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Energy and Exergy Performance Optimization on a Real Gas Turbine Power Plant

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Abstract**—**This paper presents the energy and exergy optimization of a real gas turbine power plant performance of 100 MW of power, installed in the South East of Tunisia. A simulation code is established using the EES (Engineering Equation Solver) software. The parameters considered are those of the actual operating conditions of the gas turbine thermal power station under study. The results show that thermal and exergetic efficiency decreases with the increase of the ambient temperature. Air excess has an important effect on the thermal efficiency. The emission of NOx rises in the summer and decreases in the winter. The obtained rates of NOx are compared with measurements results.

Keywords**—**Efficiency, exergy, gas turbine, temperature.

I. Introduction

TODAY, gas turbines are one of the most widely-used power generating technologies. In recent years, gas turbines (GT) are widely developed for several industry applications. Despite their advantages, their thermal efficiency is sensibly affected by the ambient temperature which varies greatly throughout the year.

Significant efforts have been made by researchers to improve the efficiency of the cycle and control the pollution of the air due to gas turbine emission, in particular by reducing the nitrogen oxide (NOx) emissions [1]-[3].

Anupam and Sanjay [2] studied the influence of cycle operating parameter variations such as pressure ratio (PR), combustor primary zone temperature, turbine inlet temperature (TIT), equivalence ratio and residence time on the performance of two different gas turbine power plants: A basic gas turbine cycle and a two-stage compression cycle with intercooled heat exchanger. The gas emission rate is analyzed too. Obtained results show that the gas turbine efficiency increases with the PR to reach a maximum value of 35% for PR equal to 28 and then decreases. Gas turbine specific work increases with the increase of the TIT for a fixed value of PR. For an equivalence ratio equal to 0.95 and residence time of 0.5 ms, the primary zone temperature in the combustion chamber and the NOx emission rates increase simultaneously with the PR. Moreover, for the same value of equivalence ratio and a PR of 22, increasing the residence time leads to a decrease of the CO emission rate and a rise of the NOx release. Rahman et al. [3] analyzed the effect of PR, ambient temperature, air to fuel ratio and TIT on the performances of simple gas turbine power plant. The authors highlighted that the thermal efficiency is significantly affected by indicated parameters. Also, the increase of ambient temperature and the air to fuel ratio reduces the thermal efficiency and the net power output. In addition, it is found that the specific fuel consumption and the supplied heat increase linearly with both the ambient temperature and the air to fuel ratio.

Gülder [4] presented an approximate formula to determine the adiabatic flame temperature of jet fuel-air systems in terms of pressure, temperature, equivalence ratio and H/C atomic ratio of the fuel. Gülder compared the results obtained from the proposed formula to data obtained using chemical equilibrium code including 14 species in the combustion products. It is found that the error is about 0.4 of the estimated temperature.

For the Tunisian industry, the gas turbine power plants constitute the principal sources of electrical energy. Several gas turbine power plants of about 100 MW of net power are installed in different regions of the Tunisia. In addition, some combined cycle power plants of 400 MW are planted in the important industrial sites [5]. National programs are established in order to better control the energy conservation and reduce the gas emission in conformity with the international standards. In this context, energy and exergy optimization study is conducted on the different power plants in collaboration with the Tunisian Society of Electricity and Gas, STEG.

The effects of the operating parameters on the energy efficiency and the NOx emission rates are analyzed. Measurements performed on the NOx release are used to validate the proposed models.

II. Power Plant Operating Mode

The actual study is conducted on a GT cycle owned by the STEG and installed in Gabes (South East of Tunisia). The installation is a General Electric (MS 9001 E') Gas Turbine 165 and produces of about 100 MW of net power. It is constituted by an axial compressor with 17 stages and a turbine with 3 stages. The combustion section is of "inversed flow" type engendering 14 annular combustion chambers equipped with: flame tubes, Spark Plugs, flame detectors and fuel injectors as presented in Fig. 1.

The GT cycle arrangement diagram is presented in Fig. 2. The thermal energy is converted into mechanical work used to drive simultaneously the compressor and an electrical generator of 149.250 kVA of power. A gear is installed between the generator and the turbine shaft to provide the suitable rotation speed around 3000 rv/mn. The GT body is cooled by an appropriate refrigeration system using cold air

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circuit provided by two centrifugal fans.

