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## The Effect of Ambient Temperature on Components Performance of an In-service Gas Turbine Plant using Exergy Method

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### ABSTRACT

Exergy analysis has been applied to 138 MW in-service gas turbine plant with data obtained from plant operation. The research investigated the effect of ambient temperature on the performance of the components system of the gas turbine plant using the exergy concept, to identify components that offer significant work potential saving opportunity. The fuel used is natural gas of low calorific value of 39.4 MJ m<sup>-3</sup>. The irreversibility rate of system components, exergy efficiency and the efficiency defects were determined for each component and for the entire plant at different temperatures. The results show a moderate increase of the total irreversibility rate of plant from approximately 415 to 421 KW while the rational efficiency decreased from 16.53 to 15.44% at same temperature range of 295 to 307 K. Furthermore, a 1°C increase in ambient temperature lead to 0.43 MW increase in the total irreversibility rate and 0.3% decrease in rational efficiency. The overall breakdown of plant, efficiency defect for ambient temperature range (295-307 K) stood at 83.475-84.56, 65.06-65.18, 3.23-3.33, 8.19-8.99 and 7.0-7.06% for combustion chamber, compressor, turbine and exhaust manifold, respectively. The exergy destruction rate in the combustion chamber and the turbine was about 36.4 and 5.4%, respectively. Our computation shows exergy efficiency value of 45.43 and 68.4% for both the combustion chamber and the turbine at 14°C increase in temperature. Overall suggestions for plant improvement are made as results show that the combustion chamber and turbine are the main sources of irreversibilities in the plant.

**Key words:** Irreversibility, efficiency defect, energy, exergy efficiency, rational efficiency, gas turbine

### INTRODUCTION

The growing interests in energy efficiency and conservation have led to the development of advanced techniques for analysis of complex energy systems based on the second law of thermodynamics. One such performance analysis is based on the concept of exergy. An exergy-based system analysis is one that overcomes the limitations inherent in an energy-based analysis. Exergy is not conserved as energy but destroyed in the system, hence an exergy analysis assessing the degree of exergy destruction, identifies the location, magnitude and sources of system inefficiencies, (Flavio and Segioil, 2000; Zhang *et al.*, 2000; Ebadi and Gorji-Bandpy, 2005; Rosen, 2001; Rosen and Dincer, 2004). Exergetic analysis allows thermodynamic evaluation of energy

conservation, because it provides the tool for a clear distinction between energy losses to the environment and internal irreversibilities in the process (Mehmet and Ayhan, 2007; Prasada *et al.*, 1995; Ahmet *et al.*, 2006). This advantage gives information about the possibilities of improving thermal processes and quantifies irreversibilities in system components and their contribution towards the gross irreversibility of the entire plant. This gives opportunity for economic consideration and plant modernization for performance improvement.

The gas turbine plant used as a case study for this research is located in the city of Afam in Nigeria. The Afam thermal plant is the major thermal plant contributing substantially to the national grid-system and is regarded as the backbone in the National energy utility sector. A comprehensive exergy analysis to locate and identify components that offer significant work potential saving opportunity is important. This in-depth analysis was lacking in the work of Ofodu and Abam (2001) in which components efficiencies and the effect of intensive properties of the environment to irreversible losses were not properly identified, thus making it difficult to arrive at proper thermal suggestions for system improvement. In this study the effect of ambient temperature on irreversible losses in system components and the effect of turbine inlet temperature on exergy destruction rate on the entire plant are clearly identified. Mass and energy conservation laws were applied to system components and exergy balances for each component of the plant was presented.

## **BACKGROUND OF AFAM POWER STATION AND DATA COLLECTION**

The Afam power station is located in Okoloma village in the oil rich Ndoki clan of Oyigbo in Rivers State of Nigeria. The Afam generating station is owned by the Power Holding Company of Nigeria Plc (PHCN). It has the biggest gas turbine plants in the country and is recently known as Afam Electric Power Business Unit (AEPBU). Afam power station consists of (20) gas turbines installed in five phases in sequence with the power demand increase. Afam II has 4 units BBC gas turbines, type 9C and total capacity of 95 MW. Afam III has 4 units BBC turbines, type 9C total capacity 109 MW. Afam IV has 6 units BBC turbine, type 13D and total capacity of 450 MW. Afam V has 2 units Siemens V94.2 and a total capacity of 276 MW. The turbine used for this research is GT 20 in Afam V with capacity of 138MW which was in-service at the time of this research from 2007 to 2009. During this period operational data were monitored and the daily monthly average data determined. Other data were obtained from operational log sheet in the previous year of 2006. The ISO or design data for Siemens V94.2 gas turbine plant used in this research are summarised in Table 1.

