



encit 2020



18th Brazilian Congress of Thermal Sciences and Engineering
November 16-20, 2020 (Online)

ENC-2020-0248

ENERGY AND EXERGY ASSESSMENT ON A SMALL ENGINE DRIVEN GENERATOR SET FUELLED WITH DIESEL AND SOYBEAN METHYL ESTER BIODIESEL BLEND

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Abstract. Energy demand has been for decades in the search of new alternative sources of fuels for the use in engines. Thus, it was noticed that the biodiesel of soybean oil has similar properties to the diesel oil as well as it has the advantage of being vegetal origin, renewable, biodegradable, and non-toxic. Therefore, this study brings the energy and exergetic analysis of a generator group operating with different fuels, among them the pure diesel (B0) and the methyl ester mixture transesterified of soybean oil and diesel (B25), both were submitted to different points of load. The tests were performed at steady state. Results were obtained regarding fuel energy, thermal efficiency, exhaust gas energy and heat transfer losses. Fuel exergies, exergetic efficiency, exhaust gas exergy, destroyed exergy and the exergy of heat transfer losses were also obtained. It was found that the addition of biodiesel in the mixture provided almost the same energy performance as the B0 and the exergetic performance was like the energy, thus accompanying the same performances. No significant variation of efficiencies was found with the increase of biodiesel (reduction of 0.2 to 3.2% in thermal efficiency and reduction of 2.7 to 5.6% in exergetic efficiency), accompanying the same performances. No significant variation of efficiencies was found with the increase of biodiesel (reduction of 0.2 to 3.2% in thermal efficiency and reduction of 2.7 to 5.6% in exergetic efficiency).

Keywords: diesel, biodiesel, energy analysis, exergetic analysis, generator set.

1. INTRODUCTION

Many research and studies have been conducted to develop alternative fuels from renewable resources appropriate to operate in compression ignition internal combustion (IC) engines. In papers (Santos et al., 2017), (Canakçi and Hosoz, 2006), (Monyem, 1998) presented the effects of using blends of biodiesel from soybean oil and diesel in IC engines on performance and gas emissions. The advantage of the performance presented by the authors revealed that blends with biodiesel obtained practically the same thermal efficiencies and slightly higher specific fuel consumption when compared to B0 performance. In thermal machines is desirable to convert the fuel energy into usable power at the highest rate, however due the losses which occurs in a real engine operation, the complete conversion of this input energy to produce work is not possible. The energy in a control volume can not be destroyed and the amount of output energy is the same of the amount of energy which is input. Nevertheless, the energy conservation principle assessment is not always suitable to describe some aspects of the use of the energetic resources. For the determination of waste and energy losses, their location, types, and real values, exergetic analysis is more appropriate (Terzi, 2018; Moran et al., 2018).

The benefits of exergy analysis are numerous compared to energy analysis. Some of the more significant ones follow below (Terzi, 2018):

- Exergy efficiencies are always the measures of the approach to true ideality and provide more meaningful information when assessing the performance of energy systems. Also, exergy losses clearly identify the locations, causes, and sources of deviations from ideality in a system.
- Exergy methods can help evaluate the thermodynamic values of the product energy forms in complex

systems with multiple products (e.g., cogeneration and trigeneration plants).

- Exergy-based methods can be used to improve economic and environmental assessments.
- Exergy can improve understanding of terms like energy conservation and energy crisis.
- Exergy methods can help in optimization activities

Still for (Terzi, 2018), an exergy analysis should be conducted only after the validity of the mass, and energy balances has been confirmed. The thermodynamic analysis should be evaluated that an exergy balance is obtained by combining the corresponding energy and entropy balances.

A simple procedure for performing a comprehensive exergy analysis of such a system involves the following steps:

- Subdivision of the process under consideration into each system component (process or subprocess), depending on the depth of detail and understanding desired from the analysis.
- Calculation of the conventional mass and energy balances on the process and definition of all basic quantities (e.g., work and heat) and properties (e.g., temperature and pressure).
- Based on the nature of the process, the acceptable degree of analysis complexity and accuracy, and select a reference environment model.
- Calculation of the energy and exergy values relate to the selected reference environment model.

This work aims to assess the variation of energy and exergy flows for the diesel and 25% of soybean methyl ester blend.

