ISSN 2176-5480

22nd International Congress of Mechanical Engineering (COBEM 2013)
November 3-7, 2013, Ribeirão Preto, SP, Brazil

ANALYSIS OF THE EFFICIENCY OF A GAS TURBINE USING NATURAL GAS AS FUEL

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Abstract. *This work is an evaluation of gas turbine (GT) using natural gas. A model was made by real data from a thermoelectric in northeast of Brazil. The modeling was developed for the main components using mass balance, energy balance (first law of thermodynamics) and first law of thermodynamics for systems reagents. Some parameters were determinated such as: Overall efficiency of cycle, compressor and turbine efficiencies, air/fuel ratio, the maximum temperature of the cycle. The effect of pressure ratio, the air / fuel ratio, specific mass of atmospheric air, maximum temperature of the cycle and heat lost in the turbine were discussed.*

Keywords: *gas turbine, natural gas, turbine efficiency, first law of thermodynamics.*

1. INTRODUCTION

The global demand energy is constantly growing and research sources of energy affordable and efficient are important. According to Najjar (2000) the gas turbine is known for low capital cost, high flexibility, high reliability, short delivery time.

The use of gas turbines will has an increasingly important role in the generation of electricity (Stambler, 1989 apud Najjar, 1996). For Brazil gas turbines operate in a manner complementary to hydroelectric, primarily during times of droughts.

The simple cycle gas turbine, the compressor may need to 40-80% of work turbine. This is an important factor when considering the real cycle, because the lower the work required by the compressor higher useful work of the turbine.

At International Standard Organization (ISO) under ambient conditions, the efficiency of the natural gas turbine is typically around 30% (Cohen et al., 1987 apud Popli et al., 2012). However, the energy efficiency of the gas turbine is hampered by high temperature ambient and relative humidity (RH), because the turbines are constant volume machines, the power decreases when the air density and the mass flow rate decreases, this because of increase of temperature ambient (Popli et al., 2012).

De Sa and Al Zubaidy (2011) described that for each increase of one degree in temperature above ISO conditions the Gas Turbine loses 0.1% in terms of thermal efficiency and 1.47 MW of its (useful) Power Output.

This work was performed an analysis of the efficiency of the thermal cycle; efficiency of the compressor and turbine. The data were obtained from a natural gas turbine installed in northeastern Brazil. A model was developed to reach objectives. A previous discussion of their behavior with the variation of temperature, pressure and air/fuel ratio was also conducted.

2. METHODOLOGY

The gas turbine study has three main equipments for their operations are: compressor, combustion chamber and turbine. The atmospheric air comes into the compressor through the intake and it is conducted at the combustion chamber. Natural gas is mixed with the compressed air in chamber combustion. The combustion gases leave the chamber with maximum temperature of the cycle (T_4). The combustion gases at the point (4) are expanded in the turbine, converting thermal energy into work shaft, which is converted into electrical energy through the generator. Figure 1 shows a schematic representation of the gas turbine.

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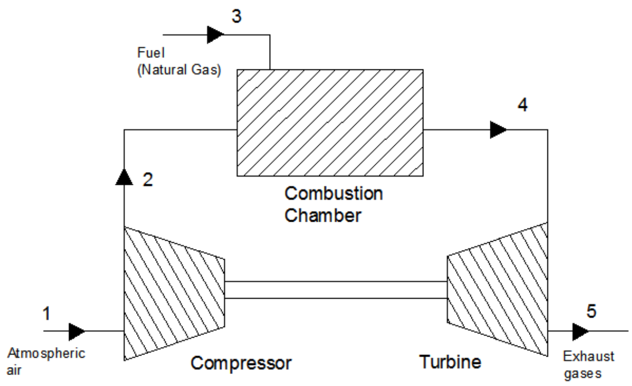

