{p}\left[\left(T-T_{0}\right)-T_{0}\ln\frac{T}{T_{0}}\right] \quad (2)$$

and

$$\dot{E}^{P} = \dot{m}RT_{0}\ln\frac{P}{P_{0}} \quad (3)$$

where P_0 and T_0 are the pressure and temperature, respectively, at standard state; $\dot{m}$ is the mass flow rate of the working fluid; R is the gas constant; c_p is the specific heat at constant pressure.

The exergy change of a system during a process is equal to the difference between the net exergy transfer through the system boundary and the exergy destroyed within the system boundaries as a result of irreversibilities. The exergy destroyed is proportional to the entropy generated and is positive for all actual processes. Hence, the general exergy equation applicable to all the components of the gas turbine plant may be written, utilizing the decomposition defined in equation (1) as follows:

$$\dot{E}^{W} = \dot{E}^{CHE} + \left(\sum_{inlet}\dot{E}_{i}^{T} - \sum_{exit}\dot{E}_{e}^{T}\right) + \left(\sum_{inlet}\dot{E}_{i}^{P} - \sum_{exit}\dot{E}_{e}^{P}\right) + T_{0}\left(\sum_{exit}\dot{S}_{e} - \sum_{inlet}\dot{S}_{i} + \frac{\dot{Q}_{CV}}{T_{in,CV}}\right) \quad (4)$$

where $\dot{E}^{W}$ $\dot{W}$ represents the exergy rate of power output; the term E^{CHE} denotes the rate of exergy flow of fuel in the plant; $\dot{S}$ is the entropy transfer rate; $T_{in,CV}$ is the temperature of the source from which the heat is transferred to the working fluid; the fourth right-hand term is the exergy destroyed in the component; and $\dot{Q}_{CV}$ in the fourth right-hand term denotes the heat transfer rate between the component and the environment.

In heat engines, such as the Brayton cycle considered, the exergy input to the system is the difference between the exergy of the positive heat interaction between the system and the high temperature thermal source, and that of the negative heat interaction between the system and the surroundings, the recovered exergy in the process is the network of the reversible heat engine cycle (Oko, 2008). The exergy destroyed in the cycle is the sum of the exergy destructions of the processes that compose the cycle. Hence, the exergy-balance equations for each component in the gas turbine power plant can be derived from the general exergy balance equation given in equation (4). The exergy destroyed during each process is calculated separately and then summed up as the total exergy destruction in all the processes in the cycle.

The exergy-balance equations and the exergy destroyed during each process and for the whole plant are written as follows:

$$\dot{E}_{D_{AC}} = T_{0}\left(\dot{S}_{2}-\dot{S}_{1}\right) = \dot{m}T_{0}\left[c_{p1-2}\ln\left(T_{2}/T_{1}\right) - R\ln\left(P_{2}/P_{1}\right)\right] \quad (5a)$$

$$\dot{E}^{WAC} = \left(\dot{E}_{1}^{T}-\dot{E}_{2}^{T}\right) + \left(\dot{E}_{1}^{P}-\dot{E}_{2}^{P}\right) + T_{0}\left(\dot{S}_{2}-\dot{S}_{1}\right) \quad (5b)$$

Combustion chamber:

$$\dot{E}_{D_{CC}} = T_{0}\left(\dot{S}_{3}-\dot{S}_{2}+\dot{S}_{2^{1}}+\frac{\dot{Q}_{2-3}}{T_{in,CC}}\right)$$

$$\dot{m}T_{0}\left[\left(c_{p2-3}\ln\left(T_{3}/T_{2}\right) - R\ln\left(P_{3}/P_{2}\right)\right) + \left(c_{p2^{1}}\ln\left(T_{2^{1}}/T_{0}\right) - R\ln\left(P_{2^{1}}/P_{0}\right)\right) + \frac{c_{p2-3}\left(T_{3}-T_{2}\right)}{T_{in,CC}}\right] \text{ -------- } \quad (6a)$$

$$\dot{E}^{CHE} + \left(\dot{E}_{2}^{T}+\dot{E}_{2^{1}}^{T}-\dot{E}_{3}^{T}\right) + \left(\dot{E}_{2}^{P}+\dot{E}_{2^{1}}^{P}-\dot{E}_{3}^{P}\right) + T_{0}\left(\dot{S}_{3}-\dot{S}_{2}+\dot{S}_{2^{1}}+\frac{\dot{Q}_{2-3}}{T_{in,CC}}\right) = 0 \quad (6b)$$