2. METHODOLOGY

2.1 Fuels characterization

It was used Diesel (B0) in the first test. In the second one, it was used the B25 blend, consisting of transesterified soybean oil methyl ester (25% m/m) and of-the-shelf diesel. Table 1 summarizes the physical-chemical properties for both fuels. The soybean oil was transesterified at the Renewable Energies Lab. of UFGD.

Table 1 - Fuels physical-chemical properties

Property	Diesel	Biodiesel
Formulae	C _{14,09} H _{24,78}	C _{18,74} H _{34,43} O ₂
Average Molecular Weight (kg kmol ⁻¹)	194,20	291,77
Higher Heating Value (kJ kg ⁻¹)	45339	39871
Lower Heating Value (kJ kg ⁻¹)	42640	37388
Chemical Exergy (kJ kg ⁻¹)	45427	40105

(Canakci e Van Gerpen, 2003)

2.2 Test rig

The tests were accomplished in a Diesel generator set, shown in Fig. 1, model GB3500, Gera Power Brasil®, composed by a four-cycle, air cooled Diesel engine, model ATIMA 178F.

In data collection, the generator set was instrumented according to the test bench's schematic design in Fig. 2.

Figure 2 shows: a bank of resistors (De Lorenzo®); a wattmeter (Politem®); a tank (Lucadema®), a stopwatch; a scale; a type-k thermocouple; a Pitot Tube anemometer (AOIP®) and generator set (Gera Power Brasil®). The accuracies of the test rig are show in Table 2.



Figure 1. Test bench.

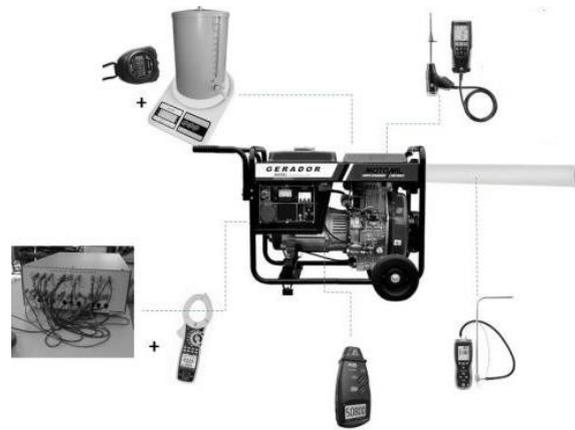


Figure 2. Schematic of the test bench.

Table 2 – Instrumentation accuracies

Measurement	Device	Accuracy	Unity
Mass	Digital scale	$\pm 1,00$	g
Time	Digital stopwatch	$\pm 0,01$	s
Temperature	Type -K thermocouple	$\pm 0,10$	$^{\circ}\text{C}$
Inlet airflow	Pitot tube digital anemometer	$\pm 1,00$,	$\text{m}^3\text{min}^{-1}$,
Brake power	Digital Wattmeter	$\pm 1,00$	kW

2.3 Test procedures

It was waited a time of 1800 seconds prior to the start of the tests, to ensure that the engine was warm and in steady-state operating condition. It was determined that the generator set was subjected to three different load points: 0.35 kW; 0.70 kW; 1.05 kW.

The demanded load was controlled through a bank of resistors. For each demanded load point, it was established a time of 300 seconds for data collection and an interval of 180 seconds was established between a point and another, to assure the stabilization of engines' operation.

The fuel's initial mass and final mass were recorded by the tank, stopwatch, and the scale, at the beginning and at the end of each data collection point to measure the fuel mass flow.

These equipment were used to measure: mass flow, speed, and temperature of the intake air (anemometer); temperature of the exhaust gases (type-k thermocouple), effective power of the generator set (wattmeter).

The environmental conditions at the time of the tests like atmospheric pressure, temperature and humidity were collected by the UFGD's Meteorological Station.

2.4 Energy analysis

The energy assessment provides the calculation of the energy rates which are transferred through the boundary of the volume control. These energy rates are: electric power, heat losses and the enthalpy related to the mass flow rate of the fuel and exhaust flue gases.

The control volume adopted in this work is shown in Fig. 3. The control volume shown on the figure is at steady state. The change in potential and kinetic energy from inlet to exit were neglected. It was adopted the standard reference state 25°C (298.15 K) and 1 atm. The measured temperatures of the intake air were close to the reference temperature; therefore, its energy and contribution were neglected in calculations. It was also considered that exhaust gases were regarded as ideal gases. The energy analysis, shown in Eq. (1), was calculated based on the control volume adopted.

