ENERGETIC AND EXERGETIC ANALYSES OF A DIESEL ENGINE OPERATING IN A DUAL FORM

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Abstract. *Given the latest global oil crisis which caused a consequent increase in its price and also the expected shortage combined with an increasing awareness of the need for using this fuel in the most rational way possible, new energy sources and more efficient and less polluting combustion processes have been sought to be used. Among the cleaner fuels is natural gas whose consume has presented a very large growth in recent years.*
Even considering the new technologies of electronic injection system of fuel, the use of natural gas in diesel engines is limited to low power systems. Moreover, due to complexity of the issue, new studies have been conducted to improve the performance of the engine when operating in dual form (fed by a blend of diesel and natural gas).
In this sense, the aim of this work is to investigate theoretical and experimentally the performance characteristics of a commercial diesel engine being operated with natural gas and diesel. Experimental facility (thermal system) is composed by a diesel engine coupled to an electronic generator with measuring sensor for temperature, pressure, air, natural gas and diesel flow meters, gas transducers, gas analyzer and power absorption system, constituted by an electric charge bank and its controlling system. For energetic and exergetic analysis of such dual engine, a mathematical model based on the thermodynamics concepts was developed. Numerical and experimental results concerning the effect of air conditions, the type and quantity of fuel used and the exhaustion gases over the engine performance and environmental impact are presented and analyzed. In this work, the diesel engine operated with powers ranging from 10 to 150 kW and replacement rates from 60% to 85%.

Keywords: *Diesel engine, dual engine, energy, exergy*

1. INTRODUCTION

The world has great interest concern over the environmental impact and/or exhaust emissions of fossil fuel combustion mainly those related to internal combustion engine. Natural gas appear as an attractive energy source to be used as a fuel in combustion process for power production in internal combustion engines due to the physical and chemical properties and it is available in great quantities in many location of the world, thus resulting in efficient combustion and substantial reduction of emissions (Nwafor, 2000). Due to this important characteristic natural gas has been used as alternative fuel in diesel engine so called as dual-fuel engine. In dual-fuel engine a mixture of air, diesel and natural gas is compressed and them fired by ignition at the end of compression phase. According to Mansuor et al. (2001) the advantage of this type of engine resides in the fact that it uses the difference of flammability of two fuels. In addition we can cite the economical and environmental potential benefits of using natural gas in diesel engine.

The performance of dual-fuel engine (gas-diesel) has been investigated for many researches with promising results (Mansour et al., 2001; Papagiannakis and Hountalas, 2004; Uma et al., 2004 Papagiannakis and Hountalas, 2003), but no works has been carried out to energy and exergy analysis.

Szargut (1988) defines exergy as the amount of work obtainable when some matter is brought to a state of thermodynamic equilibrium with the common components of the natural surroundings by means of reversible process, involving interaction only with this components of nature. The exergy is the energy that can be completely converted into mechanical energy. It can be established as the most appropriate standard for evaluating the variation in the quality of energy in the analysis of thermal systems (Kotas, 1985). When exergetic analysis is carried out, it is possible to identify the points where losses occur, ie where is the destruction of exergy. This destruction of Exergy is a function of the irreversibility of the process or degradation of the quality of energy resources (Kotas, 1985). Them exergy analysis can be used to indicate possible ways of improving of thermal and chemical processes and therefore, what areas should receive special technical. However, exergy analysis cannot state whether as not the possible improvement is practicable (Szargut et al., 1988).

In this sense, the aim of this work is to describe a theoretical and experimental procedure to analysis the energetic and exergetic performance of a dual-fuel diesel engine when liquid diesel is partially substituted by natural gas under ambient intake temperature.

2. EXPERIMENTAL METHODOLOGY

2.1 Description of natural gas and diesel fuel composition used in this experiment.

The chemical composition of the natural gas and liquid diesel fuel used in the experiment are presented in Table 1. These values are representatives of typical commercial fuels supplied in Campina Grande City, Paraíba State, Brazil.

