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EXPERIMENTAL AND EXERGETIC COMPARISON OF A DIESEL ENGINE FUELED WITH SUNFLOWER OIL AND TUNG OIL

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Abstract. *The vegetable oils can be an energetic alternative for diesel generators. This paper presents an experimental and exergetic analysis of a diesel engine fueled with sunflower oil, tung oil and diesel oil. The straight vegetable oils, respective blend 50/50 v/v with diesel oil and the diesel oil were tested on a dynamometric bench with a single cylinder, four strokes, direct injection diesel engine and 14.7 kW of power. The physical-chemical properties of the tested fuels and the engine parameters as torque, power and break specific fuel consumption (BSFC) were measured. The biofuels were heated to reduce their viscosity and specific mass. The thermodynamic analysis was carried out for each fuel at 1800 rpm. The standard fuel was diesel oil. Two statistical techniques, Analysis of Variance (ANOVA) and the Dunnett´s test were utilized to analyze the results. Reduction in power, increase in break specific fuel consumption, and similar exergetic efficiency were found in the engine performance with vegetable oils in comparison with diesel oil. These results demonstrated the technical feasibility to operate diesel engines with these biofuels.*

Keywords: *biofuels, vegetable oils, tung oil, sunflower oil, compression-ignition engine.*

1. INTRODUCTION

Problems related of the fossil fuels dependency have motivated several studies about the applicability of biodiesel and of vegetable oils in compression-ignition engines. The vegetable oils are chosen with regard to the production of oilseed of each region. Noteworthy, soybean and sunflower oils in the United States and rapeseed oil in Europe. Palm, jatropha, cotton and coconut oils have been tested in Asia (Mondal *et al*., 2008). In Brazil, greater interest has been reserved for soybean oil, palm oil and castor oil.

An alternative fuel must present physical and chemical properties close to standard fuel. For the use in compression-ignition engines, the key properties are viscosity, specific mass and Lower Heating Value (LHV). Vegetable oils are mostly polyunsaturated triglycerides. The unsaturation bonds of its carbon chain and the large molecular size produce high viscosity, high specific mass, low volatility and less LHV when compared to diesel oil (Agarwal *et al*., 2008; Rakopoulos *et al*., 2006). High viscosity causes poor atomization, large droplet size and high penetration of the jet. The fuel is not well distributed and mixed for burning into the combustion chamber, leading to poor combustion with power loss (Agarwal *et al*., 2008; Franco and Nguyen, 2011).

Different procedures are found in the literature to reduce the viscosity of vegetable oils. Some researchers tested the heating of the vegetable oil before of its injection in the engine (Almeida *et al*., 2002; Hartmann *et al*., 2012; Nwafor, 2004; Pugazhvadivu and Sankaranarayanan, 2010; Venkanna *et al.*, 2009). Others researchers blended the vegetable oils with diesel oil in different proportions without heating (Agarwal *et al*., 2008; Chalatlon *et al.*, 2011; Nwafor and Rice, 1996). Another proposed solution is to increase the injection pressure to improve the spray characteristics (Sarada *et al.*, 2010; Venkanna *et al.*, 2009).

The assessment of alternative fuels involves tests in dynamometric bench and thermodynamic analysis of engine performance. Statistical techniques are useful to compare results. Tat (2011) used Student's t-test to compare the exergetic efficiency of four types of biodiesel with different cetane numbers. Nevertheless, there are few studies in the literature where these statistical techniques have been applied with regard to testing of vegetable oils in a diesel engine. In this work, an experimental and exergetic comparison on the performance of a direct injection diesel engine is presented. The diesel engine was operated with 100% raw sunflower oil, 100% raw tung oil and respective blends with diesel oil in volumetric proportion 1:1. These vegetable oils are well suited in southern Brazil. The biofuels were tested in dynamometric bench and the preheating was the mechanism used to reduce the viscosity. The engine efficiency operating with each fuel was evaluated through of the first and second laws of thermodynamic. The results were compared with those obtained for diesel oil using two statistical techniques: analysis of variance (ANOVA) and Dunnett's test. Dunnett's test is appropriate for the comparison with a standard parameter, which in this case was diesel oil. The statistical comparison of the exergetic efficiency and of the exergy destruction did not reveal differences between all the fuels tested in the diesel engine.

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2. CHARACTERIZATION AND TESTING

2.1 Fuels

The fuels tested were raw sunflower oil, raw tung oil, commercial Brazilian diesel oil and respective blends with diesel oil in volumetric proportion of 1:1. The diesel oil was standard fuel. Commercial Brazilian diesel oil has a volumetric addition of 5% of biodiesel in accordance with national regulations. The vegetable oils used were raw, that is, they had undergone no refining process. The dynamic viscosity of the fuels was measured at different temperatures with a viscometer of concentric cylinders (Thermo Electron Corporation, model HAAKE VT550). Figure 1 presents the results of the dynamic viscosity. These results were utilized to determinate the heating temperature of each fuel before entering the injection pump with aim to approximate its viscosity to that of diesel oil at room temperature. The injection temperature and the label of each fuel are presented in Tab. 1. The 100D oil was admitted into the injection pump at room temperature. It can be observed that the 100TG oil presents the highest viscosity value, about sixty times the viscosity value of 100D oil. The blends presented reduction of the viscosity in comparison of the straight vegetable oils. The viscosity variation with the heating was more significant for the vegetable oils.

