MODELING AND OPTIMIZATION OF A TURBO CHARGED DIESEL ENGINE TO OBTAIN A HIGHER AMOUNT OF ENERGY FROM THE EXHAUST GASES

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Abstract. *This work presents the theoretical results obtained with the analysis and optimization of a turbo charged diesel engine in operational conditions. The main objective is to calculate the higher amount quantity of energy that could be extracted from the exhaust gases of an internal combustion engine, without any modification in the engine and turbo compressor performances. The work was done in two stages, where in the first stage, the mathematical modeling of four main components of the engine and turbo compressor (compressor, intercooler, combustion chamber and turbine) was implemented. The modeling was done based on thermodynamical, physical and black box models, where, most of them were founded in the literature. The second stage is constituted the by system model optimization process. The gases pressure released from turbine was considered as the predominant parameter in the optimization process. This pressure was defined as the variable to be maximized in the optimization and the other variables were defined as the restrictions of the system. The method of Lagrange Multiplier Equations was used, which is based on the calculation of the restrictions gradient. The input parameters to the components of the engine received a statistical treatment. These parameters were treated as constant and later they received different values with the purpose to obtain a several optimal points. The results of this work can be used to design and implement a controller that is possible to maintain the engine in optimum performance conditions, obtaining larger amount of energy from the exhaust gases. As results the optimum range was about 181 to 229 kPa for different engine speeds between 1500 and 2900 rpm.*

Keywords*: modeling, optimization, turbo charged diesel engine, exhaust gases.*

1. INTRODUCTION

Currently, the automotive sector accounts for more than 10% of everything that Brazil generates of wealth. In spite of all economic growth, the sector contributes negatively to the degradation of the environment. The data from the report of Sustainable Development Indicators - Brazil 2010, published by IBGE (Brazilian Institute of Geography and Statistics) show that the emission of greenhouse gases in Brazil rose from 1.35 billion to 2.20 billion tons of CO_2 equivalent between 1990 and 2005, an increase of 62%.

The energy analysis of internal combustion engines (ICE) shows a large inefficiency in the use of energy produced in the process of burning fossil fuels. All energy produced in combustion can be divided into three almost equal terms: the net power, the portion equivalent to the mechanical and thermal losses and finally, the aggregate amount of power on the exhaust gases.

The objective of this work is to calculate the higher amount quantity of energy that could be extracted from the exhaust gases of an internal combustion engine, without any modification in the engine and turbo compressor performances.

A higher efficiency in the use of resources is obtained with the co-generation. For the ICE, the exhaust gases are the co-generation source and auxiliary systems, such as the turbocharger, increase the pressure of air entering the combustion chamber, improving engine performance.

Pitillo (2006) studied the possibility of holding up co-generation of the energy from exhaust gases in a turbocharged diesel ICE. He developed a semi-empirical mathematical model that enabled him to estimate the free power contained in the exhaust fumes, this model was validated with experimental data of a diesel engine found in the literature.

In his work Martins (2004) developed a methodology for modeling of turbo compressors considering the behavior of mass flow and efficiency. He used semi empirical models of the ASHRAE Toolkit (1994), which are applied to obtain the curves of pressure ratio against mass flow and mass flow against efficiency in order to compare with experimental data provided by manufacturers of turbo compressors.

Pimenta et al (2004) brought a viable application for the energy of the exhaust gases. They made the modeling of an absorption cycle driven by the exhaust gases from an ICE for refrigeration of food in a refrigerated truck. The computer simulation was performed on the software EES (Engineer Equation Solver) and they obtained a refrigeration capacity of 3.64 kW.

2. METHODOLOGY

Initially, it was developed a mathematical model which considers the four main components of the ICE. Through the method of Lagrange multipliers has become possible to know the optimal point of operation of the ICE, seeking the maximum amount of energy contained in exhaust gases. The obtained model consists of 40 equations and 49 variables.

All data were obtained from three sources: thermodynamic processes, black box models and parameterizations of equipments.

The behavior of each component studied was taken from manufacturers' catalog, with the exception of the intercooler, whose behavior was estimated by the principle of conservation of energy.

2.1. Mathematical Model

Figure 1 represents the four studied components and points of interest for each process. Thus, the conditions of entry and exit will be found, representing the values of the thermodynamic properties of fluids.

