TERMODINAMIC SIMULATION OF A DIESEL ENGINE USING DIFERENTS BLENDS OF DIESEL OIL BIODIESEL

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Abstract. *Thermodynamic modeling of a single cylinder Diesel Engine working with diesel oil and blends of biodiesel and diesel oil. The computational models can help optimization and development of engines using different fuels, allowing time and resources reduction on experimental tests, consequently meaning a helpful tool for research centers. The research for using renewable fuels in diesel engines has been developed in the least years due to huge necessity to use these alternative fuels. The present work presents a thermodynamic simulation (zero-dimensional) of a single cylinder Diesel Engine, working with different blends of diesel and biodiesel. The engine simulator is validated in a test bench using a single cylinder engine. Just of all some definitions and concepts related to internal combustion engines are given. Following thus, an explanation of the problem and the aims are exposed. The zero-dimensional model, to be more precise, it includes the variation of the specific heat ratio with temperature of the working fluid and cylinder wall temperature heat exchange. Some simplifications assumptions are adopted according to the literature, as ideal gas law for intake and exhaust gases, etc. The engine geometry representation establishes the displacement volume as a function of the crank angle. It is assumed that a polytrophic transformation occurs between the intake valve closure and the start of the combustion. The combustion modeling uses Wiebe function, which describes the fuel heat released rate during premixed and mixing controlled combustion. It calculates the fuel mass fraction burned as function of combustion efficiency and the energy introduced by the fuel mass. The whole thermodynamic formulation model is defined for the cylinder compression, combustion and expansion processes, meaning that intake and exhaust valves are closed. A differential equation is obtained for the work output inside the cylinder together with other equations. Simultaneously solving of the system leads to several variables as function of the crank angle. Woschni equation is used for the heat transfer coefficient. Specific heat ratio-calculation is done for the reagents and products of the fuel specification. The different fuel modeling is performed in the combustion equation by obtaining its chemical composition. To solve the differential equation system the MATHEMATICA application software is implemented. Finally the curve of internal cylinder pressure versus the crank angle is obtained for the simulation process using different blends of fuel. Experimental test, obtained for the same engine connected to a dynamometer with data acquisition system. Results of simulation and test are compared.*

Keywords*: Engine Diesel, Performance, Biodiesel*

1. INTRODUCTION

In the present energy scenario, there is an increase in blends of diesel oil and biodiesel research as alternative fuels for internal combustion engines. In Brazil, the soy bean and the palm oil biodiesels are been produced for this purpose.

When those fuel blends are used as a substitute for diesel oil, it’s highly essential to understand the parameters that affect the operational phases of the engine and results in performance modification and emissions.

The biodiesel properties depend on the type of the vegetable oil used in the production process. Experimental analyses of the engine with various fuel blends require much effort, time and consequently great amount of money. Hence, a thermodynamic model of the engine was developed to analyze the performance characteristics of the engine fueled with those different blends.

2. RESEARCH METHODOLOGY

A zero-dimensional simulation model is developed for engine simulation. The combustion simulation is based in Lyn (1962) and Watson *et al.* (1980a) and consist of three different phases, as followed:

- Ignition delay;
- Pre-mixed combustion;
- Diffuse combustion.

The thermodynamic study and development of these phases results in a system of ordinary differential equations, solved with the MATHEMATICA software package. The variables of the equations described in the test are defined in the nomenclature in the item 9 of the paper.

3. THERMODYNAMIC MODEL

A model based on the first law of thermodynamics for closed cycle has been developed to study the performance of the engine, including compression, combustion and expansion processes.

The compression process is analyzed with the ideal gas equation and polytropic process. The combustion process is modeled using the equation of double Weibe's function relation. According to Watson *et al.* (1980b) and Miyamoto *et al.* (1985) these functions are used to adequately model the combustion in the premixed and diffuse zone.
The ignition delay is also taken into account in the model and is calculated using the Hardenberg function.

3.1. Engine Geometry

The basic geometry of the engine cylinder is shown in the Figure 1.

