International Journal of Mechanical, Industrial and Aerospace Sciences
ISSN: 2517-9950
Vol:11, No:7, 2017

Investigation of Main Operating Parameters Affecting Gas Turbine Efficiency and Gas Releases

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***Abstract*—**This work presents a study on the influence of the main operating variables on the gas turbine cycle. A numerical simulation of a gas turbine cycle is performed for a real net power of 100 MW. A calculation code is developed using EES software. The operating variables are taken in conformity with the local environmental conditions adopted by the Tunisian Society of Electricity and Gas. Results show that the increase of ambient temperature leads to an increase of Tpz and NO_x emissions rate and a decrease of cycle efficiency and UHC emissions. The CO emissions decrease with the raise of residence time, while NO_x emissions rate increases and UHC emissions rate decreases. Furthermore, both of cycle efficiency and NOx emissions increase with the increase of the pressure ratio.

***Keywords*—**CO, efficiency, gas turbine, NO_x, UHC.

I. Introduction

THE technology of gas turbines (GT) has been significantly developed in recent years in terms of material choice, conceptual characteristics, and operating conditions. The improvement of the GT cycle efficiency constitutes the major aim of the research works. Furthermore, the reduction of gas release is taken into consideration to preserve the environment [1]. Several previous works are developed on the GT cycles in the purpose to analyze the influence of the operating parameters on the power plant efficiency [2]-[4].

II. Power Plant Presentation

The actual study is conducted on a simple GT cycle. Fig. 1 illustrates the considered GT design. This installation is of about 100 MW of net power and it is mainly constituted by an axial compressor with 17 stages and a turbine with three stages. The combustion section is of "inversed flow" type engendering 14 annular combustion chambers equipped with: flame tube, spark plugs, flame detectors and fuel injectors. The GT cycle arrangement is presented in Fig. 2. Ambient air is cleaned and compressed through the compressor before entering into the combustion chamber where the fuel is injected according to an appropriate ratio.

The combustion reaction occurs at practically constant pressure. From the combustion chamber, the hot gases are expanded through the turbine. The thermal energy is converted into mechanical work used to drive simultaneously the compressor and the alternator. The used fuel is the Tunisian natural gas with thermo-physical properties given in Table I.

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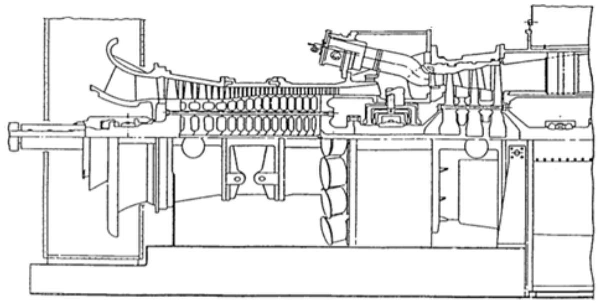

Fig. 1 GT design [5]

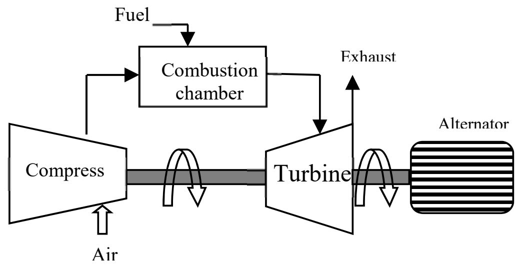

Fig. 2 GT cycle

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

The operating parameters are indicated in Table II. These parameters are defined according to the local environmental conditions.

III. Modeling

A. Power Plant Performances

1. Compressor

The air is introduced into the compressor at the ambient conditions. The isentropic and real outlet temperatures of the compressor are determined from:

$$T_{2is} = T_1 . PR^{\frac{ka-1}{ka}} \quad (1)$$

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

So, the actual work consumed by the compressor is given by

$$\dot{W}_{comp,air} = cp_{air}.\dot{m}_{air}.(T_2 - T_1)/R_{comp} \quad (3)$$

with: Cp_{air}: The specific heat of air which can be determined using the equation proposed by Ibrahim and Rahman [6]:

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

$T_a = \frac{(T_2+T_1)}{2}$ in K. This equation is valid for 200 K<T <800K

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%

2. Combustor

The combustion efficiency and the relative pressure losses are indicated in Table II. The energy balance established on the combustion can be expressed as:

$$\dot{m}_{air}.cp_{air}.T_2 + \dot{m}_{fuel}.PCI + \dot{m}_{fuel}.cp_{fuel}.T_f = \left(\dot{m}_{air} + \dot{m}_{fuel}\right).cp_{gas}.T_3 \quad (5)$$

where T_3 is the hot gas temperature leaving the combustion chamber.

$$T_3 = \frac{\dot{m}_{air}.cp_{air}.T_{2r} + \dot{m}_{fuel}.PCI + \dot{m}_{fuel}.cp_{fuel}.T_f}{\left(\dot{m}_{air} + \dot{m}_{fuel}\right).cp_{gas}} \quad (6)$$

The specific heat of hot gases, Cp_{gas}, is given according to the following equation [3]:

$$cp_{gas} = 1.8083 - 2.3127.10^{-3}.T_3 + 4.045.10^{-6}.T_3^2 - 1.7363.10^{-9}.T_3^3 \quad (7)$$

The heat power provided by the combustion chamber is given by:

$$\dot{Q}_h = \dot{m}_{fuel}.PCI.R_h \quad (8)$$

3. Turbine

The isentropic outlet temperature of the turbine is determined by:

$$T_{4is} = T_3/(PR^{\frac{kg-1}{kg}}) \quad (9)$$

The real turbine outlet temperature T_4 is determined by:

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

The total mass flow rate through the turbine is given by:

$$\dot{m}_{tot} = \dot{m}_{air} + \dot{m}_{fuel} \quad (11)$$

The real power produced by the turbine is:

$$\dot{W}_t = \dot{m}_{tot}.cp_{gas}.(T_3 - T_4).R_t \quad (12)$$

The net power of the GT power plant is expressed as:

$$\dot{W}_{net} = \dot{W}_t - \dot{W}_{comp,air} \quad (13)$$

The thermal efficiency of the GT power plants defined as:

$$\eta_{cycle} = \frac{\dot{W}_{net}}{\dot{Q}_h} = \frac{\dot{W}_t - \dot{W}_{comp,air}}{\dot{Q}_h} \quad (14)$$

B. Pollutant Gases Formation

As indicated earlier, the combustion process produces different pollutant gases such nitrogen oxides (NO_x), carbon monoxide (CO), and unburned hydrocarbon (UHC).

NO_x are present in all combustion processes. They arise from the oxidation of the nitrogen contained in the air, as well as from the organic nitrogen contained in the fuel. Their formation process requires high temperatures [7]. Consequently, one of the possibilities to reduce the NO_x concentration is to maintain the temperature of the burner and the metal surfaces inside the combustion chamber as low as possible. This can be assured using several methods. The water steam injection in the combustion chamber is one of the processes used to reduce the NO_x release [8]. Other methods are also investigated in the purpose to decrease the NO_x emission in the GT power plants such as Dry Low NO_x combustor [9] and reburning process [10]. The CO and UHC emissions are due to the incomplete combustion reaction.

The main parameters affecting the pollutant gas emissions form GT power plant are the equivalence ratio and the primary zone temperature.

C. Equivalence Ratio

The equivalence ratio ϕ is defined by the ratio of real fuel-air ratio to the stoichiometric fuel-air ratio. That can be expressed as:

$$\phi = \frac{\left(\frac{\dot{m}_{fuel}}{\dot{m}_{air}}\right)actual}{\left(\frac{\dot{m}_{fuel}}{\dot{m}_{air}}\right)\text{stoichiometric}} \quad (15)$$

D. Primary Zone Temperature

Primary zone temperature Tpz is the important parameter which affects directly the GT emissions. It represents the higher temperature value reached in the combustor. Primary zone temperature is expressed by [2]:

$$T_{PZ} = A.\sigma^{\alpha}.e^{\ (\beta(\sigma+\lambda)^2)}.(\frac{P_3}{P_0})^{x^X}.(\frac{T_3}{T_0})^{y^X}.(\frac{H}{C})^{z^X} \quad (16)$$

with

$$\sigma = \phi \text{If} \phi \leq 1$$

$$\sigma = \phi - 0.74 \ \text{ If } \phi \geq 1$$

$$x^X = a_1 + b_1.\sigma + c_1.\sigma^2$$

$$y^X = a_2 + b_2.\sigma + c_2.\sigma^2$$

$$z^X = a_3 + b_3.\sigma + c_3.\sigma^2$$

and $\frac{\mathrm{H}}{\mathrm{C}}$ is the hydrogen carbon ratio of fuel.

