

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

Farhat Hajer, Khir Tahar, Ammar Ben Brahim

**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  $T_{pz}$  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  $NO_x$  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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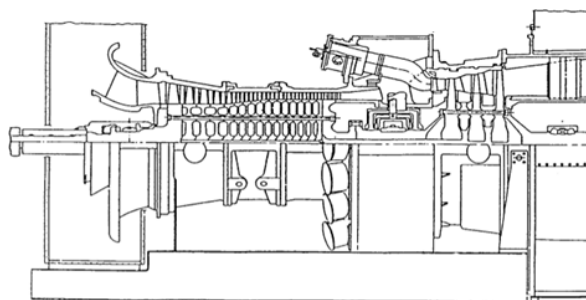


Fig. 1 GT design [5]

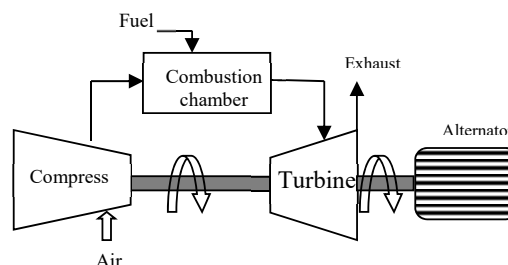


Fig. 2 GT cycle

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:

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

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

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

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

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

$$cp_{air} = 1.0189 \cdot 10^3 - 0.1378 \cdot T_a + 1.9843 \cdot 10^{-4} \cdot T_a^2 + 4.2399 \cdot 10^{-7} \cdot T_a^3 - 3.7632 \cdot 10^{-10} \cdot T_a^4 \quad (4)$$

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

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

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

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

$$T_3 = \frac{\dot{m}_{air} \cdot cp_{air} \cdot T_2 + \dot{m}_{fuel} \cdot PCI + \dot{m}_{fuel} \cdot cp_{fuel} \cdot T_f}{(\dot{m}_{air} + \dot{m}_{fuel}) \cdot cp_{gas}} \quad (6)$$

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

$$cp_{gas} = 1.8083 - 2.3127 \cdot 10^{-3} \cdot T_3 + 4.045 \cdot 10^{-6} \cdot T_3^2 - 1.7363 \cdot 10^{-9} \cdot T_3^3 \quad (7)$$

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

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

## 3. Turbine

The isentropic outlet temperature of the turbine is determined by:

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

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:

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

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  $\phi$  is defined by the ratio of real fuel-air ratio to the stoichiometric fuel-air ratio. That can be expressed as:

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

*D. Primary Zone Temperature*

Primary zone temperature  $T_{pz}$  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]:

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

with

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

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

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

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

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

and  $\frac{H}{C}$  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  $\left(\frac{T_3}{T_0}\right)$  [2].

*E. NO<sub>x</sub> Emission Model*

The NO<sub>x</sub> release rate can be calculated according the equivalence ratio and the residence time as [5], [2]:

For  $\phi \leq 1.08$

$$NO_x = 1^{13} \cdot (P_3 \cdot \frac{10^{-5}}{1.46})^{aa} e^{(-\frac{71442}{T_{pz}})} \cdot (7.56 \cdot \phi^{-7.2} - 1.6) \cdot \tau^{0.64} \quad (17)$$

For  $\phi > 1.08$

$$NO_x = 1^{13} \cdot (P_3 \cdot \frac{10^{-5}}{1.46})^{aa} e^{(-\frac{71442}{T_{pz}})} \cdot (5.21 \cdot \phi^{-2.99} - 1.6) \cdot \tau^{0.64} \quad (18)$$

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

*F. CO Emission Model*

Carbon monoxide emission rate is given by:

$$CO = \exp\left(-\frac{CE}{T_{pz}}\right) \cdot cph \cdot \left(\frac{P_3}{1.46}\right)^{a1} \cdot \left(\frac{\tau}{0.5}\right)^{a2} \quad (19)$$

where:

$$CE = 6.23^4 \cdot \phi^{3.8} \cdot \exp\left(\left(\frac{-\phi}{0.56}\right)\right)^{1.75}$$

$$cph = 4.54^3 \cdot \phi^4 \cdot \exp\left(\left(\frac{-\phi}{1.02}\right)\right)^{2.23}$$

