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# ENERGY AND EXERGY ANALYSIS OF THE USE OF A BIOFUEL IN AN INTERNAL COMBUSTION ENGINE

### Lucas Rhuan Prudente

Graduate Program in Mechanical Engineering, School of Mining, Federal University of Ouro Preto, Minas Gerais, Brazil  
lucas.prudente@aluno.ufop.edu.br

### Elisângela Martins Leal

### Claudio Marcio Santana

Department of Mechanical Engineering, School of Mining, Federal University of Ouro Preto, Minas Gerais, Brazil  
elisangelamleal@ufop.edu.br; claudio.santana@ufop.edu.br

**Abstract.** Biofuels play an important role in meeting the world demand for liquid fuels, mainly for environmental reasons, largely due to their lower level of pollution with their burning and processing. This paper aims to perform a theoretical analysis of a compression-ignition internal combustion engine using the bio-oil produced from a microwave-assisted pyrolysis process. The combustion process of the bio-oil is modeled in the Chemical Equilibrium for Applications (CEA NASA) software. Physical characterization and gas chromatography coupled with mass spectrometer of the bio-oil are performed, which shows a chemical composition of carbon (39.1%), hydrogen (4.8%), oxygen (20.2%), and an LHV of 7781.98 kJ/kg. From these results, the energy and exergy modeling in a diesel engine is performed, varying the  $\lambda$ -factor from 0.8 to 1.2 for two situations: one using the bio-oil/diesel blends in the ratio of 25%-75% and pure diesel oil. The results show the feasibility of energy and exergy performance for the bio-oil used.

**Keywords:** Energy analysis; Exergy analysis; Internal combustion engine; Pyrolysis oil (bio-oil); Eucalyptus Citriodora.

## 1. INTRODUCTION

The Diesel engine is a thermal machine that can be operated from the combustion of diesel oil and biodiesel oil / diesel blends inside the combustion chamber. Biodiesel is a biofuel processed from biomass, organic matter of plant or animal origin, such as plants, vegetable oils and animal fat. Research indicates that its main purpose will be to replace diesel oil used in internal combustion engines with compression ignition or to generate another type of energy that can partially or totally replace fossil fuels.

According to Oliveira et al. (2019) the concept of sustainability originated in the context of natural renewable resources. The dimension of sustainability is linked to the effective use of existing resources in the various ecosystems with minimal environmental deterioration, both the production and consumption model must be compatible with the natural environment and the available economic structure. Oil has long been the most used natural resource to meet global energy consumption, says Reis (2016). The pressure placed on these fuels by the pursuit of sustainability, however, provoked a reaction from their industries, which currently develop and propose solutions with lower environmental impacts and identified as collaborators in the search for a sustainable model of development.

Barbosa and Ibrahim (2014) cite that biomass consists of organic compounds from agricultural residues, vegetation leftovers, wood bark, aquatic plants, as well as animal and food waste. It is used in the production of biofuels, through a thermal process called pyrolysis, in which the waste is sent to a pyrolytic reactor, where the organic compounds gain gaseous (volatile) characteristics as a function of heat, gradually decomposing. Boggavarapu and Ravikrishna (2013) argue that renewable nature is one of the most important advantages, in addition to being known for reducing greenhouse gases (GHG), fuels derived from biomass are called biofuels.

According to Siraj et al. (2017), the differences in physical and chemical properties between fossil fuels and biofuels cause several changes in the operation of diesel engines. The low heating value of biofuels leads engines to higher fuel consumption compared to diesel and lower fuel injection rate due to high viscosity. Anand et al. (2011) claim that nitrogen oxide (NO<sub>x</sub>) emissions are higher in biofuels (4,382 g/kWh with soy methyl ester) than with diesel (3,570 g/kWh with

S10 diesel oil) in compression ignition engines. This behavior is due to the cumulative effect of the ignition delay, which varies according to parameters such as load, rotation, and fuel injection advance.

Moran and Shapiro (2018) say that exergy transferred from a system to its surroundings and not used is usually a waste. Better use of energy resources can be achieved by reducing exergy destruction within a system and/or reducing losses. According to Wall (2009), it is possible to verify what happens with the exergies (availability) in an ICM, through the pV or p- $\alpha$  diagram of the real operation of the thermodynamic cycle, according to Fig. 1:

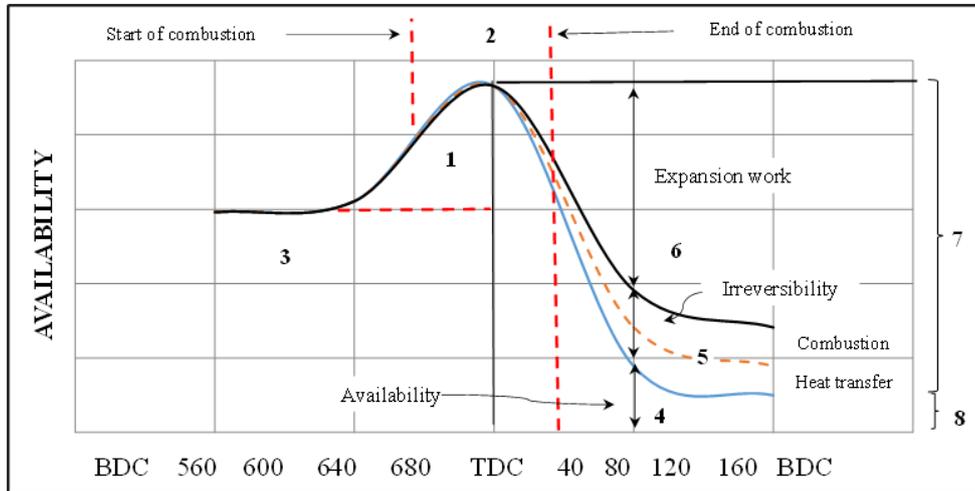


Figure 1. Availability versus angle of the crankshaft (Wall, 2009).

In Figure 1, the number (1) indicates the gain of availability due to the compression work, and the number (2) specifies the occurrence of the combustion process. Number (3) indicates the exergy of the fuel, while number (4) designates the mechanical work of the engine. Number (5) shows the irreversibilities generated, and number (6) indicates the exergy of the exhaust gases. Number (7) designates the losses of the whole process, and number (8) indicates the efficiency of the cycle, the fraction used. The black line represents the energy balance (first law), the blue line represents the exergy balance (second law) and the red line represents the irreversibility due to the combustion and the heat exchange. The red line is located between the black and the blue line.

Faced with the world scenario, where less polluting energy sources are increasingly discussed, the use of alternative fuels has become of great importance due to the uncertainties of the use of fossil fuels. The objective this work is to evaluate the behavior of a compression ignition internal combustion engine, through the energetic and exergetic analysis of the pyrolyzed oil of *Eucalyptus Citriodora*.

## 2. METHODOLOGY

Santana et al. (2021) defines the automotive ICE as a heat engine that, through the combustion of fuel, releases heat energy for the operation of the engine and consequently the displacement of the vehicle. Felipe Filho (2014) claims that the second law of thermodynamics imposes that heat must transit between thermal reservoirs with different temperatures, and only a part of this heat in transit can be converted into work.

Abbasi and Emanverdi (2014) and Hariharan and Reddy (2011) state that one of the main parts of a diesel cycle ICE is the fuel injection system. Features such as injection pressure, number and diameter of injection holes directly affect the combustion process and engine performance. Performance and emissions also depend on factors such as the amount of fuel injected, combustion chamber shape, fuel injection time, spray type, air turbulence, injector position and diameter, among others.

Ferguson (2001) states that the performance of an ICE is characterized by some geometric and thermodynamic parameters, such as the compression ratio (CR), the engine displacement ( $V_d$ ) and the fuel lower heating value (LHV). Pulkrabek (1997) and Heywood (1988) state that the heat provided by the combustion of the air-fuel mixture gives the power in an ICE is according to Eq. (1):

$$\dot{Q} = \dot{m}_f LHV \quad (1)$$

Where, in SI units,  $\dot{Q}$  is the heat transfer rate (kW);  $\dot{m}_f$  is the fuel mass flowrate (kg/s) and  $LHV$  is the lower heating value of the fuel (kJ/kg).

According to Heywood (1988), a parameter measured in the engine test, the specific fuel consumption ( $sfc$ ) is more useful than fuel consumption  $\dot{m}_f$ . The  $sfc$  is the fuel consumption per unit of power, and measures how efficiently an

engine uses the fuel to produce work. By Eq. (2) it is possible to identify that low *sfc* values are desirable. In spark ignition engines, the mean value of the *sfc* is 270 g/kWh and in the compression ignition engines the mean value of the *sfc* is lower (200 g/kWh).

$$sfc = \dot{m}_f / P \quad (2)$$

Where, in SI units, P is the engine power (kW).

According to Giacosa (1970), engine performance is defined by thermal efficiency ( $\eta_{th}$ ) or fuel conversion efficiency, according to Eq. (3). The thermal efficiency is empirically defined as the ratio of the indicated power (P) and the heat flux  $\dot{Q}_f$  supplied by the fuel, or the ratio between the indicated work ( $W_i$ ) and the heat supplied by the fuel ( $\dot{m}_f LHV$ ).

$$\eta_{th} = \frac{P}{\dot{Q}_f} = \frac{W_i}{\dot{m}_f LHV} = \frac{P(n/N)}{\dot{m}_f(n/N)LHV} = \frac{1}{sfc \times LHV} \quad (3)$$

According to Boggavarapu and Ravikrishna (2013), the application of biofuels in diesel engines shows advantages and disadvantages since these engines are not designed for this purpose and several components need modifications. A comprehensive study of various aspects of biofuels is required before practical application. The fuel injection process impacts combustion and exhaust gas emissions.