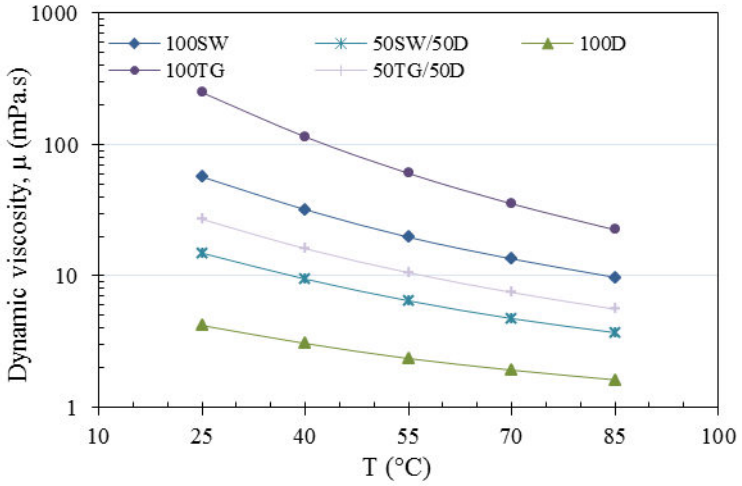

Figure 1. Dynamic viscosity of the fuels tested.

Table 1. Label and injection temperature of the fuels tested.

Fuel	Label	Injection temperature [°C]
Sunflower oil	100SW	85
Tung oil	100TG	95
50% Sunflower oil-50% Diesel oil	50SW/50D	65
50% Tung oil - 50% Diesel oil	50TG/50D	85
Diesel oil	100D	25

Table 2. The physical and chemical properties of the fuels tested[a].

Property	100SW	50SW/50D	100TG	50TG/50D	100D
C (wt. %)	77.61	80.94	77.51	80.88	84.6[b]
H (wt. %)	11.29	13.25	11.40	13.29	15.4[b]
O (wt. %)	11.10	5.81	11.09	5.83	-
S (mg/kg)	0.6	862	1.49	854	1800[c]
ρ (25°C) (kg/m^3)	919	882	934.8	889.9	845[d]
LHV (kJ/kg)	36212	39122	35810	38885	42287[c]

[a] Proprieties measured by National Institute of Technology, Rio de Janeiro.
[b] Taken as $C_{12}H_{26}$.
[c] Estimated by National Petroleum Agency.
[d] Measured at Laboratory of Combustion and Thermal Systems Engineering, Federal University of Santa Catarina.

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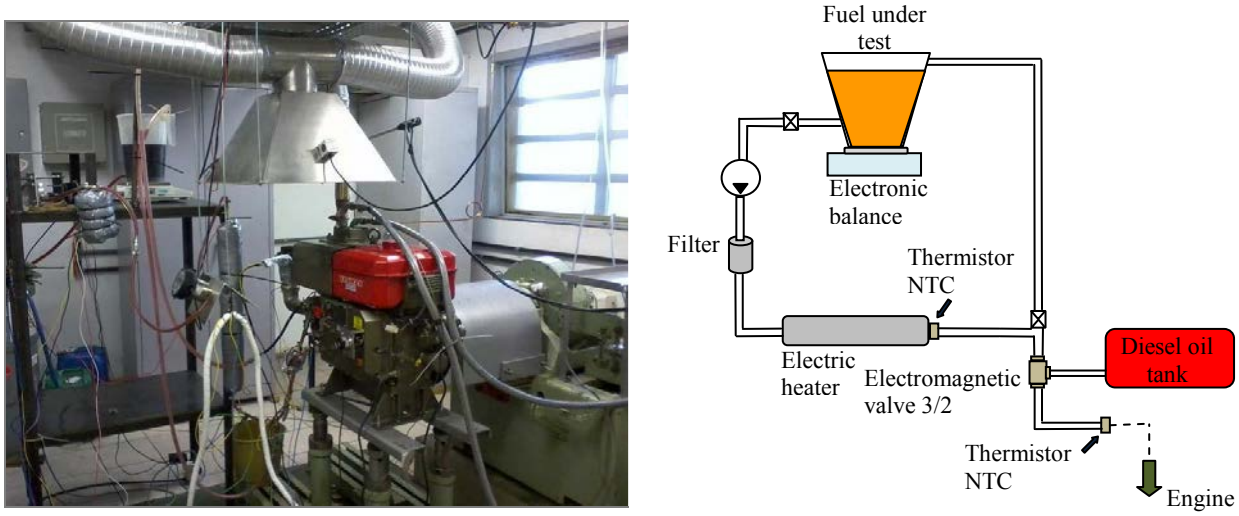
The main physical and chemical properties of the sunflower oil, tung oil, diesel oil and the blends are presented in Tab. 2. The 100SW oil and the 100TG oil present contents of carbon and hydrogen very close. The LHV for 100SW and 100TG oils is 14.4% and 15.3% lower, respectively, when compared with diesel fuel.