Figure 1 - Schematic representation of the gas turbine

According to Boyce (2006) power output and efficiency of the actual cycle are considerably smaller compared to the ideal cycle. The simple cycle has real open compressor inefficiencies (η_c) and turbine (η_T). The gas turbine was developed using the relationships:

2.1. Compressor

The compressor was considered adiabatic. His energy balance was described as:

$$\dot{m}_1 h_1 = \dot{m}_2 h_2 + \dot{W}_c \tag{1}$$

The efficiency of the compressor was defined for the equation below:

$$\eta_c = \frac{\dot{W}_{cs}}{\dot{W}_c} = \frac{(h_{2s} - h_1)}{(h_2 - h_1)} \tag{2}$$

Where $\dot{m}_1$ and $\dot{m}_2$ are the mass flows rates of air before and after being compressed; h_1 and h_2 are the enthalpies of the state of the gas; $\dot{W}_c$ is the work consumed by the compressor and the subscript s is for the isentropic.

A model for efficiencies isentropic of compressor was cited by Wilson (1984 apud Najjar, 1996), represented by equation below:

$$\eta_{cs} = 1 - [0.04 + \frac{(r_c - 1)}{150}] \tag{3}$$

2.2 Combustion Chamber

The combustion chamber is where fuel is added with compressed air. The composition of natural gas is important for modeling of the combustion chamber. It was considered composition according to table 1.

Table 1 - Composition and Typical Properties of Natural Gas (Source: Santos, 2005)

Component	% Volume
Methane	88.82
Ethane	8.41
Propane	0.55
Nitrogen	1.62
Carbon dioxide	0.6
Density relative to air	0.62
Higher Calorific value (kJ/m³)	39355.9
Lower Calorific value (kJ/m³)	35169.1

Some hypotheses were adopted:

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- The efficiency of the combustion chamber (η_{cc}) is 99%.
- The proportion of each mole of oxygen is involved 79.0/21.0 moles of nitrogen (Van Wylen et al., 2003).

The efficiency of combustion chamber was related with his heat lost

$$\eta_{cc} = \frac{\dot{m}_3 .LHV - \dot{Q}_{loss,CC}}{\dot{m}_3 .LHV} \tag{4}$$

Where $\dot{m}_3$ is the mass flow rate of fuel; LHV is the lower calorific value of natural gas and $\dot{Q}_{loss,CC}$ is the heat loss in the combustion chamber.

The lower heating value (LHV) of the fuel is important for evaluating the efficiency of the first law of thermodynamics and heat exchanged in the combustion chamber. It is determinate as the follow equation:

$$LHV = \sum_P n_{out}\left(\bar{h}_f^0 + \Delta\bar{h}\right)_{out} - \sum_R n_{in}\left(\bar{h}_f^0 + \Delta\bar{h}\right)_{in} \tag{5}$$

Where n is the number of moles of the combustion reaction; $\bar{h}_f^0$ is the enthalpy of formation and $\Delta\bar{h}$ is the variation of enthalpy of formation.

As determined by enthalpy of formation shown in Eq. (6) represents the amount of heat transferred in the chamber during combustion at constant temperature and water in the product in the vapor state (Van Wylen et al., 2003).

The combustion reaction was determinate for composition of natural gas

$$[(0.8882)\mathrm{CH_4} + (0.0841)\mathrm{C_2H_6} + (0.0055)\mathrm{C_3H_8} + (0.0162)\mathrm{N_2} + (0.006)\mathrm{CO_2}] + [(\varphi X)(\mathrm{O_2} + 3.76\,\mathrm{N_2})] \longrightarrow \mathrm{CO_2} + 2.\mathrm{H_2O} + (\varphi)3.76\,\mathrm{N_2} + (\varphi - 1)X.\mathrm{O_2} \tag{6}$$

The important parameter related with excess air is the theoretical air

$$\varphi = \frac{\mathrm{AC_s}}{\mathrm{AC}} \tag{7}$$

Where AC is the air fuel ratio and the subscript s is stoichiometric.