## 2.1 Pyrolysis process

Borsano and Barbosa (2019) analyze that a clean development mechanism that has been publicized and developed for some time, and that can be a good alternative for waste disposal, is pyrolysis. It consists of the chemical decomposition of organic waste by heat, in the absence of oxygen. According to Vieira et al. (2014) pyrolysis generally occurs at a temperature ranging from 400 °C until the beginning of the gasification system. Pyrolysis stands out, when compared to different types of processes, for generating products (liquid, solid and gaseous), with added values that can be used as a source of fuels or among other uses.

According to the Sairem group (2022), microwave-assisted pyrolysis (MAP) is an efficient and controllable way to convert biomass into chemicals or fuels. Due to the unique characteristics of dielectric heating caused by electromagnetic microwave irradiation, a higher quality of pyrolysis products and high efficiency of the process is expected in relation to conventional heating (LEITE, 2021).

Although microwave-assisted pyrolysis can provide several advantages over conventional pyrolysis, it still presents several challenges. Among them, the bio-oil yield of approximately 40% by weight, a lower value than other configurations, non-homogeneous heating and the quality of the bio-oil produced are issues that still need to be explored (PIMENTA, 2022).

Energy production from biomass in Latin America contributed between 10% and 14% of the world's renewable sources (MOYA et al., 2019). The redistribution of the products resulting from the pyrolysis process makes it possible to verify the balance of the conversion of wood into charcoal and by-products, thus being called mass and energy balance, which can be used as an alternative to verify mass and energy yields that were not used. in the process (JESUS, 2016).

Pires et al. (2017) state that forest residue is characterized as the material that is discarded in the wood collection stage, both in natural forests and forests and in reforestation, and by sawdust and shavings produced in wood processing. Araújo et al. (2018) say that the main species cultivated in Brazil for energy use belong to the genus *Eucalyptus*. According to the Brazilian Tree Industry (IBÁ, 2016) eucalyptus plantations occupy 5.6 million hectares of the country's planted tree area. *Eucalyptus Citriodora* is one of the most important species of eucalyptus, being cultivated all over the world. In Brazil, its plantations were started with a view to physiological adaptation, growth, and wood production for charcoal production.

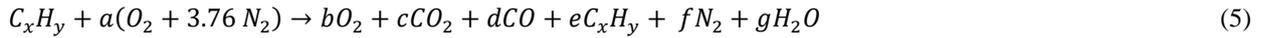
## 2.2 Energy and exergy analysis of an internal combustion engine

Alkidas (1988) and Abbasi and Emamverdi (2014) argue that First Law analysis (or energy analysis) of a system does not make it possible to determine energy distribution across the system boundary and does not provide the means to assess responsible energy degradation. by reducing the work produced by the system. The maximum (available) work of a system in a given thermodynamic state related to a given ambient condition can be determined by Second Law analysis (or exergy analysis). To assess the performance of an ICE, exergy analysis can be applied to assess the inefficiencies associated with various processes (combustion, friction, fuel injection, for example).

Sayin et al. (2007) applied the first law of thermodynamics to perform an energy analysis in an ICE. Some simplifying assumptions were adopted, such as: the engine runs steadily, combustion air and exhaust gases are ideal gas mixtures, and the kinetic and energy potential effects of combustion air, fuel flow and exhaust gases are insignificant. Thus, the first law equation adapted to the ICE is shown in Eq. (4):

$$\frac{Q_{ICE}}{n_{fuel}} - \frac{W_{ICE}}{n_{fuel}} = \bar{h}_p - \bar{h}_r = \sum_p n_{out}(\bar{h}_f^0 + \Delta\bar{h}) - \sum_r n_{in}(\bar{h}_f^0 + \Delta\bar{h}) \quad (4)$$

Where:  $n_{fuel}$  is the fuel molar flowrate,  $\bar{h}_p$  and  $\bar{h}_r$  represents the products and reactants enthalpies per mol of fuel, respectively. The subscripts *out* and *in* represent the air, fuel (reactants) and combustion gases (products) flows that enter and leave the control volume.  $n_{out}$  and  $n_{in}$  represent the number of moles entering and leaving the control volume and are calculated previously by Eq. (5) relative to the combustion reaction. Finally,  $\bar{h}_f^0$  and  $\Delta\bar{h}$  represent the enthalpy of formation and the change in enthalpy due to a given state change in constant composition.



Sayin et al (2007) state that the specific exergy of a fluid flow can be found by adding the specific thermomechanical (tm) and chemical (ch) exergies. This relationship is described by Eq. (6):

$$ex_{tot} = ex_{tm} + ex_{ch} \quad (6)$$

The specific thermomechanical exergy can be defined by Eq. (7):

$$ex_{tm} = (h - h_0) - T_0(s - s_0) \quad (7)$$

Where:  $h$  and  $s$  are the fluid specific enthalpy and entropy, respectively, while  $h_0$  and  $s_0$  are the corresponding values of enthalpy and entropy when the fluid reaches the dead state at 25°C ( $T_0$ , reference temperature) and 101325 kPa ( $P_0$ , reference pressure).