Gas Turbine:

$$\dot{E}_{D_{GT}} = \dot{m}T_{0}\left[c_{p3-4}\ln\left(T_{4}/T_{3}\right) - R\ln\left(P_{4}/P_{3}\right)\right] \quad (7a)$$

$$\dot{E}^{WGT} = \left(\dot{E}_{3}^{T}-\dot{E}_{4}^{T}\right) + \left(\dot{E}_{3}^{P}-\dot{E}_{4}^{P}\right) + T_{0}\left(\dot{S}_{3}-\dot{S}_{4}\right) \quad (7b)$$

Exhaust:

$$\dot{E}_{D_{EXH}} = T_{0}\left[\left(\dot{S}_{4}-\dot{S}_{1}\right)+\frac{\dot{Q}_{4-1}}{T_{0}}\right] = \dot{m}T_{0}\left[\left(c_{p4-1}\ln\left(T_{4}/T_{1}\right) - R\ln\frac{P_{4}}{P_{1}}\right) + \frac{\left(c_{p4-1}\left(T_{4}-T_{1}\right)\right)}{T_{0}}\right] \quad (8)$$

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where $\dot{E}_{D_{AC}}, \dot{E}_{D_{CC}}, \dot{E}_{D_{GT}}$ and $\dot{E}_{D_{EXH}}$ represent the exergy destroyed in the air compressor, combustion chamber, gas turbine, and exhaust, respectively; $\dot{E}^{WAC}$ and $\dot{E}^{WGT}$ represent the exergy flow rate of the power output from the air compressor and the gas turbine, respectively; $\dot{E}^{CHE}$ is the exergy flow rate of fuel in the combustion chamber.

1. *Second-law efficiency of the gas turbine power plant*

Since exergy is more valuable than energy according to second law of thermodynamics, it is useful to consider both input and output from the plant in terms of exergy. From the above, the general definition of the exergy or second-law efficiency for a system may be written as

$$\eta_{II} = \frac{Exergy\ recovered}{Exergy\ supplied} = 1 - \frac{Exergy\ destroyed}{Exergy\ supplied} \quad (9)$$

Hence, the second-law efficiency of the gas turbine power plant under study is evaluated for the various components and for the overall plant from the following equations

Air compressor:

$$\eta_{II,AC} = 1 - \frac{\dot{E}_{D_{AC}}}{\dot{E}^{WAC}} \quad (10)$$

Combustion chamber:

$$\eta_{II,CC} = 1 - \frac{\dot{E}_{D_{CC}}}{\dot{E}^{CHE}} \quad (11)$$

Gas turbine:

$$\eta_{II,GT} = 1 - \frac{\dot{E}_{D_{GT}}}{\dot{E}^{WGT}} \quad (12)$$

Overall plant:

$$\eta_{II,PLANT} = \frac{\dot{E}^{WPLANT}}{\dot{E}^{CHE}} \quad (13)$$

Where $\dot{E}^{WPLANT}$ is the net power output from the plant?

2. *Power-to-Heat ratio and Specific Fuel Consumption (SFC)*

The Power –to-Heat ratio for the simple cycle is given by

$$R_{PH} = \frac{\dot{E}^{WPLANT}}{\dot{Q}_{2-3}} \quad (14)$$

where $\dot{Q}_{2-3}$ is the process heat supply rate.

The Specific Fuel Consumption (*SFC*) for the cycle is given by

$$SFC = \frac{3600 f}{\dot{E}^{WPLANT}} \quad (15)$$

where *f* is the fuel-air ratio.

III. RESULTS AND DISCUSSION

Table 1 contains a record of the online data collected for the running power plant. The mass flow rates, temperatures and pressures were obtained directly from the speedtronics control system. The reference temperature and pressure were taken as $25°C$ and 1.0132bar, respectively, at relative humidity of 60%.

