HEAT LOSS AND PERFORMANCE OF A DUAL FUEL DIESEL ENGINE: A THEORICAL AND EXPERIMENTAL STUDY

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Abstract. The purpose of this article is to evaluate the performance of an internal combustion engine when the same has part of its source of energy, in the case the diesel oil, substituted by natural gas. This substitution is accomplished in a partial way in function of the load, thanks to introduction of a conversion kit that allows to the engine to operate in the dual way with diesel gas. The objective is to observe the behavior of the engine working with maximum amount of gas substituting the diesel in relation to the pure diesel for several of loads to a constant rotation of 1800 rpm. The diesel engine system is equipped with electric charge bank, pressure and temperature sensors, gas, air and diesel, flow meters, gas analysis probe and data acquisition system. The obtained results of a series of parameters and of the analyses of the combustion products of the engine are used in the thermodynamic model to studying their effect on the heat loss (losses by heat to the environment) and on the efficiencies energetic and exegetic. The analysis of the results allowed observing an improvement of the efficiencies of the engine operating in dual way after certain power. Similar behavior is observed in relation the heat loss of the engine to the environment.

Keywords: Dual engine, Exergy, Energy, Internal Combustion Engines.

1. INTRODUCTION

The use gaseous fuel in diesel engine is increasing worldwide. The use of gaseous fuel is incentivated by its combustion cleaner compared to conventional liquid fuels and its increasing availability and attractive prices. The lower maintenance costs, the engine life longer following developments in cryogenic technology to store and transport the fuel and economically in liquefied form have motivated the use of this fuel. Moreover, it measly uniformly with air, resulting in efficient combustion and substantial reduction of emissions in the exhaust gas (Papagiannakis and Hountalas, 2003).

Natural gas combustion is characterized by the long ignition team delay (Mbarawa et al. 2001).

In dual-fuel engines, the carbureted mixture of air and gaseous fuel (natural gas) is compressed and then fired by the liquid fuel injection which ignites spontaneously at the end of compression phase, using this way the difference of flammability of two fuels. The presence of the gaseous fuel influences the pre-ignition and post-ignition process in the complex manner, depending mainly on the fuel used, its concentrations, and operating conditions (Karim, 1980). Many researches have been related in the literature about dual-fuel engines (Reitz and Rutland, 1995; Bi and Agrawal, 1998; Mansour et al., 2001; Mbarawa et al., 2001; Papagiannakis and Hountalas, 2003; Uma et al., 2004; Sheti and Salariya, 2004; Payri and Gaurdiola, 2005; Costa, 2007).

In Brazil, great part of the produced diesel oil is used in the national fleet of heavy vehicles and for electric power generation. The costs of this type of fuel and the concern with the environment have been taking the adaptation of these engines to operate with natural gas, using the diesel oil just as an element to promote the ignition of the mixture of air and natural gas. That is possible due to use of conversion kits that allows the natural gas to be used in the diesel engines. The great advantage of this system is in the reduction of the noxious pollutant gases and of the costs of electric power production from this source of energy.

This article has the purpose of examining an diesel engine, model CUMMINS 6CTA8.3 l with 188 kW that operates from a stationary way to 1800 rpm, for electric power production.

2. THE STUDIED PROBLEM

A schematic diagram of the problem studied in this article is presented in the Figure 1. The electric-mechanical system is composed by an engine diesel CUMMINS 6CTA 8.3 l with mechanical power of 188 kW to 1800 rpm, coupled to the generator Onan Genset of 150 kW. The unit totally equipped with air, gas and diesel flow meters, temperature and pressure sensors in several points of the system and a probe to analysis of gases. All the date are collected in real time through dates acquisition system.

The dual fuel mode use compressed natural gas the primary fuel and quantities of diesel pilot fuel goes ignition the engine is equipped with the dual-fuel commercial conversion kit (Figure 2). The commercial kit allows the engine to operate with 100% diesel fuel or in the dual-fuel mode. The natural gas is introduced into the intake air system. This way, the original characteristic of the diesel engine are not modified.

Figure 1. Diesel engine and electrical generator system.

Figure 2. Commercial conversion kit

2.1. Description of natural gas and diesel fuel compositions.

The main properties of the natural gas and liquid diesel fuel used in the present work is shown in table 1. These values shown represent of typical commercial fuels supplied in the city of Campina Grande, Paraíba State, Brazil. It is observed that the methane is the main constituent of the natural gas resulting to the relatively big octane number, which makes it suitable to engine of high compression rate.