2.2 Experimental setup

The experimental setup is composed for a single cylinder, four strokes, direct injection diesel engine (Yanmar, model YT22) and 14.7 kW of power coupled to electromagnetic dynamometer (Schenk, model W70). The dynamometric bench is equipped with a fuel supply system, emission measurement system and control and data acquisition system. An electric heater comprised of an aluminum tube (12.7 mm diameter), electric resistance of 108 Ω and ceramic insulation was manufactured to heat the biofuels before its entry to the engine. Type-J thermocouples were installed at the engine cooling water inlet and outlet, in the exhaust manifold and the intake manifold of the engine. Resistive sensors of the type thermistor Negative Temperature Coefficient (NTC) were installed at the heater outlet and the injection pump inlet. An electromagnetic valve 3/2 was installed to switch the fuel employed. This enables the passage of the diesel oil from the original tank or of the fuel under test. The fuel consumption was measured using an electronic balance (Shimadzu, model 8200S UX) with serial communication. The instant reading of the data allowed the calculation of the fuel flow. The engine torque was measured with an extensometer type load cell installed on the dynamometer arm. The speed was measured with an incremental encoder 60 pulses/second coupled to the dynamometer shaft. The concentration of the exhaust gases (CO, CO_2 and NO_x) was measured with a portable gas analyzer (Testo, model 350-XL). The probe of the exhaust gas analyzer was installed in the exhaust pipe. The tests were controlled through of an electronic control system developed with software LabVIEW 7.1. Figure 2 presents the dynamometric bench and its fuel supply system.



(a) (b)

Figure 2. (a) Dynamometric bench. (b) Fuel supply system of the dynamometric bench.

2.3 Experimental procedure

The tests were performed at the maximum flow rate of the injection pump. In each test the engine operation was started with diesel fuel until the heating period was completed, that is, when the cooling water temperature reached 70 °C. At same time, the biofuel was heated by recirculation on the fuel supply system. The diesel oil was then replaced by the fuel under test. After stabilizing the engine, the brake process was started with the dynamometer. The load was applied during the speed range of 2200 rpm to 1400 rpm. For each load the engine operated until steady state, which was verified by the stabilizing of the emissions. Measurements of torque, speed, power output, break specific fuel consumption and emissions of CO, CO_2 and NO_x in exhaust gases were recorded in the steady state. The measurements were registered at intervals of 10 seconds making fifteen readings under each load condition. The tests with each fuel were performed three times to verify the repeatability. For to prevent the clogging of the fuel supply system, in the case of the biofuels, before the end of each test the fuel was changed again, leaving the engine to operate with diesel fuel for a period of 20 minutes. The average results were statistically analyzed applying the ANOVA and Dunnett tests. The statistical analysis was performed considering a confidence interval of 95% using the software Minitab 14. The expanded uncertainties of each measurement are shown in Tab. 3.

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Table 3. Expanded uncertainty of each measurement*.

Measurements	Expanded Uncertainty [%]
Engine Speed	± 2.5
Break Torque	± 2.8
Power	± 2
Fuel Mass Rate	± 5
Specific Fuel Consumption	± 4
CO	± 9
CO_2	± 5
NO_x	± 8
Exhaust gas temperature	± 0.8

*Expanded uncertainty as a percentage of the mean value for a probability of 95%.

3. THERMODYNAMIC ANALYSIS

The thermodynamic analysis was based in the first law and the second law of the thermodynamic. The control volume used is presented in Fig. 3. The measurements obtained in the tests as well as the chemical composition of the fuels were used in the thermodynamic calculations, which were executed using the software Engineering Equation Solver (EES). The following assumptions were made:

- Reference environment T_o = 25°C and p_{ref} = 101325 Pa.
- Steady-state open system.
- Negligible kinetic and potential energy effects.
- The combustion air and exhaust gases are ideal gas mixtures.
- Superficial engine temperature is mean temperature of the cooling water of the engine.

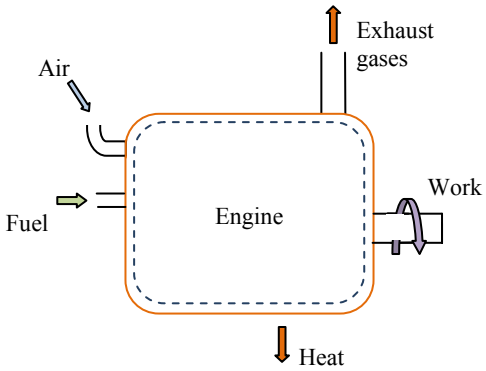

Figure 3. Control volume for the thermodynamic analysis of the diesel engine.