Table 1 Operating data for the 33-MW gas turbine power plant

S/N	Operating parameter	Value	Unit
1	Mass flow rate of air through compressor	136.5	kg/s
2	Temperature of inlet air to compressor	302	$°K$
3	Pressure of inlet air to compressor	0.10132	*MPa*
4	Outlet temperature of air from compressor	603	$°K$
5	Outlet pressure of air from compressor	0.835	*MPa*
6	Fuel-gas(natural gas) mass flow rate	2.80	kg/s
7	Fuel- air ratio at full load(on mass basis)	0.02	-
8	Inlet Temperature of fuel-gas	302	$°K$
9	Inlet pressure of fuel-gas	0.2279	*MPa*
10	Inlet temperature to gas turbine	1087	$°K$
11	Exhaust gas temperature	644	$°K$
12	Exhaust gas pressure	0.1032	*MPa*

The exergy flow rates at the inlet and outlet of each component of the plant were evaluated based on the values of measured properties such as pressure, temperature, and mass flow rates at various states. These quantities are used as input data to the computer program written to perform the simulation of the performance of the components of the gas turbine power plant and the overall plant. The values obtained for the chemical, thermal and mechanical exergy flow rates at various state points in the gas turbine plant are shown in Table 2.

An exergy balance for the components of the gas turbine plant and of the overall plant is at this point performed and the net exergy flow rates crossing the boundary of each component of the plant, together with the exergy destruction in each component are calculated and are as shown in Table 3. The product of a component corresponds to the added exergy whereas the resource to the consumed exergy (Mahamad and Mofid, 2005). The sum of the exergy flow rate of products, resources and destruction equals zero for each component. Hence, for each component, the sum of the values of the thermal and mechanical exergy components and the exergy destruction are substituted in the respective exergy balance equation and equated to the value of the output exergy as shown in the table.

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Table 2 Exergy flow rates and entropy generation rates at various state points in the gas turbine plant

State	$\dot{m}(kg/s)$	$T(K)$	$P(MPa)$	$\dot{E}^T(MW)$	$\dot{E}^P(MW)$	$\dot{E}^{CHE}(MW)$	$\dot{S}(MW/K)$
1	136.5	302	0.10132	0.0452	0.0000	0.0000	0.0000
2	136.5	603	0.8350	14.6604	23.7966	0.0000	0.0144
2^1	2.80	302	0.2279	0.0021	0.7205	112.8035	0.0014
3	139.3	1087	0.8100	67.0999	23.9347	0.0000	0.0365
4	139.3	644	0.1075	18.3703	0.2117	0.0000	0.0015

This zero sum indicates that the exergy balance for the compressor, combustion chamber, the gas turbine and the overall plant are satisfied. The value of the total exergy destruction in the plant calculated from the addition of the individual exergy destructions in the components of the cycle is compared with the calculated value of the exergy destruction in the exhaust.

The exergy flow rate of the power output of the gas turbine power plant is found from the exergy balance to be 30.2 MW. The exergy flow rate of fuel in the combustion chamber is found to be 116.9MW. The total exergy destruction in the plant is found to be 69.83MW. The gas turbine is found to have the highest efficiency of 99.3%. The exergy efficiency of the combustion chamber is much lower than that of other

Table 3 Net exergy flow rates and exergy destruction in the gas turbine plant

Component	$\dot{E}^W(MW)$	$\dot{E}^{CHE}(MW)$	$\dot{E}^T(MW)$	$\dot{E}^P(MW)$	$\dot{E}_D(MW)$
Air Compressor	42.5290	0.0000	13.2241	24.9533	4.3515
Combustion Chamber	0.0000	116.8655	50.9338	0.6106	65.0313
Gas Turbine	72.7329	0.0000	47.4040	24.8762	0.4527
Plant	30.2	116.8655	16.7539	0.5335	69.8355
Exhaust	0.0000	0.0000	0.0000	0.0000	81.9193

plant components due to the high irreversibility in this section. Its value is calculated as 44.3%. The exergy efficiency of the axial flow air compressor is calculated as 89.7%. The exergy efficiency of the overall plant at compressor inlet air temperature of $29°C$ and turbine inlet temperature of $1087K$ is found to be 25.8%.