## **DESCRIPTION OF GAS TURBINE PLANT**

The schematic of the 138 MW gas turbine system is shown in Fig. 1. The system consists of an Air-Compressor (AC), Combustion Chamber (CC) and the Turbine (GT). An exergy stream from the atmosphere in state 1 with ambient properties  $T_0$  and  $P_0$  enters the system through the axial flow compressor. Exergy in the form of mechanical work is supplied to the compressor through the turbine shaft. Due to irreversibilities, part of the exergy amount in the compressor is consumed in the process of work addition to the working fluid. The working fluid enters the combustion chamber at very high pressure delivery exergy. In the combustion chamber the fuel is another contributor of exergy.

The atomization of the fuel under high turbulence results into complex chemical process of the combustion chamber. The resultant effect of this is the flow of an exergy stream of hot gases under

Table 1: Gas turbine design parameters

Design variable	Parameter	Units	Value
Compressor	Inlet temperature $T_1$	K	298.650
	Outlet temperature $T_2$	K	601.150
	Inlet pressure $P_1$	Bar	1.013
	Outlet pressure $P_2$	Bar	9.600
	Number of stages		16.000
	Speed	rpm	3000.000
Combustion chamber	Mass flow rate of air	kg sec <sup>-1</sup>	470.000
	Inlet temperature $T_2$	K	601.150
	Inlet mass flow rate of natural gas	kg sec <sup>-1</sup>	477.000
	Maximum temperature $T_3$	K	1330.150
	Outlet mass flow rate	kg s <sup>-1</sup>	9.300
	Inlet pressure	Bar	9.600
Turbine	Inlet temperature $T_3$	K	1330.150
	Outlet temperature	K	807.900
	Number of stages		5.000
	Inlet pressure	Bar	9.600
	Inlet mass flow rate	kg sec <sup>-1</sup>	477.000
	Outlet flow rate	kg sec <sup>-1</sup>	477.000

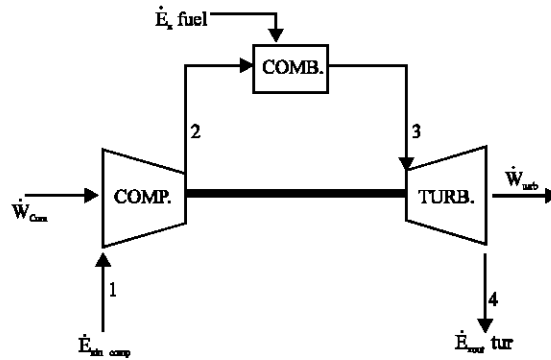


Fig. 1: Work and exergy flow in gas turbine system

a very high pressure and temperature into the gas turbine. Again part of the exergy is destroyed in the turbine due to irreversibilities and finally the balance is partly utilized in driving the compressor, while the rest becomes available as the network produced by the turbine. The exhaust combustion gasses still contain a measure of exergy flow as they emerge from the turbine exhaust hoods. For Afam power plant the stream of exergy flow in the exhaust gases is wasted, since there is no heat recovery system.

### EXERGY AND IRREVERSIBILITY ANALYSIS

The exergy balance equations and associated irreversibilities for various components system of the gas turbine plant are computed on the following assumptions of adiabatic conditions: (1) the combustion process is complete, (2) all components have adiabatic boundaries, (3) the mass flow of gas is constant throughout the cycle, (4) the exergy value of air entering the compressor is assumed zero since air is drawn from the atmosphere which is the reference state, (5) the reference state of

air is at a temperature of 25°C and 1 atmosphere and (6) the air and the combustion products are assumed ideal characteristics.

The general exergy-balance equation applicable to any component of a thermal system from Kwon *et al.* (2001) is given by:

$$\sum \dot{W} = \sum \dot{Q} + \sum_{in} \dot{E}_x - \sum_{out} \dot{E}_x - \dot{E}_{xloss} \quad (1)$$

The term  $W$  and  $Q$  denote work and heat interactions in the system and  $E_{xloss}$  is the lost work due to internal irreversibility which is give by:

$$To\Delta S \quad (2)$$

Applying Eq. 1 to the components systems of the gas turbine and considering each of the components as a control volume the analysis is carried out as follows:

**Compression process:**

$$-\dot{W}_C = \sum_{in} \dot{E}_x - \sum_{out} \dot{E}_x - E_{xloss} \quad (3)$$

$$\therefore \sum_{in} \dot{E}_x + \dot{W}_C = \sum_{out} \dot{E}_x + E_{xloss} \quad (4)$$