The energy balance for the control volume on the basis of 1 kmol of fuel is written as

$$\frac{\dot{Q}_{out}}{\dot{n}_f} - \frac{\dot{W}_{out}}{\dot{n}_f} = \bar{h}_g - \bar{h}_f - \bar{h}_a \quad (1)$$

where $\dot{Q}_{out}$ is the heat transfer rate, $\dot{W}_{out}$ is the brake engine power, $\dot{n}_f$ is the molar flow rate of the fuel and $\bar{h}_g$, $\bar{h}_f$ and $\bar{h}_a$ are the absolute enthalpies of the exhaust gases, the fuel and the air, respectively. The absolute enthalpy of the fuel is obtained from the formation enthalpy and the sensible enthalpy. The formation enthalpy of the fuel at the reference temperature was determined from the general reaction equation for complete combustion with the theoretical amount of air and considering the LHV as the net energy available under stoichiometric conditions with the reactants and products under the reference conditions. The energy efficiency is defined as the ratio of the brake power to the fuel energy input rate,

$$\eta = \frac{\dot{W}_{out}}{\dot{m}_f LHV} \quad (2)$$

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where $\dot{m}_f$ is the mass flow rate of the fuel. The exergy balance for the control volume on the basis of 1 kmol of fuel is written as

$$\dot{n}_f\left(\overline{e}_f+\overline{e}_a-\overline{e}_g\right)+\dot{W}_{in}-\dot{Q}_{out}\left(1-\frac{T_o}{T_m}\right)-\dot{W}_{out}=\dot{E}_D \tag{3}$$

where $\overline{e}_f$, $\overline{e}_a$, and $\overline{e}_g$ are the specific flow exergies of the fuel, combustion air and exhaust gas, respectively, $\dot{W}_{in}$ is the system power input related to the electric power of the preheating, $\dot{E}_D$ is the exergy destruction rate and T_m is the superficial engine temperature. Because the fuel enters the system in a condition relatively close to the reference state, the specific physical exergy of the fuel was disregarded and was only considered the specific chemical exergy, which was calculated according to Szargut *et al*. (1988) as

$$\overline{e}_f^{ch}=LHV\left(1.0374+0.01594\frac{h}{c}+0.0567\frac{o}{c}\right) \tag{4}$$

The specific flow exergy of the combustion air was neglected due to its entering the system in a state relatively close to the reference state. The specific exergy of exhaust gases is constituted by the specific physical exergy and the specific chemical exergy of the gaseous mixture. The specific physical exergy of the gaseous mixture was calculated as

$$\overline{e}_g^{ph}=\sum_{i=1}^{N}n_i\left\{\overline{h}_{i,(T)}-\overline{h}_{i,o}-T_o\left[\overline{s}_{i,(T,p_o)}-\overline{s}_{i,o}-\overline{R}ln\frac{p_i}{p_o}\right]\right\} \tag{5}$$

where $\overline{e}_g^{ph}$ is specific physical exergy of the exhaust gases, $\overline{h}_{i,(T)}$ and $\overline{h}_{i,o}$ are the enthalpies of the *i*th component at temperatures T and T_o, respectively, $\overline{s}_{i,(T,p_o)}$ and $\overline{s}_{i,o}$ are the entropies of the *i*th component in (T,p_o) and (T_o,p_o), respectively, $\overline{R}$ is the universal gas constant and p_i is the partial pressure of the *i*th component in the gaseous mixture. The specific chemical exergy of the gaseous mixture was calculated as

$$\overline{e}_g^{ch}=\overline{R}T_o\sum_{i=1}^{N}n_i ln\left(\frac{y_i}{y_i^e}\right) \tag{6}$$

where $\overline{e}_g^{ch}$ is the specific chemical exergy of exhaust gases, y_i is the mole fraction of the *i*th component in the exhaust gases for (T,p) and y_i^e is the mole fraction of the *i*th component in the reference environment. The exhaust gases were considered as a gaseous mixture of CO_2, H_2O, N_2 and O_2. The reference environment was according to Szargut *et al*. (1988). The exergy efficiency is defined as the ratio between the net exergy work rate and the rate of exergy input into the system,

$$\varepsilon=\frac{\dot{W}_{out}}{\dot{E}_{in}} \tag{7}$$

where $\dot{E}_{in}$ is the rate of exergy input estimated as the sum of the fuel exergy and the system power input. The electric power $\dot{W}_{in}$ was calculated as 0.310 kW for 100TG oil, 0.224 kW for 100SW oil and 0.180 kW for 50SW/50D and 50TG/50D blends.

4. RESULTS AND DISCUSIONS

In the following figures are not presented the measurements above 2000 rpm. The measurements for speeds above 2000 rpm are affected by the governor, which is a mechanical engine component that limits its rotation. Figure 4 presents the brake power versus engine speed for the fuels tested. The highest brake power was obtained with the 100D oil and the lowest brake power was obtained with straight vegetable oils. These results are corresponding with the LHV of each fuel. Also, it is observed that the power decreased in the sequence of the viscosity increase as presented in Fig. 1. Power reduction in respect to diesel oil is presented in Fig. 5. Although the vegetable oils present approximately 15% reduction in LHV, the percentage of power reduction was lower in some speeds. The power reduction of the blends was

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nearly constant in the speed range tested. In the case of 100TG and 100 SW oils, the best results of power were presented about the low speed (1500 rpm).