The values of the different parameters indicated in (16) are given in Table III for different ranges of the equivalence ratio and the ratio $\left(\frac{\mathrm{T}_3}{\mathrm{T}_0}\right)$ [2].

E. NO_x Emission Model

The NO_x release rate can be calculated according the equivalence ratio and the residence time as [5], [2]:

For $\phi \leq 1.08$

$$NO_x = 1^{13}.(\mathrm{P}_3 . \frac{10^{-5}}{1.4^6})^{\mathrm{aa}} \,\mathrm{e}^{(-\frac{71442}{\mathrm{T_{pz}}})}.(7.56.\phi^{-7.2} - 1.6).\tau^{0.64} \quad (17)$$

For $\phi > 1.08$

$$NO_x = 1^{13}.(\mathrm{P}_3 . \frac{10^{-5}}{1.4^6})^{\mathrm{aa}} \,\mathrm{e}^{(-\frac{71442}{\mathrm{T_{pz}}})}.\left(5.21.\phi^{-2.99} - 1.6\right) * \tau^{0.64} \quad (18)$$

where $aa = 11.949 \,\mathrm{e}^{(-\phi/5.76)} - 10{,}0$

F. CO Emission Model

Carbon monoxide emission rate is given by:

$$CO = \exp\left(-\frac{CE}{T_{pz}}\right).cph.(\frac{P_3}{1.4^6})^{a1}.(\frac{\tau}{0.5})^{a2} \quad (19)$$

where:

$$CE = 6.23^4.\emptyset^{3.8}.\exp((\frac{-\emptyset}{0.56}))^{1.75}$$

$$cph = 4.54^3.\emptyset^4.\exp((\frac{-\emptyset}{1.02}))^{2.23}$$

$$a1 = -0.0447.\emptyset^{-1.87} + 0.2$$

$$a2 = -0.362.\emptyset^{-1.9} + 0.2$$

TABLE III
PARAMETER VALUES FOR TPZ 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

G. UHC Emission Model

Unburned hydrocarbon emission rate can be determined by [8]:

$$UHC = \frac{0.755^{11} * epx(9756/T_{PZ})}{P_3{}^{2.3} * \tau^{0.3} * (\Delta P_3/P_3)^{0.6}} \quad (20)$$

IV. RESULTS AND DISCUSSIONS

The cycle considered for the numerical simulation is shown in Fig. 2. To analyze the influence of the main operating variables on the cycle performances and on the pollutant gas releases, a simulation code is established using the software EES (Engineering Equation Solver) taking into consideration the real operating condition of the GT power plant which was previously mentioned.

Taking into consideration the fact that the power plant performances are significantly affected by the ambient temperature, the operating conditions of the considered cycle will be analyzed for local ambient temperature which varies from 7 to 43 °C. Fig. 3 depicts the variation of the cycle efficiency according to the ambient temperature, while Fig. 4 illustrates the variation of the primary zone temperature and the NO_x emission rate versus T_1. 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 a variation of the ambient temperature equal to 36 °C, the cycle efficiency is reduced by about 7%.