Table 1-Basic composition of diesel and gaseous fuels used.

Fuel (source)	Chemical composition (in volume)						
Diesel (Medeiros et al., 2002)	$C_{12}H_{26}$ 98,53 %	S 1,47 %					
Natural gas (PBGAS, 2006)	CH_4 89,42%	C_2H_6 7,24%	C_3H_8 0,16%	C_4H_{10} 0,18%	N_2 1,27%	CO_2 1,66%	O_2 0,08%

2.2. Energy equation

The general equation of the first law of the thermodynamics in permanent regime is given by the expression below. It is used to evaluate the amount of heat loss of the engine, when the same is considered the control volume. Its form is:

$$\dot{Q}_{vc} + \sum \dot{m}_e \left(h_e + \frac{V_e^2}{2} + gZ_e \right) = \sum \dot{m}_s \left(h_s + \frac{V_s^2}{2} + gZ_s \right) + \dot{W}_{vc} \tag{1}$$

where $\dot{Q}$ is the heat transfer for the engine (control volume), $\dot{m}$, h, $V^2/2$ e gZ are, respectively, the rate of mass flow, the specific enthalpy , the specific kinetic energy and potential energy of the mass that flows through the engine. The sub-indexes "e" and "s" designate, respectively, entrance and exit, and $\dot{W}$ is the power developed by the engine.

The energy balance for a system that involves chemical reactions and where the variations of kinetic energy and potential are negligible is:

$$\dot{Q} + \sum \dot{m}_e h_e = \sum \dot{m}_s h_s + \dot{W} \tag{2}$$

For the dual engine enters diesel, gas and air simultaneously and leave the products of the combustion. Separating the products of the combustion as the sum of dry products, supplied by the analyzer of gas, and the steam of water, the equation above can be written as:

$$\begin{aligned}
&\dot{Q} + \dot{n}_d (y_{C_{12}H_{26}} M_{C_{12}H_{26}} h_{C_{12}H_{26}} + y_s M_s h_s)_d + \dot{n}_g (y_{CH_4} M_{CH_4} h_{CH_4} + y_{C_2H_6} M_{C_2H_6} h_{C_2H_6} \\
&+ y_{C_3H_8} M_{C_3H_8} h_{C_3H_8} + y_{C_4H_{10}} M_{C_4H_{10}} h_{C_4H_{10}} + y_{N_2} M_{N_2} h_{N_2} + y_{CO_2} M_{CO_2} h_{CO_2} + y_{O_2} M_{O_2} h_{O_2})_g \\
&+ \dot{n}_{O_2} (M_{O_2} h_{O2} + 3{,}76 M_{N_2} h_{N2} + 7{,}655 w M_{H_2O} h_{H_2O})_a = \dot{n}_P (y_{CO_2} M_{CO_2} h_{CO_2} \\
&+ y_{CO} M_{CO} h_{CO} + y_{N_2} M_{N_2} h_{N_2} + y_{NO} M_{NO} h_{NO} + y_{NO_2} M_{NO_2} h_{NO_2} + y_{SO_2} M_{SO_2} h_{SO_2} \\
&+ y_{CH_4} M_{CH_4} h_{CH_4} + y_{O_2} M_{O_2} h_{O_2})_p + \left(\dot{n}_{H_2O} M_{H_2O} h_{H_2O}\right)_{H_2O} + \dot{W}
\end{aligned} \tag{3}$$

where: $\dot{n}_d, \dot{n}_g, \dot{n}_{O_2}, \dot{n}_P, \dot{n}_{H_2O}$ are the molars flows rate of diesel, gas, oxygen, combustion products and of the steam of water; w is the absolute humidity and y_i and M_i represent the mole fraction and the molecular weight of the component i.

The results of the gas analysis of the combustion products was obtained by the analyzer Kane International Limited, model KM 9106 (Figure 3).

Figure 3. Analyzer KM 9106

The energy efficiency of the internal combustion engine is calculated by the following expression:

$$\eta_t = \dot{W} / PCI \tag{4}$$

where PCI is the lower heating value in kW and $\dot{W}$ is the shaft power of the engine in kW.