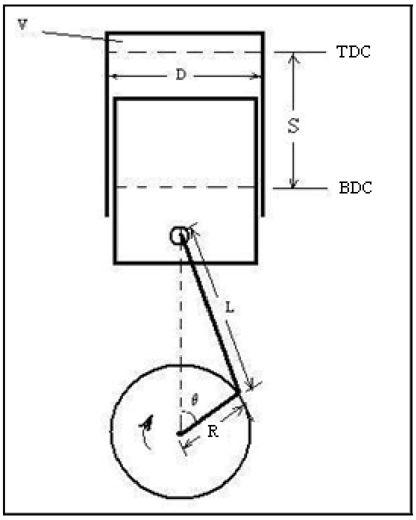

Figure 1. Engine geometry

Then, the rate of change of cylinder volume (V) with the crank angle is given by Heywood (1988a), in the equation (1):

$$V(\theta) = V\left\{1 + \frac{1}{2}(r-1)\left[BM + 1 - \cos(\frac{\theta r}{180}) - \sqrt{(BM^2 - \sin^2(\frac{\theta r}{180}))}\right]\right\} \quad (1)$$

3.2. Energy Equation

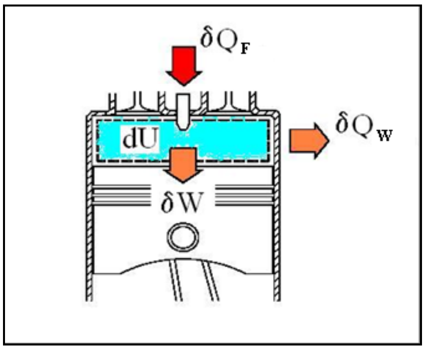

Figure 2. Cylinder energy balance

The energy equation, according to the first law of thermodynamics, after substituting the rate of change of heat release and convection heat losses, can be written as in equation (2):

$$\frac{1}{\gamma - 1}\left[\frac{1}{T}\frac{dT}{d\theta} - \frac{1}{\gamma - 1}\frac{d\gamma}{d\theta}\right] = \frac{1}{PV}\left[\frac{\delta Q_F}{\delta\theta} - \frac{\delta Q_W}{\delta\theta}\right] \quad (2)$$

3.3. Ignition Delay

The time delay between the beginning of injection and the start of combustion, defined as ignition delay period, is calculated by Handenberg and Hase (1979), in the following equation (3):

$$\tau_{ID}(\Delta\theta) = (0.36 + 0.22 S_P) e^{\left[E_a \left(\frac{1}{RT} - \frac{1}{17190}\right)\left(\frac{21.2}{P - 12.4}\right)^{0.63} \right]} \quad (3)$$

Where Ea is the energy obtained by the following expression:

$$E_a = \frac{618840}{CN + 25} \quad (4)$$

3.4. Combustion Release Rate

The total energy introduced in the system by the fuel (Q_F), can be expressed as in equation (5):

$$Q_F = m_F . LHV \quad (5)$$

The heat release rate $\left(\frac{\delta Q_F}{\delta\theta}\right)$ can be expressed by:

$$\left(\frac{\delta Q_F}{\delta\theta}\right) = Q_F \cdot \frac{dx}{d\theta} \quad (6)$$

Where $\left(\frac{dx}{d\theta}\right)$ is the rate of fuel combustion. The fraction of fuel burned, x(θ), can be expressed by the double Wiebe function, as follows (equation (7)):

$$x(\theta) = 1 - \left(x_p . e^{-a\left(\frac{\theta - \theta_{ic}}{\Delta\theta_p}\right)^{mp+1}} + x_d . e^{-a\left(\frac{\theta - \theta_{ic}}{\Delta\theta_d}\right)^{md+1}} \right) \quad (7)$$

Note that: $x_p + x_d = 1$.

3.5. Heat Transfer

For the calculation of heat transfer through the cylinder wall, h(θ), WOSCHNI (Woschni and Anisits, 1974) equation was used, as follows (equation (8)):

$$h(\theta) = 3.26 * D^{-0.2} * P^{0.8} * T^{-0.55} * v_g^{\ 0.8} \quad (8)$$

And

$$v_g = 2.28 * S_P + 0.00324 (P - P_0) * V_d * \frac{T_1}{P_1 V_1} \quad (9)$$

3.6. Specific Heat Calculation

It is presented a new proposal for the calculation of the specific heat at constant pressure (c_p), as a function of the temperature, in a polynomial equation for temperatures above 4000 K, by Lanzafame and Messina (2003):

$$c_p = a_0 + a_1(\ln T) + a_2(\ln T)^2 + a_3(\ln T)^3 + a_4(\ln T)^4 + a_5(\ln T)^5 \quad (10)$$

According to Zhou and Fet (2005), biodiesel presents the specific heat at constant pressure, cp, 40% higher than diesel oil. The coefficients a_1, a_2, a_3, a_4 and a_5 are presented in the references for biodiesel and diesel oil.