$\dot{E}_{xin}$  = Exergy of air  $\dot{E}_{xa}$

$E_{xout}$  = exergy out flow from the air compressor  $E_{xc}$ , for a mass flow rate to the combustion chamber:

$$\dot{E}_{xc} = \dot{m}_a (h_2 - h_1) - T_0 \dot{m}_a (s_2 - s_1) \quad (5)$$

$$h_2 - h_1 = \int_{T_1}^{T_2} c_p dT \quad (6)$$

$$\text{and } s_2 - s_1 = c_p \ln \left( \frac{T_2}{T_1} \right) - R \ln \left( \frac{P_2}{P_1} \right) \quad (7)$$

$$T_1 = T_0, P_1 = P_0$$

The total irreversibility in the compressor is made up of two components mechanical irreversibility and process irreversibility (Kotas, 1995):

$$\text{Mechanical irre. } Imc = \left( \frac{1}{\eta_c} - 1 \right) P_{2-1} \quad (8)$$

$$\text{Process irreversibility } I_{\text{Pirr}} = \dot{W}_c - [\dot{E}_x c + \text{Imc}] \quad (9)$$

Where:

$P_{2-1}$  = Internal power requirement of the compressor for a given mass flow rate

$\eta_c$  = Isentropic efficiency of compressor

$W_c$  = Gross power input to the compressor

$$P_{2-1} = m_a (h_2 - h_1) \quad (10)$$

$$\dot{W}_c = P_{2-1} + \text{Imc} \quad (11)$$

The total irreversibility in the compressor or exergy loss is given as follows:

$$I_{\text{CT}} = \dot{E}_x \text{loss} = \text{Imc} + I_{\text{Pirr}} \quad (12)$$

$$\text{Efficiency defect } = \delta = \frac{\text{Rate of exergy destruction}}{\text{Exergy rate of fuel}} \quad (13)$$

Exergy efficiency of compressor is given as:

$$\eta_{\text{ex comp}} = \frac{\dot{E}_{\text{zcomp}}^{\text{out}} - \dot{E}_{\text{zcomp}}^{\text{in}}}{\dot{W}_{\text{comp}}} \quad (14)$$

**Combustion process:** Applying the exergy balance equation in Eq. 1:

$$E_s = \sum_i (\dot{E}_{\text{xf}})_i + \sum_{\text{in}} \dot{E}_x c - \sum_{\text{out}} \dot{E}_x c c - \dot{E}_x \text{loss} \quad (15)$$

$E_s$  is shaft work and  $\dot{E}_{\text{xf}}$  is exergy of fuel.

The exergy of fuel is a combination of physical ( $E_{\text{xph}}$ ) and chemical ( $E_{\text{xch}}$ ) exergy.

$$\dot{E}_{\text{xph}} = \dot{m} c_p^{-h} (T - T_0) - \dot{m} T_0 \left\{ c_p^{-s} \ln \left( \frac{T}{T_0} \right) - R \ln \left( \frac{P}{P_0} \right) \right\} \quad (16)$$

Where,

$$c_p^{-h} = \frac{1}{T - T_0} \int_{T_0}^T c_p^- dT \text{ and } c_p^{-s} = \frac{1}{\ln \left( \frac{T}{T_0} \right)} \int_{T_0}^T c_p^- dT \quad (17)$$

$c_p^{-h}$  and  $c_p^{-s}$  are mean molar isobaric exergy capacity for evaluating enthalpy and entropy changes  $c_p^-$  in Eq. 17 is expressed or obtained in a polynomial form as:

$$c_p^- = a + b^2T + cT^2 + dT^3 \quad (18)$$

The values of a, b, c and d are constant characteristics of the gas considered and obtained from standard tables.

$$\dot{E}_{xch} = \sum_i x_i \dot{E}_x^{chem} + RT_0 \sum_i x_i \ln x_i \quad (19)$$

where,  $x_i$  = Mole fraction of constituent.

$$\dot{E}_{xf} = \dot{E}_{xph} + \dot{E}_x^{che.} = c_p \left[ (T - T_0) - T_0 \ln \left( \frac{T}{T_0} \right) \right] + RT_0 \ln \left( \frac{P}{P_0} \right) + \sum_i x_i \dot{E}_x^{chem.} + RT_0 \sum_i \ln \gamma_i x_i \quad (20)$$

The exergy loss or irreversibility in the combustion chamber is obtained from (Nag, 2003):

$$\dot{E}_x^{loss} = I_{cc} = T_0 \left[ (s_p)_3 - (s_R)_2 \right] \quad (21)$$

$$(s_R)_2 = (s_A)_2 + (s_F)_0 \quad (22)$$

The subscripts P, R, A and F represents, product, reactants, air and fuel and  $s_P$ ,  $s_R$ ,  $s_A$ ,  $s_F$  are entropies respectively. Substituting Eq. 22 in Eq. 21 gives:

$$I_{cc} = T_0 \left\{ \left[ (s_P)_3 - (s_P)_0 \right] + (s_P)_0 - \left[ (s_A)_2 - (s_A)_0 + (s_F)_0 + (s_A)_0 \right] \right\} \quad (23)$$

$$\Delta s_0 = (s_P)_0 - \left[ (s_F)_0 + (s_A)_0 \right] \quad (24)$$

Substituting Eq. 24 in Eq. 23:

$$I_{cc} = T_0 \left\{ \left[ (s_P)_3 - (s_P)_0 \right] - \left[ (s_A)_2 - (s_A)_0 \right] + \Delta s_0 \right\} \quad (25)$$

