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## **ASSESSMENT OF SUSTAINABILITY INDICATORS OF TWO GAS- TURBINE PLANTS WITH NAPHTHA AND NAPHTHA-RFG MIXTURE AS FUELS**

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### **Abstract**

*To enhance sustainability of any energy system exergy based sustainability indicators (exergy efficiency, waste exergy ratio, environmental effect factor and exergetic sustainability index) are used. In the present paper sustainability aspects of two GT based power plant are carried out using sustainability indicators. For this purpose, two GT1) configurations, case A (Naphtha based GT power plant) and case B (Naphtha-Residual fuel gas mixture GT 2) are taken up as case study. Results show that exergetic sustainability index obtained as for case A is higher as compared to case B.*

### **Keywords**

Exergy, Naphtha, Residual Fuel Gas, Sustainability Analysis

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## **1. Introduction**

Driven by social change and environmental degradation, sustainability (Rosen and Dincer 2001) has become most talked topic among academicians, policy regulators and researchers in the last decade and it requires a sustainable supply of energy services without causing negative or less environmental impacts.

The basic ethics of modern life depends upon energy services and it becomes more evident that clean and climate-friendly energy technology must be deployed as rapidly as possible in order to have energy access for all while minimizing greenhouse gas (GHG) emissions. For this it specifically means to adopt various renewable source of energy like solar, wind, biomass etc. However, the fact must be acknowledging that an immediate shift to substantial use of renewable energy is not feasible. Besides the increased deployment of renewable energy; an efficient technology (Ibrahim et al. 2017) which can use energy resources more efficiently for power generation is of great relevance.

Based on new challenges in commercial applications of energy services, gas turbine (Ibrahim et al. 2017; Pattanayak 2015) has become a preferred choice because of high efficiency and small installation time. In fact, seeing the aspect of environment, economy and efficiency, gas turbine will play an important role in power generation market. However, the fact must be understood that utility of any energy system in connection to environment is best guided by thermodynamic principle particularly second law of thermodynamics (exergy). First law of thermodynamics talks about energy conservation does not provide us true magnitude of losses in energy system. But exergy analysis (Butcher and Reddy 2007; Kaushik, Reddy, and Tyagi 2011; Zhu, Deng, and Qu 2013) on the other side provides a true picture of true potential of energy system. It reveals the inefficient thermodynamic process and can be used to assess the sustainability of energy systems. Further there are exergy based sustainability indicators which is used to qualify whether as energy system is sustainable or not.

A detailed literature review has been performed on various energy systems for exergy based sustainability aspect. Exergetic sustainability analysis of LM6000 gas turbine power plant is carried out by plant (Aydin 2013) and reported decrease in waste exergy by employing steam turbine to GT. These exergetic performance parameters and sustainability metrics are also used in other energy system such as fuel cell (Midilli and Dincer 2009) , aviation sector (Aydin 2013; Aydin et al. 2015; Turan et al. 2014; Turan and Aydin 2016) etc.. However, there are many researchers which are working in this field, but few of them has been cited.

In the present context sustainability analysis of two gas turbine power plant (GT1 and GT2) has been taken up as case study for exergy based sustainability parameters. GT1 (case A) is being powered by Naphtha and GT2 (case B) is powered by Naphtha-Residual fuel gas mixture (RFG). Residual fuel gas (1.47 kg/sec, 19.6 bar, 92 °C RFG) is a process gas blended with Naphtha. This decreases the amount of Naphtha in combustion chamber which may benefit in the aspect of efficiency and economy, but still environmental aspect has gain a lot of importance in today's scenario. Hence, exergetic based sustainability indicators are important to for analysis of energy system. It may further have noted that so far exergetic sustsiability analysis have been considered mostly for utility power plants operating on natural gas as the fuel; power plants using other fuels or fuel mixes have rarely been considered. In this paper exergetic sustainability analysis of two gas turbine power plants is analyzed through energetic and exergetic perspective using the actual operating data.