The energy flow rates which passes through the control volume are: the fuel energy flow rate (input), the electric power (output), the exhaust gases energy flow rate (output) and the heat losses due the heat transfer from the engine driven generator to the environment (output). The mass fuel flow rates were measured to compute the fuel energy flow rates, which, in turn is the result of the multiplication between the mass fuel flow rate and the Lower Heating Value (LHV) of this fuel. The power factor of the generator was assumed as 1.0 and its efficiency as 100%. So, the effective brake power was considered the same value as the electrical power measured. The combustion

inefficiencies were neglected. The combustion was assumed as complete regarding the air/fuel ratio measured by the scale and pitot tube. Thus, the mass fractions of CO₂, H₂O, N₂ and O₂ in the flue gases were estimated to compute the specific enthalpy of the exhaust gases. The exhaust gases mass flow rate was considered the sum of the air and fuel mass flow rates, regarding the mass conservation principle. The heat losses was computed as the difference of the input energy and the sum of the remaining energy outputs.

In the Eq. (1), Q_c and $m_c \cdot LHV$ represent the energy released by the fuel, where m_c is the mass flow of the fuel and LHV is the lower heating value. The exhaust gas energy is represented by \dot{Q}_{ge} and $\dot{m}_{ge} \cdot h_{ge}$, where \dot{m}_{ge} is the mass flow of the exhaust gases and h_{ge} represent the specific enthalpies of the products, weighted by the mass fraction (y_i) computed by Eq. (2). The mass fraction was computed assuming that the combustion is complete and regarding the air/fuel ratio measured. And W_e and \dot{Q}_p are, respectively, the generator's effective power and the losses that are represented by the energy that is transferred by heat to the environment.

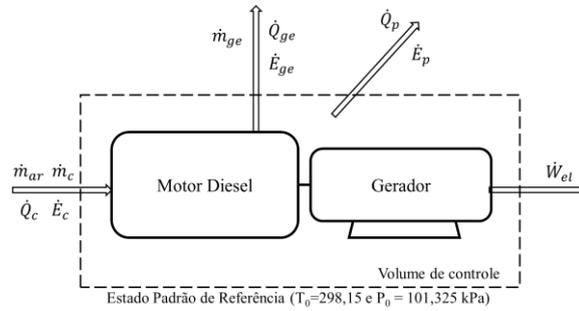


Figure 3. Mass, energy and exergy flow into and out of the control volume.

$$\dot{Q}_c = \dot{m}_c \cdot LHV = \dot{Q}_{ge} + \dot{Q}_p + \dot{W}_e = \dot{m}_{ge} \cdot h_{ge} + \dot{Q}_p + \dot{W}_e \quad (1)$$

$$h_{ge} = \sum_{i=1}^j y_i \cdot (h_i) \quad (2)$$

2.5 Exergy analysis

The exergy rates which goes through the volume control follows the energy rates behavior. So, at this time the specific entropies were taken into account. For the fuel, the chemical calculated exergy related to the LHV was assumed. The specific exergy of the exhaust gases were assume as the sum of the thermomechanical and chemical exergies of the weighted composition of the flue gases by the mass fraction of the specimens. The destroyed exergy was assumed as the remaining output exergy regarding the exhaust gases exergy and the power measured.

The exergy rate balance for the adopted control volume (Fig. 3) is shown in Eq. (3). It was considered exergy transfers accompanying mass flow, control volume at a steady-state, the intake air exergy was disregarded. And the exhaust gases were modeled as ideal gases. Dead state was considered as the reference state, therefore 25 °C and 1 atm were used as reference. Equation (3) demonstrates the rate in which exergy is transferred into and out to the control volume boundaries. The fuel chemical exergy (e_c^{ch}) was computed using Eq. (4). The thermomechanical specific exergy (e_{ge}^{th}) of the flue gases is given by Eq. (5) and its specific chemical exergy (e_{ge}^{ch}) by Eq. (6)

$$\dot{E}_c = \dot{m}_c \cdot e_c^{ch} = \dot{E}_{ge} + \dot{E}_p + \dot{W}_{el} = \dot{m}_{ge} \cdot (e_{ge}^{th} + e_{ge}^{ch}) + \dot{E}_d + \dot{W}_e \quad (3)$$

$$e_c^{ch} = [1,0401 + 0,1728 \cdot H_c + 0,0432 \cdot O_c + 0,2169 \cdot S_c \cdot (1 - 2,0628 \cdot H_c)] \cdot PCI \quad (4)$$

$$e_g^{th} = \sum_{i=1}^j y_i \cdot \{h_i(T) - h_i(T_0) - T_0 \cdot [s(T) - s(T_0)] - R \cdot \ln(P/P_0)\} \quad (5)$$

$$e_g^{ch} = \sum_{i=1}^j y_i \cdot e_i^{ch} + R \cdot T_0 \cdot \sum_{i=1}^j y_i \cdot \ln(y_i) \quad (6)$$