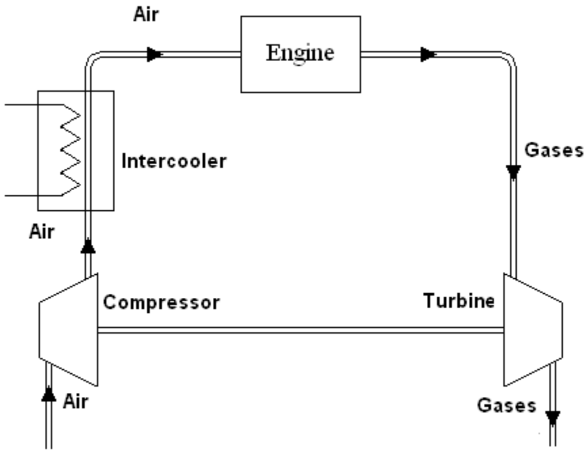

Figure 1. Schematic representation of the engine components.

2.1.1. Compressor

The compressor was modeled using a black box model and the thermodynamic properties. The compressor behavior curve was lent by AlliedSignal Turbocharging Systems, Torrance, California, USA. According to Bermudez (1995) and Pitillo (2006) the efficiency and the compressor pressure ratio can be represented in terms of its rotation speed and mass flow of air. It had been performed two regressions of the curve of the manufacturer, resulting in the Eq. (1) and Eq. (2).

$$\begin{aligned}\eta_{cp} &= -2.88*10^{-5} N_{cp,co} + 2.07*10^{-9} N_{cp,co}^{2} - 4.029*10^{-14} N_{cp,co}^{3} - 6.637*10^{-10} \dot{m}_{ar,co} N_{cp,co}^{2} + \\ & 3.277*10^{-19} N_{cp,co}^{4} - 7.223*10^{-8} \dot{m}_{ar,co}^{2} N_{cp,co}^{2} + 1.177*10^{-13} \dot{m}_{ar,co} N_{cp,co}^{3} - \\ & 1.142*10^{-24} N_{cp,co}^{5} - 1.663*10^{-8} \dot{m}_{ar,co}^{3} N_{cp,co}^{2} + 9.302*10^{-13} \dot{m}_{ar,co}^{2} N_{cp,co}^{3} - \\ & 1.399*10^{-18} \dot{m}_{ar,co} N_{cp,co}^{4} + 1.246*10^{-30} N_{cp,co}^{6} - 4.665*10^{-7} \dot{m}_{ar,co}^{4} N_{cp,co}^{2} + \\ & 1.267*10^{-12} \dot{m}_{ar,co}^{3} N_{cp,co}^{3} - 4.07*10^{-18} \dot{m}_{ar,co}^{2} N_{cp,co}^{4} + 4.766*10^{-24} \dot{m}_{ar,co} N_{cp,co}^{5}\end{aligned} \tag{1}$$

$$\begin{aligned}Rp_{cp} &= 4.6857*10^{-5} N_{cp,co} - 8.4186*10^{-10} N_{cp,co}^{2} + 4.7509*10^{-5} \dot{m}_{ar,co} N_{cp,co} + \\ & 8.9625*10^{-15} N_{cp,co}^{3} - 2.2927*10^{-9} \dot{m}_{ar,co} N_{cp,co}^{2} - 4.6702*10^{-20} N_{cp,co}^{4} - \\ & 3.3294*10^{-9} \dot{m}_{ar,co}^{2} N_{cp,co}^{2} + 2.7101*10^{-14} \dot{m}_{ar,co} N_{cp,co}^{3} + 1.0853*10^{-25} N_{cp,co}^{5} - \\ & 6.0062*10^{-8} \dot{m}_{ar,co}^{3} N_{cp,co}^{2} + 1.2571*10^{-13} \dot{m}_{ar,co}^{2} N_{cp,co}^{3} - 1.4971*10^{-19} \dot{m}_{ar,co} N_{cp,co}^{4}\end{aligned} \tag{2}$$

The equations above were generated from experimental data obtained by the manufacturer; however, they should be corrected to actual conditions for the application of the compressor. Reason why, are also part of the model the relationship between the mass flow of air and corrected mass flow of air and the relationship between rotation speed and corrected rotation speed. The pressure ratio for the compressor was represented by the ratio between the suction pressure and the discharge pressure.

Three thermodynamic relations complement the group of equations related to the compression process. The first represented the relationship of pressures and temperatures in the suction and discharge for an isentropic process, with the value of k equal to 1.4 (value corresponding to the dry air). The second equation quantified the amount of energy per unit of time transferred to the air during compression. And finally, the third defined the efficiency as the ratio between the energy involved in an isentropic process and the energy in the actual energy.