$$a1 = -0.0447 \cdot \phi^{-1.87} + 0.2$$

$$a2 = -0.362 \cdot \phi^{-1.9} + 0.2$$

TABLE III  
PARAMETER VALUES FOR TPZ DETERMINATION

Parameter	$0.3 \leq \phi \leq 1.0$		$0.3 \leq \phi \leq 1.0$	
	$0.92 \leq \left(\frac{T_3}{T_0}\right) \leq 2$	$2 \leq \left(\frac{T_3}{T_0}\right) \leq 3.2$	$0.92 \leq \left(\frac{T_3}{T_0}\right) \leq 2$	$2 \leq \left(\frac{T_3}{T_0}\right) \leq 3.2$
A	2361.76	2315.75	916.82	1246.17
$\alpha$	0.1157	-0.049	0.288	0.381
$\beta$	-0.948	-1.114	0.145	0.347
$\lambda$	-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]:

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

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  $\text{NO}_x$  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  $T_1$ . For a variation of the ambient temperature equal to  $36^\circ\text{C}$ , the cycle efficiency is reduced by about 7%.

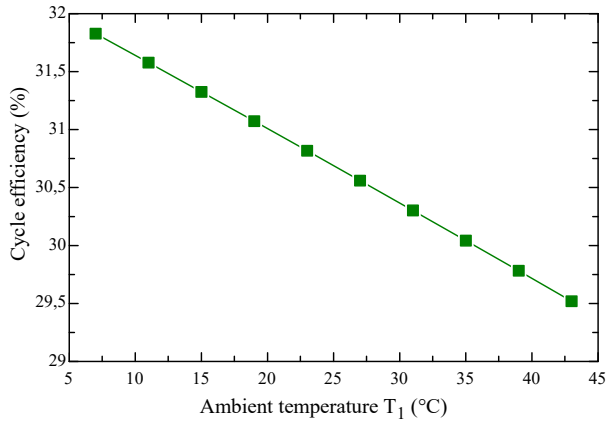


Fig. 3 Variation of cycle efficiency with ambient temperature

Fig. 4 shows the influence of ambient temperature on the  $\text{NO}_x$  emission rate and on the primary zone temperature. As a result, the variation of ambient temperature for  $36^\circ\text{C}$  leads to increase the  $\text{NO}_x$  emission rate by 40 ppm and the primary zone temperature of about  $18^\circ\text{C}$ .

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.

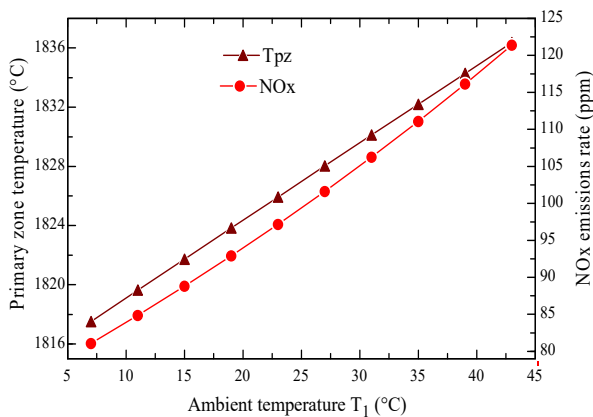


Fig. 4 The affection of ambient temperature on  $\text{NO}_x$  emission and primary zone temperature

In the purpose to analyze the residence time effect on pollutant gas release, a chosen value of air intake temperature is  $10^\circ\text{C}$ . Fig. 6 depicts the variations of UHC and  $\text{NO}_x$  emission rates according to the residence time. The

equivalence ratio is taken equal to 0.35.