3.7. Equivalent Fuel

The equivalent fuel definition (Heywood, 1988b) was used in the research, denoted by $C_oH_pO_q$, where C, H and O are the atoms of carbon, hydrogen and oxygen. Then,

$$C_O H_p O_q \rightarrow \%D*(C_{13}H_{28}) + \%B*(C_{20}H_{37}O_2) \tag{11}$$

And the values for “o”, “p” and “q” are as follows:

$$o = \%D*13 + \%B*20 \tag{12}$$

$$p = \%D*28 + \%B*37 \tag{13}$$

$$q = \%B*2 \tag{14}$$

Where “%D”, and “%B” are the percentage of diesel and biodiesel presented in the mixture.

3.8. Performance Parameters

The simulation program calculates the indicating work (W_i), the mean indicating pressure (Pm_i), the indicating power (Pot_i), the indicated torque (Tq), the thermal efficiency, the engine efficiency and the specific fuel consumption (SFC), as defined in Heywood (1988c) .

4. INPUT DATA

The input data for the simulation model are:

- Engine geometry characteristics;
- Initial conditions of operation;
- Combustion data, as pre-mixed and diffusive combustion duration, fuel burned rate, etc.;
- Fuels composition data;
- Fuels blends varying from 3% (B3) to 100% (B100) of biodiesel mixed with diesel oil.

In this research, soy biodiesel was used in blends. Table 1 presents the soy biodiesel analysis results.

Table 1. Soy biodiesel analysis results

Characteristics	**Unity**	**Range**	**Results**
Density at 20 °C	kg/m^3	850 - 900	882.1
Viscosity at 40 °C	mm^2/s	3.0 - 6.0	4.132
Water content	mg/kg	max 500	< 500
Total Contamination	mg/kg	max 24	x
Cetane Number			45
Flash Point	ºC	100	153
Carbon content	% mass	max 0.02	0.00724
Pour Point	ºC	-	-3
Acidity Index	mgKOH/g	max 0.50	0.1

5. SIMULATION RESULTS

Figure 3 presents the pressure versus crank angle simulation results for the operation point with 50% of the maximum load and 2500 rpm. Different curves are presented for several blends used.

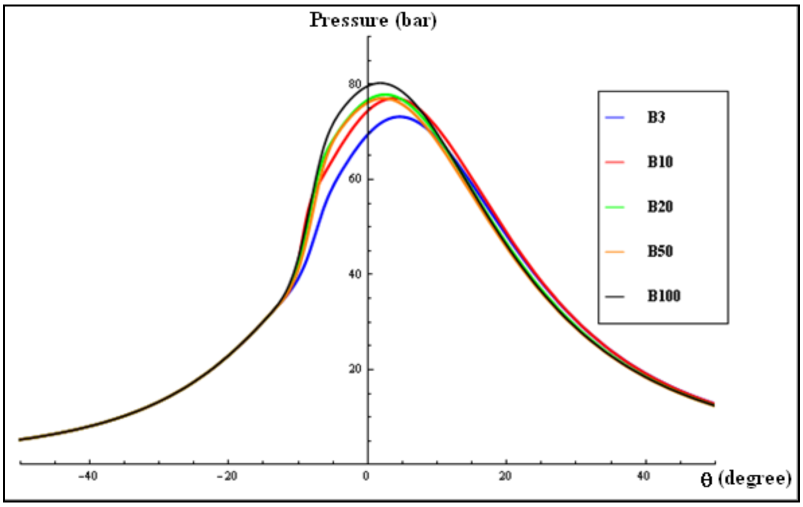

Figure 3. Pressure versus crank angle simulation results

6. EXPERIMENTAL TEST APPARATUS

Table 2 presents the main characteristics of the engine connected to the dynamometer.

Table 2. Engine main data

Manufacturer / Model	AGRALE/M95W
Type	04 strokes - Diesel
Cylinder	01 cylinder vertical, direct injection
Cylinder diameter	95 mm
Stroke	105 mm
Admission Valve diameter	42 mm
Displacement volume	744 cm3
Ratio of connecting rod length to crank radius (L/R)	3.24
Compression ratio	21:1

The test facilities of the Thermal Engines Laboratory of the Federal University of Rio de Janeiro were used in tests, as dynamometers and acquisition system for acquiring performance parameters data.

Figure 4 presents a simplified diagram of the equipments used during the tests.