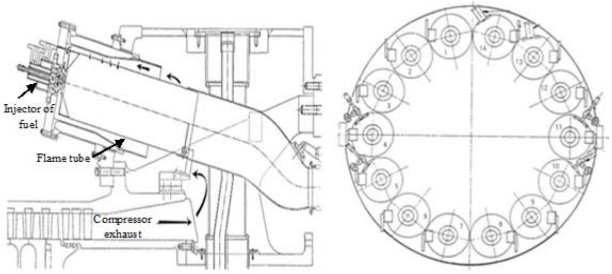

Fig. 1 Combustion chamber arrangement [4]

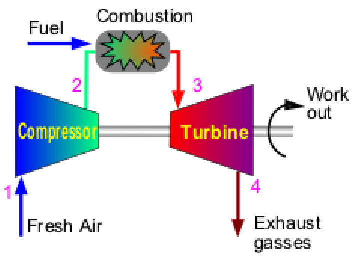

Fig. 2 GT cycle

The Tunisian Natural Gas is used as fuel. The thermo-physical properties of this fuel are given in Table I. The operating parameters used in this project are indicated in Table II.

TABLE I
FUEL THERMO-PHYSICAL PROPERTIES

Elements	Mass composition
Nitrogen	1.51
Methane	77.02
Carbon dioxide	1.15
Ethan	15.96
Propane	4.20
Méthyl-2-propan	0.04
Butane	0.12

TABLE II
OPERATING PARAMETERS

Symbol	Quantity	unit
	Compressor	
T_1	Air inlet temperature	40 °C
R_{comp}	Isentropic efficiency	85%
ka	Specific heat ratio of air	1.4
PR	Pressure ratio	10 bars
	Combustor	
R_h	Efficiency of combustion chamber	90%
kg	Exhaust gas specific heat ratio	1.332
$(\Delta P/P_2)$	Pressure drops ratio	5%
	Turbine	
R_t	Isentropic efficiency	85%

III. MODELING

A. Energetic Analysis

1. Compressor

As the compression was considered adiabatic and the ideal gas, the work of compressor is written by:

$$\dot{W}_{comp,air} = cp_{air}.\ \dot{m}_{air}.\ (T_2 - T_1)/R_{comp} \quad (1)$$

The isentropic efficiency of the compressor is expressed as:

$$R_{comp} = \frac{T_{2is} - T_1}{T_2 - T_1} \quad (2)$$

According to (2), the temperature T_2 is expressed as:

$$T_2 = T_1 + \frac{T_{2is} - T_1}{R_{comp}} \quad (3)$$

with Cp_{air}: the air specific heat which is calculated using the Ibrahim and Rahman equation [3]:

$$cp_{air} = 1.0189.10^3 - 0.1378.T_a + 1.9843.10^{-4}.T_a^2 + 4.2399.10^{-7}.T_a^3 - 3.7632.10^{-10}.T_a^4 \quad (4)$$

$T_a = \frac{(T_2+T_1)}{2}$ is given in Kelvin and valid for 200K < T < 800 K

2. Combustor

The specific heat of hot gases, Cp_{gas}, is obtained by (5):

$$cp_{gas} = 1.8083 - 2.3127.10^{-3}.T_3 + 4.045.10^{-6}.T_3^2 - 1.7363.10^{-9}.T_3^3 \quad (5)$$

The heat to be supplied by the combustion chamber can be expressed by:

$$\dot{Q}_h = \dot{m}_{fuel}.PCI.R_h \quad (6)$$

Chemical reactions: The global combustion reaction occurring in the combustion chamber can by expressed as:

$$aCH_4 + bC_2H_6 + cC3H_8 + dC_4H_{10}(1) + eC_4H_{10}(2) + fN_2 + gO_2 + hCO_2 + iA_r \dashrightarrow lCO_2 + mN_2 + nO_2 + oH_2O + pNO + qA_r \quad (7)$$

The different product factors are determined according to chemical equilibrium of the reaction and taking into consideration the mass flow rates and chemical composition of air and fuel.