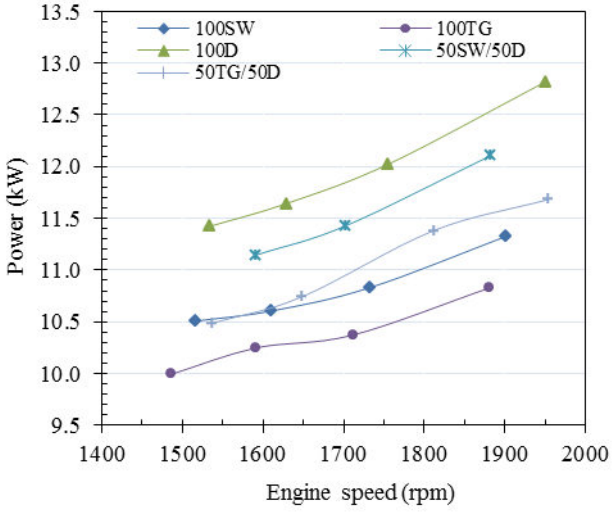

Figure 4. Power versus engine speed.

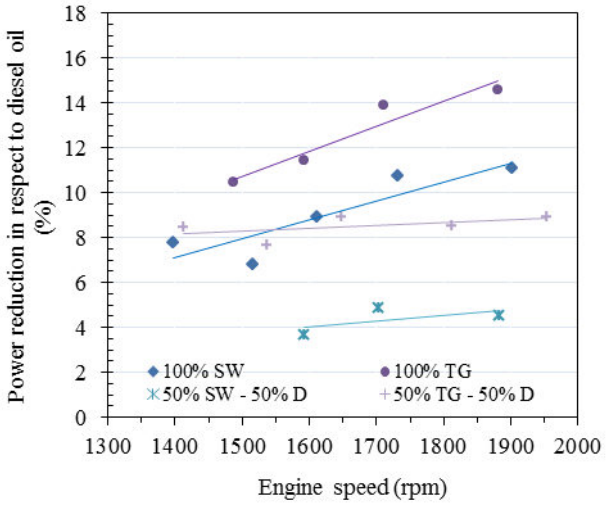

Figure 5. Power reduction in respect to diesel oil.

Figure 6 presents the results of mean power at 1800 rpm for all fuels tested. These results were compared statistically. The engine power for the 100TG, 100SW, 50TG/50D and 50SW/50D oils was 12.7%, 8.6%, 4.6% and 2.2%, lower than the power produced with the diesel oil, respectively. The ANOVA test identified a statistically significant difference ($p=0.00$) between the mean power of the fuels tested. The Dunnett´s test found statistically significant differences for these results in comparison with 100D oil ($p<0.05$).

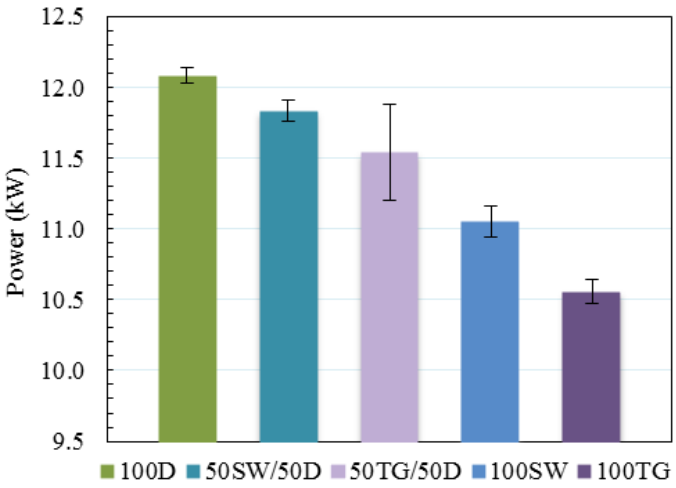

Figure 6. Power at 1800 rpm.

The break specific fuel consumption is presented in Fig. 7. All vegetable oils and blends resulted in higher BSFC when compared to diesel oil. This result is consistent with LHV values and specific mass of the fuels. The mechanical mechanism of fuel injection tends to maintain a constant volumetric displacement at each speed. In this condition, the fuel mass injected is proportional to the specific mass, increasing for the biofuels and compensating the difference in LHV value. This behavior is also commonly reported in the literature (Altin *et al*., 2001; Chalatlon *et al*., 2011; Nwafor, 2004; Sarada *et al*., 2010; Venkanna *et al*., 2009). At 1800 rpm, the BSFC for the 100TG, 100SW, 50TG/50D and 50SW/50D oils was 17.6%, 12.5%, 7.6% and 3.6%, higher than the BSFC of the diesel oil, respectively. ANOVA and Dunnett's test found that these differences were statistically significant ($p<0.05$), except for the 50SW/50D blend. The result of mean BSFC at 1800 rpm is presented in Fig. 8.