The stoichiometric of the reaction provides the theoretical quantity of air. Where φ is the theoretical air quantity and X is the number of moles of air for complete combustion without excess (Van Wylen et al., 2003).

The maximum temperature of the cycle (T_4) is at the entrance of the turbine. This temperature has been the primary means used to improve the efficiency of the gas turbine. Historically there has been increased mainly due to the development of new materials more resistant to high temperatures (Çengel & Boles, 2006).

The maximum temperature of the cycle was determined by the first law of thermodynamics to reagent systems at steady state. The temperature of air inlet (T_2) and the fuel (T_3) were considerate at the determination (Van Wylen et al., 2003).

$$\frac{\dot{Q}_{loss,CC}}{\dot{m}_3} + \sum_R n_{in}\left(\bar{h}_f^0 + \Delta\bar{h}\right)_{in} = \sum_P n_{out}\left(\bar{h}_f^0 + \Delta\bar{h}\right)_{out} \tag{8}$$

2.3 Turbine

The Turbine power is determined as:

$$\dot{W}_T = \dot{W}_{Liq} + \dot{W}_C \tag{9}$$

The net power ($\dot{W}_{Liq}$) is a given real gas turbine.

Heat lost in the turbine

$$\frac{\dot{Q}_T}{\dot{m}_3} + \sum_R n_{in}\left(\bar{h}_f^0 + \Delta\bar{h}\right)_{in} = \sum_P n_{out}\left(\bar{h}_f^0 + \Delta\bar{h}\right)_{out} + \frac{\dot{W}_T}{\dot{m}_3} \tag{10}$$

Turbine efficiency was defined as:

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$$\eta_T = \frac{\dot{W}_T}{\dot{W}_{Ts}} \tag{11}$$

The isentropic work is determinate as:

$$\dot{W}_{Ts} = c_{p,gases}.\dot{m}_4(T_4 - T_{5s}) \tag{12}$$

Where T_{5s} and $\dot{W}_{Ts}$ temperature and turbine work isentropic respectively; R_{gases} constant flue gas that is directly related to the universal gas constant ideal (R); $c_{p,gases}$ specific heat at constant pressure of the exhaust gases and k_{gases} being the ratio of specific heats for the exhaust gases.

The state isentropic is defined as the relation of ideal gases:

$$\left(\frac{P_4}{P_5}\right)^{\left(k_{gases}-1/k_{gases}\right)} = \frac{T_4}{T_{5s}} \tag{13}$$

The isentropic constant of exhaust gases were defined by the relation of the specific heat of exhaust gases:

$$k_{gases} = \frac{c_{p,gases}}{c_{p,gases} - R_{gases}} \tag{14}$$

The specific heat of exhaust gases were determinate by the first law of thermodynamic

$$\dot{Q}_T + \dot{m}_4.c_{p,gases}.(T_4 - T_5) = \dot{W}_T \tag{15}$$

A model for efficiency isentropic for turbine was cited by Wilson (1984 apud Najjar, 1996), represented by equation below:

$$\eta_{ts} = 1 - [0.03 + \frac{(r_t - 1)}{180}] \tag{16}$$

2.4 Thermal efficiency of the cycle

The logical way to evaluate the performance of an internal combustion engine is to compare the actual work performed with the maximum work to be carried out in a reversible state change in accordance with the second law of thermodynamics. However, in practice, is defined as the ratio of the actual work, and the value of the enthalpy of combustion, defined as thermal or thermal yield efficiency (Van Wylen et al., 2003).

$$\eta_{Cycle} = \frac{\dot{W}_{Liq}}{\dot{m}_3.LHV} \tag{17}$$

According to Boyce (2006) the thermal efficiency of the cycle is increased with the maximum temperature and pressure ratio. When higher the temperature of combustion gases, higher the cycle efficiency, moreover the pressure ratio increase improves the cycle efficiency until a certain value, after the cycle efficiency starts reducing. This effects can be showed at figure below.

