CO-GENERATION USING DIESEL ENGINE EXHAUST GASES, POTENTIALS DETECTED USING PARAMETRIC MODELING

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Abstract: *It was modeled and simulated a turbocharged diesel engine, using parametric modeling and data obtained in literature (Bermudez,1975). In order to simplify the modeling it was used an Otto cycle representing the diesel engine. This cycle was properly adjusted and models found in literature for different components .The model obtained was validated using experimental data found in literature. The simulation work done shows that for rotation velocities above 1600 RPM ,it is possible to obtain from turbocharger a power of the order of 3 kW that can be used independently of normal turbocharger operation without interfering in engine thermal efficiency. It is also possible to see the effect on intake and exhaust gases of pressure and temperature and analyze the effects caused when this gases conditions are well controlled to obtain best thermal efficiency.*

Keywords: Computational simulation, Turbocharged engine, Semi-empirical Modeling,

1. INTRODUCTION

The employ of modeling through calculation taken from its average operational values (MVEM) of and overfed diesel engine, allows to obtain the characteristic curves of the engine operation where parameters such as, power and torque developed by the engine and its fuel specific consumption may be identified. In order to allow the dimensioning of turbomachinery, state conditions of the working fluid for different stages of operation, these identified in the surroundings of the engine circuit, may be obtained through a simulation of its components, analyzing its different regimes of operation. Nowadays, one has emphasized pertinent concerns referred to an eventual lack of generating sources of energy. To identify alternative solutions of generating sources of energy and the optimization of the use of actual systems are one of the main focus of study of the scientific community. However, these initiatives must be always followed by an analysis of technical and economic viability. The availability of an evaluated and appropriated modeling of an operation circuit of an engine consists on the basic and fundamental tool for execution of these initiatives.

The literature provides many authors, which could cooperate to this work.

Pettersson, F. (2000) promoted the development of models based on controled systems, new tests, strategic controls and diagnosis, being later applied on engines simulation, several conditions of operation in which the engine was submitted to experiments, one observed considerable oscillations in operational properties among simulations. But the static models has presented a good behavior when applied in simulation.

Ritzén, J. (2003), modeled and simulated by fixed stages an overfed diesel engine. The goal was to create a average valued model that could be simulated with individual solutions by stages of the cycle. The variables, mass flux and turbo efficiency has presented accentuated deviations when compared to the experimental values defined to turbine velocities up to 75.000 [RPM].

Bermudez, V. T. (1995) developed a analysis of the internal aerodynamics of a overfed diesel engine, tracing the influence of the composing equipments of two engines of four and six cylinders with intercooler, using turbomachinery equipped with waste-gate for the four cylinders engine, and with double inlet turbines (VGT) with variation of the effective section for the six cylinders engine, results observed fit well in curves of performance of these engines,.

Bermudez, V. T. (1995), for the development of the compressor's physical model of a turbomachinery coupled in a IC, 06 cylinders engine defined with simplifying hypotheses that the work conditions of the compressor were almost stationary, and that the loss of air pressure in the inlet of the compressor was considered constant for each operational condition applied (rotation regime; mass flow). It was possible to obtain the characteristic curve of the compressor's operation, showing the inter-relation amongst the operational variables, such as, pressure relation, efficiency, rotation and mass flux, which made it viable mainly to establish a relation between power given by the turbine and the power actually taken by the compressor.

Martins (2004) studied and evaluated through semi-empiric models proposed by the ASHRAE – T.C 4.7 document, the behavior characteristics of the air mass flux and of the efficiency (in the application of the refrigeration field) of usual turbo compressors, used in overfeeding of diesel engines. Initially with the operation data supplied by Brazilian manufacturers, the radial turbines and the centrifugal compressors were simulated through the models obtained by ASHRAE, and the comparing results were satisfactory.

2. ENGINE MODELING

The choice and conceivement of the models that may present simultaneously characteristics of simplicity and applicability consist in one of the challenges of this work. The presence or not of a physical model is what differences the semi-empiric modeling from statistic modeling, being characterized the semi-empiric modeling as the one that uses a physical model, and parameters of the real component or system, but with a middle level of complexity (mathematical equations of easy solution). The statistic modeling in turn is characterized working with a wide bank of data without, however, giving up a pre-established physical model. In both, one may get models which when simulated presents a, good approach with measured data.