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

Fuel (source)	Chemical composition (in volume)						
Diesel				$C_{12}H_{26}$		S	
(Medeiros et al., 2002)				98.53 %		1.47 %	
Natural Gas	CH_4	C_2H_6	C_3H_8	C_4H_{10}	N_2	CO_2	O_2
(PBGAS, 2006)	89.42%	7.24%	0.16%	0.18%	1.27%	1.66%	0.08%

2.2 Experimental apparatus and procedures

The electro-mechanical system studied consists of a commercial engine (Cummins 6CTA), with mechanical power of 188 kW to the 1800 rpm, coupled to a electric generator Onan Genset of 150 kW. The unit is totally scored with air, gas and diesel flow meters, temperature and pressure sensors in several points of the system and probe for gas analysis. Experimental data of mass flow rates, temperature and pressure of air, gas and diesel, and exhaust emissions are collected in real time through data acquisition system. Figure 1 presents the electromechanical system. Detail about the experimental apparatus and procedure can be found in: Costa, (2007) and Costa, et al (2008).

Figure 1 – Electro-mechanical system

3 MATHEMATICAL MODELING

Consider the Figure 2, which shows an engine schematic form. In the analysis, it is considered that the fuel enters in the engine, with mass flow rate $\dot{m}_c$ and it mixed with a quantity of air $\dot{m}_a$. Both the air and fuel has kinetic and potential energy variations negligible. The fuel enters for the engine at temperature T_c and pressure P_c, while the air enters at temperature T_a and pressure P_a. The mixture burns completely and the combustion products leave the engine at temperature T_p and pressure P_p with mass flow $\dot{m}_f$. The engine develops a power output and transfers a quantity of heat to the environment $\dot{Q}$. All the exchanges of energy between the lubrification oil and refrigeration water with the body of the engine, and air with the turbo-compressor and cooler, are contained within the control volume that involves the engine, and therefore their effects are contained in $\dot{Q}$. The internal combustion engine operate at steady-state.

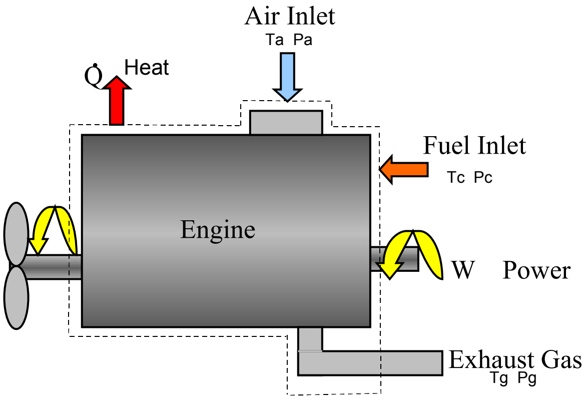

Figure 2 - Schematic of the fuel diesel engine

3.1 Mass conservation

For steady state situation, all properties are unchanging in time. In this case, for a control volume presented in Figure 2, we can write:

$$\sum_e \dot{m}_e = \sum_s \dot{m}_s \tag{1}$$

where $\dot{m}_e$ and $\dot{m}_s$ represent inlet and outlet mass flow rate, respectively.

3.2 Energy conservation

The equation of the first law of Thermodynamics is used to determine the heart transfer involved in the analysis of the engine. The energy rate balance at steady start can be writhen as:

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

where $\dot{Q}, \dot{m}$, h, $V^2/2$ and gZ are respectively: heat transfer rate, mass flow rate, specific enthalpy, specific kinetic energy and specific potential energy, and $\dot{W}$ is the useful power developed by the engine.

The equation (2) allows an assessment of the heat transfer rate that is lost by the engine to the environment. Kinetic and potential energy effects are small and therefore are neglected. So, the equation (2) can be written as follows:

$$\dot{Q} + \sum \dot{m}_e h_e = \sum \dot{m}_s h_s + \dot{W} \tag{3}$$

In dual-fuel engines, diesel natural gas and air burns completely given as results combustion products. In terms of diesel, natural gas, air and combustion products composition on dry basis, the equation (3) can be written as:

$$\begin{aligned}
&\dot{Q} + \dot{n}_d(y_{C_{12}H_{26}}M_{C_{12}H_{26}}h_{C_{12}H_{26}} + y_s M_s h_s)_d + \dot{n}_g(y_{CH_4}M_{CH_4}h_{CH_4} + y_{C_2H_6}M_{C_2H_6}h_{C_2H_6} \\
&+ y_{C_3H_8}M_{C_3H_8}h_{C_3H_8} + y_{C_4H_{10}}M_{C_4H_{10}}h_{C_4H_{10}} + y_{N_2}M_{N_2}h_{N_2} + y_{CO_2}M_{CO_2}h_{CO_2} + y_{O_2}M_{O_2}h_{O_2})_g \\
&+ \dot{n}_{O_2}(M_{O_2}h_{O2} + 3{,}76M_{N_2}h_{N2} + 7{,}655M_{H_2O}h_{H_2O})_a = \dot{n}_P(y_{CO_2}M_{CO_2}h_{CO_2} \\
&+ y_{CO}M_{CO}h_{CO} + y_{N_2}M_{N_2}h_{N_2} + y_{NO}M_{NO}h_{NO} + y_{NO_2}M_{NO_2}h_{NO_2} + y_{SO_2}M_{SO_2}h_{SO_2} \\
&+ y_{CH_4}M_{CH_4}h_{CH_4} + y_{O_2}M_{O_2}h_{O_2})_p + \left(\dot{n}_{H_2O}M_{H_2O}h_{H_2O}\right)_{H_2O} + \dot{W}
\end{aligned} \tag{4}$$

where: $\dot{n}_d, \dot{n}_g, \dot{n}_{O_2}, \dot{n}_P, \dot{n}_{H_2O}$ are molar flow of diesel, natural gas, oxygen, combustion products and water vapor, respectively. In the equation (4) w is the absolute humidity of air, and y_i and M_i are the molar fractions and molecular weights of each component i involved in the combustion process. The molar fractions for the diesel and natural gas are provided in Table 1. The molar fractions of combustion products on the dry basis, y_{CO2}, y_{CO}, y_{NO}, y_{NO2}, y_{CH4} and y_{O2} are measured with the gas analyzer. The other coefficients of the equation are obtained from a stoichiometric balance for each chemical element involved in the process.

In the molar basis equation (4) can be written as follows:

$$\dot{Q} + \dot{n}_d\left(\sum_{i=1}^{\hat{i}} y_i\overline{h}_i\right)_d + \dot{n}_g\left(\sum_{j=1}^{\hat{j}} y_j\overline{h}_j\right)_g + \dot{n}_{O_2}\left[\overline{h}_{O_2} + 3{,}76\overline{h}_{N_2} + 7{,}655\,w\,\overline{h}_{H_2O}\right] = \dot{n}_p\left(\sum_{k=1}^{\hat{k}} y_k\overline{h}_k\right)_{\text{Dry Products}} \quad n_{H_2O}\,\overline{h}_{H_2O} + \dot{W} \quad (5)$$

The specific enthalpy and molar basis is calculated by:

$$\overline{h} = \overline{h}_f^0 + \Delta\,\overline{h} \tag{6}$$

where $\overline{h}_f^0$ is the enthalpy of formation on molar basis and the specific heat capacity in molar basis $\Delta\,\overline{h} = \int_{T_{ref}}^{T} \overline{c}_p \Delta\,T$

beig $\overline{c}_p = Mc_p$.

The second term accounts for the change in enthalpy from the temperature T_{ref} to the temperature T.

The molar specific heat capacity and enthalpy formation can be foundfor example, in Van Wyley and Sonntag, (1976); Incropera and DeWitt, (2002) and Naterer, (2003); Moran and Shapiro (2000); Kotas (1985); Szargut et al. (1988) and Li (1996).

The thermal efficiency of the engine is calculated by the following expression:

$$\eta_t = \dot{W} / PCI \tag{7}$$

where PCI is the Lower Heating Value of the mixture. So, the energy efficiency of the engine can be written more appropriately as:

$$\eta_t = \frac{\dot{W}}{\dot{n}_d\left[\sum_{i=1}^{k}\left(y_i M_i PCI_i\right)\right]_d + \dot{n}_g\left[\sum_{j=1}^{r}\left(y_j M_j PCI_j\right)\right]_g} \tag{8}$$

where i and j denote the components of diesel and natural gas, respectively, and is the molecular weight.