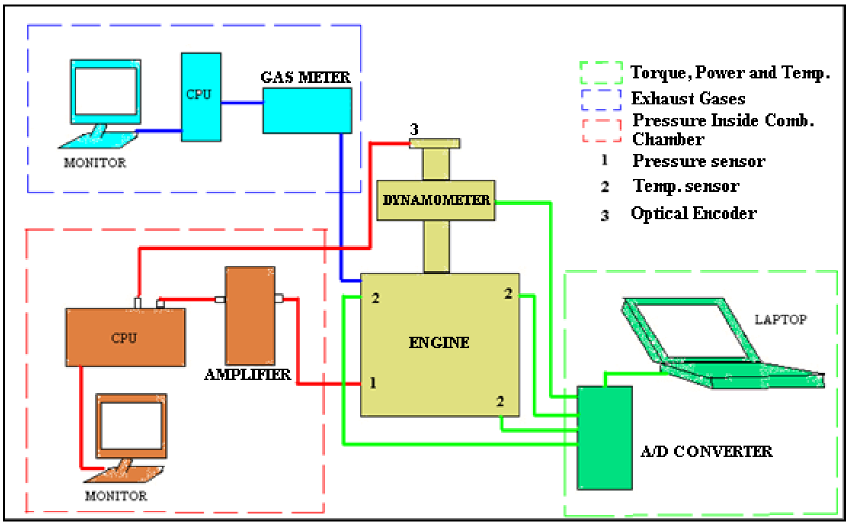

Figure 4. Equipments used during the test

A DINAMATIC dynamometer (Foucault current) was used with the DinMon data acquisition system.

Several instruments for fuel consumption, pressure, temperature and torque measurements belonging to the bench test facilities were used. Figure 5 shows the location of the temperature sensors in the engine.

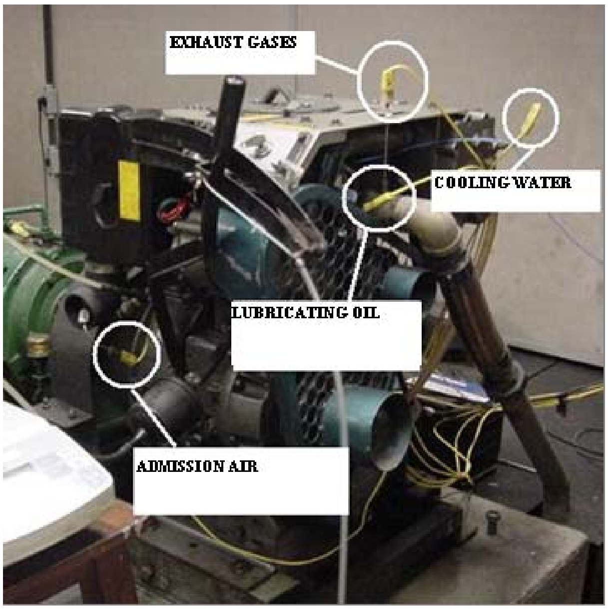

Figure 5. Position of temperature sensors inside the engine

7. SIMULATION AND EXPERIMENTAL TEST ANALYSIS

Several data using different blends were analyzed by comparing the simulation and experimental tests results, for the performance parameters. Figure 6 presents the simulation results and the experimental data of B20 blend for 50% load and 2500 rpm.