Applying ideal gas equation:

$$I_{cc} = T_0 \left\{ \left[ \dot{m}_g c_{pg} \ln \left( \frac{T_3}{T_0} \right) - \dot{m}_g R_g \ln \left( \frac{P_3}{P_0} \right) - \dot{m}_a c_{pa} \ln \left( \frac{T_2}{T_0} \right) - \dot{m}_a R_a \ln \left( \frac{P_2}{P_0} \right) \right] + \Delta s_0 \right\} \quad (26)$$

The rate of exergy loss is related with the Gibbs function and enthalpy of formation as:

$$T(\Delta s)_0 = \Delta G_0 - \Delta H_0 \quad (27)$$

and

$$\frac{\Delta G_0}{\Delta H_0} = 1.0401 + 0.1728\beta \quad (28)$$

$$\Delta H_0 = mf \times CV \quad (29)$$

where,  $\beta$  is the mass ratio of hydrogen to carbon in the fuel and mf and CV are mass flow rate of fuel and calorific value of natural gas. The exergy of combustion chamber which is the exergy out Excc can be evaluated by direct substitution in Eq. 15. Thus, combustion chamber exergy efficiency is evaluated as:

$$\eta_{\text{ex combt}} = \frac{\dot{E}_{x \text{ combt}}^{\text{out}}}{\dot{E}_{x \text{ combt}}^{\text{in}} + \dot{E}_{x \text{ Fuel}}} \quad (30)$$

**Expansion process:** The exergy balance equation in the turbine is as follows:

$$\dot{E}_{x \text{ cc}} = \dot{E}_{x \text{ T}} + \dot{W}_{\text{T}} + \dot{E}_{x \text{ loss}} \quad (31)$$

- $\dot{E}_{x \text{ cc}}$  = Exergy inflow to the turbine
- $\dot{E}_{x \text{ T}}$  = Exergy outflow to the turbine
- $\dot{W}_{\text{T}}$  = Power output from the turbine
- $\dot{E}_{x \text{ loss}}$  = R rate of exergy loss in the turbine

The exergy inflow into the turbine is evaluated from Eq. 31. The internal power generated in the turbine is given by:

$$P_{\text{T3-4}} = \dot{m}_g (h_3 - h_4) \quad (32)$$

$$h_3 - h_4 = \int_{T_4}^{T_3} c_p dT \quad (33)$$

The mechanical irreversibility in the turbine is given by:

$$I_{\text{mT}} = (1 - \eta_t) P_{\text{T3-4}} \quad (34)$$

Thus, the effective power output from the turbine is:

$$\dot{W}_{\text{T}} = P_{\text{T3-4}} - I_{\text{mT}} \quad (35)$$

Exergy output from the turbine is gotten as:

$$\dot{E}_{x \text{ T}} = \dot{m}_g c_{p g} (T_4 - T_3) - \dot{m}_g T_0 \left[ c_p \ln \left( \frac{T_4}{T_3} \right) - R_g \ln \left( \frac{P_4}{P_3} \right) \right] \quad (36)$$

The rate of exergy dissipation or irreversibility in the turbine is given as:

$$I_{\text{GT}} = \dot{m}_g T_0 (s_4 - s_3) = \dot{m}_g T_0 \left[ c_{p g} \ln \left( \frac{T_4}{T_3} \right) - R_g \ln \left( \frac{P_4}{P_3} \right) \right] \quad (37)$$



The rate of exergy loss due to exhaust flue gases is given as:

$$I_{\text{exhaust}} = \int_{T_4}^{T_0} \left( \frac{T - T_0}{T} \right) dQ = \dot{m}_g c_{pg} [T_4 - T_0] - \ln T_0 \left( \frac{T_4}{T_0} \right) \quad (38)$$

$$\text{Turbine exergy efficiency} = \eta_{\text{ex turbine}} = \frac{\dot{W}_{\text{turbine}}}{\dot{E}_{x \text{ turb}}^{\text{in}} - \dot{E}_{x \text{ turb}}^{\text{out}}} \quad (39)$$

The total exergy destruction or overall exergy loss in the plant is given as:

$$\dot{E}_x \text{ loss pt} = \dot{E}_{\text{comp}}^{\text{in}} + \dot{E}_{\text{Fuel}} - \dot{E}_{\text{turb}}^{\text{out}} - (\dot{W}_{\text{turb}} - \dot{W}_{\text{comp}}) \quad (40)$$