3. Turbine

For adiabatic detente, we proceed as for compression:

$$\dot{W}_t = cp_{gaz} * (T_3 - T_4) \quad (8)$$

The isentropic efficiency of the turbine is:

$$\eta_t = \frac{T_3 - T_4}{T_3 - T_{4is}} \quad (9)$$

We can express T4 by:

$$T_4 = T_3 - \eta_t * (T_3 - T_{4is}) \quad (10)$$

From where:

$$T_3 - T_4 = \eta_t * (T_3 - T_{4is}) \quad (11)$$

and T_3 is determined by applying the energy balance on the combustion chamber. Therefore, real power produced by the turbine is:

$$\dot{W}_t = \dot{m}_{tot} * \eta_t * cp_{gaz} * (T_3 - T_{4is}) = \dot{m}_{tot} * \eta_t * cp_{gaz} * T_3 * \left(1 - \frac{T_{4is}}{T_3}\right) \quad (12)$$

where $\dot{m}_{tot}$ is the total mass flow rate through the turbine:

$$\dot{m}_{tot} = \dot{m}_{air} + \dot{m}_{fuel} \quad (13)$$

The net power of the gas turbine power plant is determined as:

$$\dot{W}_{net} = \dot{W}_t - \dot{W}_{comp,air} \quad (14)$$

The thermal efficiency of the Gas Turbine power plant is expressed as:

$$\eta_{cycle} = \frac{\dot{W}_{net}}{\dot{Q}_h} = \frac{\dot{W}_t - \dot{W}_{comp,air}}{\dot{Q}_h} \quad (15)$$

B. Exergy Analysis

As a definition, exergy is the maximum theoretical useful work that may be received from energy in a system of ideal machines that can be destroyed due to irreversibility. Exergy is composed of the physical exergy and the chemical exergy. In this research, the kinetic and potential parts of exergy are neglected due to their dispensable rates. The physical exergy can be defined as the maximum theoretical useful work obtained as a system interact with an equilibrium state as well as the chemical exergy is associated with the departure of the chemical composition of a system from its chemical equilibrium. In combustion process, the chemical exergy is an important part of exergy [6], [7]. Exergy analysis can determine the deficiencies of energy systems by quantifying the entropy-generation of all components based on the first and second laws of thermodynamic. By applying the exergetic balance the exergy, exergetic losses and exergetic efficiency of different points of the cycle, which are defined in Fig. 1, were determined and were calculated b for each element: We noted by $\dot{E}_p$ the exery product and $\dot{E}_f$ the exergy fuel.

1. Compressor

The exergy product and fuel of the compressor is calculated by applying the exergy balance of the control volume cross in it:

$$\dot{E}_{p,c} = \dot{E}_2 - \dot{E}_1 \quad (16)$$

$$\dot{E}_{f,c} = \dot{W}_c \quad (17)$$

$$\dot{E}_{p,c} = \dot{m}_a * ((c_{p,a} * (T_2 - T_1) - T_0 * ((c_{p,a} * \ln\left(\frac{T_2}{T_1}\right) - R_a * \ln\left(\frac{P_2}{P_1}\right) \quad (18)$$

The exergy destruction in compressor has been determined by:

$$\dot{E}_{D,c} = \dot{E}_1 + \dot{w}_c - \dot{E}_2 \quad (25)$$

The exergy efficiency can be calculated with the relation as under:

$$\eta_{ex,c} = \frac{\dot{E}_{p,c}}{\dot{E}_{f,c}} = (\dot{E}_2 - \dot{E}_1)/\dot{W}_c \quad (19)$$

2. Combustion Chamber

The exergy balance of combustor:

$$\dot{E}_{f,cc} = \dot{E}_{fuel} \quad (20)$$

$$\dot{E}_{p,cc} = \dot{E}_3 - \dot{E}_2 \quad (21)$$

$$\dot{E}_2 = \dot{m}_a * (c_{p,a} * (T_2 - T_0) - T_0 * \left(c_{p,a} * \ln\left(\frac{T_2}{T_0}\right) - R\ln\left(\frac{P_2}{p_0}\right)\right)) \quad (22)$$

$$\dot{E}_{fuel} = \dot{m}_f * ex_{fuel} \quad (23)$$

To evaluate the fuel exergy, the following formula can be used which define the corresponding ratio of simplified exergy:

$$\xi = \frac{ex_{fuel}}{PCI} \quad (24)$$

The ratio of chemical exergy to lower heating value is usually close to 1, for the majority of regular gaseous fuels, so:

$$\dot{E}_{f,cc} = \dot{m}_f * PCI \quad (25)$$

$$\dot{E}_3 = ex_3^{ph} + ex_3^{chim} \quad (26)$$

$$ex_3^{ph} = \dot{m}_g * (c_{p,g} * (T_3 - T_0) - T_0 * \left(c_{p,g} * \ln\left(\frac{T_3}{T_0}\right) - R\ln\left(\frac{P_3}{p_0}\right)\right)) \quad (27)$$

$$ex_3^{chim} = \sum_{i=1}^{n} X_i * ex^{chi} + R * T_0 \sum_{i=1}^{n} X_i \mathrm{Ln} X_i \quad (28)$$

where X_i: the molar fraction of compound i; ex^{chi}: The specific exergy of each compound.

3. Turbine

$$\dot{E}_{p,t} = \dot{W}_t \quad (29)$$

$$\dot{E}_{f,t} = \dot{E}_3 - \dot{E}_4 \quad (30)$$

$$\dot{E}_{p,t} = \dot{W}_t = \dot{m}_g * c_{p,g} * (T_4 - T_3) \quad (31)$$

$$\dot{E}_{f,t} = \dot{m}_g * ((c_{p,g} * (T_3 - T_4) - T_0 * ((c_{p,g} * ln\left(\frac{T_3}{T_4}\right) - R_a * ln\left(\frac{P_3}{P_4}\right)))) \quad (32)$$

Exergy destruction of the turbine can be illustrated by:

$$\dot{E}_{D,t} = \dot{E}_3 - \dot{E}_4 + \dot{w}_t \quad (33)$$

The exergy efficiency of the turbine can be determined by:

$$\eta_{ex,t} = \frac{\dot{E}_{p,t}}{\dot{E}_{f,t}} = (\dot{W}_t/(\dot{E}_3 - \dot{E}_4)) \quad (34)$$

The exergy efficiency of the gas turbine cycle can be expressed by:

$$\eta_{ex,cycle} = \frac{\dot{W}_{net}}{\dot{E}_1+\dot{E}_{fuel}} \tag{35}$$

C. NOx Formation

During combustion, the formation of NOx is carried out according to three different reaction mechanisms: thermal mechanism, fuel and fast. Each training path has its own characteristics as indicated in Fig. 3.

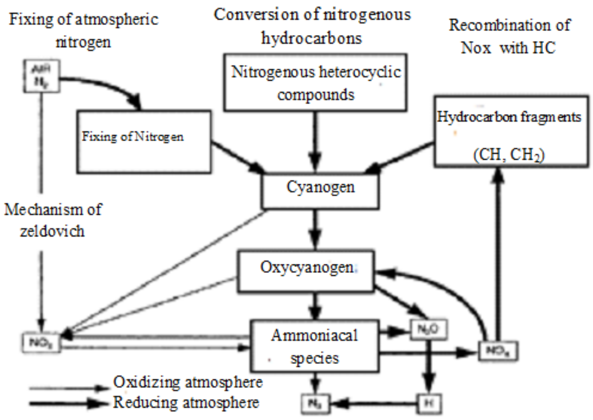

Fig. 3 The different NOx formation paths [8]

The formation of thermal NOx is at the origin of the reaction of nitrogen and oxygen in the atmosphere at high temperature. The formation of the NO fuel is due to the oxidation of the nitrogen derivatives contained in the fuel. Under hydrocarbon-rich conditions, hydrocarbon moieties such as C, CH, CH_2 can react with the nitrogen of air to form according to Fenimore, NH, HCN, H2CN, and C. These compounds are oxidized to NO in the flame zone which is poor in hydrocarbons.