## 2. System Description

The general mass balance and energy balance equation when applied for a system can be expressed in the rate form as

$$\dot{m}_{in} = \dot{m}_{out} \quad (1)$$

$$\dot{E}_{in} = \dot{E}_{out} \quad (2)$$

Fig.1 and Fig.2 presents the schematic diagram of GT1 (case A), GT2 (case B). In both GT cycle, air (109.2 kg/sec, 30°C, 1bar) is compressed to higher pressure (109.2 kg/sec, 360°C, 8.9 bar) and used to burn fuel inside the combustion chamber. The resulting combustion product enters the turbine, gets expanded producing power, and some power is being taken up to drive compressor also. For carrying out calculation some of the assumptions have been taken up (Ersayin and Ozgener 2015a) and actual data is being used for analysis (Table 1) .

### *Assumptions*

For the energy and exergy analysis of the naphtha based combined cycle power plant following assumptions have been made.

- The system operates under steady state conditions.
- Ideal gas properties are applied for combustion products and air.
- Reference condition for the ambient and also the inlet condition of the gas turbine power plants is :  $P_0 = 101.325 \text{ kPa}$  and  $T_0 = 273 \text{ K}$

Further, we use the following data: -

- The fuel naphtha has following composition: - C (0.8392), H<sub>2</sub> (0.1583), S (0.001) with lower heating value (44079 kJ/kg).
- The residual fuel gas has following composition: - H<sub>2</sub> (0.3674), CO (0.0005), H<sub>2</sub>S (0.0001), CH<sub>4</sub> (0.4986), C<sub>2</sub>H<sub>4</sub> (0.0144), C<sub>2</sub>H<sub>6</sub> (0.0096), C<sub>3</sub>H<sub>8</sub> (0.0173), C<sub>3</sub>H<sub>6</sub> (0.0073) with lower heating value (51660 kJ/kg).

The top cycle works on Brayton cycle and the calculations are done using the plant data (Table 1).

*Compressor:*

$$\frac{T_{2s}}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{g_a-1}{g}} \quad (3a)$$

$$h_{AC} = \frac{T_{2s} - T_1}{T_2 - T_1} \quad (3b)$$

$$W_{AC} = \dot{m}_a c_{pa} (T_2 - T_1) \quad (3c)$$

$$c_{pa}(T) = 1.048 - \frac{3.837T}{10^4} + \frac{9.45T^2}{10^7} - \frac{5.49T^3}{10^{10}} + \frac{7.92T^4}{10^{14}} \quad (3d)$$

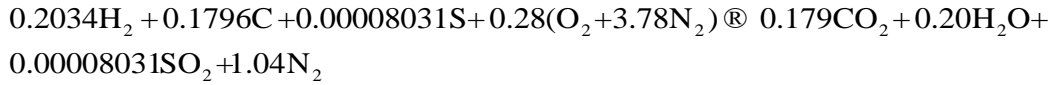
$T_1$  and  $T_2$  represent the temperature at inlet and outlet section of compressor whereas  $T_{2s}$  represents the corresponding temperature after an isentropic compression at compressor outlet, and  $g_a (=1.4)$  represents the ratio of specific heats for air. Eq. (3c) is applied to calculate the power consumed by compressor. Eq. (3d) presents the variation of specific heat of air as a function of temperature (Ersayin and Ozgener 2015b).

*Combustion chamber:*

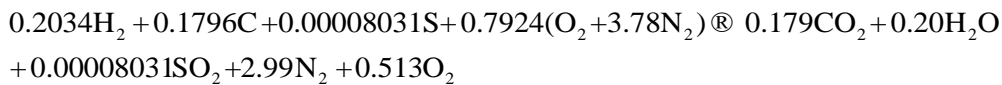
The energy balance for the combustion chamber (GT1) can be written as follows: -

$$\dot{m}_2 h_2 + \dot{m}_3 LHV_5 = \dot{m}_8 h_3 + (1 - h_{CC})(\dot{m}_2 h_2 + \dot{m}_3 LHV_5) \quad (4)$$

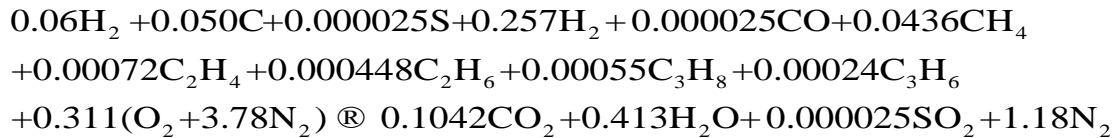
In Eq. (4),  $h_2$  and  $h_3$  are the enthalpies of the gas at compressor outlet and combustion chamber outlet.  $LHV_5$  is lower heating value of naphtha. The energy balance for the combustion process is written as:



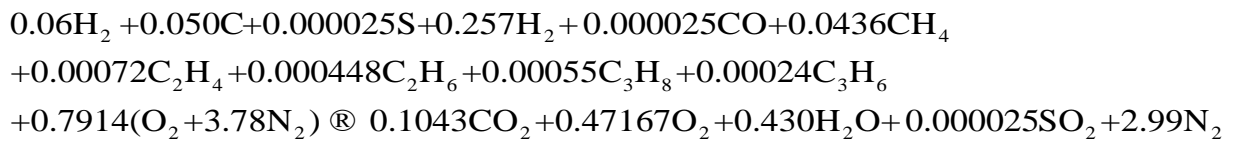
Excess air in percentage is expressed as,  $(W_{actual} - W_{theo}) / (W_{theo}) = 183\%$ ; and the reaction with excess air yields



Similarly, for combustion of naphtha and residual fuel gas of given composition with excess air (Nag 2008), combustion equation may be written as



Excess air in percentage is expressed as,  $(W_{actual} - W_{theo}) / (W_{theo}) = 154.5 \%$ ; and the reaction with excess air yields



Gas turbine:

$$\frac{\gamma T_3}{\gamma T_{4s}} = \frac{\gamma P_3}{\gamma P_4} \quad (5a)$$

$$h_{GT} = \frac{T_3 - T_4}{T_3 - T_{4s}} \quad (5b)$$

$$\dot{W}_{GT} = \dot{m}_g (h_3 - h_4) \quad (5c)$$

$T_3$  and  $T_4$  represent the temperature of gas at the inlet and the outlet section of the gas turbine whereas  $T_{4s}$  represents the corresponding temperature after an isentropic expansion after

gas turbine outlet with  $g_R (=1.27)$  for GT2 and  $g_R (=1.24)$  for GT1. Eq. (5c) is used to calculate the power produced by turbine.

The specific heat of flue gas considering the composition of the combustion products with temperature for GT2 is given below (Boles n.d.).

$$c_{pg}(T) = 1.031 + 0.0000858T + 0.000000195T^2 \quad (6)$$

Similarly, the variation of specific heat of flue gas with temperature for GT1.

$$c_{pg}(T) = 0.9840 + 0.0001262T + 0.000000146T^2 \quad (7)$$

**Table 1: Technical Specification of GT1 and GT2**

Parameters	GT1 (Case A)	GT2 (Case B)
Flow rate	109.2 kg/sec	109.2 kg/sec
CPD	8.9 bar	8.9 bar
CPT	366 °C	366 °C
Fuel Property	2.57 kg/sec, 35.1 bar, 34 °C (Naphtha)	-----
Fuel Property	0.8 kg/sec, 35.1 bar, 34 °C (Naphtha)	1.47 kg/sec, 19.6 bar, 92 °C (RFG)
Rated work	34.5 MW	34.5 MW
Actual work	24.45 MW	29.98MW
LHV	10495 kcal/kg	-----
LHV	10495 kcal/kg	12300 kcal/kg

### 3. Exergy Analysis

Exergy predicts quality and it provide the basis to judge the impact on environment as there is always degradation of exergy due to entropy generation. The general exergy balance which consists of exergy destruction and exergy loss can be written as (Ibrahim et al. 2017):

$$\dot{a} \dot{E}x_{in} - \dot{a} \dot{E}x_{out} = \dot{a} \dot{E}x_{dest} + \dot{a} \dot{E}x_{loss} \quad (8a)$$

$$\dot{E}x_{heat} - \dot{E}x_{work} + \dot{E}x_{mass,in} - \dot{E}x_{mass,out} = \dot{E}x_{dest} \quad (8b)$$

$$\dot{E}x_{heat} = \dot{a} \left(1 - \frac{T_0}{T_i}\right) \dot{Q}_i \quad (8c)$$

$$\dot{E}x_{Work} = \dot{W} \quad (8d)$$

$$\dot{E}x_{mass,in} = \dot{m}_y y_{in} \quad (8e)$$

$$\dot{E}x_{mass,out} = \dot{m}_y y_{out} \quad (8f)$$

In order to calculate physical exergy of water/steam phases, Eq.9 is used (Ahmadi and Toghraie 2016).