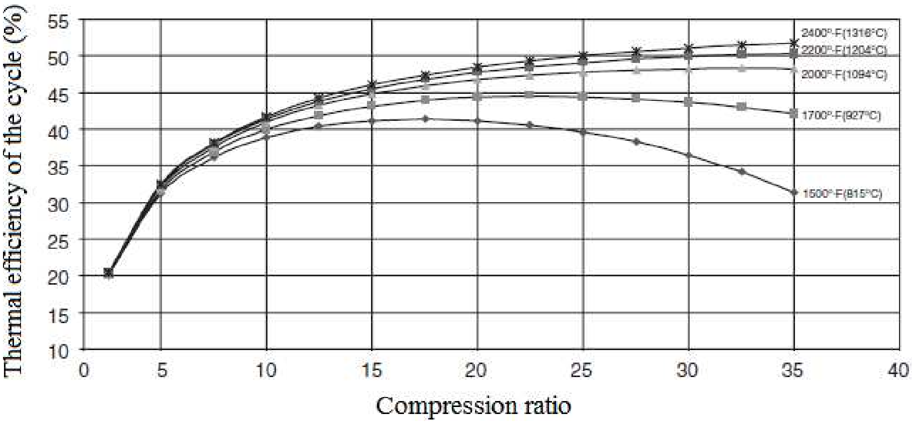

Figure 2 - The thermal efficiency of the cycle relative to pressure ratio based on compressor efficiency of 87% (Source: Boyce, 2006)

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3. RESULTS

Modeling results obtained from the data held actual inlet temperature T_1, pressure ratio r_p and power generated $\dot{W}_{Liq}$ by the gas turbine. These results were graphed for best viewing results.

The figures 3.a and 3.b, demonstrate the relative efficiencies of the compressor and the turbine in relation to the compression ratio realizes that their values approximate the efficiency of Represented by the following eq. 3 and 16.