In this work it became more convenient the use of the engine modeling through the average values (MVEM), based on the fact that the torque on the handle axis produced by individual combustions in cylinders is leveled to a average value. One seek to obtain the ideal parametric model that may describe a more faithful and coherent manner, the behavior of the variables to be analyzed, submitting afterwards the selected model to a process of validation, allowing a first evaluation over the simulation of this model.

From the right application of the physical models identified, one has promoted the simulation through a semi-empiric modeling, feeding the process with experimental data taken from literature "Sintesis De La Aerodinamica Interna Del Motor Diesel Sobrealimentado", Vicente. R. Bermudéz. 1995, in order to viable its validation, due to the impossibility of mounting a test bench for structural issues of the laboratory of energy and thermal systems where this work has been developed.

The complete circuit of engine operation taken from Peterson(2000) (was submitted to computational programs (MatLab), in order to identify the constraints of adjustment present in the models, as well as to promote the manageability of a block diagram defined to make viable the simulation of the models. Figure 1 introduces the scheme of a block diagram used to characterize the applied modeling as well as to define the engine component.

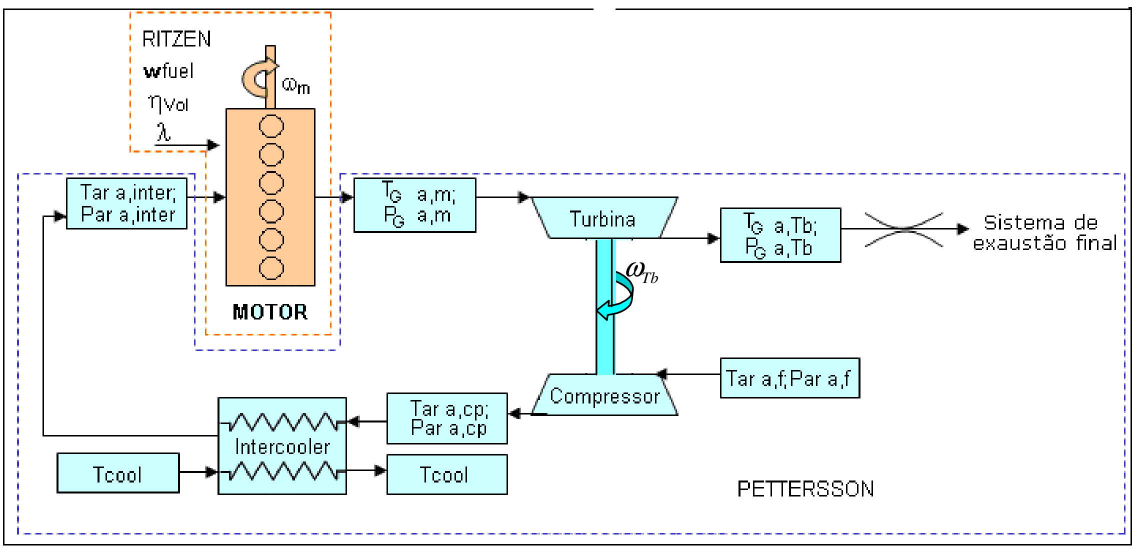


Figure 1 – Block diagram of the engine´s working circuit

The subsystems of the engine were individually modeled in the sub items below:

2.1 Compressor models

Knowing the operation curve of the compressor, statistic adjustments are proposed of the points that represent it, in order to characterize its behavior doing the interpolation among the points presented in the map.

The compressor's pressure relation $Rp_{(cp)}$ was modeled determining a 4th order polynomial with 8 (eight) constants of adjustments, with experimental data taken from literature (Bermudez, 1995),given in the following form:

$$Rp_{(cp)} = \frac{P_{ar,acp}}{P_{ar,ecp}} = f(\dot{w}_{ar,cp}, N_{cp}) \tag{1}$$

where:

$Rp_{(cp)}$ = compressor's pressure relation

$P_{air,acp}$ = air pressure after the compressor Pa

$P_{air,icp}$ = air pressure in the inlet of the compressor Pa

$\dot{W}_{air,cp}$ = air mass flux in the compressor $\frac{Kg}{s}$

N_{cp} = number of compressor's rotations rpm

The efficiency of the compressor η_{cp} was modeled determining a 4th order polynomial with 15 (fifteen) constraints of adjustment, with experimental data taken from the literature (Bermudez, 1995), given in the following form:

From thermodynamics, we have:

$$\eta_{cp} = \frac{T_{2Sar,acp-}T_{ar,ecp}}{T_{ar,acp} - T_{ar,ecp}} = f(\dot{w}_{ar,cp}, N_{cp}) \tag{2}$$

The analysis of the polynomial's grade and the number of coefficients to be used, was based on the statistic behavior of the influence of each coefficient, using the normalization process. In the calculation of η_{cp} e Rp_{cp}, efficiency and compressor's pressure relation, respectively, it's necessary to feed the models with the following inlet data initially adopted, $\dot{w}_{ar}$, P_{atm}, T_{amb}, N_{cp}, and yet, previously known. Also, in order to obtain the outlet state conditions, thermodynamic equations were used. Therefore, it is possible to determine the mechanical power developed by the compressor.

$$M_{cp}\omega_{cp} = \frac{\dot{w}_{ar,cp}.Cp_{ar}.T_{amb}.\left[Rp_{cp}^{\frac{\gamma-1}{\gamma}} - 1\right]}{\eta_{cp}} \tag{3}$$

where:

M_{cp} = torque developed in the compressor Nm

Cp_{air} = air specific heat at constant pressure $\frac{KJ}{KgK}$

γ = relation between the air specific heats

2.2 Modeling the Intercooler

A parametric modeling (Pettersson 2000), based on d'Arcy's equation for laminar regime, allows to determine the loss of pressure that occurs in the air mass flux that flows through the intercooler.

$$P_{ar;ainter} = P_{ar;\,acp} - K_{inter} \,.\, T_{ar;\,acp} \,.\, \dot{w}^{2}_{ar,\mathrm{int}\,er} \tag{4}$$

where:

$P_{air,a\,\mathrm{int}\,er}$ = air pressure after the intercooler Pa

$K_{\mathrm{int}\,er}$ = adjustment constraint of loss of pressure in the intercooler

$\dot{W}_{air,\mathrm{int}\,er}$ = air mass flux in the intercooler $\frac{Kg}{s}$

For calculation of the heat exchange in the intercooler (Pettersson 2000), adjustments were made of data measured to a black box model, presenting satisfactory results ,obtaining the value of the heat exchange in the intercooler as the following equation:

$$\varepsilon_{\mathrm{int}\,er} = K_{\varepsilon 1} + K_{\varepsilon 2}\left(\frac{T_{ar,acp} + T_{cool}}{2}\right)\dot{w}_{ar,\mathrm{int}\,er} + K_{\varepsilon 3}\,\dot{w}_{ar,\mathrm{int}\,er} \tag{5}$$

where:

$K_{z1}, K_{z2,} K_{z3}$ = adjustment constraints of efficiency of the intercooler

T_{cool} = cooling temperature of the air in the intercooler K

2.3 Modeling the engine

With the intention to determine the outlet temperature of the engine gases $T_{G,am}$, one employed the state definitions of the Otto cycle, establishing the following equations:

$$x_r = \frac{v_2}{v_1}\left(\frac{P_{G,am}}{P_{ar,em}}\right)^{\frac{1}{\gamma}}\left(1 + \frac{q_{in}}{CvT_1\left(\frac{v_1}{v_2}\right)^{\gamma\text{-}1}}x_{cv}\right)^{\frac{-1}{\gamma}} \tag{6}$$

Through a physical modeling (Ritzen 2003), it´s possible to obtain important independent variables that define extensively the operational field of the engine, such as the specific energy present in the charge by unit mass, the fraction of residual gas and the initial temperature of the gas in the engine cycle.

$$q_{in} = \frac{w_{fuel}q_{HV}}{w_{ar,em} + w_{fuel}}(1 - x_r) \tag{7}$$

$$x_r = \frac{1}{r_c}\left(\frac{P_{G,am}}{P_{ar,em}}\right)^{\frac{1}{\gamma}}\left(1 + \frac{q_{in}}{C_v T_1 r_c^{\gamma\text{-}1}}\right)^{-\frac{1}{\gamma}} \tag{8}$$

$$T_1 = x_r T_{G,am} + (1 - x_r)T_{ar,em} \tag{9}$$

where:

r_c = relation of engine compression

T_1 = initial temperature of the gas in the engine cycle K

x_{ic} = fraction of residual gas

$P_{G,ae}$ = gas pressure in the inlet of the engine Pa

$P_{air,ie}$ = air pressure in the inlet of the engine Pa

q_{in} = specific energy present in the charge by mass unity $\frac{J}{Kg}$

The temperature of the exhaust gases of the engine is modeled according to an ideal Otto cycle (Ritzen, 2003), being determined by:

$$T_{G;am} = T_1\left(\frac{P_{G;am}}{P_{ar;em}}\right)^{1-\frac{1}{\gamma}}\left(1+\frac{q_{in}}{C_v T_1 (r_c)^{\gamma\text{-}1}}\right)^{\frac{1}{\gamma}} \tag{10}$$

To determine the engine power, the following parametric model based on thermodynamic laws was developed, once it was not possible to find in literature a simple model that allows the dimensioning of this variable.

$$\dot{W}_m = K_{qin} q_{in}\left[1 - r_c^{(1\text{-}\gamma_G)}\right] \tag{11}$$

Where:

$\dot{W}_{in}$ = power developed by the engine W

K_{qin} = adjustment constant of the specific energy released

γ_G = politropic relation of the gases

2.4 Modeling the Turbine

The model developed is focused on the study of radial turbines used in engines of internal combustion. Martins (2002) (Toolkit TC 4.7 ASHRAE) observed that one could approach the operational behavior of a radial turbine to the behavior of a axial elemental turbine, known as D'Laval wheel, in order to establish the modeling of the pressure relation and of the efficiency.

The isentropic efficiency η_{Tb} is given in the form:

$$\eta_{Tb} = \frac{W_{eixo}}{W_{adiabatico}} = \frac{2 \cdot U \cdot \left(C_2 \cdot \cos(\alpha_2) - U\right)}{\frac{1}{2}\cdot C_2^2} \tag{12}$$

where the velocity C_2 may be calculated by a isentropic expansion of the nozzle, given by:

$$C_2 = \sqrt{2 \cdot Cp \cdot T_{G,eTb} \cdot \left[1 - \left(\frac{P_{G,aTb}}{P_{G,eTb}}\right)^{\frac{\gamma-1}{\gamma}}\right]} \tag{13}$$

where:

$P_{G,eTb}$ = gas pressure in the inlet of the turbine Pa

$T_{G,eTb}$ = gas temperature in the inlet of the turbine K

$P_{G,aTb}$ = gas pressure after the turbine Pa

C_P = specific heat of the gas at constant pressure $\frac{KJ}{KgK}$

The components of a turbo machinery may be represented by ducts and a simplified configuration for a nozzle, assuming that the static and total states are approximately the same in the inlet of the nozzle, results in the following equation:

$$\dot{w}_{gas} \cdot \frac{\sqrt{\gamma \cdot R_{gas} \cdot T_{SU}}}{A_{Thr} \cdot P_{SU} \cdot \gamma} = M_2 \cdot \left[1 + \frac{\gamma - 1}{2} \cdot M_2^2\right]^{-\frac{1}{2} \cdot \frac{\gamma+1}{\gamma-1}} \cdot \left[1 - \frac{\gamma - 1}{2} \cdot \zeta_{Thr} \cdot M_2^2\right]^{\frac{\gamma}{\gamma-1}} \tag{14}$$

where:

M_2 - Mach number in the outlet of the nozzle

ζ_{Thr} - Coefficient of losses in the nozzle

The proposed equations in this model shows itself quite simplified in relation to other known models, making necessary to identify only 3 parameters to characterize the turbines: nozzle area (A_{thr}); rotor diameter (D) and inlet angle of the rotor's gas flux (α_2). This procedure is easily done, being for that purpose, sufficient the use of the operation curves of the turbo machinery.