$$e_{x,physical} = (h - h_0) - T_0(s - s_0) \quad (9)$$

where  $h_0$  and  $s_0$  depicts dead-state conditions. Assuming air to be a perfect gas, the specific physical exergy of air is calculated by Eq.10.

$$y_{a,per} = c_{p,a} \ln \frac{T}{T_0} - T_0 \ln \frac{T}{T_0} + R_a T_0 \ln \frac{P}{P_0}$$

(10)

chemical exergy (Kaushik and Singh 2014) of Naphtha, Residual fuel gas have important role in performance assessment. The chemical exergy of liquid fuel (Naphtha and Carbon black fluid stock) is found out by multiplying 1.06 to lower heating value of fuel (Ersayin and Ozgener 2015a). For gaseous fuel Eq. (13) can be used.

$$ex_{ch,i} = \sum_{i=1}^n x_i ex_i^{ch} + RT_0 \sum_{i=1}^n x_i \ln(x_i) \quad (11)$$

The value of specific chemical exergy of each component is taken from (Szargut 2007). The same procedure is followed in the case of flue gas where molar composition of the combustion gases is known by chemical balance and presented in Table 3. Since, Residual fuel gas is introduced in the combustion chamber at 92°C, physical exergy is calculated by multiplying mass fraction with Eq. 11.

### Exergy Efficiency of GT2 and GT1

$$h_{II,GT1} = \frac{W_{net,GT1}}{\dot{m}_5 e_5} \quad (12)$$

$$h_{II,GT2} = \frac{W_{net,GT2}}{\dot{m}_6 e_6 + \dot{m}_5 e_5} \quad (13)$$

## 4. Exegetic Sustainability Indicators

Using theoretical background of exergy analysis, the sustainability parameters are considered and rearranged for the present GT power plant configuration: -

- Waste exergy ratio
- Environmental impact factor
- Exergetic sustainability index

#### 4.1 Waste Exergy Ratio ( $r_{we}$ )

During the operation of electrical power generation some of the exergy is destructed in the engine components and some of exergy is lost during this process. So waste exergy (Aydin 2013) can be described by Eq.(13).

$$r_{we}^{Case A} = \frac{\dot{Ex}_{we}^{case}}{\dot{Ex}_t} = \frac{\dot{Ex}_{dest,out}^{ng} + \dot{Ex}_{lost,out}^{ng}}{\dot{Ex}_{ch}^{ng} + \dot{Ex}_{ch}^{air}} \quad (14)$$

#### 4.2 Environmental Effect Factor ( $r_{eef}$ )

Environmental effect factor (Aydin 2013) is performed by ratio of waste exergy ratio to the exergy efficiency.

Environmental effect factor = waste exergy ratio/ Exergy efficiency

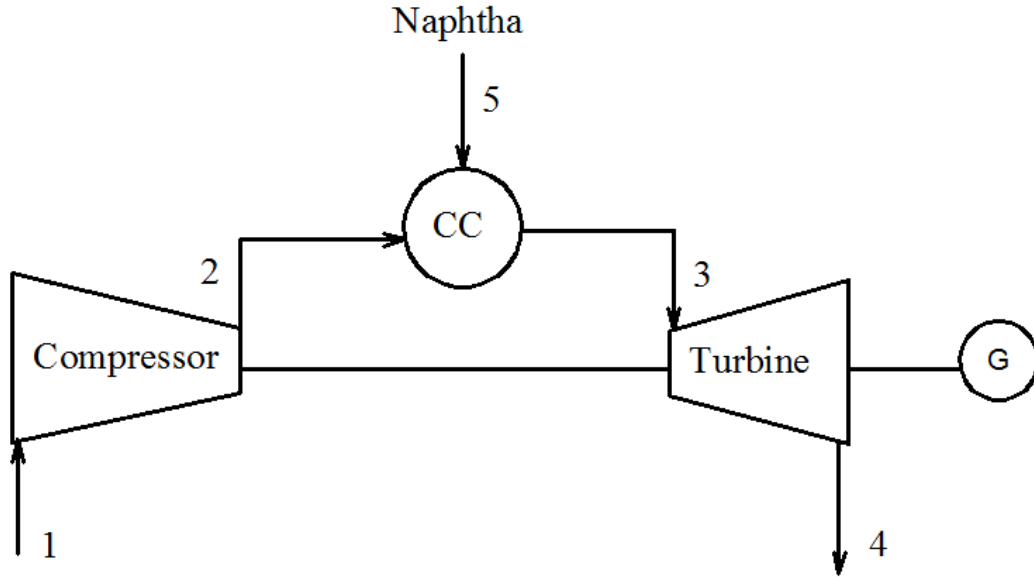
$$r_{eef}^{Case} = \frac{\dot{Ex}_{we}^{case}}{h_{ex}^{caseA}} \quad (15)$$