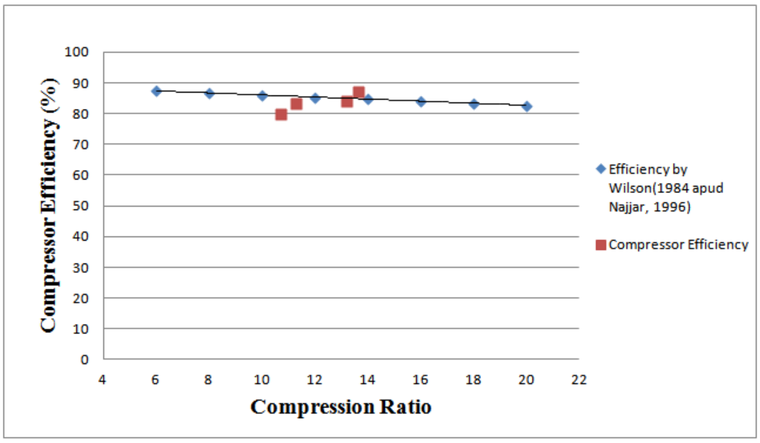

Figure 3a - Compressor Efficiency

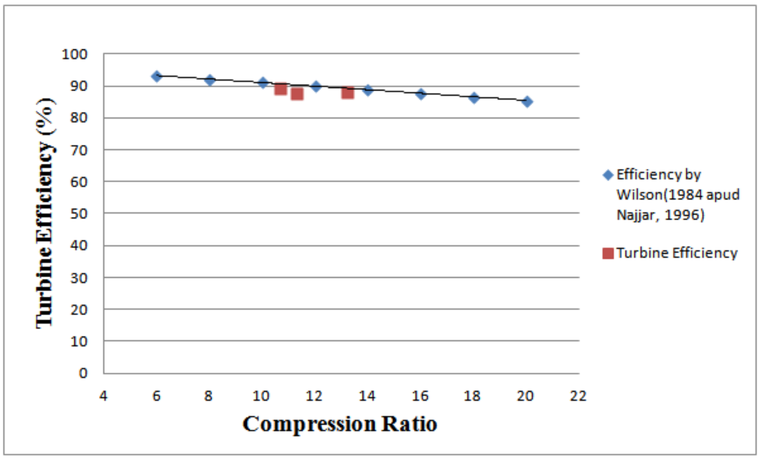

Figure 3b - Turbine Efficiency

The experimental date is closer that theoretical efficiency. The compressor and turbine efficiencies decrease with the higher compression ratio. These performances are similar with efficiency of literature.

The specific mass of atmospheric air impact on the efficiency of the compressor. For as mentioned by Popli et al. (2012), gas turbines are volumetric machines. The higher specific mass of atmospheric air at the compressor intake (at 1), increases air mass, which will be compressed, consequently resulting in increased efficiency and compression ratio of compressor, as shown in fig. 4, 5 and 6.

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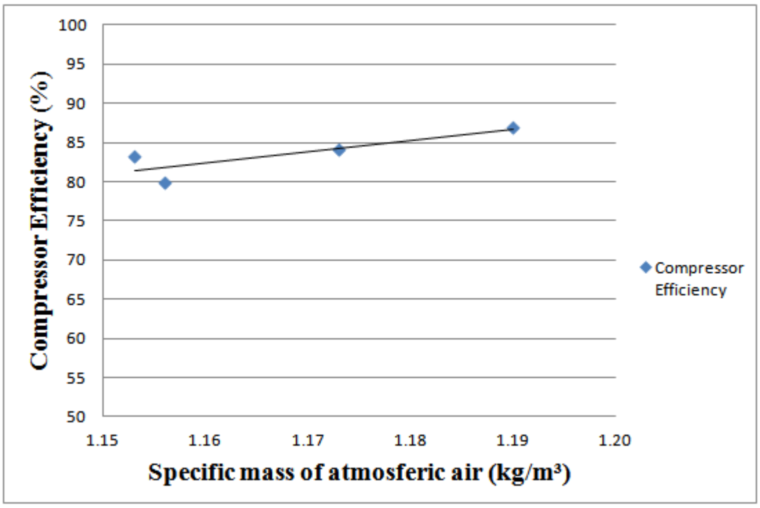