3.3 Exergetic efficiency

In many applications, the means of work consists of a mixture of ideal gases, for example, gaseous fuels, combustion products, etc. When a hydrocarbon fuel C_aH_b or another substance is a component of a mixture of ideal gases in the standard state (T_0, P_0), the hydrocarbon fuel or another substance is in the state (T_0, y_i P_0). In this case that, the chemical exergy of the fuel or substance is given by:(Li, 1996; Kotas 1985).

$$\overline{x}_i^{\,chem}(T_{0i} y_i P_0) = \overline{x}_i\left(T_0, P_0\right) + \overline{R}T_0 \ln(\mu_i y_i) \qquad \text{with } \mu_i = 1 \tag{9}$$

In the equation (9) y_i is the molar fraction of component i in the mixture of hydrocarbon fuel. In this equation, we consider the coefficient of activity $\mu_i = 1$

In the ambient conditions (standard reference state), the thermodynamic exergy is null. Then, the total exergy for a fuel is exactly equal to the chemical exergy. In the case studied, the exergy is given by:

$$x_{\text{diesel or gas}}^{chem} = \frac{\overline{x}_{\text{diesel or gas}}^{chem}}{M_{\text{diesel or gas}}} = \left(\frac{\sum_{i=1}^{n} y_i \overline{x}_i^{chem}}{M}\right)_{\text{diesel or gas}} \tag{10}$$

For the determination of the chemical exergy of the exhaust gas, we use the equations 9 and 10, and the molar fraction of components obtained experimentally.

$$\dot{X}_{chem} = \dot{n}_p \sum_{i=1}^{\hat{i}} y_i \bar{x}_i + \dot{n}_{H_2O} \bar{x}_{H_2O}^{chem} \tag{11}$$

The exergetic efficiency for the engine is determined with the aid of a exergy balance. At steady state the rate at which exergy enters the engine equals the rate at which exergy exist plus the rate at which exergy is destroyed within the engine. In this statement, the air used for combustion enters at the environmental conditions, and consequently with a value zero of exergy, only fuel provides exergy to the engine. The exergy exist the engine accompanying heat and work, and with the combustion products. If the engine power developed is taken to be the product of the engine, and the heat transfer and gases produced in the output are seen as losses, an expression for the exergetic efficiency ε, which measures how much exergy at the entrance of the engine is converted to the products, is of the form:

$$\varepsilon = \frac{\dot{W}_{vc}}{\dot{X}_c} \tag{12}$$

where $\dot{X}_c$ denotes the rate at which exergy enters with the fuel. This parameters is given by:

$$\dot{X}_c = \dot{n}_d \left(\sum_{i=1}^{\hat{i}} y_i \bar{x}_i \right)_d + \dot{n}_g \left(\sum_{j=1}^{\hat{j}} y_j \bar{x}_j \right)_g \tag{13}$$

For obtain the exhaust gas, heat, work and destroyed exergy we use the following equation:

$$\dot{X}_{Thermo} = \dot{n}_p \sum_{k=1}^{\hat{k}} y_k M_k [(h_{\hat{k}} - h_{k0}) - T_0 (s_{\hat{k}} - s_{k0})] + \dot{n}_{H_2O} [(h_{h_2O} - h_{h_2O_0}) - T_0 (s_{H_2O} - s_{H_2O_0})] \tag{14}$$

$$\dot{X}_{gas} = \dot{X}_{thermo} + \dot{X}_{chem} \tag{15}$$

$$\dot{X}_{heat} = (1 - T_0 / T_m) \dot{Q} \tag{16}$$

$$\dot{X}_{work} = \dot{W} \tag{17}$$

$$\dot{X}_{destroyed} = \dot{X}_{total} - \dot{X}_{gas} - \dot{X}_{work} - \dot{X}_{heat} \tag{18}$$

In the equations (13) the entropy was obtained in the partial pressure of each component in the mixture. We considers T_m = 313.15 K. (Temperature of the system boundary where heat transfer)

4 RESULTS

4.1 Replacement rate and exhaust emissions.

The experiments analyzed in this study were those related to higher replacement rate for each power. These values were then used to evaluate engine performance. The average values of the replacement rate was 83.78%. The replacement rate was calculated as follows (Uma et al., 2004):

$$\text{Replacement rate (\%)} = \left(\dot{m}_d - \dot{m}_{dg} \right) / \dot{m}_d \times 100 \tag{19}$$

where $\dot{m}_d$ is the mass flow rate of the fuel when the engine operating with pure diesel, and $\dot{m}_{dg}$ is the mass flow rate of fuel when the engine operates in a dual form (gas and diesel). Information about exhaust emissions can be found in Costa et al. (2008). Figure 3 presents the replacement rate as function the power.