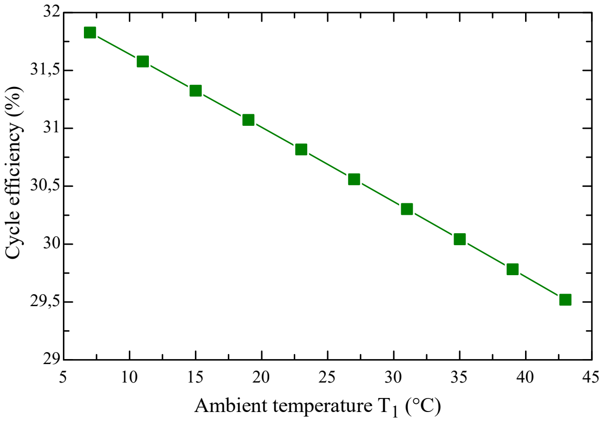

Fig. 3 Variation of cycle efficiency with ambient temperature

Fig. 4 shows the influence of ambient temperature on the NO_x emission rate and on the primary zone temperature. As a result, the variation of ambient temperature for 36 °C leads to increase the NO_x emission rate by 40 ppm and the primary zone temperature of about 18 °C.

The variation of unburned hydrocarbon emission rate with the variation of ambient temperature is presented in Fig. 5. The UHC formation decreases linearly with T_1 by about 5%.

The decrease of UHC release can be explained by the fact that rising of input temperature T_1 leads to the increase of combustion temperature. That promotes a complete combustion reaction.

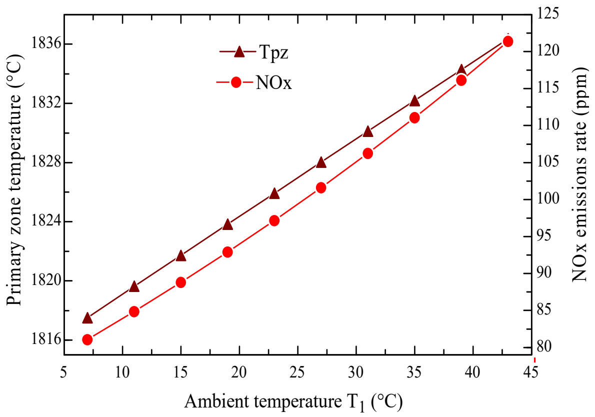

Fig. 4 The affection of ambient temperature on NO_x emission and primary zone temperature

In the purpose to analyze the residence time effect on pollutant gas release, a chosen value of air intake temperature is 10 °C. Fig. 6 depicts the variations of UHC and NO_x emission rates according to the residence time. The equivalence ratio is taken equal to 0.35.

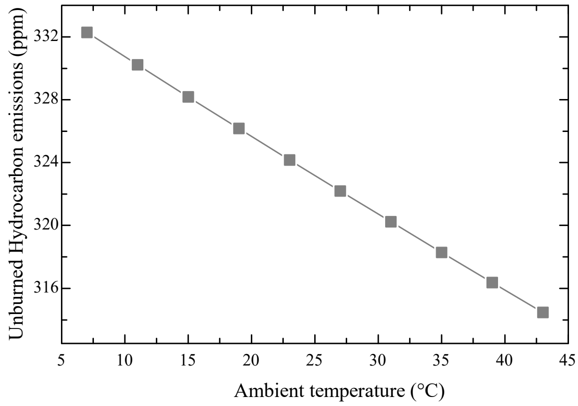

Fig. 5 Variation of UHC emissions with ambient temperature

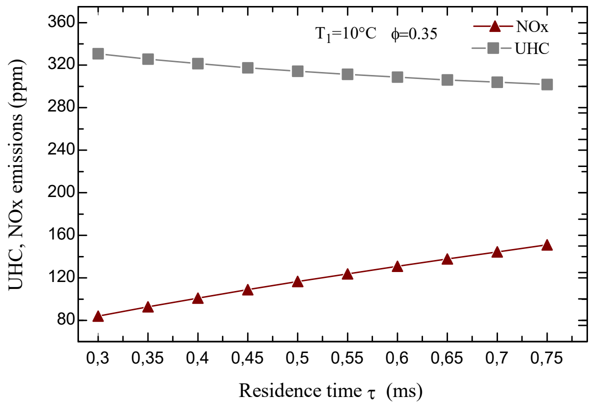

Fig. 6 Influence of residence time on UHC and NO_x emission rate

The UHC emission decreases slightly, while the NO_x formation rate increases sensibly with τ. As indicated earlier, the NO_x release rises with T_1. The decrease of UHC emissions is about 8.7%. Indeed, more residence time indicates more suitable conditions for fuel burning. For the same variation range of the residence time, the NO_x formation increases by 79%.