2.1.2. Intercooler

The heat exchanger received a simplified treatment. The equations used are based on effectiveness in terms of the maximum possible rate of heat transfer and the actual rate of heat transfer. Equation 3 represents the effectiveness of the intercooler.

$$\eta_{tr} = \frac{T_{ar,scp} - T_{ar,emo}}{T_{ar,scp} - T_{ar,ecp}} \tag{3}$$

This process was considered isobaric and the value of effectiveness was admitted as 80%.

2.1.3. Combustion Chamber

The combustion process was modeled according to the normal diesel cycle, illustrated in Figure 2.

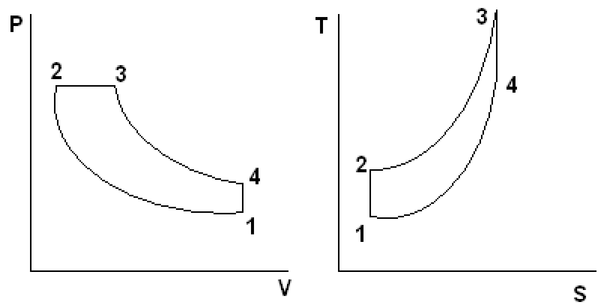

Figure 2. Diagrams of the normal diesel cycle.

The assessment of the combustion process started from two regressions of curves referring to the performance of the ICE OM-364 A. As a result was obtained the Eq. 4, which relates the fuel consumption to the speed of rotation.

$$\begin{aligned}\dot{m}_c &= -9,5*10^{-20} N_{mo}^5 + 8,59*10^{-16} N_{mo}^4 - 2,57*10^{-12} N_{mo}^3 + 2,083*10^{-9} N_{mo}^2 + \\ &\quad 3,68*10^{-6} N_{mo} - 2,81*10^{-3}\end{aligned} \tag{4}$$

Furthermore, a stoichiometric analysis was performed to relate the fuel consumption to the air consumption (Eq. 5). In this equation it was considered an excess of air of 15% and the adopted fuel had the molecular formula of light diesel oil, $C_{12}H_{26}$.

$$\dot{m}_{ar} = 14.934 * (1 + e) * \dot{m}_c \tag{5}$$

The type of fuel in a combustion process brings its own characteristics able to qualify and quantify the phenomenon of burning. One of these features is the high calorific power; the value of this parameter was made possible by Eq. 6 to estimate the flame temperature inside the combustion chamber.

$$PCS = \int_{T_i}^{T_f} \sum_{i=1}^{l} (n_i \bar{c}_{p_i} dT) \tag{6}$$

Based on Eq. 6 and on the stoichiometric balance of the combustion process was obtained Eq. 7, which relates the temperature after the compression process with the flame temperature (temperature from gas exhaust chamber).

$$
\begin{aligned}
&PCS_c = \Delta H_c = \int_{T_{2M}}^{T_{3M}} \left(12\bar{c}_{p_{CO_2}} + 13\bar{c}_{p_{H_2O}} + 69,56\alpha\bar{c}_{p_{N_2}} + 18,5\left(\alpha-1\right)\bar{c}_{p_{O_2}}\right)dT \\
&1151270 = \int_{T_{2M}}^{T_{3M}} 12\left(10,34+0,00274T-\frac{195500}{T^2}\right)dT + \\
&\int_{T_{2M}}^{T_{3M}} 13\left(8,22+0,00015T+0,00000134T^2\right)dT + \\
&\int_{T_{2M}}^{T_{3M}} 69,56\alpha\left(6,5+0,001T\right)dT + \int_{T_{2M}}^{T_{3M}} 18,5(\alpha-1)\left(8,27+0,000258T-\frac{187700}{T^2}\right)dT
\end{aligned} \tag{7}
$$

2.1.4. Turbine

The modeling of the turbine started with the assessment of specific heats, both at constant pressure as the constant volume for the exhaust gases. According to the products obtained in the stoichiometric balance and the ratio of specific heats was possible to express the value of k for the exhaust gases.