2.6 Performance Analysis

The effective brake power is the power which is delivered by the engine. For this study, the generator efficiency and load factor were neglected, thus the brake power is assumed the same as the electrical power measured by the wattmeter.

The brake specific fuel consumption (*bsfc*) is an efficiency indicator given by the ratio between the effective brake power and the fuel mass flow ratio (Eq. (7)). It represents the amount of fuel which is used to generate one unity of power.

$$bsfc = \dot{m}_c / \dot{W}_e \quad (7)$$

Thermal efficiency is the ratio of the output power and input energy rate. It works inversely the *bsfc*. The lower the *bsfc* the higher is the thermal efficiency. For engine driven generators thermal efficiency can be computed by the ratio between brake power and the fuel energy rate (Eq. (8)).

$$\eta = \dot{W}_e / \dot{Q}_c \quad (8)$$

The second law or exergetic efficiency is the ratio between the exergetic fuel rate (Eq. (9)). Once thermal efficiency does not consider the effects of irreversibilities, it is expected that the exergetic efficiency presents lower values.

$$\eta_{II} = \dot{W}_e / \dot{E}_c \quad (9)$$

2.7 Results

The LHV from B0 and B25 were calculated. The value of 42643.67 kJ.kg⁻¹ was found for B0. This result was like papers (Canakçi and Van Gerpen, 2003). For the B25 blend, LHV was 41330.46 kJ.kg⁻¹.

Figure 5a presents the engine behavior as it was demanded by the resistive bench. The engine is not sufficient to comply with the demand at all points. However, this behavior is sharper for the blend. It occurs due the higher viscosity of the blend that leads to a greater effort of the fuel pumping device.

The *bsfc* occurs inversely to thermal efficiency and the lower values means better performance. Figure 5b shows that the *bsfc* decreases as the engine is demanded. It means that the engine works better as it approaches the design point for both fuels. Figure 5b also demonstrates that *bsfc* is slightly higher for B25, but this difference may be neglected. This behavior implies that the performance is pretty the same for both fuels.

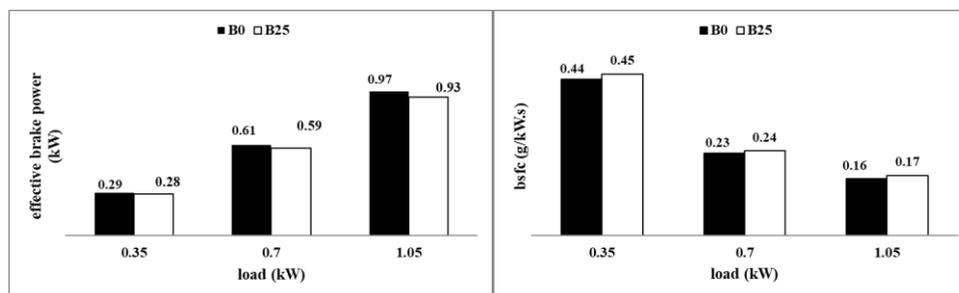


Figure 5. 5a Effective brake Power. 5b. Brake specific fuel consumption

The fuel energy flow rate is the product of the mass fuel flow rate times fuel energy (LHV) and it is shown in Fig. 6a. As expected, the energy supplied by the fuel increases as the engine was demanded for both fuels. Although the LHV of B25 is near 3% lower than the LHV of the diesel, an increase of the mass fuel flow rate occurs to comply with the demanded load. It was noticed that the energy supplied by B0 was higher than the energy supplied by B25 blend, at all assessed points, indeed. In the first and second point B0 had the energy of the fuel 3.2% larger than B25's energy. In the third point it was 1.0% larger than B25 and this fact corroborates to the *bsfc* behavior.