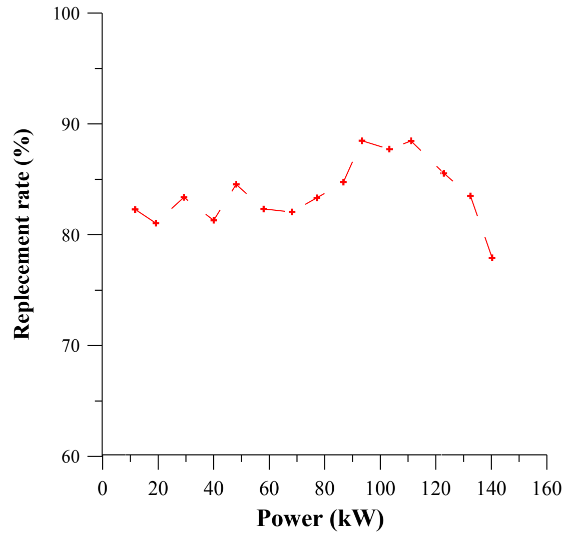

Figure 3 – Replacement rate as a function of the load of engine.

4.2 Combustion efficiency as a function of load applied

The combustion efficiency shown in Figure 4 is calculated by the gas analyzer as follows:

$$\eta_{comb} = 100\% - DFGL \quad (20)$$

where $DFGL = 20.9 \times K_{1n} \times Tnet/[K_2 \times (20.9 - \%O_2)]$, being $K_{1n} = 0.515$ and $K_2 = 15.51$ constants of the analyzer when pure diesel is used as combustible and $K_{1n} = 0.393$ and $K_2 = 11.89$ when using natural gas. In the present work we use the constants of the natural gas for the engine operating of dual form. In this figure, we can see that for low powers, the combustion efficiency is lower (dual form), when compared to pure diesel. This efficiency increases with applied load , by presenting a value of approximately 78% in the power of 150 kW. This behavior of the combustion confirms the results of the emissions obtained during the tests.

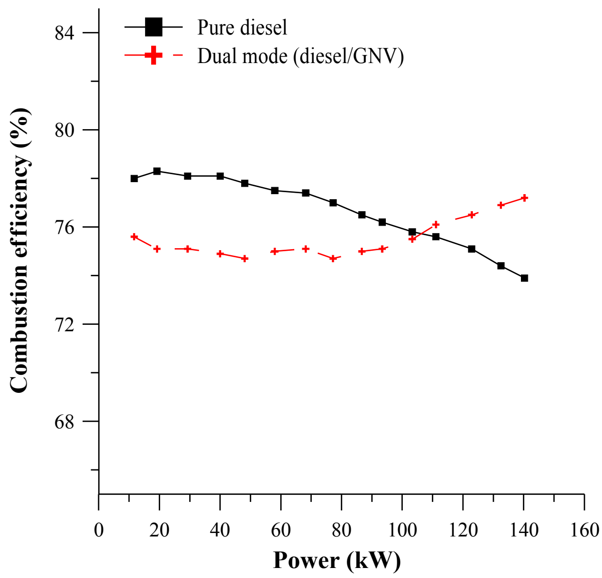

Figure 4 - Combustion efficiency as a function of the power.

4.3 Energy analysis

Figure 5 shows the thermal efficiency of the engine in similar working conditions, changing only the fuel supplied to the engine. In the operation with pure diesel, we can see that the efficiency of the engine change from 30% (70 kW) to 35% (150 kW). When the engine is operated with the mixture of gas / diesel, there is an increase in the efficiency of the engine. At higher loads, the efficiency of the engine with mixture diesel/gas is higher, reaching values of 53%. For lower powers, the efficiency decreases with the power for the condition of dual-mode operation.

Data published by Braga et al. (2006) shows that the thermal efficiency of an engine of 135 kVA operating with pure diesel, presented value of 22% for loads smaller than 50 kW and reaches the maximum value of 34% for load of 100 kW. Henham and Makkas (1998) reports thermal efficiency of 28.2% when the engine operates at 2000 rpm and 40 Nm of torque. These differences are related T_v the form to calculate this parameter. It can be observed that the power output exceeding 100 kW and operating in dual mode, the combustion efficiency of diesel is over. This increase in efficiency occurs because the ease that this fuel has to be mixed with air, leading to better combustion.

