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Exergy Analysis of Afam IV Gas Turbine Power Plant.

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ABSTRACT

In this study an exergy analysis of 75 MW gas turbine is carried out. Exergy analysis based on second law was applied to the gas cycle and individual components through an off design point modeling approach. The analysis shows that the highest exergy destruction occurs in the combustion chamber (CC), and the gas turbine is significantly affected by the ambient temperature; increase in temperature leads to decrease in GT power output. The compressor has the largest exergy efficiency of 99% as compared to the other components (combustion chamber – 76%, Turbine – 95%). The highest destruction in exergy was recorded in the combustion chamber. As a result of this destruction, 2.23kJ of energy is lost in every 1MW output of power produced by the plant.

Introduction

Energy is the capacity to do work. Any system/body above absolute zero condition possesses energy. However, this energy is not available for work if the systems temperature/pressure is below that of its environment. A departure in temperature between the system and its environment increases this available energy. As the systems energy is been converted to other forms, any irreversibility in the system will cause some of this available energy to become unavailable. The available energy is called exergy, while the unavailable energy is called anergy. From the 2nd law of thermodynamics, exergy could be conceptualized to mean the maximum amount of energy available / maximum work that can be done by a system existing at a given state, while anergy is the maximum amount of destroyed exergy due to systems irreversibility. Exergy analysis of a system helps quantify losses in a system and identify locations where these losses occur. Exergetic analysis also helps in identifying causes of irreversibilities in the plants, and is a more meaningful assessment of plant individual components efficiency [1]. Hence, the purpose of the study is to identify the irreversibilities in the system and quantify them. The study concluded that exergy is destroyed most in the combustion chamber due to increase in temperature difference between the inlet and outlet conditions which leads to the generation of a higher entropy value.

Research Elaborations

Afam IV power station was commissioned in 1982 it has six generating units (GT13 – GT18) with installed capacity of 75MW each. These turbines were built and installed by Brown Boveri. The gas turbine power plant is an open cycle single shaft system. The power plant uses natural gas of low heat value (LHV) in kJ/kg. The simplified schematic diagram of the plant is shown in Figure 1. The system consists of an air-compressor (C), combustion chamber (CC) and a gas turbine (T).

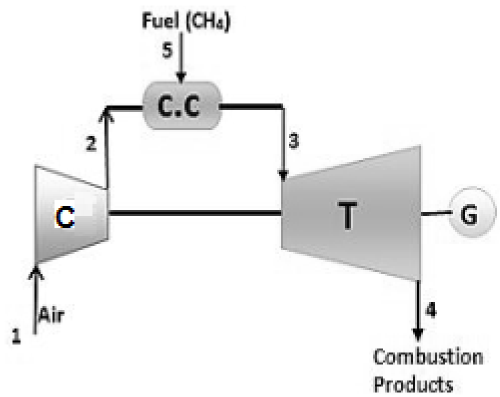

Figure 1.0. Schematics of a Gas Turbine Plant.

Data used in the study are those of GT18 gotten from Afam power station annual report 2005 to 2013 and is given in table 1.

Table 1. Technical data of GT18 gas turbine.

Parameter	Value
Ambient Temperature, T_1(K)	301.48
Compressor Outlet Temperature, T_2 (K)	595.82
Turbine inlet Temperature, T_3 (K)	1192.82
Turbine outlet Temperature, T_4 (K)	723.75
Temperature of exhaust gas, T_{exh} (K)	664.65
Compressor inlet Pressure, P_1 (bar)	1.013
Compressor outlet Pressure, P_2 (bar)	9.80
Pressure Ratio	9.67
Mass flow rate of fuel (kg/s)	6.40
Inlet mass flow rate of air (kg/s)	359.00
Power Output (MW)	58.00
LHV of fuel(kJ/kg)	48948.30
Pressure drop in combustion chamber	3%

The exergy analysis formulation in this study is developed using the methodology established by [1]-[5]. The exergy balance for a thermal system as stated by [1] is expressed as:

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$$\psi_W = \sum_{k=1}^{n}\left(1-\frac{T_o}{T_k}\right)Q_k + \sum_{k=1}^{r}[(\dot{m}\psi)_i - (\dot{m}\psi)_o] - T_o\dot{S}_{gen} \quad (1)$$

Where:

$\sum_{k=1}^{n}\left(1-\frac{T_o}{T_k}\right)Q_k$ represents the exergy summation supplied through heat transfer.