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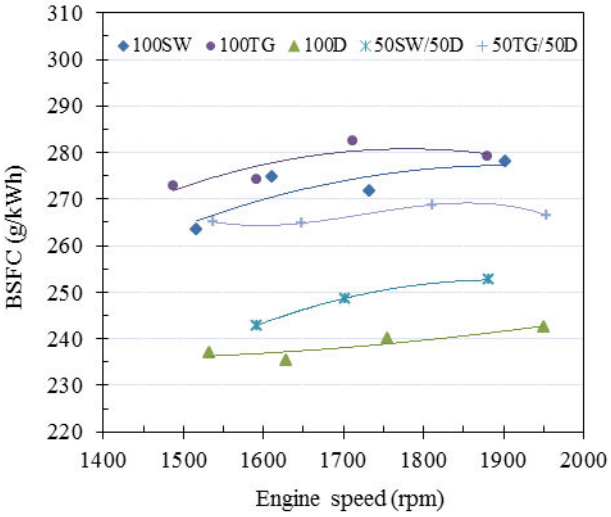

Figure 7. Break specific fuel consumption versus engine speed.

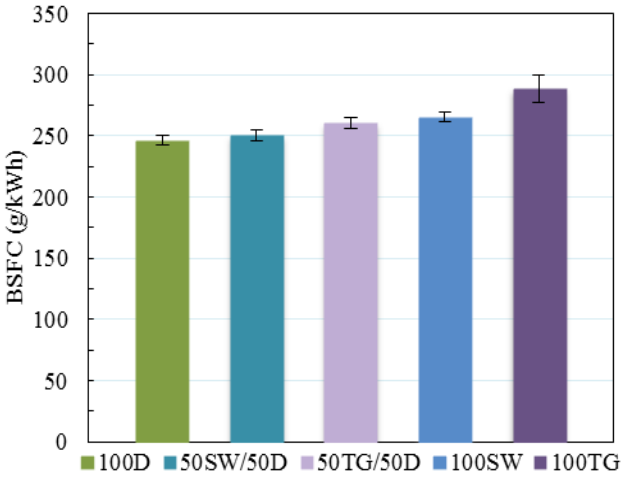

Figure 8. Break specific fuel comsuption at 1800 rpm.

Figure 9 presents the specific CO_2 emissions. The behavior of the specific CO_2 emissions was consistent with the power results and with the specific CO emissions presented in Fig. 10. At low speeds were observed high CO/kWh emissions with low CO_2/kWh emissions. In contrast, at high speeds, there were high CO_2/kWh emissions with reduction in the CO/kWh emissions. The specific CO_2 emissions were higher for vegetable oils than diesel oil because to the lowest H/C ratio. The specific CO_2 emissions of the fuels were also consistent with the air/fuel ratio. Figure 10 presents the specific CO emissions. An increase in the CO emissions at low engine speed was observed for all fuels, as a consequence of the greater mass of fuel injected. This indicates that the mixture became too rich at low speed. The 100TG oil and the 50TG/50D blend produced the highest CO/kWh emissions as result of poor atomization. The specific CO emissions for 100SW oil were equivalent to 100D oil, while the 50SW/50D blend increased the CO/kWh emissions.

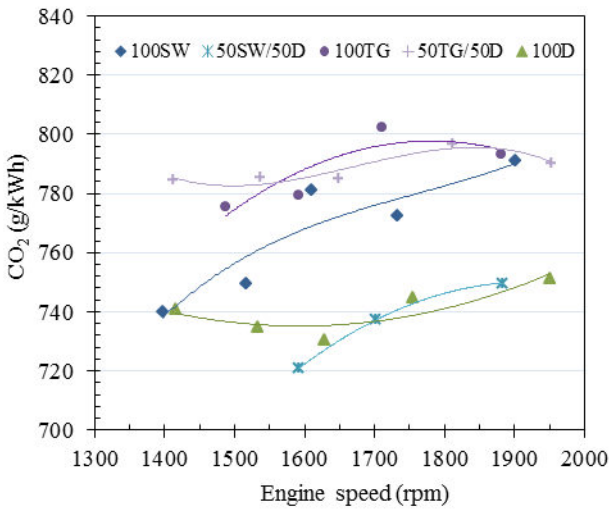

Figure 9. Specific CO_2 emission.

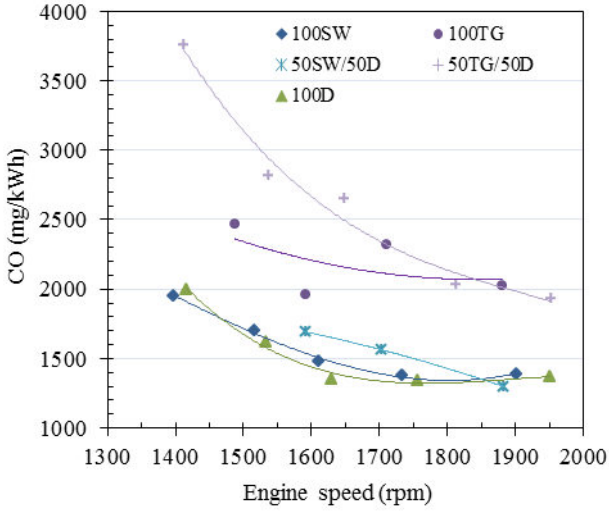

Figure 10. Specific CO emission.