Once obtained the efficiency and pressure relation, one may determine from the 1st law of thermodynamics and from the definition of isentropic efficiency for the turbine, the power developed by the turbine may be determined by:

$$M_{Tb}\omega_{Tb} = \dot{w}_{gas} . Cp_{gas} . T_{G,eTb} \cdot \left(Rp_{Tb}^{\frac{1-\gamma_G}{\gamma_G}} - 1 \right) \eta_{Tb} \tag{15}$$

2.5 Modeling the Final System of Exhaustion

The loss of pressure in the exhaust gases of the engine in the final system of exhaustion,is promoted due to physical and mechanical restrictions, it is calculated through the following physical model (Bermudez 1995).

$$P_{G,aTb} = \left(\frac{\dot{w}_G}{C_{Dexhaust}} \right)^2 \frac{R_G T_{G,aTb}}{2P_{G,aTb} + P_{atm}} \tag{16}$$

where:

$C_{Dexhaust}$ = Discharge coefficient of the exhaust gases

R_G = Gas constant

2.6 Modeling the Mass Flux

From the consideration of the swept volume in the cylinder and using the equation of ideal gas, we found as model of the air mass flux that flows through the engine the following expression:

$$\dot{w}_{ar,m} = \frac{\eta_{vol} P_{ar,ainter} V_{desl} n_{cilin} N_m}{60 n_{rev/cicl} R_{ar,em} T_{ar,ainter}} \tag{17}$$

where:

$\dot{W}_{ar,m}$ = air mass flux in the engine	$\frac{Kg}{s}$
η_{Vol} = volumetric efficiency	$\%$
$P_{ar,a\,inter}$ = air pressure after the intercooler	Pa
$V_{desloc.}$ = volume of the displaced air in the cylinders	m^3
N_e = engine rotation	rpm
$n_{rev,c}$ = number of revolutions by cycle	rpm
n_{cil} = number of cylinder of engine	
R_{ar} = gas constant in the inlet of the engine	$\frac{J}{KgK}$
$T_{ar,a\,inter}$ = air temperature after the intercooler	K

3. ADJUSTMENTS AND VALIDATION OF THE MODELS

In order to be adjusted, the calculation models presented needed a set of data taken experimentally from the former configuration of the engine, allowing us afterwards to make studies with models specifically in the characteristic points of the system where one wishes to optimize. A extensively used tool is the methodology of multi-variable least squares. The method of linearization of the equations consists on the gathering and substitution of the variables associated to the parameters by an only variable. The method of multi-variable least squares is the same applied for one variable, and is given by:

$$\begin{bmatrix} m & \sum X_i \\ \sum X_i & \sum X_i^2 \end{bmatrix} \cdot \begin{bmatrix} a \\ b \end{bmatrix} = \begin{bmatrix} \sum Y_i \\ \sum X_i \cdot Y_i \end{bmatrix} \tag{18}$$

where:

a - Coefficient of adjustment
b - Coefficient of adjustment
c - Coefficient of adjustment
X_i - Dependent variable
Y_i - Independent variable

With the statistic adjustment of the models, it was made the normalization of the dimensions of the variables involved, in order to assure that the coefficients of adjustments assume an order of magnitude really representative, according to the following general expression:

$$Variable \quad studied = Variable \quad normalized(Lim_{\max\_}Lim_{\min}) + Lim_{mim} \tag{19}$$

3. RESULTS AND SIMULATION

A flowchart has been created, represented by a block diagram, serving as the basis to build the calculation algorithm used in the simulation of the process as shown in Figure 2.