Efficiency defect of the plant is the summation of the efficiency defects of all the system components given as:

$$\delta_{\text{pl}} = \sum_{i=1}^n \delta_i \quad (41)$$

$$\delta = \frac{I}{E_x \text{ fuel}} \quad (42)$$

Where:

$I$  = Irreversibility rate

$E_x \text{ fuel}$  = Exergy of fuel

**Fuel utilization efficiency:** The fuel utilization efficiency is the ratio of all the useful energy extracted from the system to the energy of the fuel input.

$$\eta_{\text{fuel}} = \frac{\text{Useful energy}}{E_{\text{Fuel}}} \quad (43)$$

The energy of the fuel input is given as:

$$E_{\text{fuel}} = \dot{m}_g c_{pg} (T_3 - T_2^1) \quad (44)$$

**Rational efficiency:** The rational efficiency is also called second law efficiency.

It is given as:

$$\varphi = 1 - \sum_{i=1}^n \delta_i \quad (45)$$

Exergy of fuel is obtained in Eq. 19.

**RESULTS AND DISCUSSION**

The design variables or ISO data for the 138 MW Siemens V94.2 gas turbine plant is summarised in Table 1. The composition of fuel (natural gas) used is also summarised in Table 2, with the approximate values of composition in mole percent and their standard exergy in KJ Kmol<sup>-1</sup>.

Figure 2 shows the irreversibility rates of the system components: compressor, turbine, exhaust manifold (diffuser) plotted against the ambient temperature. The irreversibility rate of compressor increases from 16090 to 1650 KW, combustion chamber from 324030 to 324620 KW, turbine from 40770 to 44 810 KW, diffuser from 34871 to 35182 KW and the total irreversibility rate of plant increases from 415766 to 42113 KW for 14°C increase in temperature. The result shows a high exergy destruction rate in the combustion chamber. However, a similar result is obtained in the work of Ahmet *et al.* (2006), which recorded large value of irreversibility in the combustion chamber for combine-cycle power plant.

Figure 3 depicts the percentage efficiency defect plotted against ambient temperature.

The percentage defects for compressor, combustion chamber, turbine and diffuser increased from 3.23 to 3.33, 65.06 to 65.18, 8.19 to 8.99 and 7.00 to 7.06, respectively. While the overall rational efficiency decreased from 16.53 to 15.44 for 14°C increase in ambient temperature. The variation of exergy efficiencies of components system of gas turbine with ambient temperature is

Table 2: Composition of natural gas as it applies to Afam gas turbine plant

Specie	Formula	Mole (%)	Standard chemical exergy (KJ Kmol <sup>-1</sup> )
Methane	CH <sub>4</sub>	88.0	836510
Ethane	C <sub>2</sub> H <sub>6</sub>	5.0	1504360
Propane	C <sub>3</sub> H <sub>8</sub>	2.0	2163190
Butane	C <sub>4</sub> H <sub>10</sub>	0.6	2818930
Pentane	C <sub>5</sub> H <sub>12</sub>	N	3455800
Carbon dioxide	CO <sub>2</sub>	4.4	20140

Ofodu and Abam (2001)