The turbine was parameterized according to the work of Martins (2004) that validated the model of ASHRAE Toolkit (1994) where the turbine was considered one wheel of D'Laval. This model was corrected according to the procedure of Withfield (1976) to estimate the losses in the turbine nozzle (Eq. 8).

$$
\dot{m}_g \frac{\sqrt{R_g T_{g,smo}}}{A_{th} P_{g,smo}\sqrt{k_g}} = M_2\left(1+\frac{k_g-1}{2}M_2^2\right)^{-\frac{1}{2}\left(\frac{k_g+1}{k_g-1}\right)}\left(1-\frac{k_g-1}{2}\xi_{th}M_2^2\right)^{\frac{k_g}{k_g-1}} \tag{8}
$$

The curve behavior of the turbine was supplied by AlliedSignal Turbocharging Systems Company, Torrance, California, USA. An algorithm (created in Matlab) related the data of the manufacturer's curve to estimate the parameters of the turbine (the equivalent area, the equivalent diameter, the losses in the nozzle and the angle between the absolute velocity vector and the tangential velocity vector in the output of the rotor) for each speed of rotation found in the curve. The estimation of these parameters allowed the development of Eq 9 to Eq.12, which can be applied to any regime of rotational speeds of the turbine.

$$
A_{th} = -1x10^{-22} N_{tb}^4 + 3x10^{-17} N_{tb}^3 - 3x10^{-12} N_{tb}^2 + 1x10^{-7} N_{tb} - 0,0012 \tag{9}
$$

$$
D_{eq} = 2x10^{-16} N_{tb}^3 - 4x10^{-11} N_{tb}^2 + 2x10^{-6} N_{tb} + 0,0189 \tag{10}
$$

$$
\alpha_2 = -9x10^{-20} N_{tb}^4 + 2x10^{-14} N_{tb}^3 - 2x10^{-9} N_{tb}^2 + 1x10^{-4} N_{tb} - 0,7283 \tag{11}
$$

$$
\xi_{th} = -8x10^{-15} N_{tb}^3 + 2x10^{-9} N_{tb}^2 - 2x10^{-4} N_{tb} + 3,344 \tag{12}
$$

The model is complemented by the equation of absolute speed (Eq. 13) and the equation of the turbine efficiency (Eq. 14).

$$
C_2 = M_2\sqrt{k_g R_g T_{g,smo}} \tag{13}
$$

$$
\eta_{tb} = \frac{2U\left(C_2\cos\left(\alpha_2\right)-U\right)}{C_2^2/2} \tag{14}
$$

To calculate the temperature and pressure at the inlet of the turbine, the two are calculated for the exhaust manifold given the expansion that occurs when gases pass from the piston engine to the manifold, this is calculated using the ideal gas equation and the equation for an adiabatic expansion (Eq. 15).

$$\int_{T_{g,smo}}^{T_{g,etb}} \left(\frac{c_{v,o}}{T} \right) dT + R_g \ln \left(\frac{v_{g,etb}}{v_{g,stb}} \right) = 0 \tag{15}$$

2.2. The Lagrange Multiplier Equations

The optimization model was only possible by considering, initially, the operation of each component of the engine by establishing restrictions on its parameters. This analysis was extended to the engine completely, but greater attention was applied to the operation range of the turbine (speed of rotation) and compressor (compression ratio). The conditions of the engine inlet air were adopted according to the CNTP (normal conditions of pressure and temperature) also considering the condition of dry air. The last parameter fixed was the engine compression ratio (18:1, according to the manufacturer).

Once established the parameters, it became possible to perform the simulation model of the turbocharged engine in order to find operating points and compare them to literature values. The chosen parameters were: maximum allowable pressure in the combustion chamber and flame temperature, these values were similar to Pitillo (2006), Santos (2005) and Bermudez (1995). The simulations of this work had three input parameters (engine speed, excess of air in the mixture and the compressor pressure ratio) which were varied in a systematic way in order to validate different points of operation. Since then, was applied the method of surface responses to the simulated data and it might to analyze the influence of each of the three parameters in the behavior of the thermodynamic state of gases at the outlet of the turbine. Finally, was used the method of Lagrange multipliers, that is governed by Eq. 16, which relates the gradient of the objective function and the gradients of each of the constraints multiplied by their coefficients of Lagrange.

$$\nabla y - \lambda \nabla \phi = 0 \tag{16}$$

3. RESULTS

3.1. Response Surface Analysis

By using the software STATISTICA, it has varied three factors: the engine speed, the excess of air in the mixture and the compressor pressure ratio. The factors were adopted according to Table 1.

Table 1. Values at lower, central and upper levels of the engine speed, the excess of air in the mixture and the compressor pressure ratio.

Experimental design factors								
N_{mo} [rpm]			e [-]			Rp_{cp} [-]		
Levels of factors								
Lower	Central	Upper	Lower	Central	Upper	Lower	Central	Upper
1804	2250	2696	0.081	0.200	0.319	1.157	1.210	1.263

The most important answer, shown in Figure 3, is the pressure of gases at the outlet of the turbine.