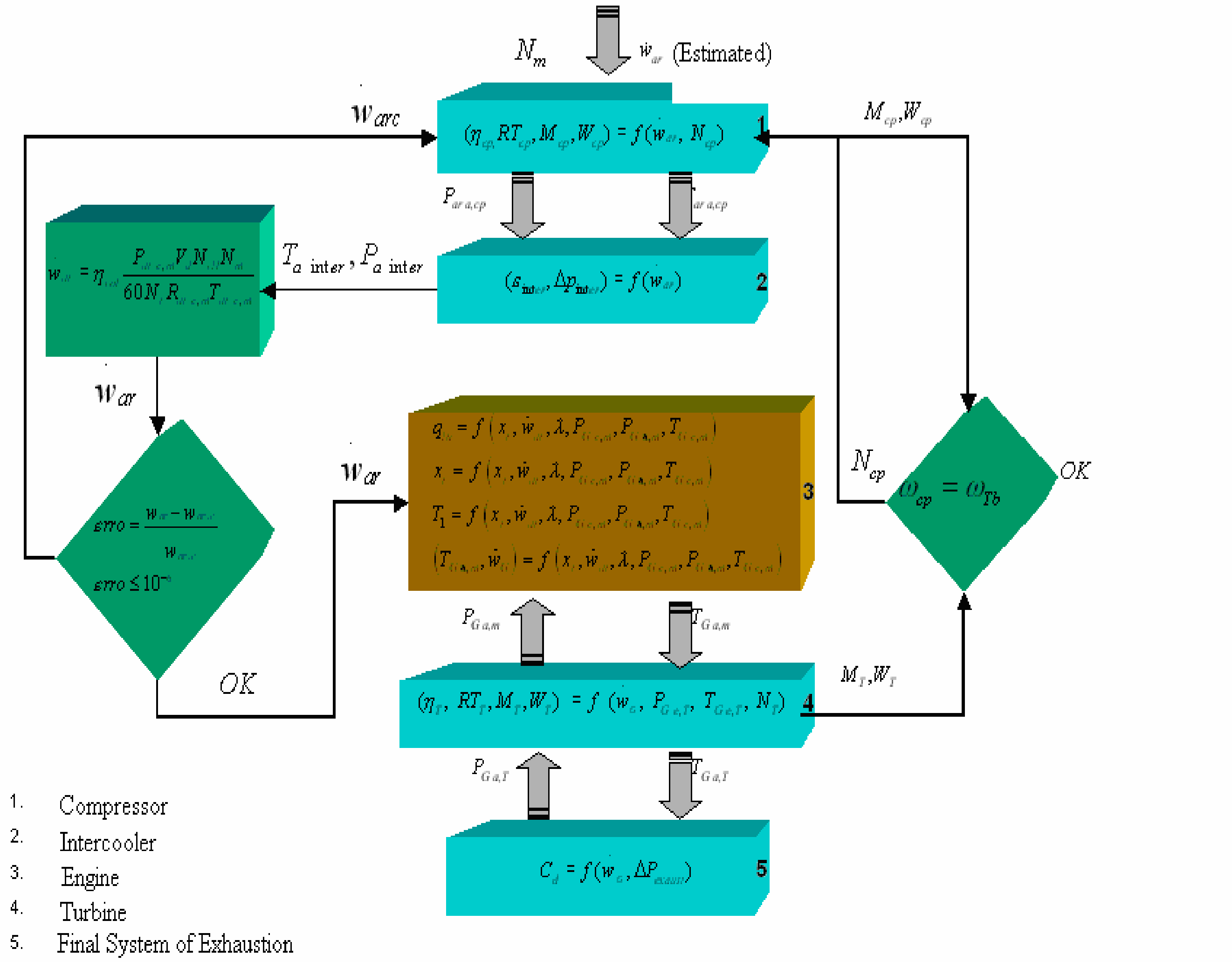
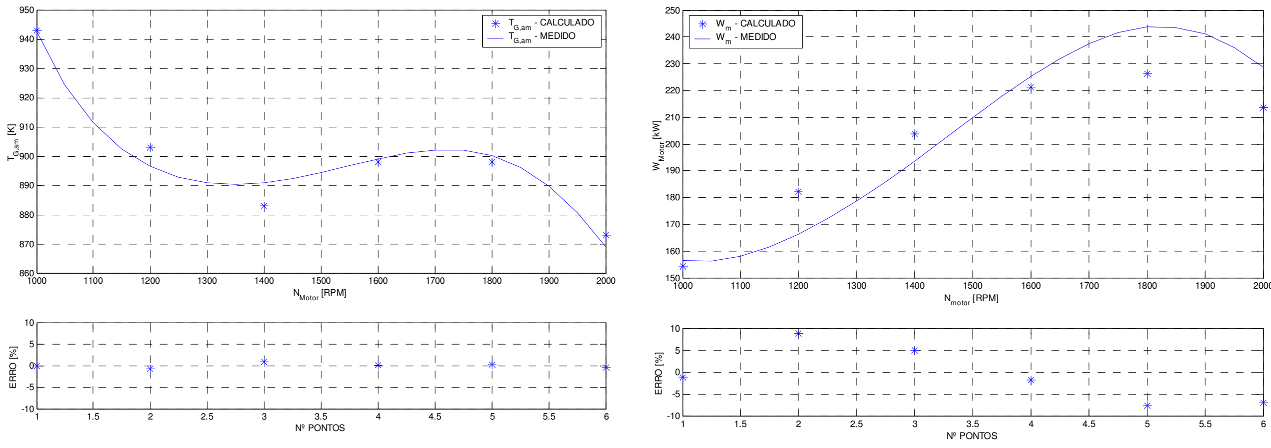