The main parameters affecting the NOx emissions form Gas Turbine power plant are the equivalence ratio and the primary zone temperature. The equivalence ratio,ϕ, the ratio of real fuel-air ratio to the stoichiometric fuel- air ratio represented by:

$$\phi = \frac{\left(\frac{\dot{m}_{fuel}}{\dot{m}_{air}}\right)_{actual}}{\left(\frac{\dot{m}_{fuel}}{\dot{m}_{air}}\right)_{\text{stoichiometric}}} \tag{36}$$

The primary zone temperature T_{pz} represents the higher temperature value reached in the combustor and it is expressed by [2]:

$$T_{PZ} = A.\sigma^{\alpha}.e^{\,(\beta(\sigma+\lambda)^2)}.(\frac{P_3}{P_0})^{x^X}.(\frac{T_3}{T_0})^{y^X}.(\frac{H}{C})^{z^X} \tag{37}$$

where $\frac{H}{C}$is the hydrogen to carbon ratio:

$$\sigma=\Phi \quad \text{if} \quad \Phi<=1$$
$$\sigma=\Phi-0.74 \quad \text{if} \quad \Phi>1$$
$$x^X = a_1 + b_1.\sigma + c_1.\sigma^2$$
$$y^X = a_2 + b_2.\sigma + c_2.\sigma^2$$
$$z^X = a_3 + b_3.\sigma + c_3.\sigma^2$$

The values of the different parameters indicated in (37) are given on Table III for different ranges of ϕ and the ratio $\left(\frac{T_3}{T_0}\right)$ [2].

TABLE III
PARAMETER VALUES FOR T_{PZ} DETERMINATION

Parameter	$0{,}3\leq \phi \leq 1{,}0$		$0{,}3\leq \phi \leq 1{,}0$	
	$0{,}92\leq (\frac{T_3}{T_0}) \leq 2$	$2\leq (\frac{T_3}{T_0}) \leq 3{,}2$	$0{,}92\leq (\frac{T_3}{T_0}) \leq 2$	$2\leq (\frac{T_3}{T_0}) \leq 3{,}2$
A	2361,76	2315,75	916,82	1246,17
α	0,1157	-0,049	0,288	0,381
β	-0,948	-1,114	0,145	0,347
λ	-1,097	-1,180	-3,277	-2,0365
a_1	0,014	0,010	0,031	0,036
b_1	-0,055	-0,045	-0,078	-0,085
c_1	0,052	0,0482	0,0497	0,0517
a_2	0,395	0,568	0,025	0,009
b_2	-0,441	-0,550	0,260	0,502
c_2	0,141	0,132	-0,131	-0,247
a_3	0,005	0,011	0,004	0,017
b_3	-0,129	-0,129	-0,178	-0,189
c_3	0,082	0,084	0,098	0,1037

D. NOx Emission

The NOx release rate can be calculated as follows [2]; for Φ <= 1.08:

$$NO_x = 1^{13}.(P_3.\frac{10^{-5}}{1{,}4^6})^{aa}\, e^{(-\frac{71442}{T_{pz}})}.(7{,}56.\phi^{-7{,}2} - 1{,}6).\tau^{0{,}64} \tag{38}$$

For Φ>1.08:

$$NO_x = 1^{13}.(P_3.\frac{10^{-5}}{1{,}4^6})^{aa}\, e^{(-\frac{71442}{T_{pz}})}.(5{,}21.\phi^{-2{,}99} - 1{,}6) * \tau^{0{,}64} \tag{39}$$

where $aa = 11{,}949\, e^{(-\phi/5{,}76)} - 10{,}0$. These equations give NO rates in g per kg of fuel.

IV. RESULTS AND DISCUSSIONS

To analyze the influence of the main operating variables on the cycle performances, a simulation code is established using the software EES. The ambient temperature has an important influence on the power plant performances. The operating conditions of the considered cycle will be analyzed for two running seasons: The cold season for which the local ambient temperature varies from 7 to 25 °C and the hot season where the ambient temperature increases from 25 to 45 °C.

For cold season, Fig. 3 presents the variation of the cycle efficiencies according to the ambient temperature while Fig. 3 (b) depicts the variation of primary zone temperature and the NOx emission rate with the ambient temperature. The equivalence ratio and the residence time are taken equal to 0.35 and 0.5 respectively. The cycle efficiency decreases slightly with T1. For an ambient temperature variation range of 18 °C, the cycle efficiency is reduced by about 3.7%. The exergetic efficiency decreases with the ambient temperature by about 6%.