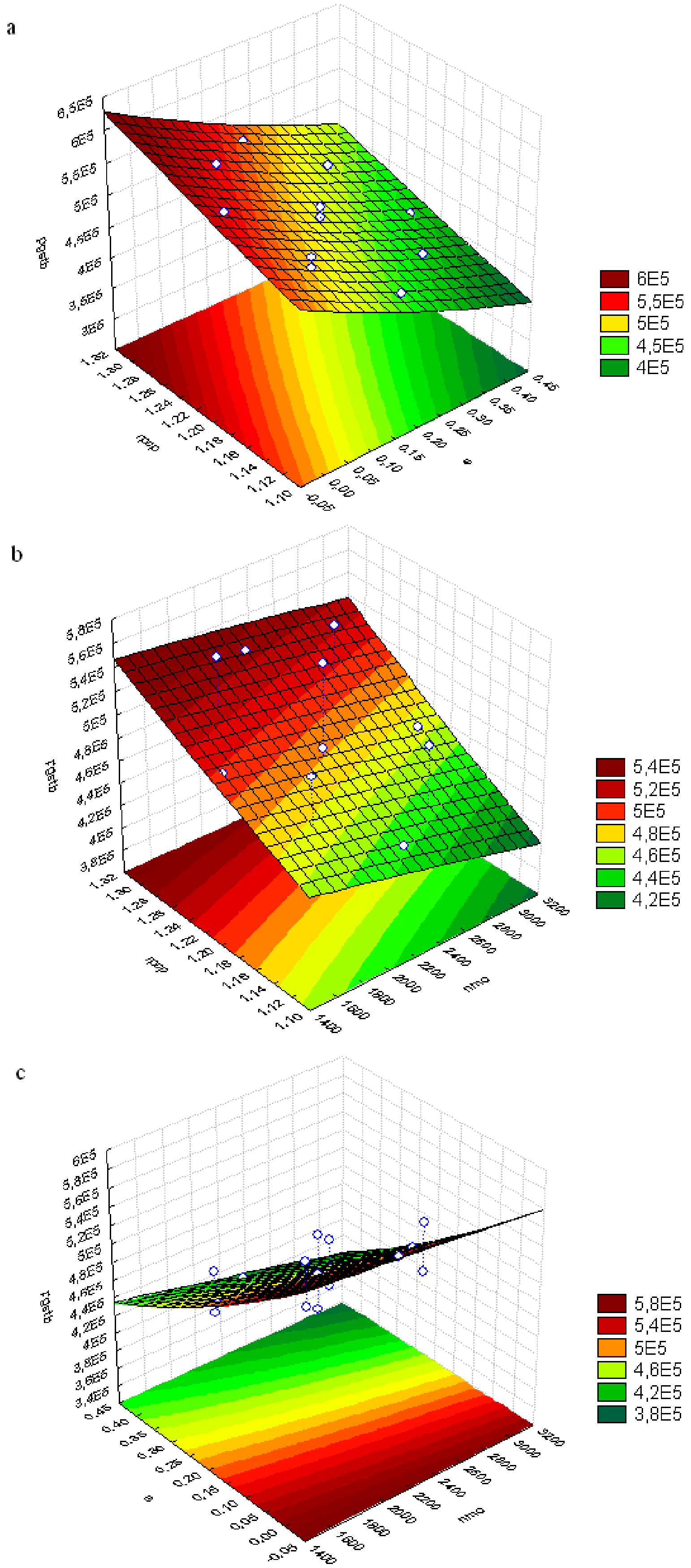

Figure 3. Response surfaces for the pressure of gases at the outlet of the turbine.

The response surfaces show the effect of three parameters on the pressure and temperature of gases in the turbine exit. The excess of air generated larger effect on the temperature, because when it hold constant the effects of other variables is negligible. By reducing the excess of air, the temperature of the gases output is high. Since this model does not consider the not-burned fuel by the lack of oxidant in the mixture, the reduction of excess of air must not exceed the practical limits (set at 14%) for the proper functioning of an engine.

It is observed in figure 3a that high values of the pressure ratios of the compressor combined with minimum excesses of air in the mixture results in high values of gas pressure at the turbine outlet. Note in figure 3b that high values of the pressure ratios of the compressor combined with minimum values of the engine speed generates high values for the response. Finally, in Figure 3c, it appears that the biggest pressures at the outlet of the turbine are the result of low excess of air and low engine speeds.