The specific chemical exergy ( $ex_{ch}$ ) of liquid fuels per unit mass can be determined by (Sayin et al., 2007), according to Eq. (8):

$$ex_{ch}^f = \left[ 1,0401 + 0,1728 \left( \frac{H}{C} \right) + 0,0432 \left( \frac{O}{C} \right) + 0,2169 \left( \frac{S}{C} \right) (1 - 2,0628) \right] \times |LHV| \quad (8)$$

According to Bilgen et al (2016), Equation (8) is obtained by statistical method that employs the result of the ratio of components C, H, O, N and S multiplied by the LHV value of fuels and liquid biofuels. Baskaran and Venkadesh (2015) state that exhaust gases can be considered with an ideal gas mixture. The specific thermomechanical exergy of the exhaust gases at a temperature ( $T$ ) and pressure ( $P$ ), can be expressed by Eq. (9):

$$ex_{tm} = \sum_{i=1}^n a_i \left\{ \bar{h}_i(T) - \bar{h}_i(T_0) - T_0 \left[ \bar{s}_0(T) - \bar{s}_0(T_0) - \bar{R} \ln \frac{p}{p_0} \right] \right\} \quad (9)$$

Where:  $a_i$  is the molar quantity of the component  $i$  in the gases mixture,  $\bar{s}_0$  is the absolute entropy at a standard pressure and temperature, and  $\bar{R}$  is the gases universal constant.

The chemical exergy of the exhaust gases can be expressed by (Baskaran; Venkadesh, 2015), according to Eq. (10):

$$ex_{ch}^{exh.gas} = \bar{R}T_0 \sum_{i=1}^n a_i \ln \left( \frac{y_i}{y_i^0} \right) \quad (10)$$

Where:  $y_i$  is the molar fraction of the component  $i$  in the exhaust gas, and  $y_i^0$  is the molar fraction of the component  $i$  in the reference environment.

The thermomechanical and chemical exergy of the combustion air can be ignored since the intake air has been admitted very close to the reference state values and the thermomechanical exergy of the fuel can also be ignored. The total exergy of the exhaust gases can be obtained by the sum of the equations 9 and 10. Thus, the exergy balance for an ICE operating at steady state can be expressed by (Ahmed and Mahanta, 2012), according to Eq. (11):

$$0 = \sum_j \left( 1 - \frac{T_0}{T_j} \right) \dot{Q}_{vc} - \dot{W}_{vc} + \dot{n}_{fuel} ex_{fuel} - \dot{n}_{exh.gas} ex_{exh.gas} - \dot{I}_d \quad (11)$$

Where:  $ex_{fuel}$  and  $ex_{exh.gas}$  represent the specific exergies of the fuel and the exhaust gas, respectively.  $T_j$  is the absolute temperature of the boundary of the control volume where the heat is rejected. The term  $\left( 1 - \frac{T_0}{T_j} \right) \dot{Q}_{vc}$  represents the rate of heat transfer that accompanies the exergy. The term  $(\dot{n}_{fuel} ex_{fuel})$  is the fuel exergy rate and

( $\dot{n}_{exh.gas} ex_{exh.gas}$ ) is the exhaust gases exergy rate. Finally,  $\dot{I}_d$  represents the exergy rate destroyed in the control volume by the irreversibilities.

Therefore, the exergy efficiency of an ICE can be expressed as (Sayin *et al.*, 2007), according to Eq. (12):

$$\varepsilon = \frac{\dot{W}_{ICE}}{\dot{n}_{fuel} ex_{fuel}} \quad (12)$$

Where:  $\dot{W}_{ICE}$  is the internal combustion engine power output.

Rakopoulos and Giakoumis (2006), Ferguson (2001), and Rakopoulos (1993) argue that studies of exergy analysis or second law analysis applied to an ICE are made based on the first law analysis or energy analysis. In the case of ICE Diesel cycle, the “zero-dimensional model” is used, which considers the gases mixture in the cylinder as a uniform mixture, also being called a “single-zone model”. In this model, it is assumed that the working fluid in the engine is the thermodynamic system, which does not exchange energy and mass with the neighborhood. The energy released during the combustion process is calculated by the energy analysis on the system.

Considerations include (Rakopoulos; Giakoumis, 2006, Ferguson, 2001, Rakopoulos, 1993):

- Spatial homogeneity of pressure and temperature;
- The working fluid is treated as an ideal gas;
- Polynomial relationships of temperature and pressure model the properties (enthalpy and internal energy) of the gas;
- The heat released by the combustion distributes uniformly throughout the cylinder;
- Blow-by losses are not taken into account;
- The enthalpy associated to fuel injection is ignored.