The thermal efficiencies are shown in Fig. 6b. At all points of load, B25 and B0 efficiencies reached almost the same values. In the first load point, compared to B25, B0 had its efficiency reduced by approximately 0.4%. The second point decreased 0.2% and in the third point it was 3.2% lower.

The exhaust gases energy is shown in Fig. 6c. It is observed that adding biodiesel in the blend contributed to a significant reduction of the exhaust gas energy. At the first load point B25 had approximately 28.5% lower energy than B0, in the second it was 35.4% lower, in the third it was 42.9%.

There were losses due to the transfer of heat, shown in Fig. 4d. It can be observed at first load point with the addition of biodiesel in the mixture that there was an increase of approximately 59.6% of the heat transfer losses. In the second point the losses increased 48.5 %, in the third 41.7%.

Figure 7a shows the fuel exergy according to the established load points. At the first and second load point, the B0 exergy is approximately 0.6% greater than that for B25 exergy, at third point it is 1.6% higher than B25.

The exergetic efficiency is shown in Fig. 7b. At the first load point B25 had exergetic efficiency approximately 3.0% lower than B0, at the second point it was 2.7% lower, at the third point it was 5.6%.

The exhaust gases exergy is shown in Fig. 7c. It is observed at the first load point that B25's exhaust gases exergy was 13.4% greater than B0, in the second point it was 7.0% higher, in the third point was 5.1% greater than B0.

Figure 7d shows the exergy loss by heat transfer. It is also observed that at the first load point, the exergy loss by heat transfer of B25 blend is 25.7% greater than B0. At the second point it was 21.1% higher and in the third point its 19.1% greater than B0.

And the exergy destruction due to irreversibilities within the control volume it was observed. At the first load point B25's destroyed exergy was 6.6% lower than B0, at second point it was 5.6% lower, at the third point it was 2.0% lower than B0.

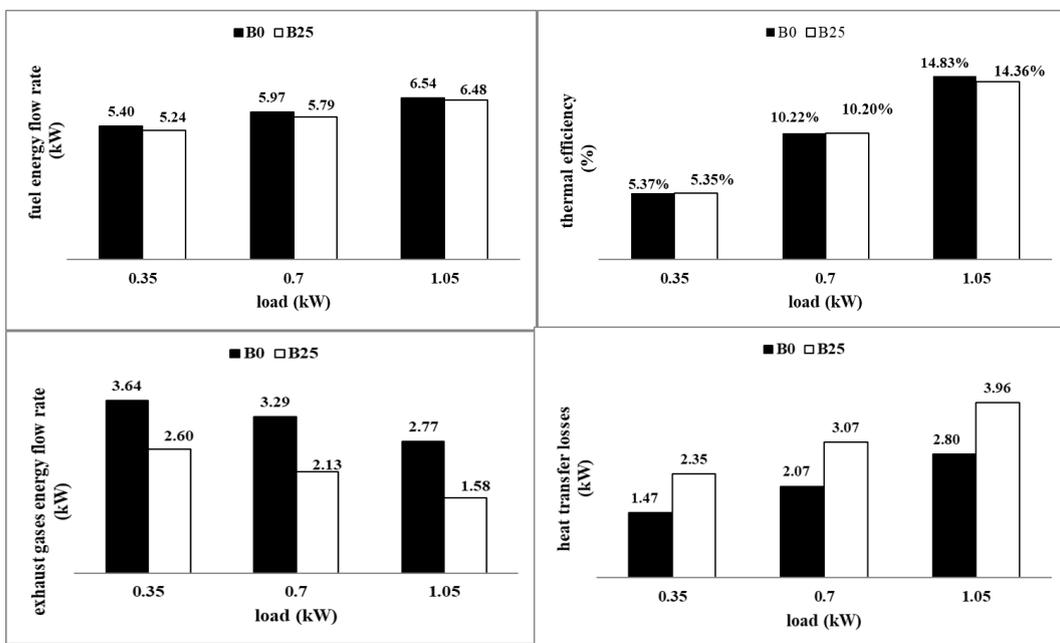


Figure 6. 6a. Fuel energy. 6b. Thermal efficiency. 6c. Exhaust gases energy. 6d. Heat transfer losses.