Figure 4 - Compressor efficiency in relation to specific mass of atmospheric air

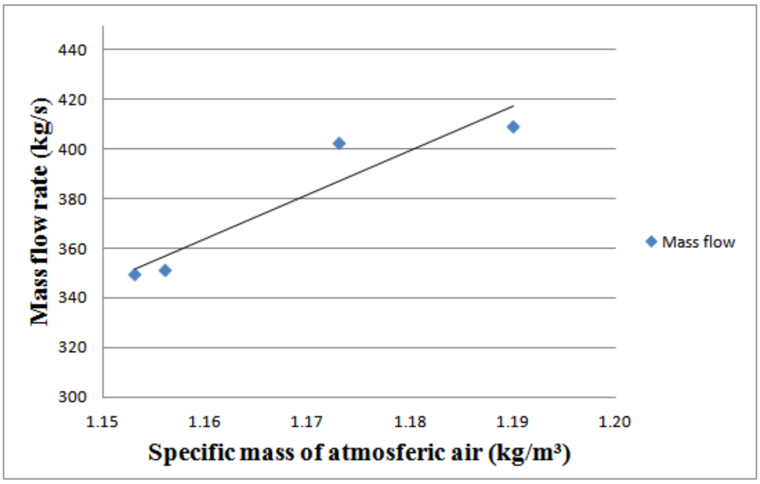

Figure 5 - Mass flow rate in relation to specific mass of atmospheric air

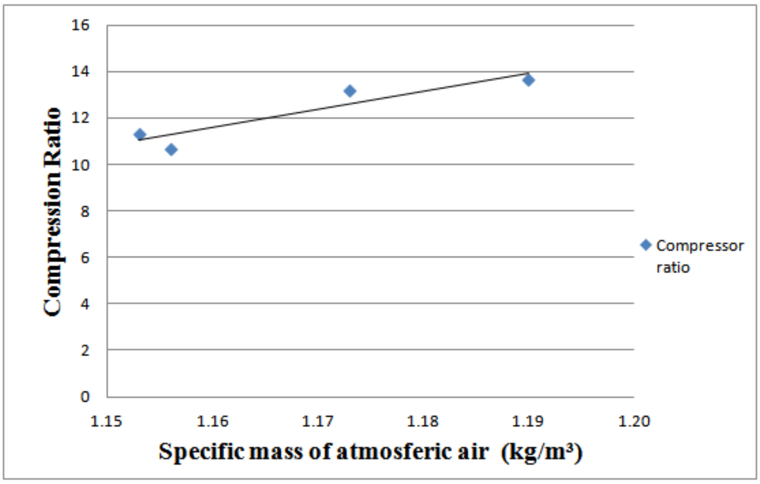

Figure 6 - Compression ratio in relation specific mass of atmospheric air

In Figures 4,5 and 6 increasing 2.94% of the specific mass atmospheric air resulted in an increase of 16.54% of mass flow rate of air, 27.47% of the compression ratio and an increase of 8.72% of the compressor efficiency.

The increased specific mass of atmospheric air is related to temperature ambient. According to Van Wylen (2003) atmospheric air can be modeled as ideal gas.

The increase of specific mass of air increase the pressure ratio however the increase of pressure ratio decrease the compressor and turbine efficiency. Boyce (2006) recommend an optimum pressure ratio for gas turbine cycle.

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The figure 7 and 8 shows the relationship between turbine efficiency and thermal cycle with the maximum temperature of the cycle, respectively.

The maximum temperature of the cycle T_4 seem has low affect at turbine efficiency. In addition, the maximum temperature of exhaust gases is determinate by the fuel flow. Into the figure 7 are the mass fuel flow between the range 7.72 to 9.13 kg/s. Higher fuel flow increases the maximum temperature. The maximum temperature has to be in the safe limit of turbine material. Soon the variation of maximum temperature is low and the effect in the turbine is not significant.

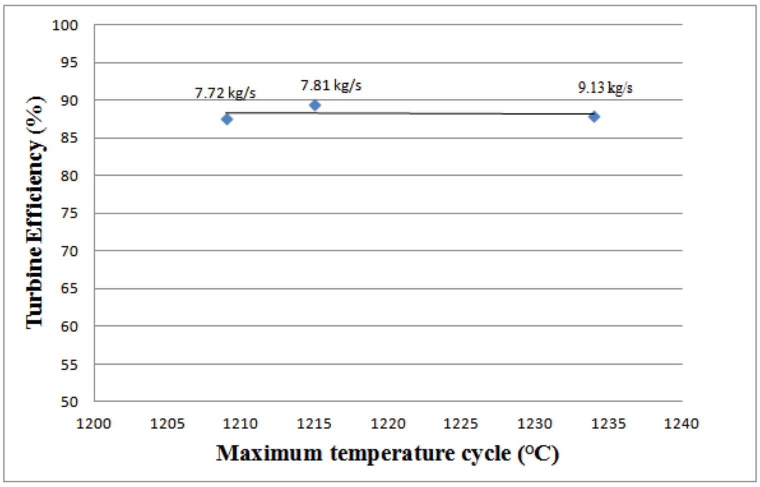

Figure 7 - Turbine efficiency in relation maximum temperature cycle

At figure 8, the maximum temperature of the cycle has influence at cycle efficiency. According the literature, higher exhaust gases temperature have an important influence on the efficiency of the cycle, Boles and Çengel (2006), Van Wylen (2003). However other factors influence the efficiency of cycle as soon as mass flow rate of combustion gases and the power output. Into the fig. 8 are the mass exhaust gases flow and the power output between the ranges 357.5 to 411.5 kg/s and 119.1 to 150.7 MW, respectively. In the higher efficiency of cycle, the mass of gases flow and the net power are higher. The reason is higher fuel flow produces higher gases flow and net power.