### 3. MATERIALS AND METHODS

The engine chosen for the analysis was the Scania 500kVa DC model 13072A.403-487-Ref 02-13, which was purchased for the project. Therefore, the technical data of this engine were used in the calculations. This engine has six cylinders and 605 hp power using commercial diesel oil. Table 1 shows the technical characteristics of this engine.

Table 1. Scania 500kVa motor data sheet at 60Hz (Scania, 2019).

MODEL	DC 13 072A.403-487 – Ref 02-13
Displacement (L)	12.7
Cylinder Diameter (mm)	130
Stroke (mm)	160
Number of cylinders	6
Standby Power at 1800 rpm (kW)	487
Power with load at 1800 rpm (kW)	445
Volumetric Compression Ratio	16.3:1
Fuel consumption - 100% (L / h)	87.7
Fuel consumption - 75% (L / h)	64.8
Fuel consumption - 50% (L / h)	43.7
Fuel consumption - 25% (L / h)	21.9

The theoretical analysis of the combustion performed by the software CEA NASA used two situations: in the first situation, the fuel adopted was Jet-A ( $C_{12}H_{23}$  available in the library of the software) and the second situation, the fuel used was a mixture of bio-oil (GC) and commercial diesel oil (S10). Jet-A has a specific mass of 800 kg/m<sup>3</sup> and lower heating value of 43000 kJ/kg, being similar to commercial diesel. The bio-oil/diesel mixture was modeled using the data obtained by gas chromatographic method. Table 2 shows the result of the gas chromatography coupled to the mass spectrometer of the bio-oil sample obtained from the process of microwave assisted pyrolysis (MAP). The diesel oil (S10) has a specific mass of 830 kg/m<sup>3</sup> and lower heating value of 45000 kJ/kg.

The calculations involving the energy and exergy analyzes were performed using the outputs of the software (combustion process), the information found in the literature, and the results obtained in the experimental analysis. These results are shown in the following section.

Table 2. Bio-oil results of gas chromatography coupled to a mass spectrometer.

Item	Molar weight (kg/kmol)	Mass quant. (kg)	% mass	% volume	LHV (kJ/kmol)	LHV (kJ/kg)
Methanol (CH <sub>3</sub> OH)	32.04	50.97	6.37	6.24	725700	22678.7
Acetone (C <sub>3</sub> H <sub>6</sub> O <sub>3</sub> )	58.08	6.32	0.79	0.43	1772000	30551.7
Methyl acetate (C <sub>3</sub> H <sub>6</sub> O <sub>2</sub> )	74.08	30.06	3.75	1.59	1583000	25532.2
Acetic acid (C <sub>2</sub> H <sub>4</sub> O <sub>2</sub> )	60.05	177.13	22.14	11.58	875600	14593.3
Tar (C <sub>6,1</sub> H <sub>7,3</sub> O <sub>0,7</sub> )	91.70	238.57	29.82	14.14	1095545.8	11947.1
Other (C <sub>6,328</sub> H <sub>7,855</sub> O <sub>2,318</sub> )	120.9	296.9	37.11	66.02	709432.6	5868.9
Total	103.15	799.95	100	100.00	802716.78	7781.98

#### 4. RESULTS AND DISCUSSION

Figures 2 illustrates the combustion simulation results by CEA software using Jet-A and the bio-oil / diesel oil mixture as the fuel. The figure show the molar fractions of CO, NO and O<sub>2</sub> to the pressure of 4500 kPa, varying the  $\lambda$ -factor from 0.8 to 1.2. The lower heating value considered for Jet-A was 43000 kJ/kg and 22303,88 kJ/kg for the mixture bio-oil / diesel oil.

It is observed that, for Jet-A, the molar fractions of the compounds CO and NO have an intersection near the equivalence ratio of 1. This behavior is not detected when the bio-oil / diesel oil mixture is used. The differences between the cases studied were due to the input parameters as well as the respective air-fuel ratios. For Jet-A, the calculated air-fuel ratio was between 18.20 and 12.13. For the bio-oil/diesel mixture, this parameter was between 8.08 and 5.39. It is verified that the behavior of the CO, NO and O<sub>2</sub> molar fractions for the bio-oil / diesel oil mixture was higher than the Jet-A ones throughout the equivalence ratio variation. For the bio-oil/diesel oil blend, CO levels increased linearly from the lean to the rich mixture. In the case of Jet-A the increase occurred, but there was intensification from the stoichiometric mixture. During poor mixing, the CO molar fractions were lower for Jet-A fuel than observed in the same range of the bio-oil / diesel oil mixture. The NO molar fractions were constant in the case of the bio-oil / diesel oil mixture. In the case of Jet-A fuel, there was a constant behavior up to the stoichiometric value. From this point, there was a drop in values approaching zero in the rich mixture.