The CO emission variation according to the residence time is presented in Fig. 7. A notably decrease of about 89% is found. This can be explained by the fact that the important residence time ensures a complete combustion reaction. Consequently, the CO formation is reduced.

Fig. 8 depicts the variation of NO_x, UHC, and CO emission rates according to the equivalence ratio ϕ.

The NO_x release rate increases to reach a maximum value of 11000 ppm for ϕ equal 0.84 then decreases sensibly to be practically negligible for ϕ higher than 1.3. This can be explained by the fact that, for important values of the equivalence ratio, the air excess which is considered for the combustion reaction becomes significant. That leads to the decrease of the primary zone temperature and consequently the decrease of NO_x release rate.

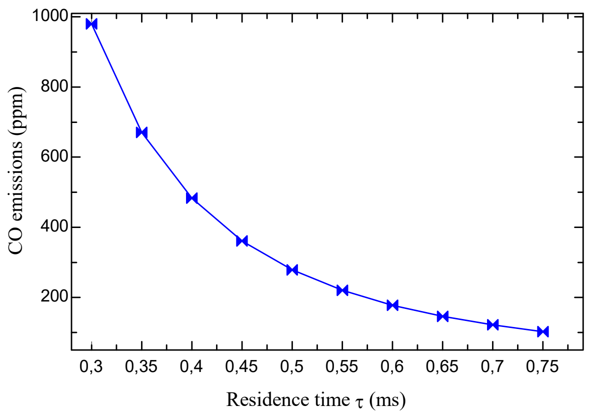

Fig. 7 The influence of CO emissions with residence time

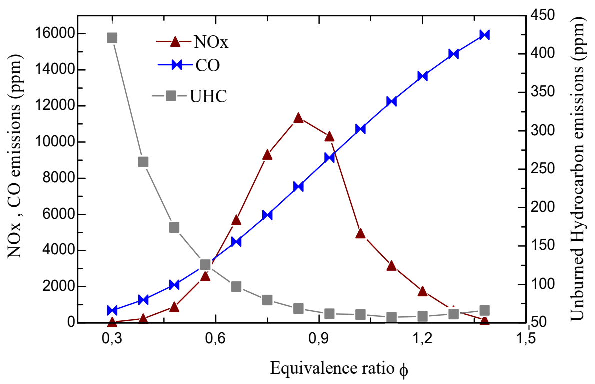

Fig. 8 The influence of equivalence ratio with NO_x, CO, and UHC emissions

To analyze the effect of pressure ratio on the cycle efficiency and gas releases, the residence time and equivalence ratio are taken equal to 0.3 and 0.35, respectively. Fig. 9 illustrates the variation of NO_x emissions rates and cycle efficiency with the variation of pressure ratio. As shown, both cycle efficiency and NO_x emissions increase with the increase of pressure ratio.

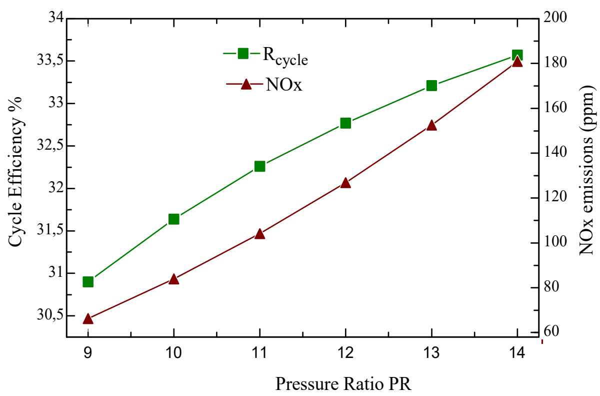

Fig. 9 The variation of cycle efficiency and NO_x emissions with pressure ratio

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
pz	primary zone
t	Turbine
Inlet compressor	
Outlet compressor	
Outlet combustor	
Outlet turbine	
Greek symbols	
Φ	Equivalence ratio
η	Efficiency
τ	Residence time

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