$\sum_{k=1}^{r}[(\dot{m}\psi)_i - (\dot{m}\psi)_o]$ represents the exergy summation of the working fluid.

$T_o\dot{S}_{gen}$ represents the irreversibility in the system.

ψ_W represents the useful work done on/by the system.

Exergy analysis on Air Compressor

Since there is no heat transfer, and work is done on the compressor by the turbine, equation 1 becomes

$$-\dot{W}_{ac} = \sum_{k=1}^{n}[(\dot{m}\psi)_i - (\dot{m}\psi)_o] - T_o\dot{S}_{gen} \quad (2)$$

The exergy of air at inlet and out let conditions can be quantified with the expression given by [2]

$$(\dot{m}\psi)_{i\ or\ 0} = \dot{m}_a(h_a - T_oS_a) \quad (3)$$

Where: h and S are the enthalpy and entropy of air at their corresponding inlet and outlet pressure and temperature conditions, and T_o is the ambient temperature.

[3] established the expression for the exergy loss (irreversibility/Anergy) in the system as:

$$I_{ac} = T_o\dot{S}_{gen} = \dot{m}_aT_o[S_2 - S_1] \quad (4)$$

$$S_2 - S_1 = C_{pa}\ln\left[\frac{T_2}{T_1}\right] - R_a\ln\left[\frac{P_2}{P_1}\right] \quad (5)$$

Where: $R_a = C_{pa}\frac{(\gamma_a-1)}{\gamma_a}$ [3]

(S_2-S_1) represents entropy change between compressor exit and inlet conditions.

γ_a= Specific heat ratio of air =1.4

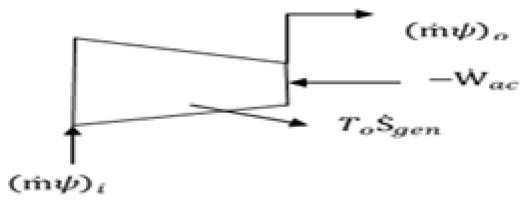

Figure 2.0. Exergy flow balance in the compressor.

Exergy Analysis on Combustion chamber

Exergy balance on the combustion chamber is carried out considering the figure 3.

At equilibrium,

$$\psi_{in} = \psi_{out} \quad (6)$$

$$\dot{m}_a\psi_a + \dot{m}_f\psi_f = \dot{m}_g\psi_g + I_c \quad (7)$$

Where ψ_a, ψ_f, ψ_g are exergy of air, fuel, and products respectively.

$\dot{m}_a\psi_a = (\dot{m}\psi)_O = -498832.44kJ$.

Expression for the exergy of fuel can be deduced from the methodology used by [2].

$$\dot{m}_f\psi_f = \dot{m}_f(h_f - T_oS_f) \quad (8)$$

Exergy destruction is expressed as;

$$I_{cc} = T_o\dot{S}_{gen} = \dot{m}_fT_o[S_3 - S_2] - \frac{Q_{cc}}{T_{ave}}\ ] \quad [4] \quad (9)$$

Where: $[S_3 - S_2] = C_{pg}\ln\left[\frac{T_3}{T_2}\right] - R_a\ln\left[\frac{P_3}{P_2}\right]$;

$Q_{cc} = C_{pg}[T_3 - T_2]$; $T_{ave} = \frac{T_3 + T_2}{2}$

Exergy Analysis on Turbine.