The specific NO_x emissions are presented in Fig. 11. Due to poor atomization, the TG100 and 100SW oils reached the lowest temperature in the combustion chamber, resulting in the lowest production of NO_x by the thermal Zeldovich mechanism (Heywood, 1988). This is consistent with the exhaust gas temperature present in Fig. 12. High exhaust temperatures were observed with the 100D oil and low temperatures with the vegetable oils. This behavior agrees with the power output, indicating better atomization and faster combustion of the diesel oil spray. The temperature increased at low speeds due to the increase of the amount of fuel injected and of the residence time, leading to high NO_x formation.

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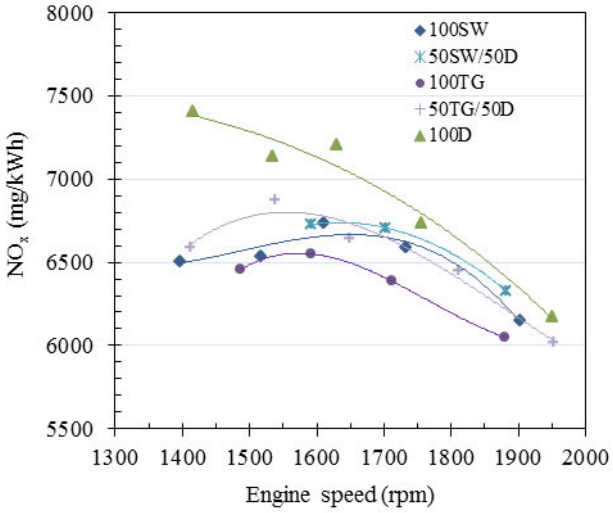

Figure 11. Specific NO_x emission.

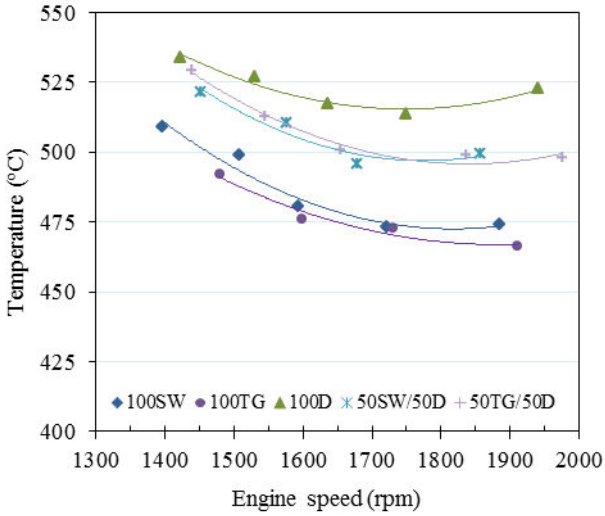

Figure 12. Exhaust gas temperature.

The exergetic efficiency and exergy destroyed at 1800 rpm are presented in Fig. 13 and Fig. 14, respectively. ANOVA test did not find statistically significant differences between the exergetic efficiency of all fuels tested ($p>0.05$). Similar result was found with the exergy destroyed, exhaust exergy and exergy loss. Figure 15 presents the exergy balance of engine performance with biofuels and diesel oil at 1800 rpm. The results were enough close showing that the engine performance was similar with all fuels tested. In this case of blends, the presence of the diesel oil decreased the ignition delay, aiding at ignition of the vegetable oils fraction. The results of the exergy balance on the vegetable oil may be favored by features of their molecular structure as bonds unsaturated, oxygen content and low cetane number. According Balafoutis *et al*. (2011), the content of unsaturated fatty acids favors the air/fuel mixture because the air oxygen reacts in the bonds unsaturated of the vegetable oils. Better blending conditions favor the efficiency of the engine. The sunflower and tung oils present 91% to 95% unsaturated fatty acids (Chen *et al*., 2010; Mehta and Anand, 2009). The vegetable oils also exhibit oxygen which may increase the rate of combustion. A low cetane number increases the ignition delay and the premixed combustion, reducing the rate of change of temperature and pressure in the combustion chamber which influences the lower exergy destruction (Benjumea *et al*., 2009; Tat, 2011).