Figure 2 – Block diagram applied in the simulation of the engine´s cycle

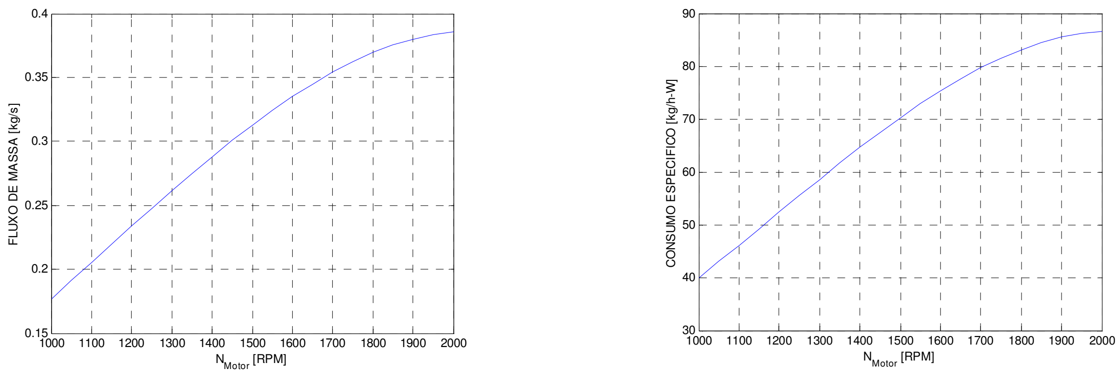
The procedure of the simulation has followed a reasoning (Ritzen 2003), of feeding the inlet and outlet surroundings, with data estimated and/or calculated such as, $\gamma_G, R_G, \dot{w}_{ar}, N_{cp}$, promoting an adjustment of pressure that, converging to the engine, allowed the determination of the temperature of exhaust gases and of the power developed by the engine. However, with the development of the simulation, we came across with some limitations in several variables, such as, air mass flow rate and rotation regime of the turbo machinery. Yet, in which concerns to the rotation of the turbo machinery, shown at right of the block diagram, it was not possible to close the simulation, for we did not have clear and complete information about the operation curve of the turbine. The process has occurred through the obtainment of data taken from the inlet and outlet surroundings, feeding the operation cycle of the engine. The evaluation of the system behavior under the action of extrapolated operational data, in relation to data obtained from the operation curves of turbo machinery as well as the confirmation of the results obtained (Bermudez 1995), are presented in Figures 3 and 4.



(a) (b)

Figure 3 – Results of the simulation in obtainment of the outlet temperature of the engine´s exhaust gases (a) and of the power developed by the engine (b) in function of itsrotation regime.

As shown in Figure 3 (a) one observed that the model represents well the temperature behaviour of the exhaust gases, in the same way of that observed in the validation showing errors of the order of 2%. In Figure 3 (b), one observed that the model represents somehow well the behaviour of the power developed by the engine. Despite in some points we have identified errors of the order of 8%, this is a characteristic of the modeling, of promoting the accumulation of errors came from the deficiency of the model used. Additionally, for the validation of the engine´s behavior, one represented the mass flow rate of fuel and the specific consumption of fuel in function of the engine´s rotation, according to Figure 4 (a) and (b).



(a) (b)

Figure 4 – Air mass flux (a) and specific consumption of fuel (b) in function of the turn regime of the engine

As shown in Figure 4 (a) one observed that the model represents well the behavior of the gas mass flow rate, showing errors of the order of 5%. Figure 4 (b) represents the behavior of specific consumption of fuel in the whole range of the engine’s velocity.

In order to search disponibility of energy in turbine, it is observed that for different engine rotations, the estimated turbine power delivered, when compared with the experimental data found in literature (Bermudez, 1995), shows that in the range of 1000 to 1600 RPM,(see figure 5 below),the accumulated errors are of the order of 4%, and above 1600 RPM,the accumulated errors are 7%,if we observe figure 3 the accumulated errors are not bigger than 2% in all rotation range, ,for this reason so high errors are not permitted.