The Grassmann diagram of the Brayton cycle power plant is shown in Figure 3. It shows the percentage exergy input and exergy loss in each device and the exhaust based on the results of the exergy analysis. Compared with other components of the power plant, the largest amount of the total exergy supplied in the plant is destroyed in the combustion chamber, the least exergy loss found in the gas turbine. It is also shown that about 43.7% of the total inlet exergy flow in the plant is destroyed and rejected in the exhaust to the atmosphere.

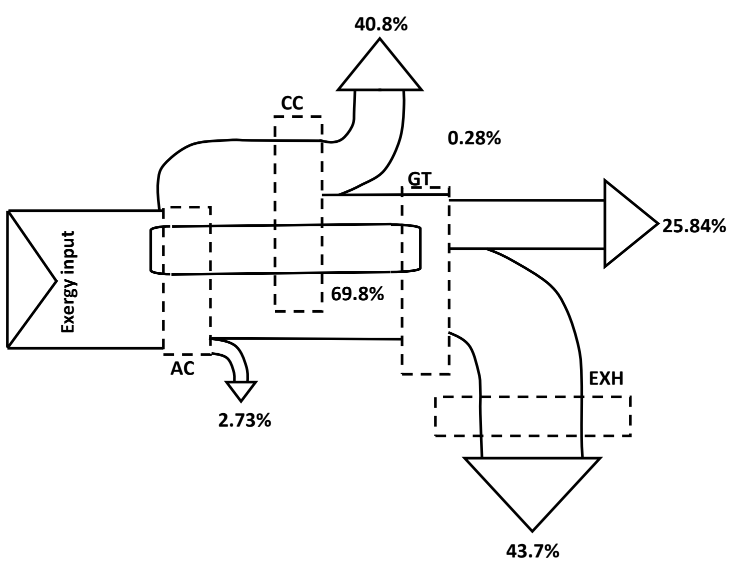

Figure 3- Grassmann diagram for the simple Brayton cycle

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The simulation of the performance of plant and components was done by varying the air inlet temperature: $15-45°C$; and the turbine inlet temperature: $1087-1800°K$, respectively. The computer program for the simulation under the conditions stated above and the results of the simulation are presented in Appendix A. Figure 4 compares the second-law efficiencies of the air compressor, combustion chamber, gas turbine and the overall plant when the ambient temperature increases. The second-law efficiencies of the plant and the combustion chamber are found to decrease more significantly with increase in the ambient temperature than the air compressor and the gas turbine, as shown in the figure. The simulation result reveals a 3.5%, 8.4%, 1.2%, and 0.07% decrease in the efficiencies of the plant, combustion chamber, air compressor, and gas turbine, respectively, for a 66% increase in the ambient temperature.

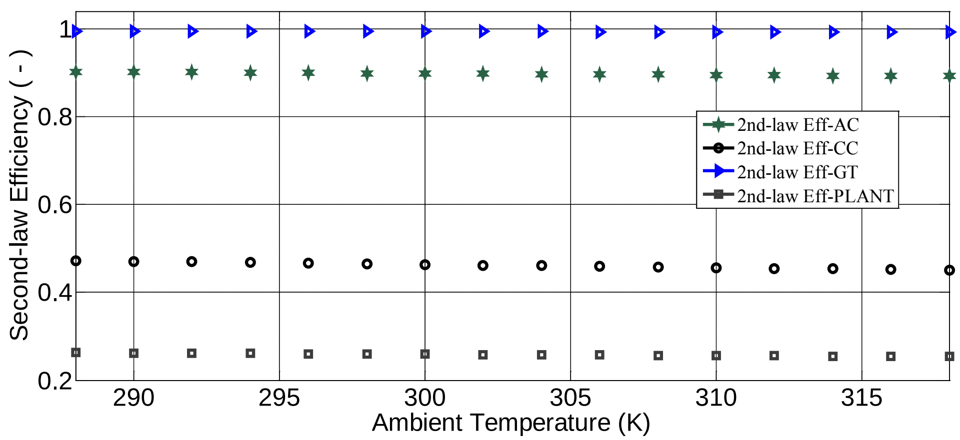

Figure4- Change in second-law efficiency with Ambient Temperature Variation

The second-law efficiency of the plant is also found to depend significantly on a change in turbine inlet temperature. Figure 5 shows that the second-law efficiency of the plant increases steadily as the turbine inlet temperature increases. The simulation result shows that the second-law efficiency of the power plant increases by about 24% for a 39% increase in the turbine inlet temperature.