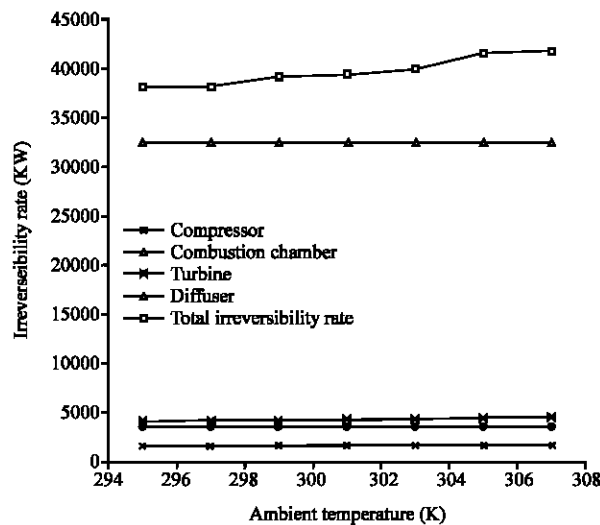


Fig. 2: Irreversibility rates of components versus ambient temperature

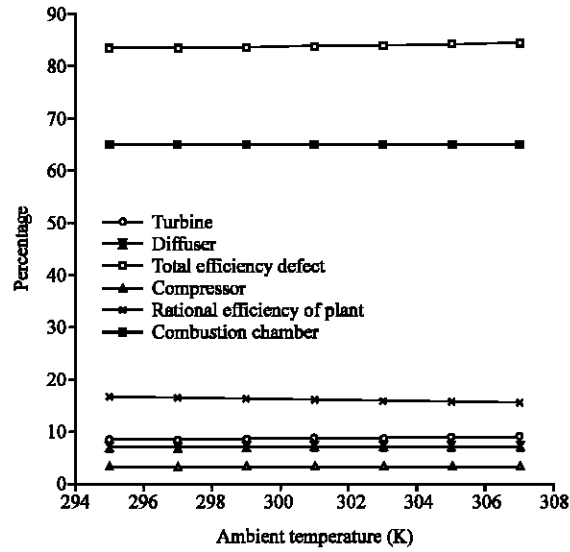


Fig. 3: Efficiency defects of components of power plant versus ambient temperature

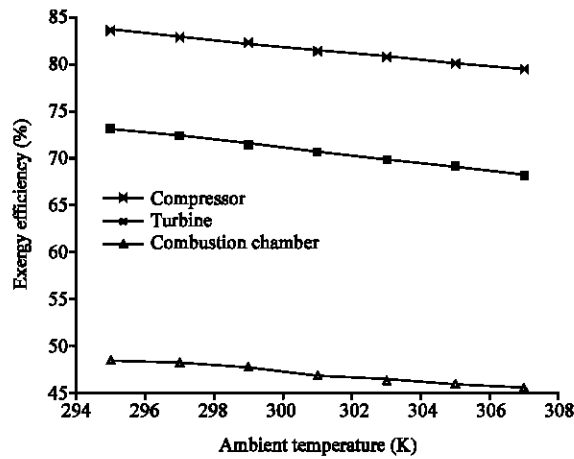


Fig. 4: Exergy efficiencies of the components versus ambient temperature

shown in Fig. 4. The exergy efficiency of compressor, combustion chamber and turbine decreased from 83.80 to 79.60, 48.60 to 45.43 and from 73.3 to 68.40, respectively. While the total irreversibility rate of the plant increased approximately from 415 to 421 MW and the rational efficiency decreased from 16.53 to 15.44%. It was observed that a 1°C increase in ambient temperature leads to a corresponding increase of 0.43 MW in the total irreversibility rate and 0.3% decrease in the rational efficiency of the plant.

Figure 5 shows the effect of 120 and 140% increase in turbine inlet temperature on plant rational efficiency and exergy destruction rate. An increase in the Turbine Inlet Temperature (TIT) caused a corresponding increase in the rational efficiency and decrease in exergy destruction rate. The change in exergy efficiency of system components of the plant for the same increase in TIT was

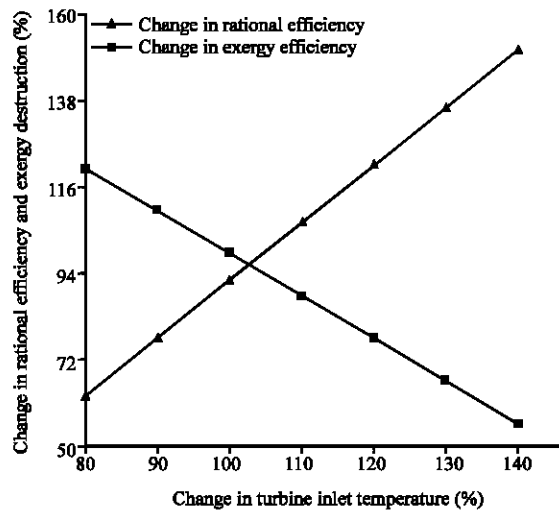


Fig. 5: The effect of change in turbine inlet temperature on the rational efficiency and exergy destruction in the plant

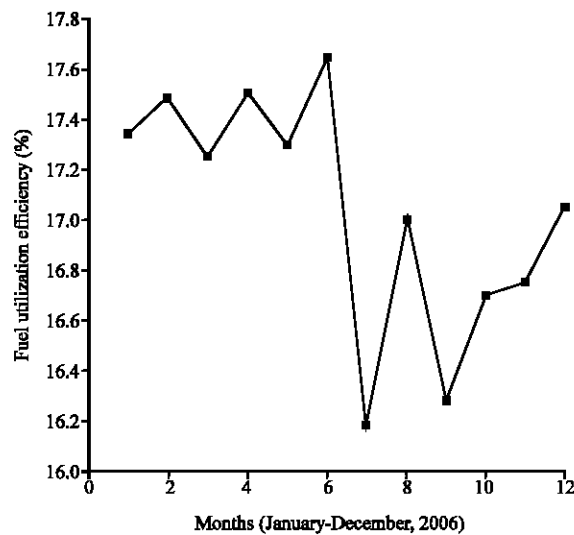


Fig. 6: Fuel utilization efficiency at full load against months of 2006

also found to cause a slight increase in the exergy efficiency of turbine while the exergy efficiency of compressor remains constant, but a good increase in the exergy efficiency of combustion chamber was observed. Also the amount of exergy destruction for 120 and 140% increase in TIT caused a decrease in the exergy destruction in the combustion chamber and the operation of the compressor was not affected. However, because of the large irreversibility in the combustion chamber, the overall exergy destruction in the plant decreases approximately of about 21.5%. It can be seen that the greatest exergy loss occur in the combustion chamber. The former findings are in concurrence with that obtained by Fiaschi and Manfrida (1998) and Habib *et al.* (1995).