Considering the exergy flow diagram in figure 4, at equilibrium, exergy flow into the turbine equals exergy flow out of the turbine. Hence,

$$\dot{m}_g\psi_g = \dot{W}_{ac} + \dot{W}_u + \dot{m}_{exht}\psi_{exht} + I_{turb} \quad (10)$$

3] expressed the relationship between the mass flow rate of fuel and gaseous product to be;

$$\dot{m}_f = F \times \dot{m}_g \quad (11)$$

Where F represents flow of fuel = $\frac{C_{pg}T_3 - C_{pa}T_2}{LHV - C_{pa}T_2}$ (12)

C_{pg}=specific heat of gaseous products = 1.148kJ/kgK.

Hence, F = 0.016, $\dot{m}_g$=401.57 kg/s

$$\dot{m}_g\psi_g = \dot{m}_g\left[C_{pg}(T_3 - T_2) - T_o\left[C_{pg}\ln\left[\frac{T_3}{T_2}\right] - R_g\ln\left[\frac{P_3}{P_2}\right]\right]\right] \quad (13)$$

$R_g = C_{pg}\frac{(\gamma_g-1)}{\gamma_g}$ [3]

γ_g = specific heat ratio of gaseous products = 1.33

Substituting values into equation 13,

$\dot{m}_g\psi_g = 182781.01kJ.$

Similarly,

$$\dot{m}_{exht}\psi_{exht} = \dot{m}_g\left[C_{pg}(T_4 - T_{exht}) - T_o\left[C_{pg}\ln\left[\frac{T_4}{T_{exht}}\right] - R_g\ln\left[\frac{P_4}{P_3}\right]\right]\right] \quad (14)$$

$$\frac{P_4}{P_3} = \frac{1}{\left[\frac{T_3}{T_4}\right]^{\gamma-1/\gamma}} = 0.883, R = 0.2848$$

Where:

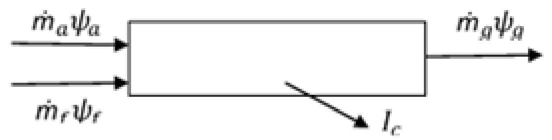

Figure 3. Exergy flow balance on combustion chamber.

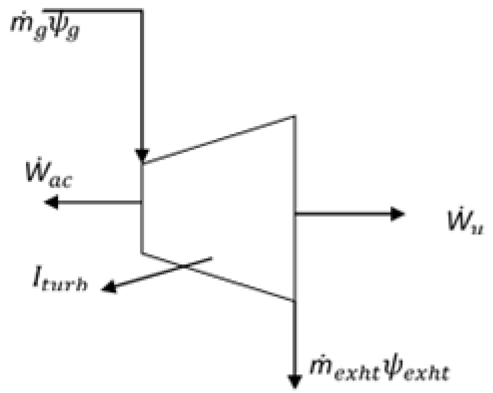

Figure 4. Exergy flow balance on turbine.

Results

Results on Compressor analysis.

Substituting values for $\dot{m}_a, T_o, and [S_2 - S_1]$ into equation 4, we get that:

$I_{ac} = 60.59kJ$

At inlet condition, P=9.8 bar; T=301.48K. From tables, the values for enthalpy and entropy of air is:

h= 301.95kJ/kg; S= 6.875kJ/kgK. Substituting into equation 3,

$(\dot{m}\psi)_i = -602790.16kJ$

At outlet condition, P=9.8 bar; T=595.82K. From tables, the values for enthalpy and entropy of air is :

h= 603.34kJ/kg; S= 6.916kJ/kgK. Substituting into equation 3,

$(\dot{m}\psi)_O = -498832.44kJ$

Substituting values of $(\dot{m}\psi)_0, (\dot{m}\psi)_i, and\ I_{ac}$ into equation 2,

$\dot{W}_{ac} = 104018.32kJ/s$

Therefore:

exergy added = $(\dot{m}\psi)_0 - (\dot{m}\psi)_i$

$= -498832.44 - (-602790.16) = 103957.72kJ$

Utilized exergy (Availability)= Exergy added – Exergy loss = 103897.13kJ

Exergetic efficiency of the compressor can be expressed as;

$$\eta_{ac} = 1 - \frac{I_{ac}}{W} = 1 - \frac{60.59}{104018} = 0.99$$

$$or \quad \eta_{ac} = \frac{utilized\ exergy}{exergy\ added} = 0.99$$

Results on Combustion Chamber.