The energy efficiency of the engine in analysis is obtained as it proceeds:

$$\eta_t = \frac{\dot{W}}{\dot{n}_d\left[\sum_{i=1}^{k}\left(y_i M_i PCI_i\right)\right]_d + \dot{n}_g\left[\sum_{j=1}^{r}\left(y_j M_j PCI_j\right)\right]_g} \tag{5}$$

where the indexes i and j refer to the component of the fuel diesel and gas, respectively.

2.3. Exergy equation

The first law of the thermodynamics declares that the energy is a conserved property . It doesn't impose any restriction to the direction in that a process can happen. This incapacity of the first law of identifying a process can happen is supplied by the introduction of another general principle: the second law of the thermodynamics. A process cannot take place unless it satisfies both the first and second laws of the thermodynamics (Çengel and Boles, 2001).

The second law assures that the energy has quality. It is also used to determine the theoretical limits for the performance of systems.

The property exergy serves as a valuable tool in the determination of the quality of the energy and it compares the potentials of work of different sources of energy or systems. The exergy is the maximum useful work that could be obtained from the system at a given state in a specified environment (Çengel and Boles, 2001).

The exergy, X, associated in a specified state it is composed by two contributions: the contribution thermomechanical, X^{term}, and the chemical contribution, X^{qui}. In a unitary base of mass, the total exergy, x, is given for:

$$x = x^{term} + x^{qui} = \left[\left(h - h_0\right) - T_0\left(s - s_0\right) + \frac{v^2}{2} + gz\right] + x^{qui} \tag{6}$$

in that the term among brackets is the contribution thermomechanical.

When an exergy variation or the flow exergy is evaluated among two states where the chemical composition of the substance is the same, the chemical contribution is canceled, just staying the difference of the contributions thermomechanical. However, in several evaluations it becomes necessary to take in consideration explicitly the contribution of the chemical exergy. Some examples are the problems that involve chemical reactions, as it is the case of the combustion engines (Moran and Shapiro, 2000).

The efficiency of devices, whose purpose is to produce w, can be evaluated as the ratio of the real work developed by the theoretical maximum work. This ratio is a type of efficiency exergetic (efficiency based in the Second Law of Thermodynamics). If the shaft power of the engine is taken as the product of the engine and if the heat transfer rate together with the gases produced in the exit is seen as losses, an expression for the efficiency exergetic is:

$$\varepsilon = \frac{\dot{W}_{vc}}{\dot{X}_c} \tag{7}$$

where $\dot{X}_c$ means the rate for the which the exergy enters with the fuel and it is given for:

$$\dot{X}_c = \dot{n}_d\left(\sum_{i=1}^{\hat{i}} y_i \bar{x}_i\right)_d + \dot{n}_g\left(\sum_{j=1}^{\hat{j}} y_j \bar{x}_j\right)_g \tag{8}$$

In the expression above is neglected the contribution thermomechanical of the exergy, tends in view that the it is small.

3. RESULTS AND DISCUSSIONS

The Figure 4 presents the diesel mass flows rate measured, in the case of the engine operating just with diesel and with diesel gas, in function of the shaft power. These mass flows rate supply the data of the diesel oil molars flows rate, which are used in the equations above. It can be observed, as it is of hoping, the diesel oil consumption increases when the shaft power produced by the engine increased.

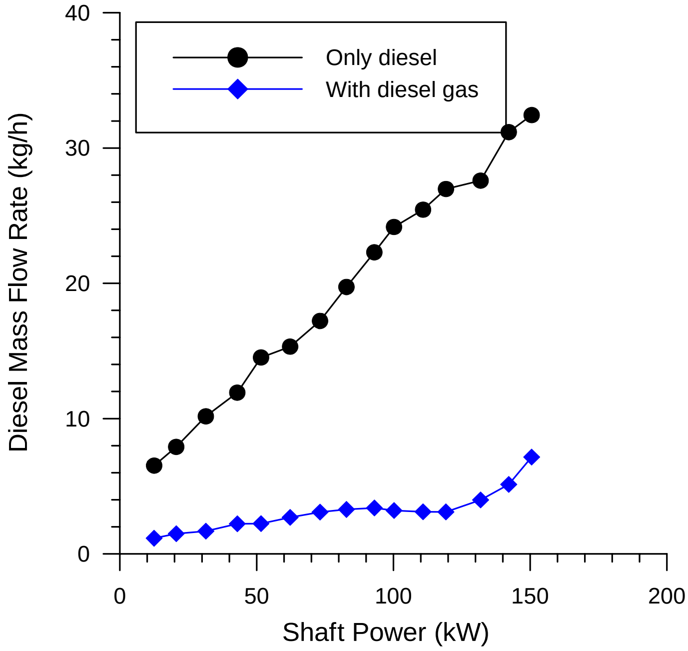


Figure 4. Consumption of the engine in function of the load just operating with diesel and with a mixture of diesel gas.