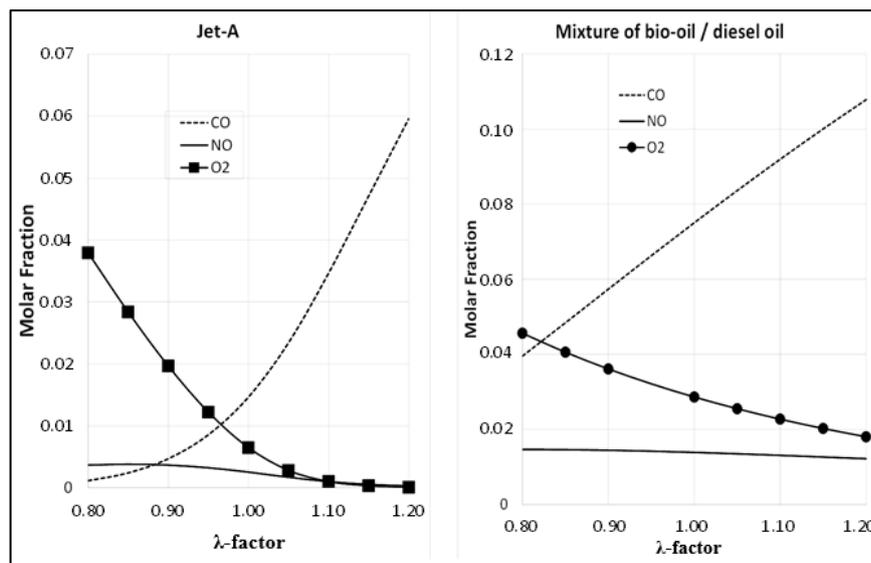


Figure 2. Combustion simulation results showing the molar fraction of CO, NO and O<sub>2</sub> regarding the  $\lambda$ -factor for Jet-A and bio-oil / diesel oil mixture as the fuel.

Figures 3 and 4 respectively show the exergy variation during the combustion process for Jet-A and for the bio-oil / diesel oil mixture, for all  $\lambda$ -factor used in the combustion modeling.

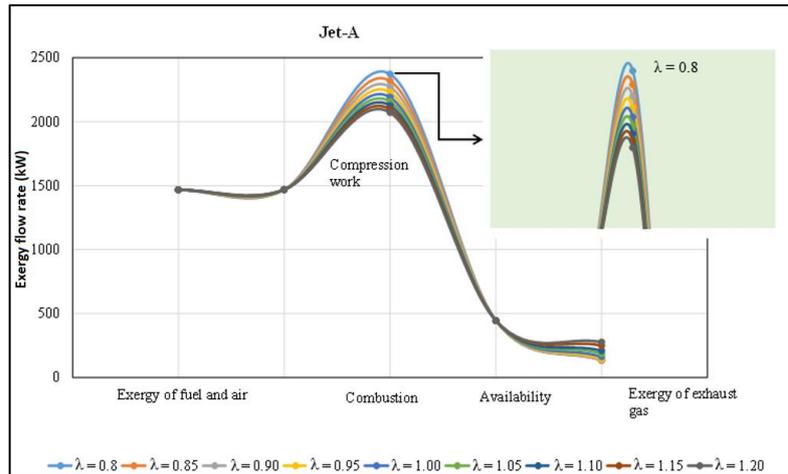


Figure 3. Results of the variation of exergy in the combustion process of Jet-A fuel.

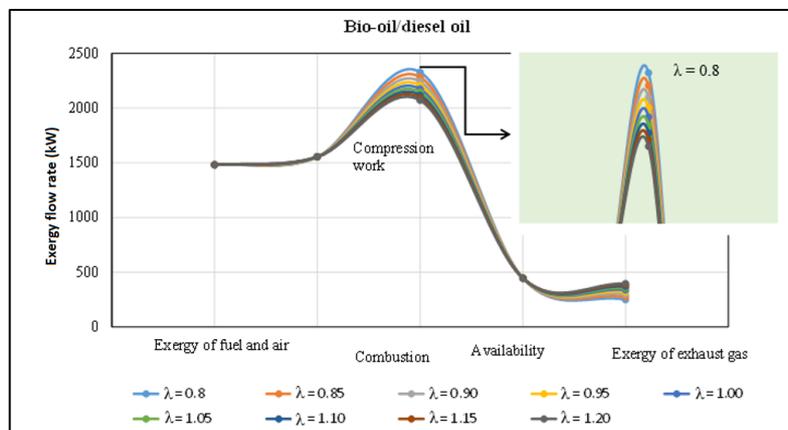


Figure 4. Results of the variation of exergy in the combustion process of the mixture bio-oil / diesel oil as the fuel.

Figures 3 and 4 show the behavior of the input (fuel and air intake) exergy and output (exhaust gases and engine power) exergy. As a function of the single zone modeling conditions, the two cases analyzed (Jet-A and the bio-oil / diesel oil mixture) can be compared within the range of the  $\lambda$ -factor (0.8 to 1.2). The peak of the curve indicates the maximum exergy achieved by the fuel and the air inside the combustion chamber. It is observed that the curve decay represents the expansion work after the end of the combustion. It is noted that there is a closeness of values throughout the exergy variation versus the  $\lambda$ -factor.