From Methane tables; at 9.8 bar and 595.82K, h_f = 805.398kJ/kg and S_f = 0.6569kJ/kgK.

Substituting values of $\dot{m}_f, h_f, T_o and\ S_f$ into equation 8,

$\dot{m}_f\psi_f = 3943.12kJ$

Since there is a pressure drop of 3% in the combustion chamber, P_3= 9.506 bar. Substituting values of $\dot{m}_f, T_o, [S_3 - S_2], Q_{cc} and\ T_{ave}$ into equation 9;

The exergy destroyed, $I_{cc} = 120340.12kJ$

Therefore:

Exergy added = $\dot{m}_f\psi_f - \dot{m}_a\psi_a$

$= 3943.12kJ - (-498832.44kJ) = 502775.56kJ$

Utilized exergy (Availability) = exergy added – exergy destroyed = 382435.44kJ

Exergetic efficiency of the combustion chamber can be expressed as;

$$\eta_{cc} = 1 - \frac{I_{cc}}{Exergy\ added} = 1 - \frac{120340.12}{502775.56} = 0.76\ or\ \eta_{cc} =$$

Results on Turbine Analysis

Substituting values of $\dot{m}_g, C_{pg}, T_4, T_{exht}, T_o, R_g and\ \frac{P_4}{P_3}$ into equation 14, $\dot{m}_{exht}\psi_{exht} = 11830.10kJ$

Substituting values of $\dot{m}_g\psi_g, \dot{W}_c, \dot{W}_u, and\ \dot{m}_{exht}\psi_{exht}$ into equation 10, $Exergy\ loss, I_{turb} = 8932.58kJ$

Therefore:

Exergy added = $\dot{m}_g\psi_g - \dot{m}_{exht}\psi_{exht} = 182781.01kJ - 11830.10kJ = 170950.90kJ$

Utilized Exergy (Availability)= $\dot{W}_{turb} = \dot{W}_c + \dot{W}_u$ = 162018.32kJ

The exergetic efficiency of the turbine is expressed below.

$$\eta_{turb} = 1 - \frac{I_{turb}}{Exergy\ added} = 1 - \frac{8932.58}{170950.90} = 0.95$$

or

$$\eta_{turb} = \frac{utilized\ exergy}{exergy\ added} = 0.95$$

From figure 4, the exergetic efficiency can also be expressed as:

$$\eta_{turb} = \frac{output\ exergy}{input\ exergy} = \frac{\dot{m}_{exht}\psi_{exht} + \dot{W}_c + \dot{W}_u}{\dot{m}_g\psi_g} = 0.95$$

Total irreversibility of the system is given as:

$$I_{Total} = I_{ac} + I_{cc} + I_{turb} \qquad (15)$$

$I_{Total} = 129333.29kJ$

Exergetic performance coefficient of the power plant gives the amount of exergy lost per unit power output, and it is expressed as;

$$\xi = \frac{I_{Total}}{\dot{W}_u} = 2.23kJ/MW$$

This means that 2.23kJ of energy is lost in every 1MW output of power produced.