The Figure 5 presents the mass flows rate gas measured of in function of the shaft power for the engine operating in the dual way with diesel gas. These values supply the molars flows rate of gas, $\dot{n}_g$, which are used in the equations above. It can be observed that as the produced shaft power increases, it increases the amount of gas substituting the diesel oil. As larger the amount of gas consumed, minor will be the amount of diesel oil that feeds the engine. There is a decrease of the consumption of gas for last potency studied due to the detonation that began to happen in this load level for amount of gas supplied. This detonation is observed by way audible.

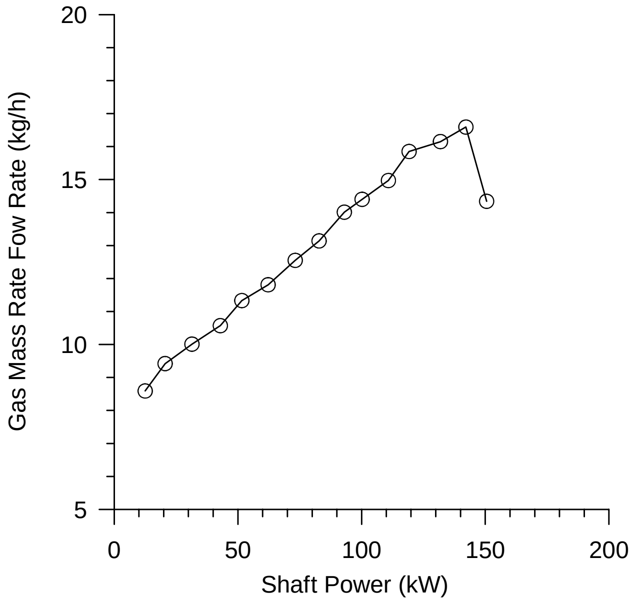


Figure 5. Gas Mass Flow Rate.

The Figure 6 presents the moist air mass flows rate measured in function of the power shaft of the engine for the case of operation with pure diesel and diesel gas. These values supply the oxygen molars flows rate of, $\dot{n}_{O2}$, which are used in the equations above. It is observed that the consumption of air increases when the shaft power increase. In low loads, the consumption of air of the engine operating in a dual way is superior to the consumption when it just operates with diesel. This happens for loads of up to 100 kW. For superior loads, the engine operating in a dual way consumes less air.

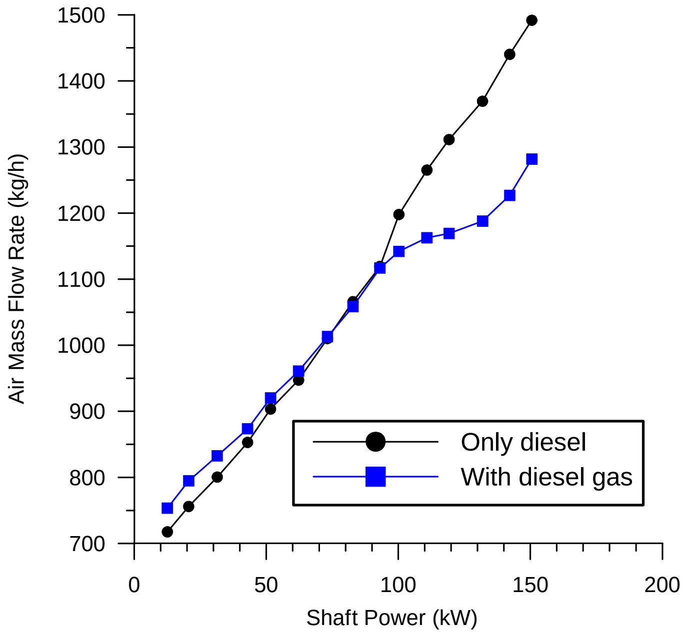


Figure 6. Air Mass Flow Rate.