#### 4.3 Exergetic sustainability index ( $\theta_{esi}$ )

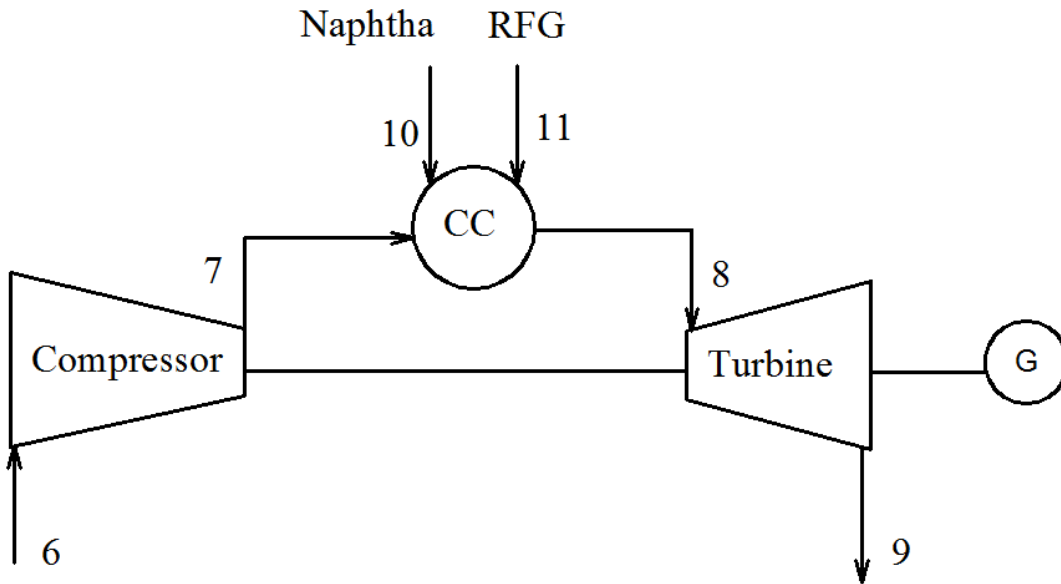
It is one of the vital parameter among all exergetic sustainability indicators (Aydin 2013) and can be found out by reverse environmental effect factor. The range of this index is between 0 and  $\infty$ .