Exergetic efficiency of the power plant is expressed as;

$$\Psi = \frac{total\ availability\ across\ the\ subsystems}{total\ added\ exergy\ across\ the\ subsystems} = 0.834$$

OR

$$\Psi = 1 - \frac{total\ irreversibility\ in\ the\ system}{total\ added\ exergy\ across\ the\ subsystems} = 0.834$$

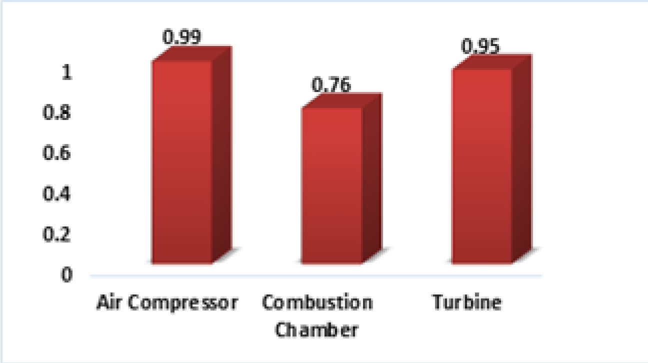

Figure 5.0. Exergetic efficiency Vs System components.

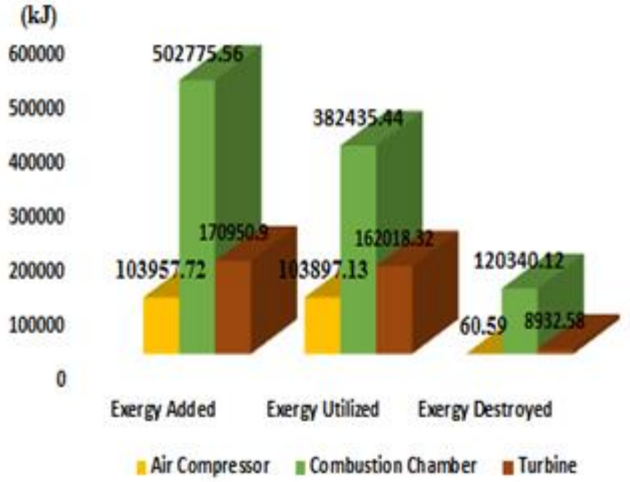

Figure 6.0. Exergy added, Exergy Utilized, and Exergy destroyed across subsystems.

Discussion

Air Compressor

From the analysis, we observe that the exergy destroyed is only 0.058% of the total exergy added to the system due to the change in exergy between air inlet and outlet conditions. This means that the air compressor is effective since it utilizes 99.9% of the supplied exergy. Exergy is least added and destroyed in the air compressor. This could be as a result of the temperature gradient across it which doesn't favour higher generation of entropy when compared to other components in the system.

Combustion Chamber.

The exergy study carried out on the combustion chamber shows that there is a significant increase in the loss of exergy when compared to that of the compressor.

Table 2 . Result of exergy analysis.

Component	Exergy Added (kJ)	Exergy Utilized (kJ)	Exergy Destroyed (kJ)	Exergetic Efficiency	Percentage Exergy Destroyed (%)
Air Compressor	103957.72	103897.13	60.59	0.99	0.058
Combustion Chamber	502775.56	382435.44	120340.12	0.76	23.94
Turbine	170950.90	162018.32	8932.58	0.95	5.23

Due to a larger departure in the exit and inlet temperatures, the exergy added was seen to be higher than its value in the air compressor which also led to an increase in the irreversibility recorded. The destroyed exergy is 23.94% of the supplied exergy leading to a lower value in the exergetic efficiency.

Turbine

The turbine recorded an exergetic efficiency of 95%, indicating that the irreversibility/exergy destruction is lower than that of the combustion chamber.

The plant exergetic efficiency connotes the real performance of the system. From the analysis carried out, this was found to be 83.4%. This means that only 16.6% of energy is destroyed within the system. From this, we can see that irreversibility which is dependent on entropy generation is also dependent on the prevailing temperature within the system. The higher the temperature, the higher the entropy generated, and subsequently, the irreversibility increases. This explains why the irreversibility is highest in the combustion chamber, followed by the turbine while in the air compressor where there is no heat addition/exhaustion the irreversibility is quite low when compared to other sub systems.

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