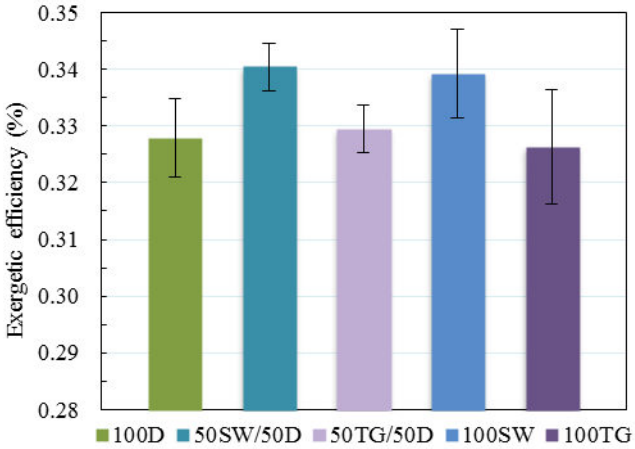

Figure 13. Exergetic efficiency at 1800 rpm.

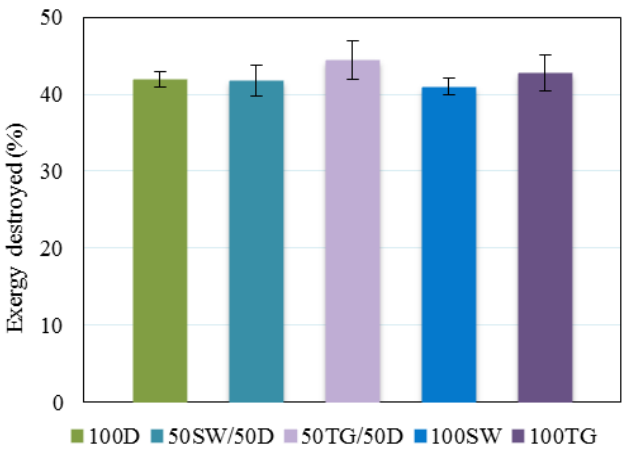

Figure 14. Exergy destroyed at 1800 rpm.

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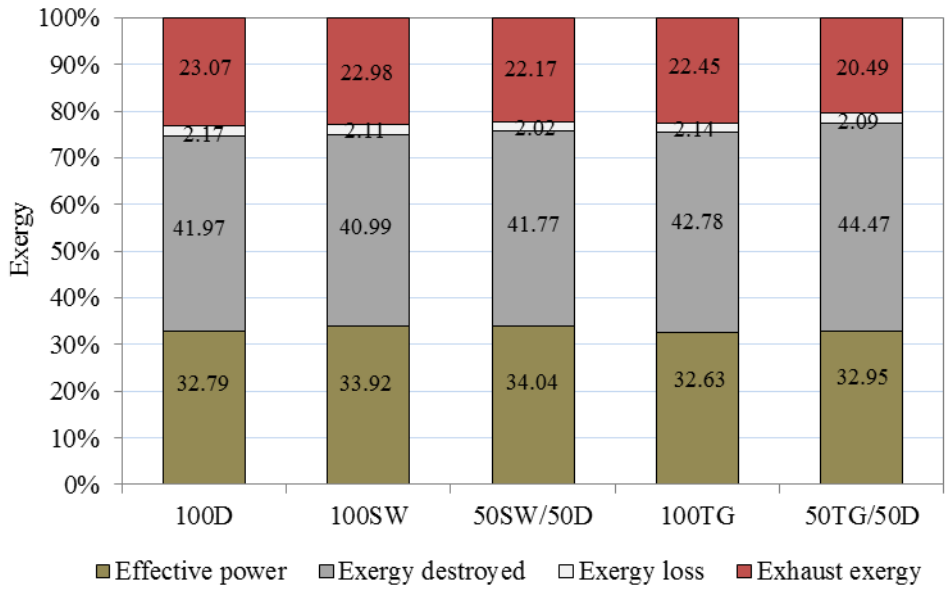

Figure 15. Exergy balance at 1800 rpm of the fuels tested.

5. CONCLUSIONS

Tung oil, sunflower oil and also blends 50/50 v/v with diesel oil were tested on a direct injection diesel engine, without any modifications on injection system of fuel. The pre-heating provided to the biofuels was effective. The biofuels showed a decrease in power as well as an increase in specific fuel consumption when compared to the diesel oil. This behavior is consistent with the physical and chemical properties of vegetable oils or respective blends.

At low speeds, the CO emissions increased as a consequence of the higher amount of fuel injected per cycle provided by the mechanical control of injection. The NO_x emission was also high at low speeds, as a result of high temperatures, long residence time and better combustion. The exhaust gas temperature was lower for the biofuels and respective blends when compared to diesel oil, showing the need of further studies in order to improve the atomization process and combustion. These results are consistent with others studies found in the literature.

The results of the thermodynamic analysis were compared at 1800 rpm using two statistical techniques (ANOVA test and Dunnett´s test). No significant difference in both, the exergetic efficiency and exergy destroyed, were identified for all fuels tested. The statistical techniques were efficient for comparison of the biofuels with the standard fuel. The similar performance can be attributed to an improvement in the combustion process due to the presence of oxygen and unsaturated fatty acids in biofuels, which promote the mixing of air /fuel.

It is expected that an increasing the injection pressure will improve the fuel atomization and, therefore, the engine efficiency. The results showed the technical feasibility of the biofuels as an alternative for diesel oil, especially if used for distributed generation and isolated communities.

6. ACKNOWLEDGEMENTS

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