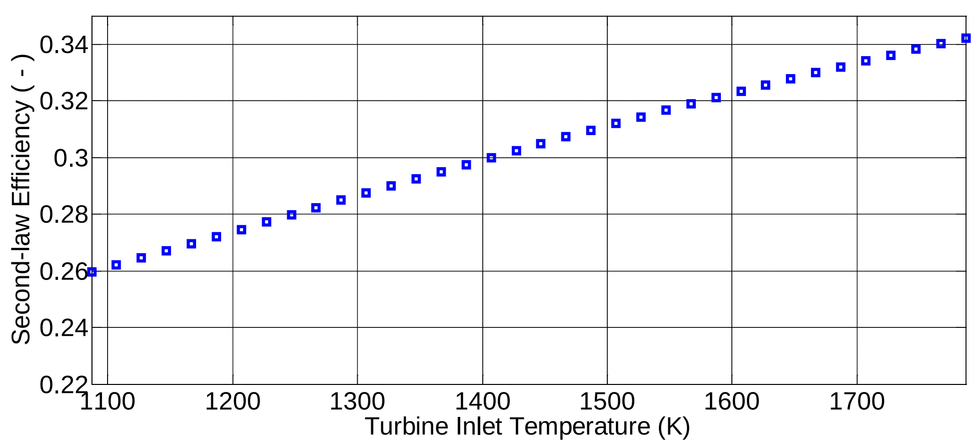

Figure5- Change in Second –Law Efficiency of Plant with Turbine Inlet Temperature Variation

The specific fuel consumption and power-to-heat ratio of the gas turbine plant were also found to change significantly with the turbine inlet temperature variation. Figure 6 shows that the power-to-heat ratio increases steadily with increase in the turbine inlet temperature. On the other hand, the specific fuel consumption decreases with increase in turbine inlet temperature. Hence, fuel energy is saved and power output from the plant enhanced as the turbine inlet temperature is increased.

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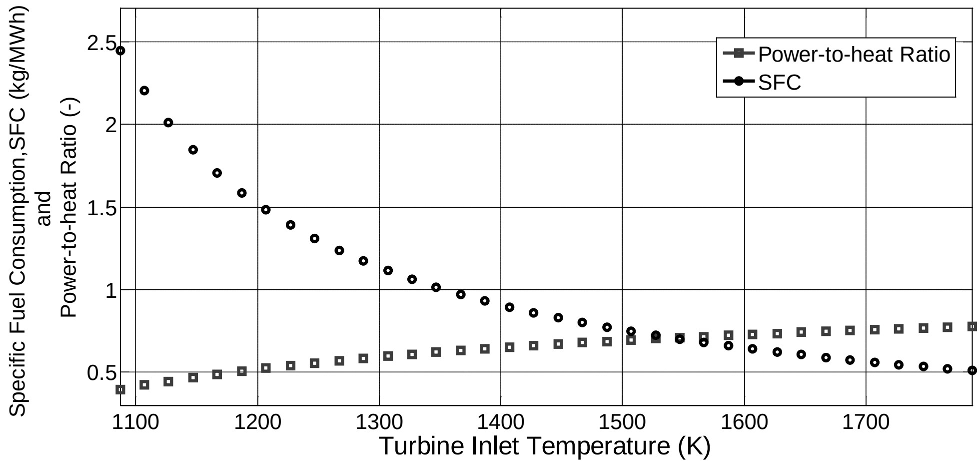

Figure6- Change in Specific Fuel Consumption and Power-to-Heat Ratio with Turbine Inlet Temperature variation

IV. CONCLUSION

An exergy analysis of a power generation gas turbine plant has been done. Exergy balance applied to each of the major components of the plant and to the overall plant reveals the amount of the total exergy generation and exergy destruction in the plant. The results from the gas turbine plant simulation reveal that the exergy destruction, exergy efficiency, exergy flow rate of the power output, power-to-heat ratio and the specific fuel consumption depend on ambient temperature and turbine inlet temperature.

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