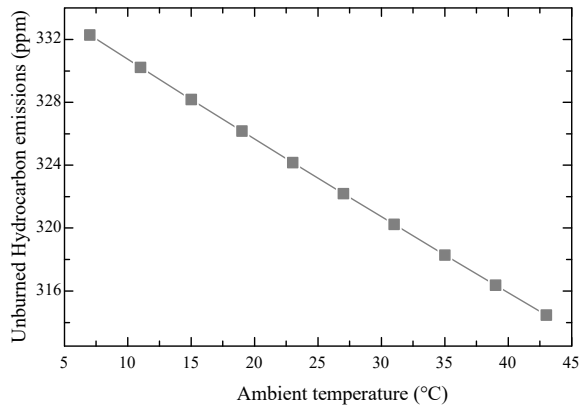


Fig. 5 Variation of UHC emissions with ambient temperature

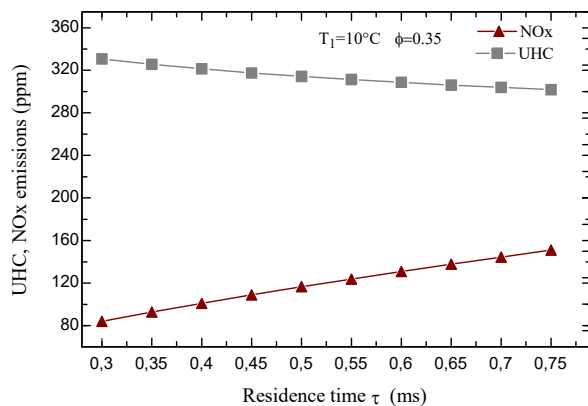


Fig. 6 Influence of residence time on UHC and  $\text{NO}_x$  emission rate

The UHC emission decreases slightly, while the  $\text{NO}_x$  formation rate increases sensibly with  $\tau$ . As indicated earlier, the  $\text{NO}_x$  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  $\text{NO}_x$  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  $\text{NO}_x$ , UHC, and CO emission rates according to the equivalence ratio  $\phi$ .

The  $\text{NO}_x$  release rate increases to reach a maximum value of 11000 ppm for  $\phi$  equal 0.84 then decreases sensibly to be practically negligible for  $\phi$  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  $\text{NO}_x$  release rate.

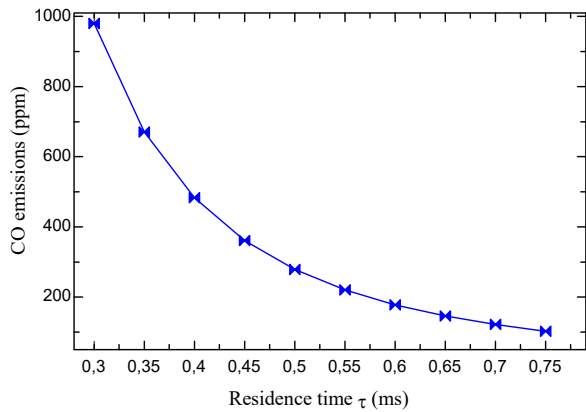


Fig. 7 The influence of CO emissions with residence time

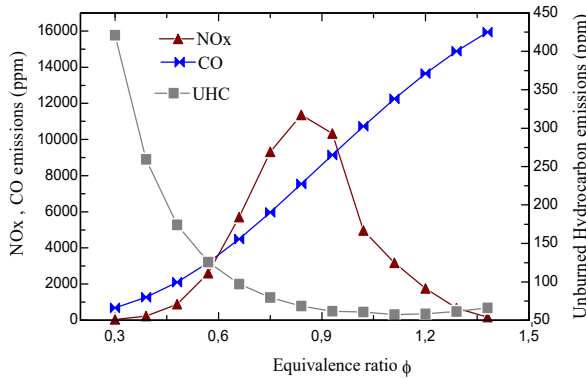


Fig. 8 The influence of equivalence ratio with NO<sub>x</sub>, CO, and UHC emissions

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<sub>x</sub> emissions rates and cycle efficiency with the variation of pressure ratio. As shown, both cycle efficiency and NO<sub>x</sub> emissions increase with the increase of pressure ratio.

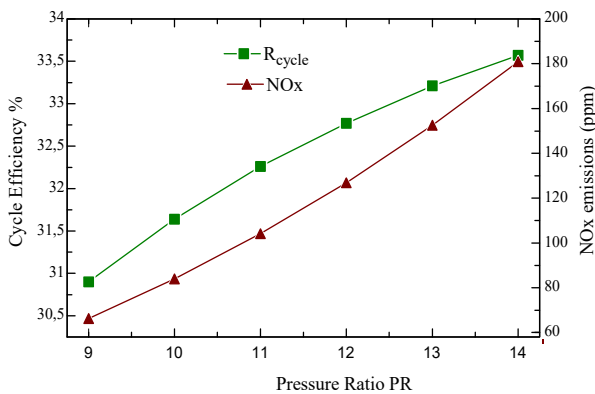


Fig. 9 The variation of cycle efficiency and NO<sub>x</sub> emissions with pressure ratio

NOMENCLATURE

C <sub>p</sub>	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
<i>Subscripts</i>	
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	
<i>Greek symbols</i>	
$\Phi$	Equivalence ratio
$\eta$	Efficiency
$\tau$	Residence time

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