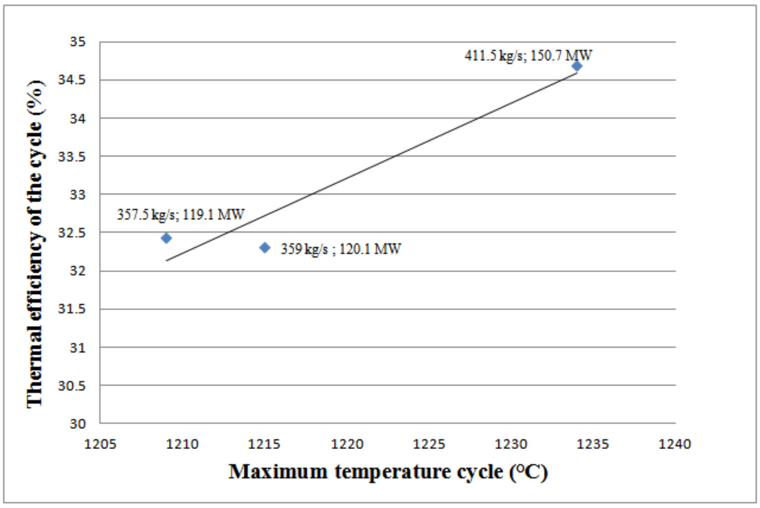

Figure 8 - Thermal efficiency of the cycle in relation maximum temperature cycle

One of the methods to increase the cycle efficiency is decreasing the air/fuel ratio of the gas turbine. The air/fuel ratio is related to the maximum temperature of the cycle, although this value should be controlled so that the maximum temperature of the cycle is not exceeding the limit of the material. In Fig. 9 can perceive the change in cycle efficiency with decreasing air/fuel ratio. The increase of air/fuel ratio, reduce the fuel flow rate for an air rate approximately constant and reducing the power output and therefore reducing the cycle efficiency.

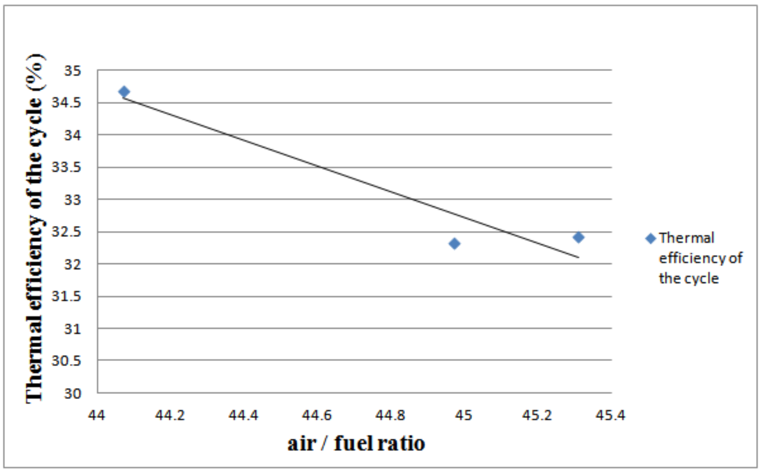

Figure 9 - Thermal efficiency of the cycle in relation air/fuel ratio

The figure 10 shows another factor that influences the maximum temperature of the cycle and consequently the efficiency of the cycle is the heat loss to the environment of the turbine. The higher heat loss to the environment reduces the amount of energy combustion gas and the net power, consequently reducing the cycle efficiency.