Figure 5 shows the variation of the exergy efficiency versus the  $\lambda$ -factor for Jet-A and for the bio-oil/diesel oil mixture as the fuel.

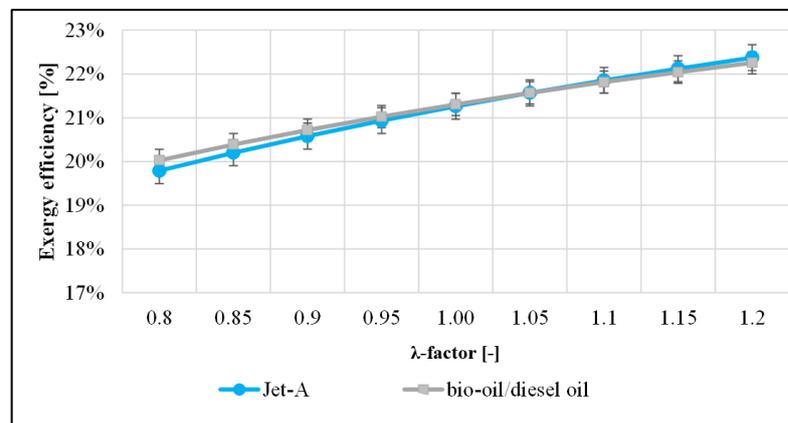


Figure 5. Exergy efficiency versus  $\lambda$ -factor.

Figure 5 shows the result of the exergy analysis, which quantifies the process along the entire exergy variation for the control volume analyzed. It is observed that there is a similar behavior of both Jet-A and the bio-oil/diesel oil mixture, as shown in figures 3 and 4. The figure shows that, in both cases, there is an increase in exergy efficiency as the  $\lambda$ -factor increases.

Figures 6 and 7 show, respectively, the exergy variation during the combustion process for Jet-A ( $\lambda$ -factor of 0.9) and for the bio-oil / diesel oil mixture ( $\lambda$ -factor of 0.8).

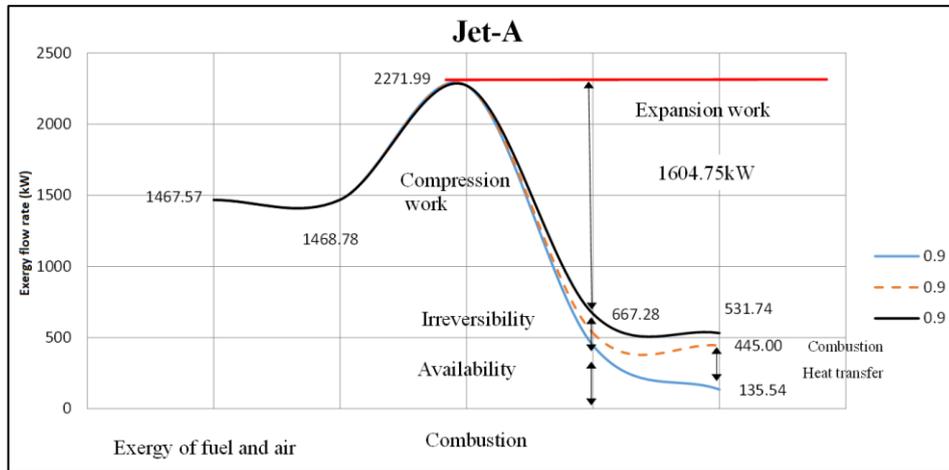


Figure 6. Exergy variation during the combustion process of the fuel Jet-A for the  $\lambda$ -factor of 0.9.

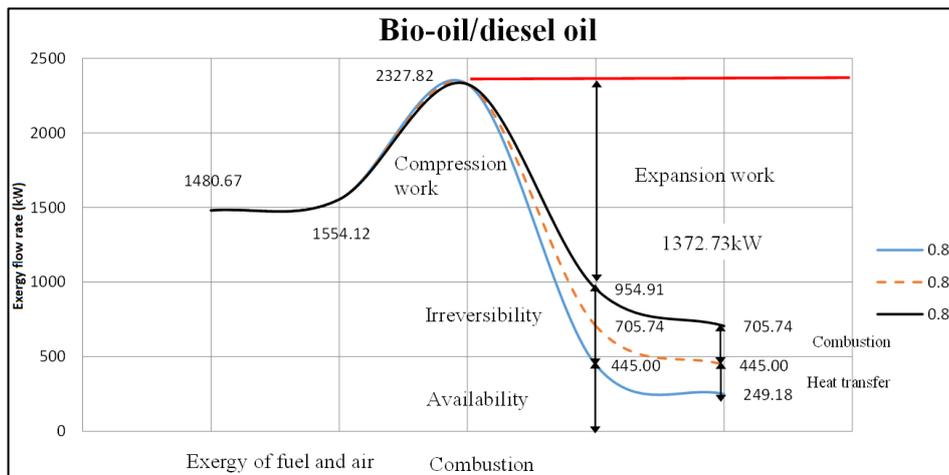


Figure 7. Exergy variation during the combustion process of the fuel bio-oil/diesel oil mixture, for the  $\lambda$ -factor of 0.8.