The Figure 7 presents the percentages in mass of diesel oil that it was substituted in function of the shaft power. It is observed that these percentages vary in agreement with the conditions of load of the engine. The limit of this rate of substitution is defined by the occurrence of detonation and characteristics of the kit. The mass of diesel substituted by gas is defined by the expression:

$$\text{Mass of Diesel Substituted by Gas} = \left(\frac{\dot{m}_d - \dot{m}_{dg}}{\dot{m}_d}\right) \times 100\ \% \tag{9}$$

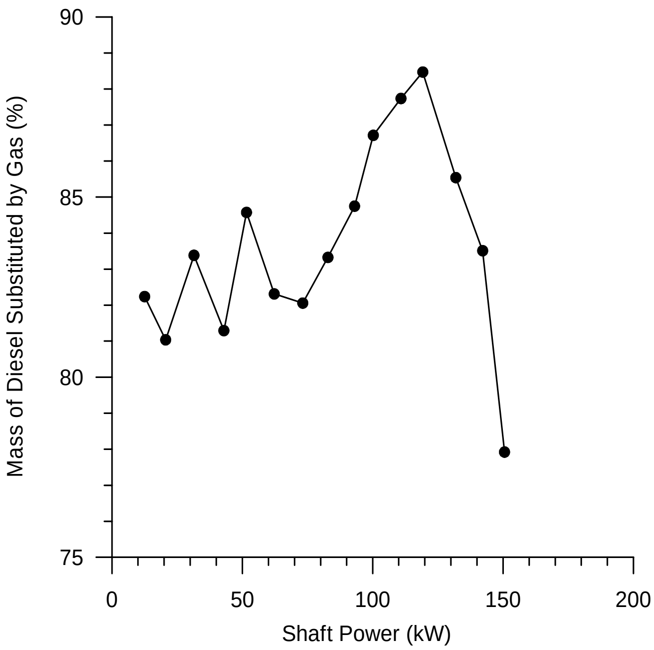


Figure 7. Percentage of the diesel mass substituted by gas.

where $\dot{m}_d$ is the rate of diesel to the engine operating with pure diesel and $\dot{m}_{dg}$ is the rate of diesel to the engine operating with mixture diesel-gas.

The Figure 8 presents the heat transfer rate to the environment in function of the shaft power of the engine for the case of operation with pure diesel and diesel gas. The heat transfer rate of the engine was calculated by the Eq. (3).

It can be observed that the losses of energy for heat of the engine operating with pure diesel is smaller than those of the engine operating in a dual way for loads to about 50 kW. For loads larger than 50 kW happen an inversion, in other words, the heat loss of the engine is the inferior when the same is operating in the dual way.

The Figure 9 presents the thermal efficiency based in PCI in function of the shaft power of the engine for the case of operation with pure diesel and diesel gas. The efficiency of the engine, as it can be observed, operating in a dual way with diesel gas overcomes the efficiency of the engine just working with pure diesel for superior potencies to 60 kW. The efficiency of the tested engine arrived the approximately 55% in the dual way the full load against the 40% for the engine operating with pure diesel.

The Figure 10 presents the efficiency exergetic in function of the shaft power of the engine for the case of operation with pure diesel and diesel gas. The efficiency of the engine can be observed operating in a dual way with diesel gas overcomes the efficiency of the engine just working with pure diesel for superior potencies to 60 kW.