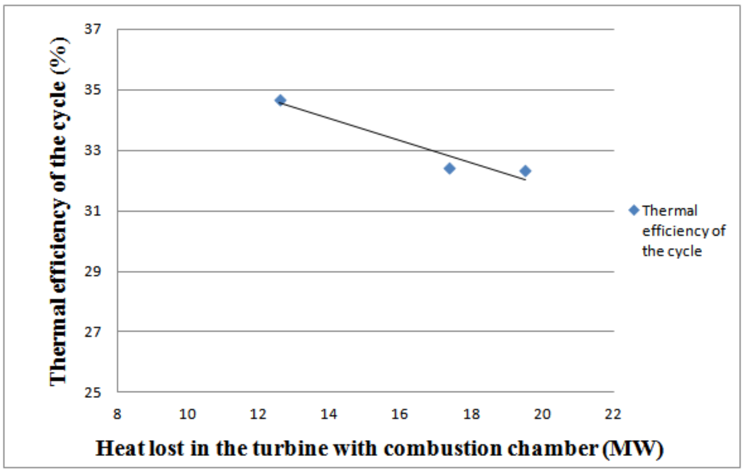

Figure 10 - Thermal efficiency of the cycle in relation heat lost in the turbine with combustion chamber

The increase of pressure ratio for compressor and turbine do not improve the efficiency of them, as showed at figure 3.a and 3.b. Although the increase of pressure ratio cycle influence directly the cycle efficiency. It can be showed at Figure 10.

It is happen for this short range of compression ratio. In the gas turbine cycle has one optimum compression ratio that increase before this value and decrease after that, according Boyce (2006). The range of compression ratio evaluated is before the optimum value. After that the efficiency of cycle will decrease.

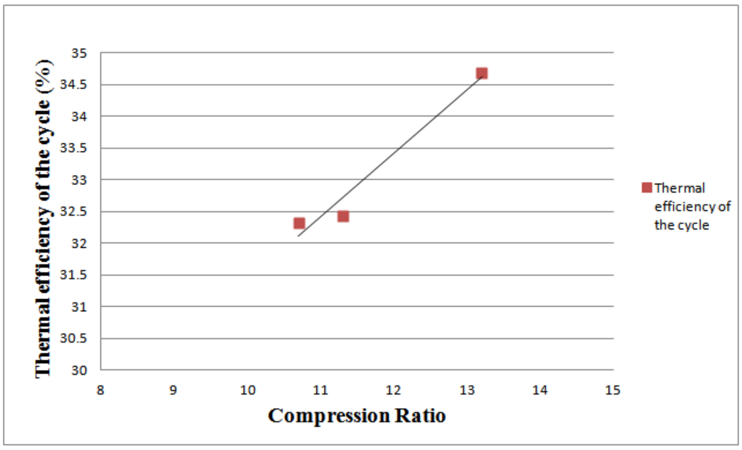

Figure 1 - Thermal efficiency of the cycle in relation compression ratio

ISSN 2176-5480

4. CONCLUSION

A model was developed using mass, energy balance. The parameters were determinate as: air/fuel ratio, maximum temperature of the cycle, overall efficiency of cycle, compressor and turbine efficiencies.

The increased pressure ratio decrease the efficiency of the compressor and turbine, however this will increase the thermal efficiency of the cycle.

The increased specific mass atmospheric air is increased mass flow rate at the compressor inlet, increased compression ratio and improving the efficiency of the compressor.

A small increase in temperature is maximum cycle is not great influence on the turbine efficiency, but the thermal efficiency of the cycle influences the increase is approximately 8%.

A factor for the increase of the maximum temperature is the air/fuel ratio. The reduction of this ratio is increasing the thermal efficiency of the cycle.

The reduction of heat loss from the turbine more combustion chamber increases the thermal efficiency of the cycle.

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