$$q_{esi} = \frac{1}{r_{eef}} \quad (16)$$





**Figure 1:** Open Cycle Diagram of GT1 (Case A) plant with Naphtha based Fuel

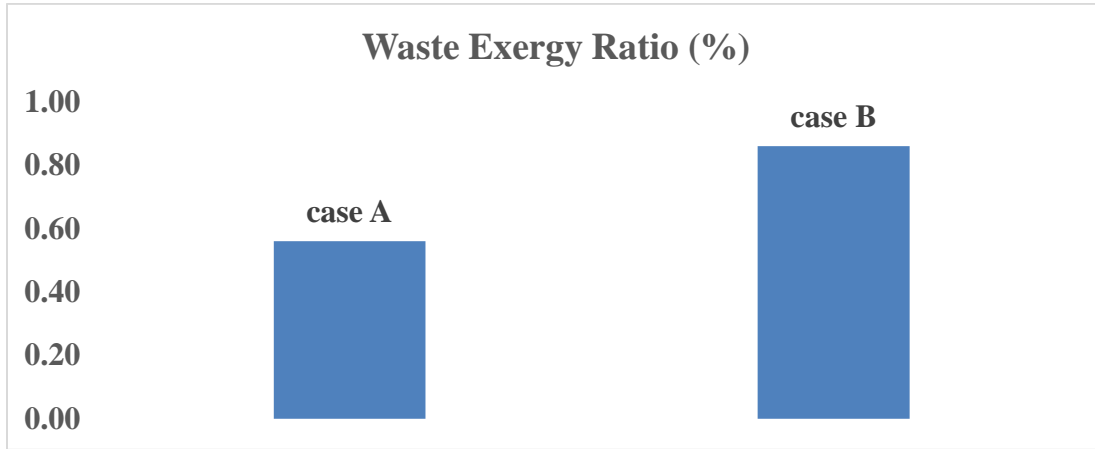


**Figure 2:** Open Cycle Diagram of GT2 (case B) plant with Naphtha-RFG Mixture based Fuel

#### 4. Results

By using the approach mentioned in Eq. 14 and Eq.15, the total energy input of fuel is 121.56 MW and 144.83 MW. Using this fuel exergy, the plant produces electrical power output of GT2 (29.98 MW) ,GT1(24.45 MW) , thus achieving cefficiency of 20% in both cases (GT1 and GT2). However with the production of electrical output, both gas turbine emit large amount of heat (593°C) GT1 and (573°C) GT2.

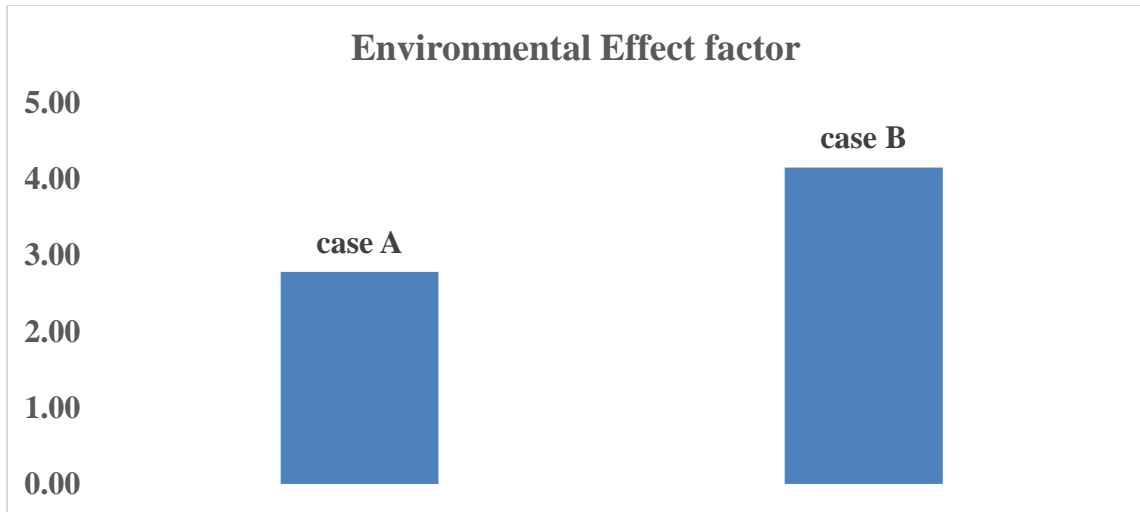
As it can be inferred from Table 3 that exergetic efficiency of both gas turbines (GT1 and GT2) are same but waste exergy ratio of GT2 is higher as compared to GT1. The waste exergy ratio (Fig.3) comprises of exergy destruction which occurs during power plant running situation and also some exergy is lost by means of hot flue gas left out open. Due to injection of Residual fuel gas at 92°C, there is high value of irreversibility in case of GT2, hence waste exergy ratio in case of GT2 is high.