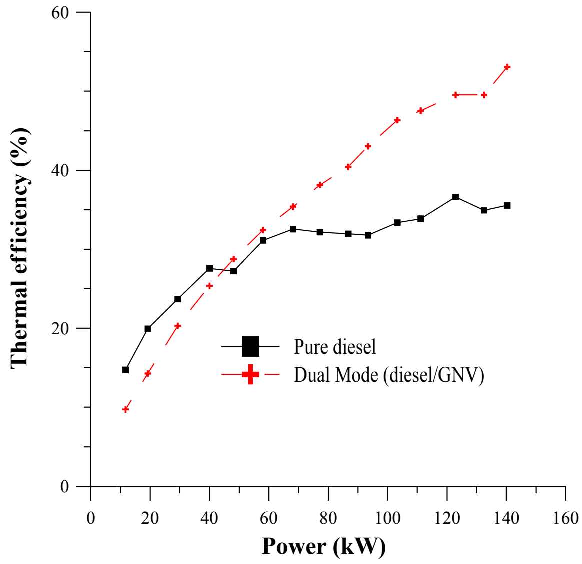

Figure 5 - Thermal efficiency as a function of the power.

Figure 6 shows the waste heat by the engine to the environment per unit of time. We can see that the system running in dual mode, provides a lower loss of heat. The heat transferred to the environment is approximately 40 kW for a power of 150 kW.

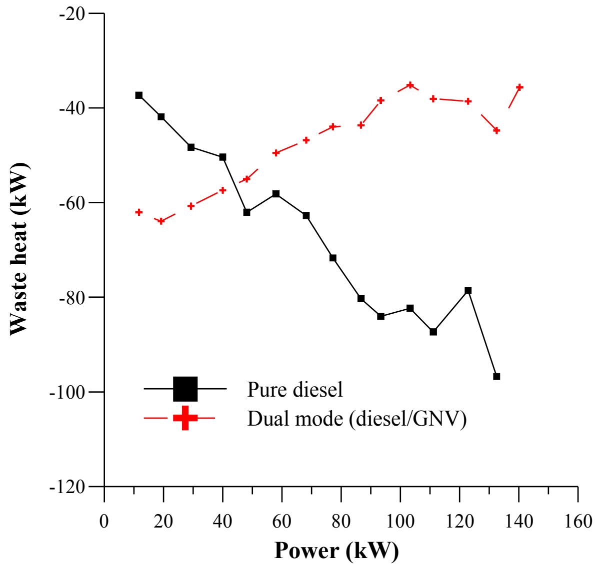

Figure 6 – Waste heat for the environment as a function of the power.

4.4 Exergetic analysis

Figures 7-10 presented shown the heat transferred to the environment, exergetic efficiency, total exergy, exergy destroyed, exergy of heat and the exergy of the combustion products, all expressed in terms of power. Data needed for calculations as molar fraction of exhaust gas, power, mass flow of gas and diesel have been measured.

Figure 7 illustrates the exergetic efficiency as a function of power. It can be seen that there is an increase in the efficiency for loads exceeding 80 kW, when operating in dual mode.

The total exergy for the engine operating in the diesel and dual condition is presented in Figure 8. It is verified that total exergy for the condition pure diesel is higher for powers of 50 kW, reaching value of 425.5 kW for the power of 150 kW. In the dual mode, total exergy is smaller for powers up to 50 kW and growing until the value of 287.3 kW for the power of 150 kW.

Figure 9 shows the behavior of destroyed exergy for the diesel and dual mode condition. For the case where we use pure diesel, it was observed that there is a large destruction of exergy, reaching up to values of 207 kW to 150 kW of power. For the dual condition it is verified that for the power of 10 kW, the values are 100 kW and growing until the value of 106.2 k W for the power of the engine of 150 kW. The heat exergy transferred to the environment as a function of power is shown in Figure 10, where we can see that pure diesel exergy reach 4.7 kW to power of 150 kW and 1.7 kW for engine power of 10 kW. For dual-mode operation there is a decrease of exergy from approximately 3 kW to 10 kW of engine power to approximately 1.7 kW to 150 kW of engine power. These results show that the heat exergy represents a small part of total exergy (≈ 2%).