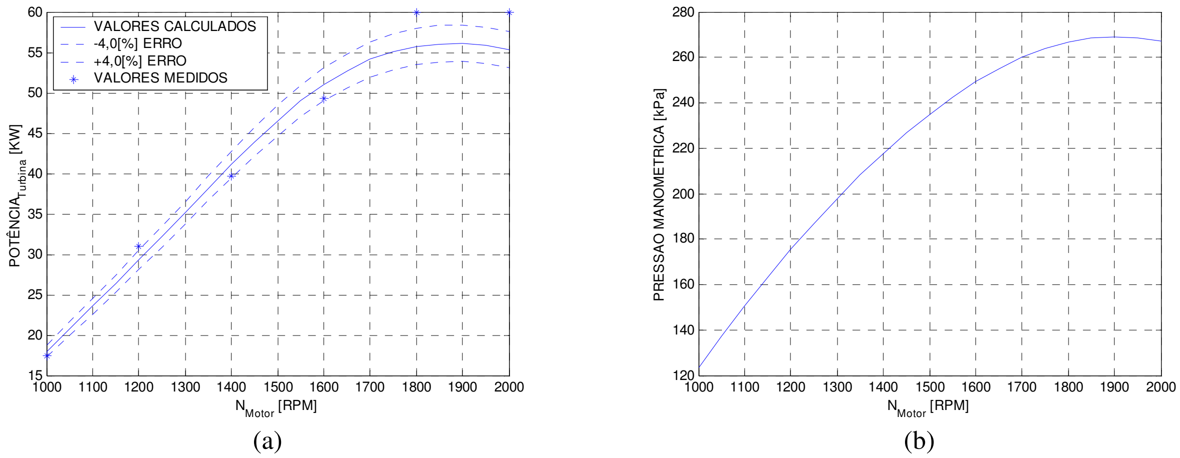


Figura 5 – (a) turbine power obtained by simulation for different engine rotations (b) Engine supply air pressure as function of engine rotation

At this point we remember that ,in order to minimize air pressure pulsations in consecutive cylinders of the motor, it was used a double turbine entrance ,causing a change in the effective turbine entrance area, this modify effective turbine power delivered by the turbine (bermudez,1995),this can be the explanation to the bigger errors found at high engine rotations(> 1800RPM). The turbine model used in this work does not change turbine entrance area , then we assume that the exceeding energy identified above 1600 RPM (2 kW), is really a disponibility of energy that can be used outside of the motor without affecting engine performance. We analyze that 5% reduction of turbine entrance area can increase disponibility of energy in 7%(4 kW).This conclusion can also be obtained ,observing that the over pressure resulting from the turbo machine, shown in figure 5 (b) is almost constant for engine rotations bigger than 1600 RPM , this proves that really the experimental data used for turbine performance model validation ,uses a double entrance turbine, properly actuated in order to control this over pressure. Then if we do not used a double entrance turbine it is possible to obtain disponibility (>2 kW) of energy from exhaust gases.

Normally refrigerated trucks used for transportation of perishable products , use refrigeration systems of 2,1 kW , with air renovation of the order of 4000 m^3/h, so this can be a possible use for this co generated energy.

4. CONCLUSION

In this work one promoted the modeling, validation and simulation of a diesel engine, 06 cylinders, 9,8 [L], 249 [kW] of maximum power, studying its behavior in stationary regime, at the expense of operational condition at full charge, including the whole range of rotation regime. The experimental data were taken from literature, due to the operational impossibility to build a tests bench, mainly in regard to the mechanical break (dynamometer). Through a ideal parametric model obtained at the expense of a large bibliography research, one evaluated a great variety of operational parameters of the engine, studied through the modeled code.

In regard to the validation, the models presented results quite satisfactory, with errors of the order of 4%. Some limitations of the modeling were identified, for instance, for the turbine, one observed that the model did not represent satisfactory the points of efficiency, however, due to the limitation in obtainment of the operation points and the relatively low errors of the order of 6%, it was possible to use this modeling. The model of the power developed by the engine in function its rotation regime, showed errors of the order of 8%, but the complex operation of the engine involves a considerable number of non linear variables, making it difficult to modeling, which allowed somehow to admit errors of this order. Nevertheless, one observed that the modeling showed a clear tendency to represent the tracing of the operation points. The literature showed itself deficient in which concerns to this modeling, so, the proposed model is definitely able of more accurate corrections and adjustments. From analyses realized it was possible to see, that cogeneration of energy is possible from exhaust gases, and that this energy in order to be used ,requires some control studies and more accurate models and experimental results in order to use this exceeding energy detected,

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