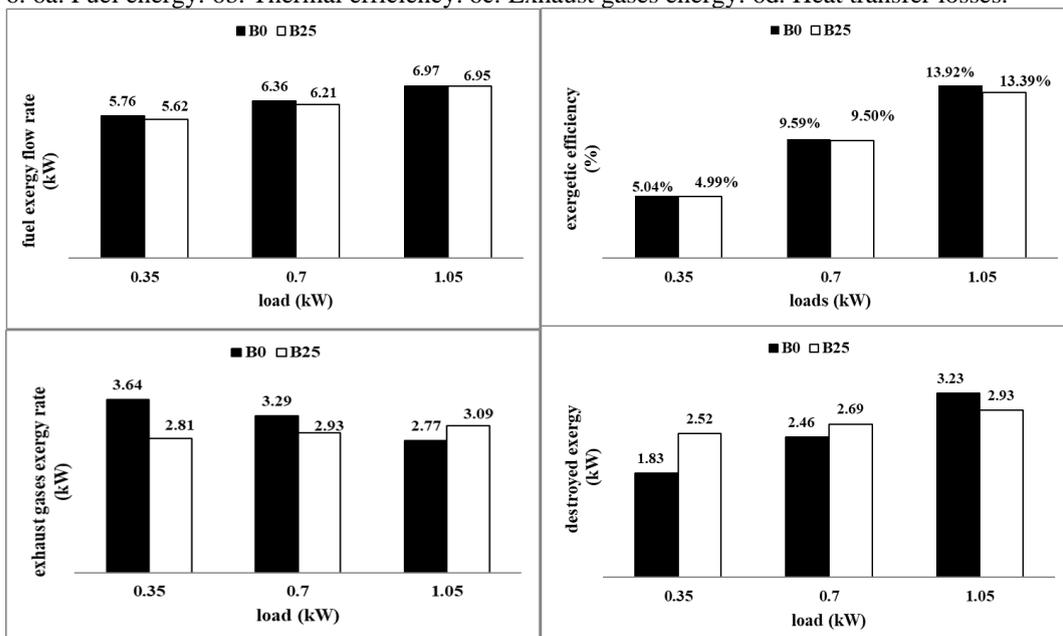


Figure 7. 7a. Fuel exergy. 7b. Exergy efficiency. 7c. Exhaust gases exergy. 7d. Destroyed exergy.

Distributions of the fuel energy for both B0 and B25 are reported in Figure 8. The energy losses in exhaust gases are higher for the pure diesel than that for the blend for all loads. It evidences better combustion efficiency related to the biofuel due the presence of oxygen in the fuel composition. Furthermore, the presence of unburned hydrocarbons and carbon monoxide is more likely if the pure diesel was used. Both UHCs and CO are products of inefficiency in the combustion process and still carry an amount of energy which must not be neglected. Also, the presence of higher amount in O₂, decreases the proportion of other specimens which are richer in enthalpy. Consequently, at the same power output the energy losses of heat transfer from the engine to the environment is higher for the operation with the blend. Also, can be noticed that the exhaust gases losses decreased as the engine was demanded for both fuels.

The distribution of the fuel exergy for both fuels is depicted in Figure 9. It can be noticed that for B0 the energetic and exergetic distributions show almost the same performance. However, it can be perceived that even the most of energy losses when the engine is operated with B25, the better opportunities for energy recovering are located at the flue gases for the blend operation.