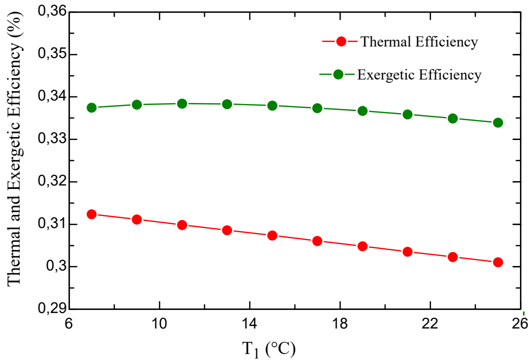

Fig. 4 (a) Variation of Thermal and Exergetic Efficiency in the cold season

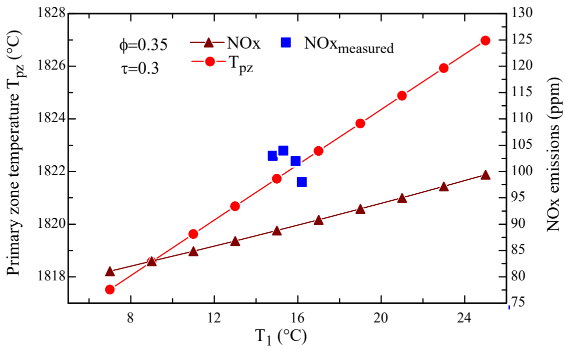

Fig. 4 (b) Influence of ambient temperature on Tpz and NOx emissions (Cold season)

The Tpz varies from 1818 to 1827 °C while the NOx emissions increase from 81 to about 100 ppm. That represents a growth of about 23.5% noting that experimental measurements of NOx rate are performed on the indicated gas turbine power plant GT and in cold and hot season where the ambient temperature is about respectively 13 °C and 35 °C. Measurements are taken using appropriate probe in the GT exit before the stack entrance. The average experimental NOx rate is about 88.8 ppm as shown on Table IV. That represents a divergence of about 4% regarding the theoretical data.

Fig. 4 (a) presents the variation of the cycle efficiency with the ambient temperature and Fig. 4 (b) shows the variation of TPZ and NOx in the hot season. For an ambient temperature variation range of 18 °C the cycle efficiency decreases of about 4% and the exergetic efficiency decreases by about 5%.

The same ambient temperature variation range leads to a raise of the NOx release of about 22.5 ppm. That represents a growth of about 22.2%. The primary zone temperature increases by 9 °C. From the indicated results, it can be shown that during the whole year, the ambient temperature variation leads to an increase of the NOx emission rate of about 45% while the primary zone temperature increases by 18 °C. At temperature equal to 35 °C, the average of measured NOx rate is about 109.35 ppm while the calculated NOx rate is about 111.1 ppm. Air excess is an important parameter and its variation influences cycle efficiency. This importance is due to the fact that excess air is necessary to ensure complete combustion. The variation of the efficiency of the cycle as a function of the excess of air is presented in Fig. 5. The PR has an important influence on the thermal and exergetic efficiency of the cycle. With an increase of air excesses from 0.05 to 0.3 the thermal efficiency notes an increase of 6%. Fig. 6 illustrates the variation of thermal and exergetic efficiency with the PR.

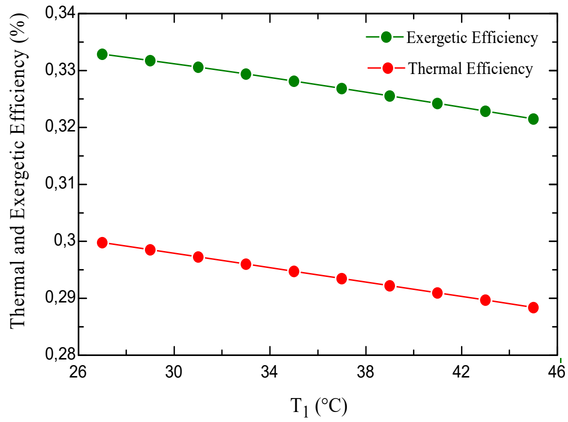

Fig. 5 (a) Variation of thermal and Exergetic Efficiency in the hot season

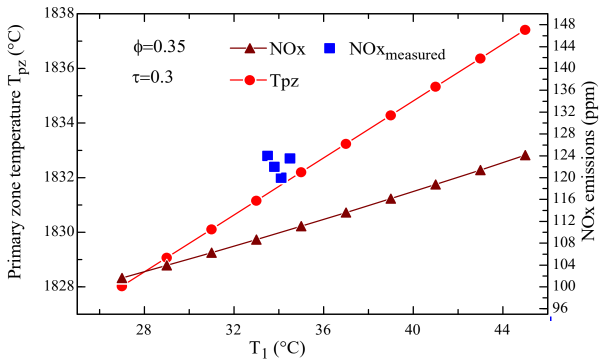

Fig. 5 (b) Variation of Tpz and NOx emission with ambient temperature (hot season)