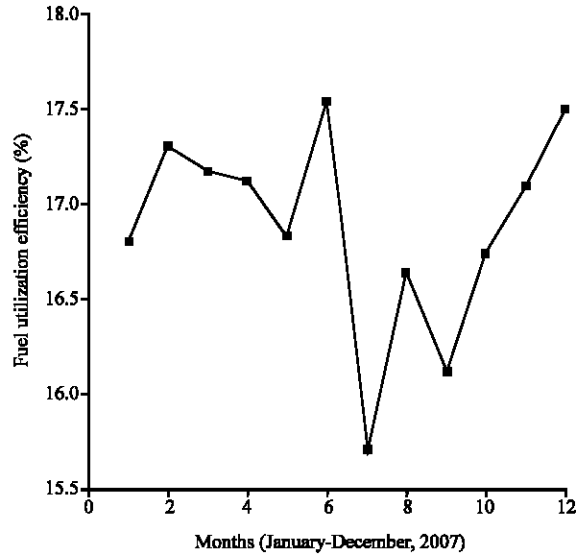


Fig. 7: Fuel utilization efficiency at full load against months of 2007

The large exergy loss in the combustion chamber is associated with large temperature variation between the flame and the combustion fuel. A reduction in this temperature difference will cause a corresponding increase in turbine inlet temperature which will reduce exergy loss. The research shows that an increase in TIT above 100% will improve efficiency of the gas turbine plant between 2-6%. These conclusions agree with the findings reported by Mahmoudi *et al.* (2009) and Ebadi and Gorji-Bandpy (2005). However, a higher value of TIT is required with technical considerations because of metallurgical limits of turbine blades material as presented by Kumar and Kale (2002).

Figure 6 shows the variation of fuel utilization efficiency at full load for the year 2006 over the months of the respective year from January to December (1-12). The fuel utilization efficiency fluctuates over the months and the highest value is obtained at the month of June with 17.69%. Figure 7 shows for 2007 with highest value of about 17.51%. Figure 8 and 9 show the fuel utilization efficiency for 2008 and 2009 with maximum values of 17.7 and 17.59%, respectively. However, there is a fluctuation in the fuel utilization efficiency across the months which must have been influenced by the variation in intensive properties of the environment and large irreversibilities in the system components. However, calculations show that the fuel utilization efficiency and second law efficiencies are higher at full load condition as compared to part load.

The fuel utilization efficiency stood at about 18.6 and 14.4% at both full and part load respectively, while the second law efficiency was found to be about 16% and 11.5% at both full and part load conditions. In a similar work conducted by Prasada *et al.* (1995) in a cogeneration plant a higher value of fuel utilization efficiency of about 82 and 79.37% was obtained for both full and part load condition. And a second law efficiency of 33 and 16% was obtained for both full and part load conditions. This shows that the efficiency of a cogeneration plant is higher than a simple gas turbine plant that operates at the same capacity. This variation can be compensated for by modification of Afam power plant.

Between the periods 2006-2009 about 45% of the total generated power yearly is lost.

This poor performance of the system is attributed to poor maintenance procedures. Investigation has revealed that since the installation of the plant about 10 years ago, only the Combustion