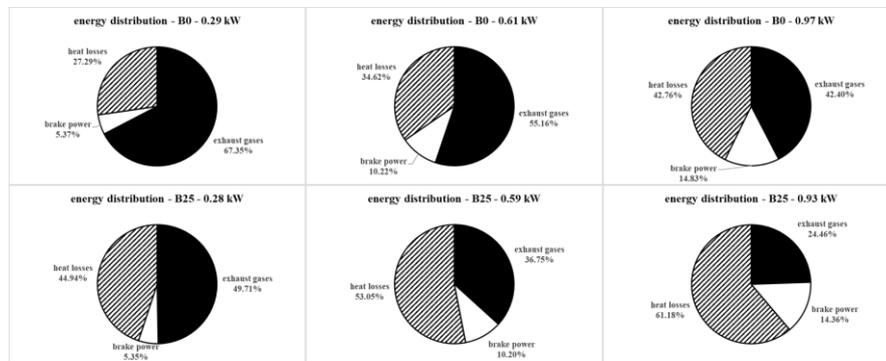


Figure 8. Distribution of the fuel energy for the fuels

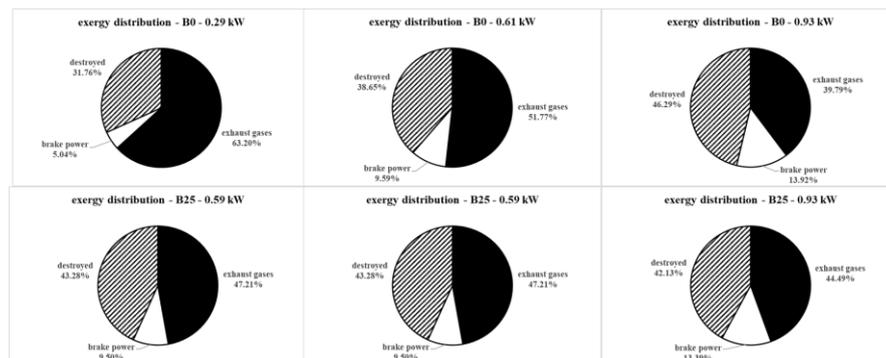


Figure 9. Distribution of the fuel exergy for the fuels

3. CONCLUSIONS

The energy and exergy analysis were employed to assess the sources of energy and exergy conversions and distribution in three different load conditions on a small compression ignition engine. Pure diesel fuel operation was used as baseline for comparison to a blend of 25% in mass of soybean methyl ester/diesel. Tests were carried out with the engine at the same conditions of load for both fuels. Some factors were revealed as follows:

1. The effective brake power for B0 presented slightly higher values than those for B25 operation. The higher viscosities and lower LHV of SBME leads that more amount of the power was spent to promote the flow by the fuel system.
2. The brake specific fuel consumption presented prettily much the same values under the same conditions for both fuels operation.
3. Also, as expected, the thermal efficiency showed the same values corroborating to the *bsfc* behavior.
4. The values presented by the fuel energy flow rate for B25 presented slightly lower values than that for B0 at all conditions. It implies that the increase in mass fuel flow rate increase did not was able to compensate the lower LHV value for B25. It also collaborates to the difference in the effective brake power as the blend was used. For both fuels it increased as the engine was demanded.
5. The exhaust gases energy flow rate decreased as the engine was demanded for both fuels. The

proportion of conversion of the fuel energy rate also decreased under the same aspects. However, this trend is more pronounced for the B25 blend. The difference between the values noticed were also quite higher as it were compared to B0. As the blend is an oxygenated fuel, the presence of O₂ on the flue gases is greater than for B0. As the amount of O₂ increases in the flue gases the proportion of CO₂ decreases leading to a lower specific enthalpy of the gases due the fact that the specific enthalpy of the O₂ is lower than that for CO₂.

6. Conversely, the heat transfer losses across the volume control increased as the engine was required. It means that the heat transfer is higher as the temperature in the cylinder is higher. The proportion of the heat losses related to the fuel energy flow rate also increases more strongly for the blend. Even the LHV for the diesel is higher, the combustion efficiency of the blend is likely to be better due the presence of oxygen in the fuel structure.
7. The fuel exergy flow rate presented the same behavior than that for the fuel energy flow rate. It also happens for thermal and exergetic efficiencies. However, for fuel exergy the values presented were slightly higher than for energy and the opposite happens to the efficiency as expected.
8. The trending of exhaust exergy rates also followed the trends of the exhaust energy rates for B0 operation. Also, the proportion of this exergy rates decreased as the engine was required. Nevertheless, for B25 the performance is inverse. Exhaust gases exergy rate increase as the engine demand increases. Its values still were lower than that for B25 in the two first loads but it overcomes at the third one. The proportion of converted fuel exergy rates remains steady from first to the last point. Although the specific enthalpy for O₂ is lower, the specific entropy is higher than the other exhaust gases specimens. So, the increase of its proportion leads to a more exergetic contribution of the oxygen. Consequently, the exergetic conversion turns more balanced than the energetic for B25.
9. As expected, the exergetic behavior for the destroyed exergy displays in the opposite way than it happens for the exhaust exergy rate.
10. The energetic assessment indicates that the most opportunity of conversion of the fuel potentiality in useful work is present in the exhaust gases for B0 and in heat losses for B25. The exergetic analysis demonstrates a more balanced results for the blend. This potential changed for the exhaust gases capability on reducing the irreversibilities and utilize of them. Finally, the more equalized exergetic performance, which in term consider the irreversibilities, leads a more realistic survey clearly locating the and sources of deviations from ideality.

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