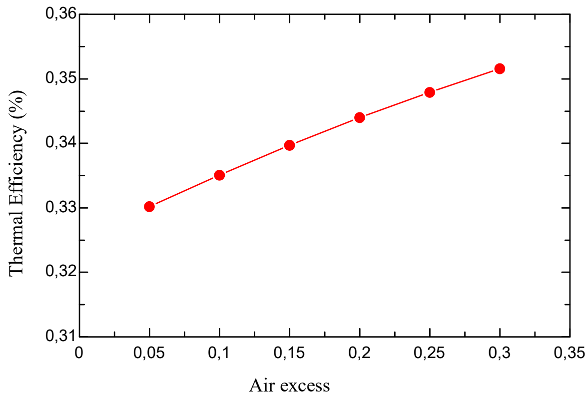

Fig. 6 Variation of thermal and Exergetic Efficiency with air excess

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TABLE IV
EXPERIMENTAL MEASUREMENTS

Power	Period	Average T_{amb}	O_2(%)	CO_2 (%)	NO (ppm)				NO_2 (ppm)	NOx (ppm)	SO_2 (ppm)
105MW	12 to 13 PM	13°C	16.1	2.8	88	88.2	89.5	90.9	0.5	87.3	<1.0
110MW	10 to 11 PM	34°C	16.1	2.8	108.9	110	109	109.5	0.4	94.6	<1.0

The results of exergetic analyses are shown in Fig. 7. The exergy destruction rate and exergy efficiency of different elements in the gas turbine cycle are computed. It is noted that the combustion chamber has the maximum rate of exergy destruction and the minimum rate of exergy efficiency among other elements. This can be explained by the chemical reactions inside the combustion chamber as well as high temperature differences between the operating fluid and flame.

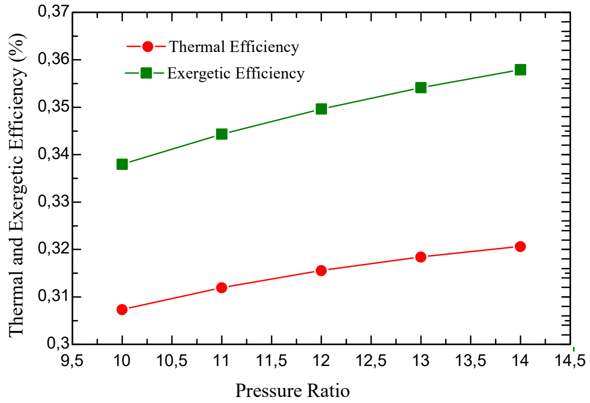

Fig. 7 Variation of thermal and Exergetic Efficiency with PR

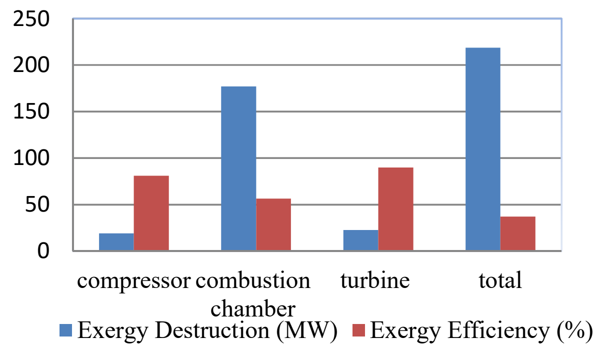

Fig. 8 Exergy Destruction and efficiency of components and of total plant

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NOMENCLATURE

Cp	Specific heat
K	Specific heat ratio
$\dot{m}$	Mass flow
P	Pressure (bars)
PCI	Calorific value
PR	pressure ratio
$\dot{Q}$	Heat supplied
R	isentropic efficiency
T	temperature
$\dot{w}$	Specific power
Subscripts	
a	Air
comp	Compressor
f	Fuel
g	Gas
h	Combustor
is	Isentropic
t	Turbine
Greek symbols	
η	Efficiency

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