However, it may be highlight that the effect of the ratio of pressures of the compressor is predominant among the other parameters about the answer. But, it should consider the limits for each engine pressure in the combustion chamber.

3.2. System Optimization

The optimum point found one that attended all the restrictions of operating parameters, always seeking the highest values of gas pressure at the turbine outlet. EES was used as a database of thermodynamic properties and as a tool for solving equation systems applied.

The values of pressure of gases in the turbine outlet was equal to 228576 Pa. The values of other parameters which determine this particular condition of operation are illustrated in Table 2.

Table 2. Values of the relevant parameters which determine this particular condition of operation.

e	η_{cp}	η_{tb}	$N_{cp} = N_{tb}$	N_{mo}	$\dot{m}_{ar}$	$\dot{m}_{co}$	Rp_{cp}	$P_{ar,scp}$
[-]	[-]	[-]	[rpm]	[rpm]	[kg/s]	[kg/s]	[-]	[Pa]
0.14	0.71	0.52	84079	2900	0.0676	0.004	1.3	131723
$P_{ar,emo}$	P_{3M}	$P_{g,smo}$	$P_{g,etb}$	$T_{ar,scp}$	$T_{ar,emo}$	T_{3M}	$T_{g,smo}$	$T_{g,etb}$
[Pa]	[Pa]	[Pa]	[Pa]	[K]	[K]	[K]	[K]	[K]
131723	7.53E+6	570465	244752	330.4	304.5	2605	1319	808

After finding the optimum point and considering that the engine, in real situations, can be operated at different rotational speeds, it was decided to find the optimal point for different engine speeds between 1500 and 2900 rpm. Figure 4 and Eq. 17 show the objective function and the behavior of gas pressure at the outlet of the turbine.

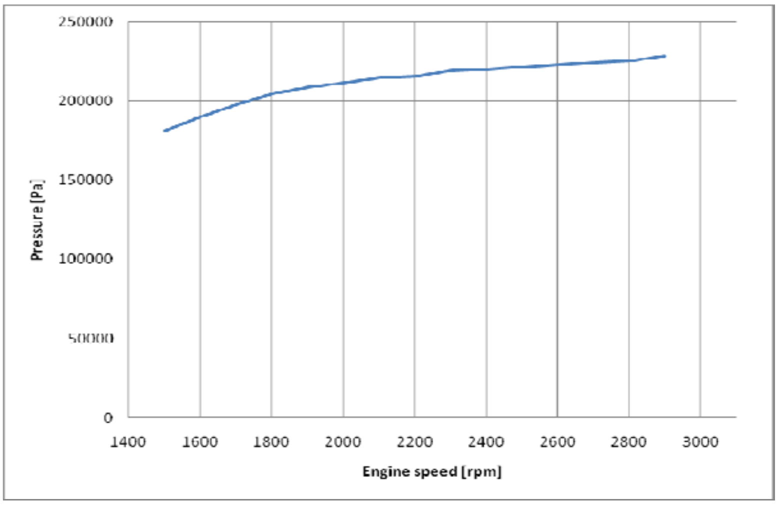

Figure 4. Behavior of gas pressure at the outlet of the turbine.

$$P_{g,stb} = 2.9326E-05N_{mo}^3 - 2.1831E-01N_{mo}^2 + 5.5394E+02N_{mo} - 2.5718E+05 \quad (17)$$

4. CONCLUSIONS

First, the model of the turbine, according to the work of Martins (2004), could validate excellent points of operation.

After the simulations, it may be highlight that the effect of the ratio of pressures of the compressor was predominant among the other parameters about the values of pressure of the gases in the turbine outlet.

Exactly at the optimal point the values of pressure of gases in the turbine outlet was, equal to 228576 Pa.

With the intention of reproducing an actual operation of this engine, an equation was proposed, which returned a range of pressures (181,000 to 229,000 Pa) of the gases in terms of engine speeds, which ranged from 1500 to 2900 rpm.

Future work will take into account the calculation of heat release in terms of the excess of air and the angle of the engine speed, in order to not only optimize the output pressure of the gases, but also, the optimize power generation in the chamber combustion.

The study showed a fairly good model for establishing a control system capable of controlling the rotational speed of the compressor in order to obtain appropriate pressure ratios of this equipment. Adding up to this pattern, an exhaust valve before the turbine.

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6. RESPONSIBILITY NOTICE

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