Figure 11 shows the behavior of the exergy of the combustion products to pure diesel and dual mode. We can check that there is growing of exergy for all conditions of operation (diesel and dual mode). For the pure diesel, exergy change from 7.47 kW in the power of 10 kW to 62.58 kW in the power of 150 kW. For operation in dual-mode, we can see that exergy starting with values of 15.12 kW for the engine power of 10 kW, reaching values of 39.41 kW to 90 kW and decreasing for values of 36.48 kW at the engine power of 150 kW. It is important to note that for engine power up 90 kW, there is now a considerable exergetic potential, which can provide a condition of use of exergy in co-generation system and of heat and cold production.

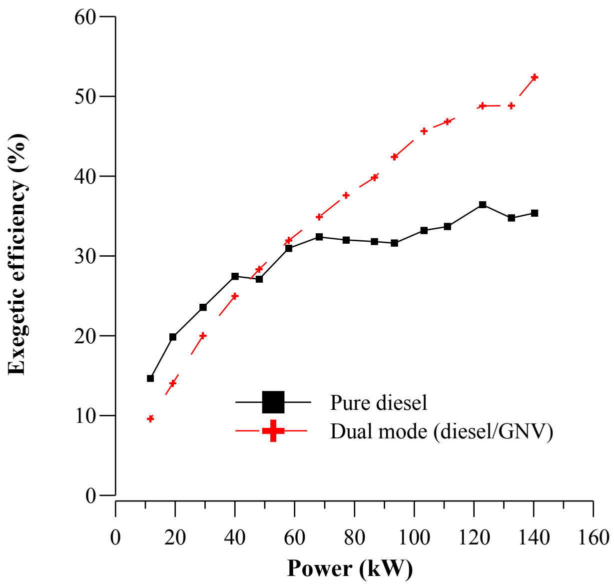

Figure 7 – Exergetic efficiency as a function of the power.

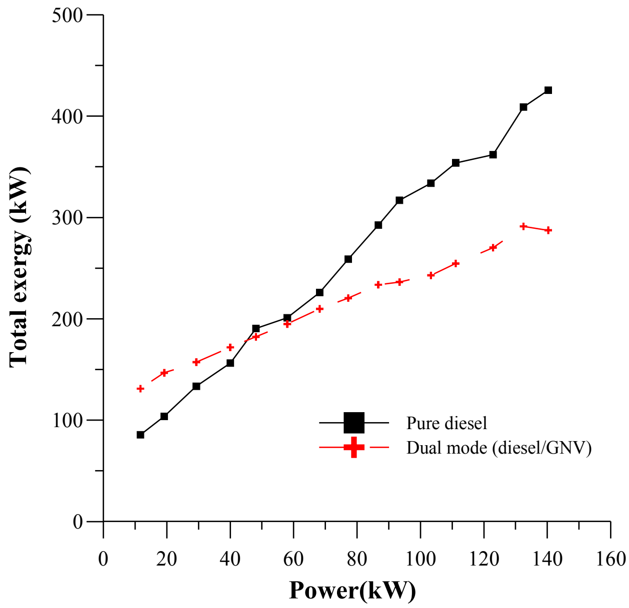

Figure 8 – Total exergy as a function of the power.

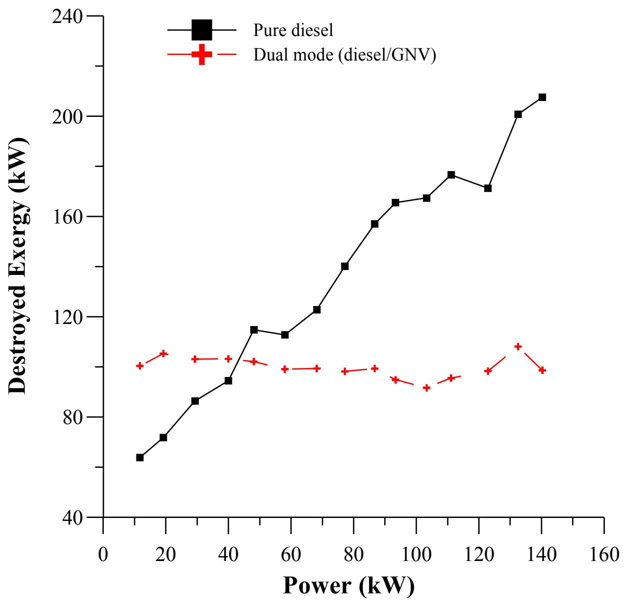

Figure 9 – Destroyed exergy for pure and dual mode as a function of the power.