Figures 6 and 7, respectively, show for Jet-A ( $\lambda$ -factor of 0.9) and for the bio-oil/diesel oil mixture ( $\lambda$ -factor of 0.8), the values obtained of exergy variation. These figures were chosen because they represent the point at which the process reaches the engine's default values (41% of energy analysis and 1084.69 kW of power). Under these conditions, Jet-A reached a maximum exergy value of 2271.99 kW and the bio-oil/diesel oil mixture a value of 2327.82 kW. The irreversibilities (combustion and heat exchange) ranged from 667.28 to 135.54 kW which corresponded to an expansion work of 1604.25 kW in the case of Jet-A. The bio-oil/diesel oil mixture, the irreversibilities varied from 954.91 to 249.18 kW and the expansion work was 1372.73 kW.

The results shown previously indicate that the bio-oil/diesel oil mixture behaves similarly to Jet-A, what is verified the extent to which the  $\lambda$ -factor was increased to 1.20. These results also show that, for poor mixing, both cases met the engine demand and the bio-oil/oil blend was the first to reach these conditions, which can be justified by the fact that the diesel oil (S10) higher than the Jet-A PCI. In addition, the 75-25% proportion was chosen properly.

In both cases, the exergy destruction (irreversibilities) was similar as previously seen, since the yield fell from 41% (energy analysis) to values close to levels of 20 to 22% (exergy analysis).

The results of the combustion modeling show that the energy performance of the engine is reached (specific fuel consumption of 195 g/kWh, heat added of 1084.69 kW and fuel flow of 0.45 kg/s) with Jet-A for  $\lambda$ -factor of 0.9 and the

bio-oil/diesel oil mixture for  $\lambda$ -factor of 0.8. These fuels produced carbon monoxide, nitrogen oxides, and oxygen molar fractions (0.0047, 0.0037, 0.0197) and (0.0395, 0.0146, 0.0456), respectively. The exergy efficiency of Jet-A and the bio-oil/diesel mixture were respectively 20.58% and 20.03%. These efficiencies were calculated for the Jet-A and for the bio-oil/diesel mixture, respectively, through the work of compression, expansion work and irreversibilities (combustion and heat exchange) obtained from the variation diagram of availability (exergies) of the internal combustion engine.

## 5. CONCLUSION

This paper performed the energy and exergy (theoretical) analysis of a compression-ignition internal combustion engine. Two fuels were modeled for these analyzes: Jet-A and a blend of 25% bio-oil, 75% diesel.

The data for the Jet-A fuel were established in the library of the software Chemical Equilibrium for Applications (CEA) while the data for the bio-oil were found with a chromatography analysis of a bio-oil produced from a microwave-assisted pyrolysis process of a biomass.

The theoretical analyzes were elaborated using the single-zone model in steady state. The combustion process of the Jet-A and bio-oil was modeled in Chemical Equilibrium for Applications (CEA NASA) software.

The calculated LHV value of the bio-oil was not similar to the literature values. There was a difference between the values. The lower heating value of the literature is about twice the value (7781.98 kJ/kg versus 19000 kJ/kg) of this parameter. This paper works with the values obtained experimentally of the microwave-assisted pyrolysis process of *Eucalyptus Citriodora*.

The emissions of CO and NO<sub>x</sub> regarding to the  $\lambda$ -factor variation were higher in the bio-oil/diesel oil mixture compared to Jet-A. The mass flowrate for Jet-A (0.0252 kg/s, specific fuel consumption of 203 g/kWh), and for the bio-oil/diesel oil mixture (0.049 kg/s, specific fuel consumption of 396 g/kWh) are above the values quoted in the literature for diesel oil (200 g/kWh).

The performance of the energy analysis with diesel oil (S10) showed a thermal efficiency of 41%, heat rejection in exhaust gases of 34.8%, heat rejection in the refrigeration system of 13.1%, heat rejection in lubrication and losses by radiation of 4.1%, demanded 0.45 kg/s of the fuel and 1084.69 kW of heat needed. The fuel Jet-A showed the performance conditions for the  $\lambda$ -factor of 0.9 (0.4334 kg/s) and the bio-oil/diesel oil mixture for the  $\lambda$ -factor of 0.8. The values are similar to the ones found in the literature. In addition, Jet-A and the bio-oil/diesel blend presented similar exergy efficiencies within the range of the equivalence ratio adopted in the combustion modeling.

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