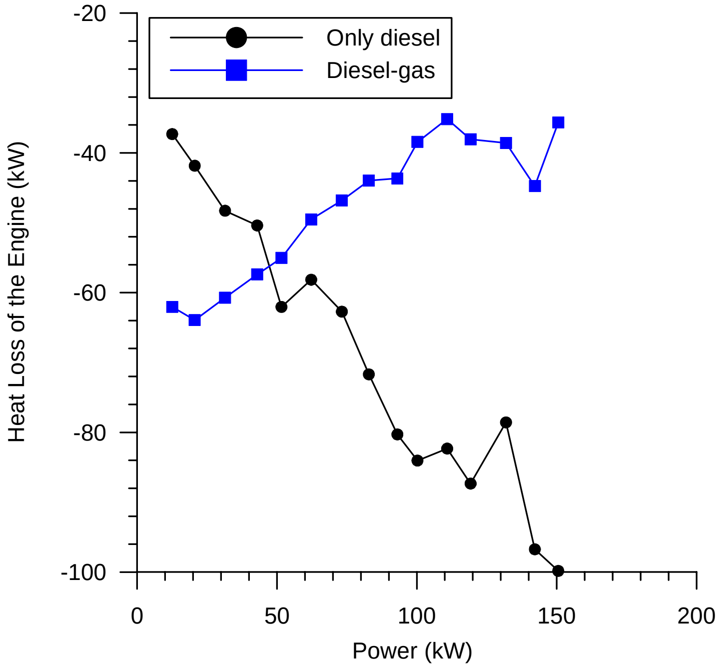
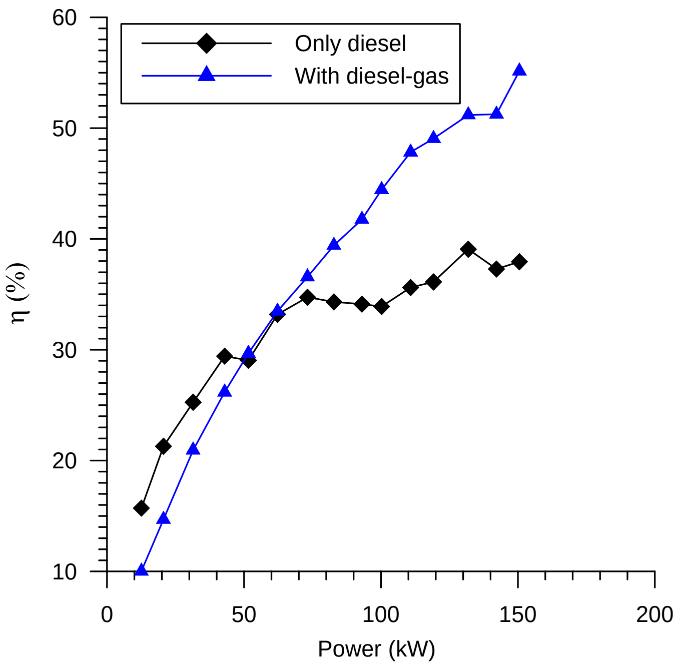


Figure 8. Heat loss of the engine to environment.



Figue 9. Efficiency based on PCI of the fuel.

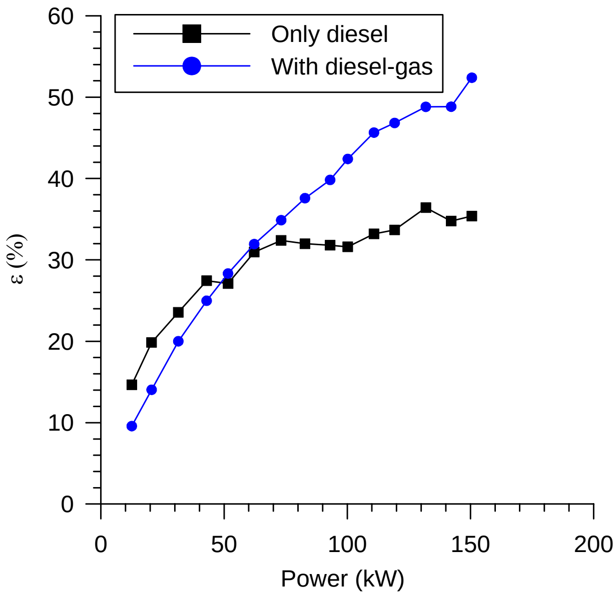


Figure 10. Efficiency exergetic.

4. CONCLUSIONS

The energetic efficiency of the engine operating in the dual way is higher that obtained in the way pure diesel, for superior shaft power to 60 kW. In the shaft power of approximately 10 kW, the values were from 15% to the condition of operation pure diesel. For the dual condition (diesel-gas), the values were of 10%. For the shaft power of 150 kW, the efficiency in the dual way is quite superior to the way pure diesel, arriving to values of 55%.

The amount of losses by heat for the atmosphere predicted by the Eq. (3), when the engine operates with pure diesel (approximately 100 kW) it was superior that obtained in the dual way (approximately 67 kW), in the shaft power of 150 kW.

The energetic efficiency predicted by the Eq. (5), it varied from 15,7 to 37,9% in the way pure diesel and from 10,02 to 55,13% in the dual way for the potency about 150 kW.

The exergetic efficiency predicted by the Eq. (7) varied from 14,6 to 35,4% in the way pure diesel and from 9,57 to 52,38% in the dual way, values considered low, indicating the need of studies to use the losses of energy for the atmosphere in the form of heat and combustion products.

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