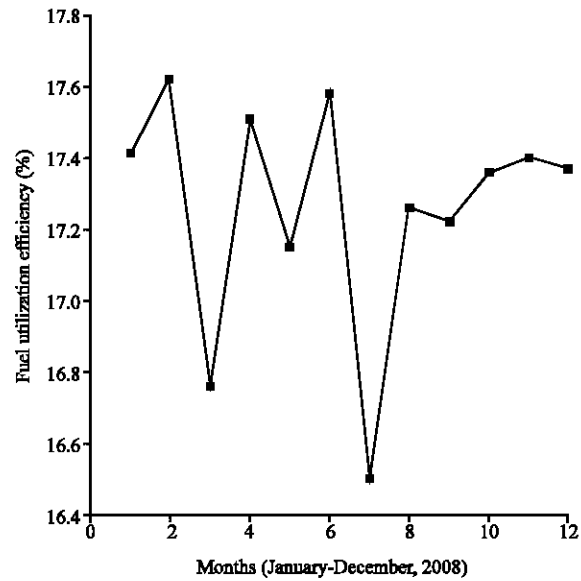


Fig. 8: Fuel utilization efficiency at full load against months of 2008

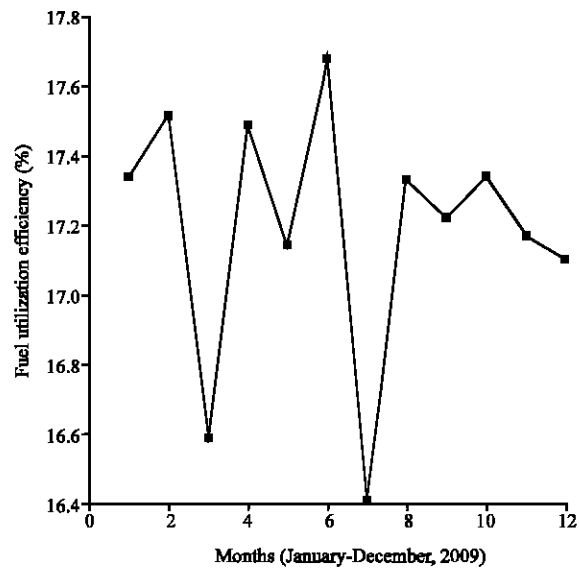


Fig. 9: Fuel utilization efficiency at full load against months of 2009

Inspection (CI) maintenance has been taking place and it is usually not regular. Up till the time of carrying out this study, none of the turbines has undergone any HGPI (Hot gas path inspection) or MI (Main inspection). It is therefore this factor that must have been responsible for the gradual thermal deterioration of the plant losses that eventually led to poor performance and occasional break down.

## CONCLUSIONS

A comprehensive study based on exergy analysis of Afam gas turbine plant and the effect of ambient temperature on performance parameters of the plant was presented. The overall exergy

loss (efficiency defect) is evaluated to be 83.47 and 84.56% for ambient temperature values 295-307 K. The irreversibility rates of the combustion chamber is approximately 35% at ambient temperature of 295 K and 36.4% at 307 K ambient temperature while the irreversibility rate for turbine is approximately 4.5 and 5.4% at 295 and 307 K, respectively. In the plant the combustion chamber is the highest exergy consumer, thus it offers the largest improvement potential. The chemical reaction between air and fuel in the combustion process is responsible for such large exergy destruction in the combustion chamber. The second biggest consumer in exergy distribution (efficiency defect) is the turbine with 8.19-8.99%, for 295 to 307 K ambient temperature.

The research shows that exergy efficiency and exergy destruction rate are affected by the ambient temperature and the turbine inlet temperature. Based on this analysis the components that should be considered for improvement in the plant are the combustion chamber and turbine unit. An air intake cooling system (IAC) can be used in Afam gas turbine plant to improve the condition of air entering the compressor. Different techniques for power boosting through IAC includes: evaporative cooling, mechanical chiller, absorption chiller and thermal energy storage. Finally, for maximum efficiency of plant a system to increase the Turbine Inlet Temperature (TIT) is proposed using special super alloys in gas turbine hot parts. However, for this plant in question, the most attractive cooling technique would be air inlet chilling, since this can be achieved without changing the present plant configuration while all aspect of chilling may be externally applied.

#### **NOMENCLATURE**

- $W_c$  = Compressor Work (KW)
- $W_t$  = Turbine Work (KW)
- $W_{net}$  = Net power (KW)
- $c_p$  = Specific heat capacity (KJ/KgK)
- $Q_{in}$  = Input heat (KW)
- $\dot{m}_a$  = Mass flow rate of air ( $kg\ sec^{-1}$ )
- $\Delta G_0$  = Gibbs function
- $\dot{m}_f$  = Mass flow of fuel ( $kg\ sec^{-1}$ )
- $c_p^{-h}$  = Mean isobaric enthalpy capacity (KJ/Kmol)
- $c_p^{-s}$  = Mean isobaric entropy capacity (KJ/Kmol.K)
- $h$  = Specific enthalpy ( $KJ\ kg^{-1}$ )
- $P$  = Pressure (kpa)
- $R$  = Universal gas constant (KJ/KmolK)
- $T_0$  = Temperature, ambient (K)
- ch = Chemical
- I = Irreversibility (KW)
- a = Air
- $\delta$  = Efficiency defect

#### **Subscripts:**

- f = Fuel
- t = Turbine
- g = Gas
- ph = Physical
- cc = Combustion chamber
- c = Compressor

**Greek symbols:**

- $\beta$  = Mass ratio of hydrogen to carbon  
 $\eta_c, \eta_T$  = Isentropic efficiency of compressor and turbine  
 $\varphi$  = Rational efficiency

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