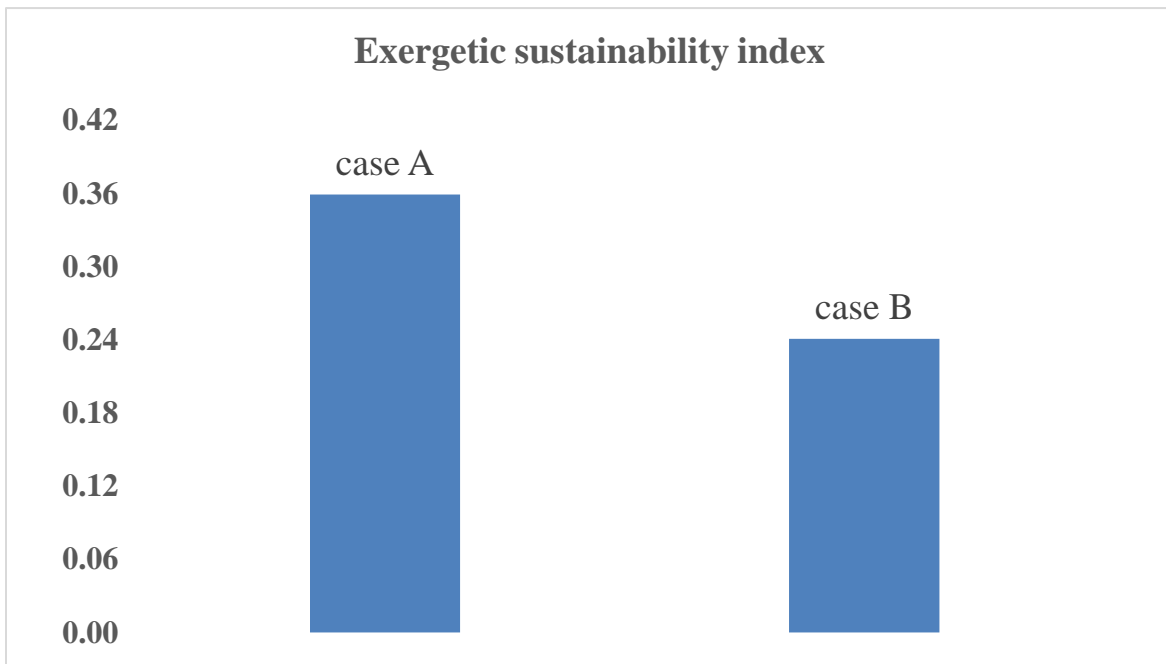
**Figure 3:** Waste Exergy Ratio for Case A (GT1) and Case B (GT2)

Since waste exergy ratio is high in case GT2 due to mixing at different temperature hence environmental effect factor is high in case of GT2. Hence it can be inferred from Eq. 15 although exergy efficiency is same in both case but due to high waste exergy ratio in case B environmental effect factor (Fig.4) is high and it predicts whether a system damages environment due to unusable waste exergy output and exergy destruction.

Since GT2 has high waste exergy ratio as compared to GT1 hence exergetic sustainability index is high in case of GT2. In other words less is waste exergy ratio less is exergetic sustainability index (Fig.5) hence a system becomes more sustainable. The measures which could be taken to increase exergy efficiency decreases the waste exergy ratio thus it makes system more sustainable.



**Figure 4:** Environmental Effect Factor for GT1 and GT2



**Figure 5:** Exergetic Sustainability Index for GT1 and GT2

**Table 3:** Exergetic Sustainability Parameters for case A(GT1) and case B(GT2)

Parameters	Case A	Case B
Fuel exergy	121.56	144.83
Useful exergy	24.45	29.98
Loss exergy	36.97	36.97

Total destruction exergy	56.55	80.11
Exergy efficiency	0.2011	0.207
Waste exergy Ratio	0.56	0.86
Environmental effect factor	2.78	4.15
Exergetic sustainability Index	0.36	0.24

## 5. Conclusions

Although efficiency is same in both gas turbine (20%), but still due to temperature difference between two fuels in GT2 (Case B), the exergy destruction (80.11 MW) is very high as compared to GT1 (56.55 MW) which leads to huge amount waste exergy ratio in case of GT2 (0.56) as compared to GT1 (0.86) which damages environment. Of course blending of Residual fuel gas along with Naphtha gives some benefit in economic aspect, but still due to high exergy destruction in case of GT2 it is not beneficial as to decrease the exergy destruction rate in combustion chamber of GT2 (case B), turbine inlet temperature has to be increased which is again a cost intensive because of material selection. Hence GT2 system has less exergetic sustainability index (0.24) as compared to GT1 (0.36). However, installation of some preheating arrangement before Naphtha injection in combustion chamber in GT2 may reduce exergy destruction.

## Nomenclature

- AC air compressor
- CC combustion chamber
- CPD compressor pressure discharge
- CPT compressor pressure temperature
- LHV lower heating value
- GT gas turbine
- RFG Residual fuel gas

## Indices

- t total
- u useful

we waste  
eef environmental effect factor  
esi exergetic sustainability index  
in inlet  
out outlet

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