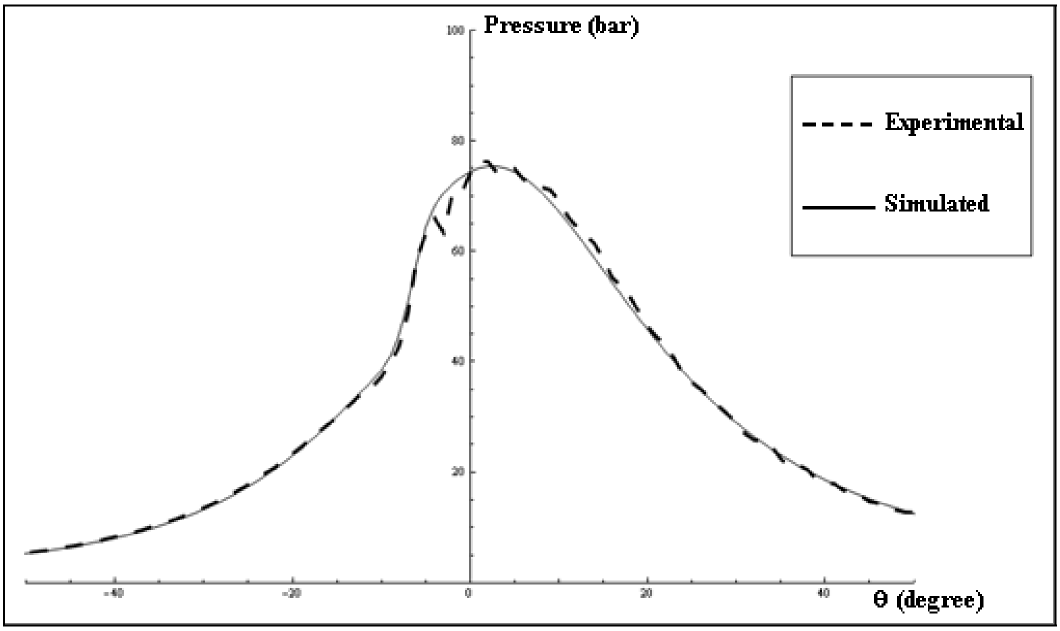

Figure 6. Simulated and experimental data of pressure versus crankshaft angle

Table 3. Performance parameters comparison for B20 blend

% Load/ RPM	***Mean Indicating Pressure* (bar)**			**Brake Effective Power (kW)**		
	Exp	**Sim.**	**Dev.(%)**	**Exp**	**Sim.**	**Dev. (%)**
25 / 1500	3.71	3.85	+3.81	1.57	1.48	-6.00
50 / 1500	5.48	5.62	+2.63	3.16	3.09	-1.99
75 / 1500	8.23	8.36	+1.61	4.74	4.63	-2.32
25 / 2000	3.84	3.98	+3.68	2.09	2.12	+1.00
50 / 2000	5.39	5.42	+0.58	4.21	4.19	-0.50
75 / 2000	8.07	8.21	+1.74	6.30	6.37	+1.00
25 / 2500	3.71	3.77	+1.66	2.67	2.85	+6.86
50 / 2500	5.39	5.55	+2.99	5.26	5.58	+5.97
75 / 2500	7.97	8.52	+6.90	7.91	8.01	+1.32

Table 3 - Performance parameters comparison for B20 blend (cont.)

% Load / RPM	**Max. Pressure (bar)**			**SFC (g/kWh)**		
	Exp	**Sim.**	**Dev. (%)**	**Exp**	**Sim.**	**Dev. (%)**
25 / 1500	65.51	66.38	+1.33	340	369	+8.56
50 / 1500	72.33	73.47	+1.58	296	303	+2.28
75 / 1500	78.73	78.75	+0.03	326	334	+2.35
25 / 2000	66.91	64.79	-3.17	378	375	-0.88
50 / 2000	75.82	73.21	+3.44	291	292	+0.48

75 / 2000	79.19	79.07	-0.15	314	311	-0.98
25 / 2500	60.54	61.35	+1.34	408	378	-7.42
50 / 2500	76.22	75.26	-1.26	308	290	-5.84
75 / 2500	78.54	76.75	-2.28	314	310	-1.41

8. RESULTS ANALYSES AND CONCLUSIONS

Table 3 presents deviations lower than 8% for simulation and experimental performance parameters results. Accepted results are obtained.

We can note higher performance results when the engine uses higher blends (B100).

The simulation models, properly developed, can predict the performance of an engine using different blends of biodiesel and diesel oil. We can, consequently, obtain a large economy in experimental tests to be developed.

9. NOMENCLATURE

BDC	bottom dead center
BM	ratio of connecting rod length to crank radius (L/R)
CN	fuel cetane number
D	cylinder bore diameter
L	connecting rod length
LHV	lower heat value of the fuel (J/kg)
P	pressure inside the combustion chamber
P_0	pressure inside the combustion chamber, when there’s no combustion
P_1	pressure value when the admission valve is closed
Q_F	rate of combustion heat release (kJ/degree crank angle)
Q_W	heat transfer to the cylinder walls (kJ/degree crank angle)
R	universal gas constant; crank radius
S_p	piston speed
T	temperature inside the combustion chamber
TDC	top dead center
T_1	temperature value when the admission valve is closed
V	volume
V_d	displacement volume;
V_1	volume value when the admission valve is closed
r	compression ratio;
c_p	specific heat at constant pressure
c_v	specific heat at constant volume
$dV/d\theta$	rate of change of cylinder volume with crank angle
$dT/d\theta$	rate of change of temperature with crank angle
mp	form factor of the combustion chamber in the pre-mixed combustion
md	form factor of the combustion chamber in the diffusive combustion
m_F	fuel mass admitted in the cylinder in a operation cycle (kg)
vg	gas speed in the admission chamber
xp	fraction of the fuel burned during the pre-mixed phase
xd	fraction of the fuel burned during the diffusive phase
γ	specific heat ratio (c_p/c_v)
$\Delta\theta_P$	duration of the pre-mixed phase
$\Delta\theta_d$	duration of the diffusive phase
θ	crank angle (angular displacement with respect to BDC)
$\tau_{ID}(\Delta\theta)$	ignition delay in crank angle, θ

10. REFERENCES

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