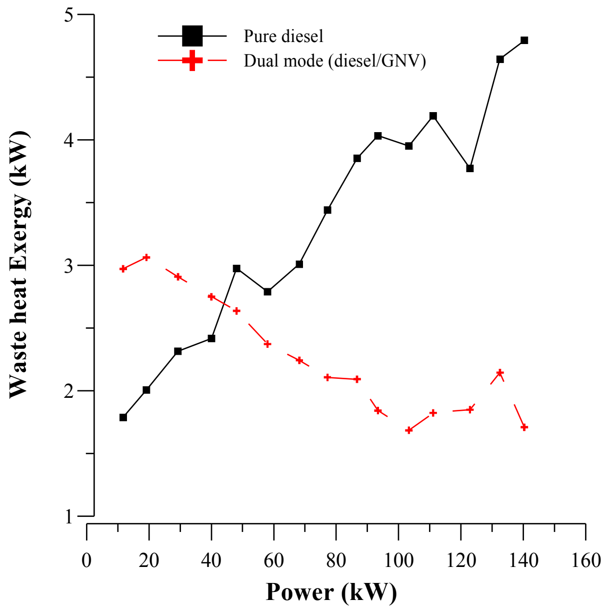

Figure 10 – Waste heat for the environment as a function of the power.

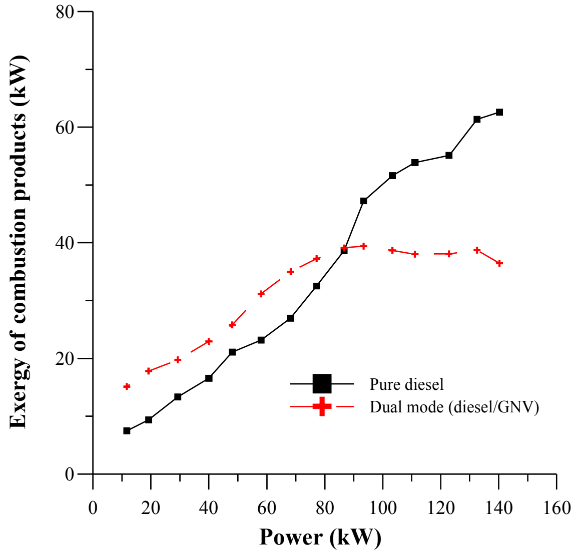

Figure 11 – Exergy of the combustion products as a function of the power.

5. CONCLUSIONS

The following conclusion can be cited:

a) Viability of use of engines diesel to operate of dual form with natural gas was verified.
b) The engine operated of satisfactory form and had been reached replacement rate more then 80% without presenting any abnormality such as detonation.
c) The mathematical model predict satisfactory the process as compared with experimental values and literature.
d) The energy efficiency ranged from 15.7 to 37.9 % in mode diesel pure and 10.02 to 55.13 % in dual mode, to power changing from 10 to 150 kW.
e) The exergetic efficiency ranged from 14.6 to 35.4 % in mode diesel pure and 9.57 to 52.38 % in dual mode, when the power engine changes from 10 kW to 150 kW. Values considered low, indicating the need for studies about lost energy to the environment in the form of heat and exhaust gas.

f) For the engine operating with pure diesel, total exergy presented values ranging from 85 kW to 425 kW, for engine power ranging from 10 to 150 kW. When operated in dual mode, the total exergy presented values 131.0 to 287.3 kW.
g) The destroyed exergy ranged from 100.4 to 98.6 kW, the exergy in the exhaust gas ranging 15.1 to 36.4 kW, while the waste heat to environment presents values – 2.9 to – 1.7 kW, respectively for power changing from 10 to 150 kW, when operated in dual mode.
h) The destroyed exergy ranged from 62 to 206 kW, the exhaust gas exergy ranged from 8 to 62 kW, while the waste heat to environment presents values – 1.7 to – 4.8 kW, respectively for power changing from 10 to 150 kW, when operated in pure diesel.

6. ACKNOWLEDGEMENTS

This work is supported by fund of GEBRA, BRASYMPE and CNPq. The authors would like to thank to the researches reported in the paper that contributes for its improvement.

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8. RESPONSIBILITY NOTICE