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Brake Thermal Efficiency and BSFC of Diesel Engines: Mathematical Modeling and Comparison between Diesel Oil and Biodiesel Fueling

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Abstract

The objective of this work was to investigate the brake specific fuel consumption (*BSFC*) of the engine, installed in an agricultural tractor, fueled before with diesel oil (B0) and then with biodiesel (B100). This was done both vs. the engine speed, which vs. engine load. To understand the influence of the fuel heating value and the brake thermal efficiency (*BTE*), a mathematical modeling of the *BTE* vs. engine speed and engine torque was developed. The bench tests on the engine, fueled with the two fuels (B0 and B100), allowed to point out that on average, the *BTE* of the engine was unchanged in the comparison between the two fuels, while the average *BSFC* was 19% higher with the B100, consistent with the lower heating value of B100 (-17.2%). However, the power output of the engine with the B100 was not reduced by the same amount, but with an average value of 11%, as there was a partial recovery due to the slight increase in the fuel consumption rate. The fitting of the mathematical model to the engine, fuelled with the two fuels (B0 and B100), allowed to draw the diagrams of the calculated *BTE* and *BSFC* contours for both fuels, with a good accuracy represented by a mean relative error of 2.1±2.2% (B0) and 2.0±1.5% (B100). Finally, the comparison of the previous results, allowed to highlight the points of the engine speed-torque plane where the biodiesel *BTE* was different than that of diesel oil *BTE*. It emerged that the engine fuelled with B100 had a higher *BTE* at low load and speed, but lower than that of B0 at high speed due to reduced inclination to evaporate compared to diesel oil, which is critical when the speed increases and the time available for combustion is reduced.

Keywords: Biodiesel, Mathematical modeling, Brake thermal efficiency, Brake specific consumption, Diesel engine

1 Introduction

As is known the use of biodiesel (B100), instead of diesel oil (B0), shows interesting environmental benefits, even though for economic reasons, today a complete replacement of fossil fuel is not possible.

To produce biodiesel, both vegetable oils and animal fat can be used. For this reason it is renewable, not-toxic and biodegradable [1, 2, 3, 4 and 5].

Oils and fats are triglycerides [6, 7 and 8], responsible of the high viscosity characterizing these materials. As this high viscosity is incompatible with Diesel-cycle engines, triglycerides need a transesterification to activate a conversion in fatty acids through reaction using a basic catalyst. In this way, two components are produced: an *ester* (methyl or ethyl ester), commonly named *biodiesel*, because it is usable as a fuel for diesel engines having a viscosity slightly greater than diesel oil; *glycerol*, thicker than biodiesel and then easily separable by settling.

With reference to particulate and carbon dioxide, the pollutants resulting from biodiesel combustion in diesel engines, are less harmful to environment and human health [9 and 10] than traditional diesel oil.

Nevertheless, engine power is reduced [11, 12 and 13] due to the lower calorific value and the higher viscosity [14 and 15]. Anyway, the engine thermal efficiency does not change and can even improve slightly [12, 14, 15, 16, 17, 18, 27 and 19].

As a consequence, the brake specific fuel consumption (*BSFC*) increases [11, 13, 14, 20, 21 and 22, 27, 28, 29] but the smokiness lowers down to 50% [22 and 23].

Overall, the modifications to implement on engines to be powered by biodiesel are minimal [24].

The aim of this study was to investigate experimentally the Brake Specific Fuel Consumption (*BSFC*) of the engine fueled before with diesel oil (B0) and then with biodiesel (B100), either vs. the engine speed and vs. engine load.

To understand the influence of the Fuel Heating Value and the Brake thermal Efficiency (*BTE*), a mathematical modeling of the *BTE* vs. engine speed and engine torque was developed.

Finally, the fitting of the mathematical model to the engine, fuelled with the two fuels (B0 and B100), allowed to highlight the points of the engine speed-torque plane where the biodiesel *BTE* was different than diesel oil one.

2 Materials and methods

2.1 Mathematical modelling of BSFC and BTE

As is well known the relationship between the brake specific fuel consumption *BSFC* (g/kWh), the brake thermal efficiency *BTE* and the fuel heating value *H*, (kJ/kg) is:

$$BSFC = \frac{3.6 \cdot 10^6}{H \cdot BTE} \tag{1}$$

The brake thermal efficiency *BTE*, in turn, is the product of mechanical efficiency *ME* and indicated thermal efficiency *ITE*. Taking account of the friction between the moving mechanical parts, fluid pumping and operation of auxiliaries, the former can be expressed by:

$$ME = \frac{1}{1 - \frac{FMEP}{BMEP}} \tag{2}$$

where: *BMEP* is brake mean effective pressure (kPa) and *FMEP* is friction mean effective pressure (kPa). The brake mean effective pressure is defined as:

$$BMEP = \frac{4 \cdot \pi \cdot T}{D} \tag{3}$$

where *T* is engine torque (Nm) and *D* is engine displacement (dm^3).
Since the friction mean effective pressure *FMPE* is not influenced by the torque, it results that the mechanical efficiency *ME* is zero when the torque *T* is zero and it grows with the torque, according to a curve with the positive, but decreasing, derivative.
Indicated thermal efficiency *ITE* is also a function of the torque *T*. In quantitative terms one can say that it remains basically constant for low to average values of the torque, but with higher values the increase in the fuel supply is such that it worsens combustion and hence reduces *ITE*.
Thus the brake thermal efficiency $BTE = ME \cdot ITE = f(T)$ is a function of the torque, according to a curve that is well represented by a second order polynolial.
With regard to the influence of rotation speed ω (s^{-1}) on the efficiency $BTE = ME \cdot ITE = f(\omega)$, it is known [25] that the friction mean effective pressure *FMPE* is partly proportional to rotation speed and to squared speed itself.
Therefore, we can assume that also the function $ME = f(\omega)$ is a second order polynomial.
About the indicated thermal efficiency *ITE*, for any given condition of torque, it diminishes with the decrease in rotation speed ω due to increased pressure and heat losses from the combustion chamber, in consequence of the longer duration of each cycle. Therefore it is reasonable to assume the function $ITE = f(\omega)$ and so the function $BTE = ME \cdot ITE = f(\omega)$, as a polynomial of second order.
Finally, combining the two polynomial equations $BTE = ME \cdot ITE = f(T)$ and

$BTE = ME \cdot ITE = f(\omega)$, we get:

$$BTE = \left(A_0 + A_1 \cdot T + A_2 \cdot T^2\right) \cdot \left(B_0 + B_1 \cdot \omega + B_2 \cdot \omega^2\right) \tag{4}$$

Or:

$$BTE = C_0 + C_1\omega + C_2\omega^2 + C_3T + C_4T\omega + C_5T\omega^2 + C_6T^2 + C_7T^2\omega + C_8T^2\omega^2 \tag{5}$$

with coefficients C_0 ... C_8 that are easily calculable by the method of multiple regression. These coefficients are specific for each engine and each fuel (B0 or B100).

2.2 Experimental tests

The experiment was performed on the engine of an agricultural tractor, with the characteristics reported in Table 1.

To determine the *BSFC* vs. speed and torque and, in turn, to determine the *BSFC* contours using the mathematical model, it was necessary to record experimentally the triplets of torque, speed and fuel consumption rate of the engine fuelled by the two fuels (B0 and B100) with the equipments reported in Table 2.

The equipment included an hydraulic mobile test and coupled to the power take-off of the tractor and, hence, recording a power output very close to the power effectively available to the tyres (differently to SAE protocols prescriptions, in which the engine is detached from the rest of the vehicle and without most of auxiliaries, used by the engine manufacturer to indicate the nominal power). The fuel consumption rate was measured through a chrono-gravimetric method.

The development of the mathematical model will allow the comparison of performance between the two fuels (B0 and B100) as will be seen, but it also permitted to realize the *BSFC* contours, with a small number of experimental measurements (30).

These 30 experimental triplets (speed, torque and fuel consumption rate) were made according to a schedule as in Figure 1.

The first 14 tests were referred to the speed-torque curve, starting from the maximum engine speed (point 1) up to the point 14, through an increase of the brake load. Thus, two additional series of tests were conducted. The first, by placing the fuel pump rack to get a speed reduced to 210 (s^{-1}) and then increasing the brake load from point 15 to 22. The second, by placing the fuel pump rack to a lower speed (150 s^{-1}) and then increasing the brake load from point 23 to 30.

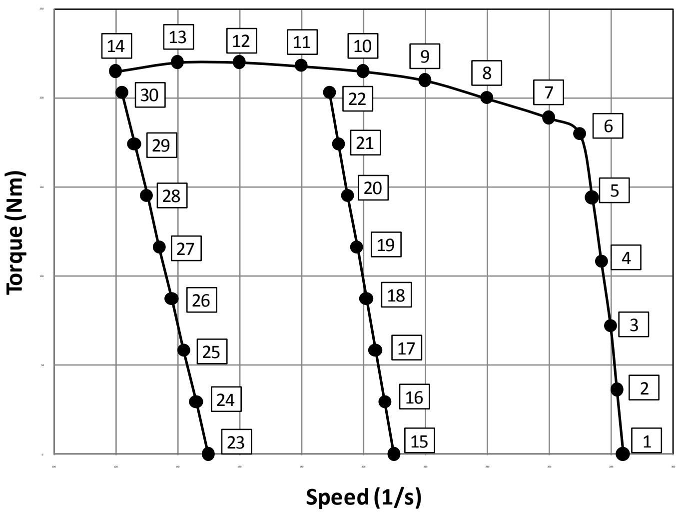

Figure 1 – A typical schedule of data points used in calibration the mathematical model.

Table 1 - Technical characteristics of the engine employed in this study.

Description	**Unit**	**Specifications**
Type	-	Diesel, turbocharged, with direct injection
Cylinders, configuration	nr.	4, straight and vertical
Total displacement	dm^3	3.905
Volumetric compression ratio	-	15.5:1
Nominal power (SAEJ1995)	kW	58.8
Nominal engine speed	rpm	2,500

The biodiesel (B100) used during the tests is a pure commercial fatty-acid methyl-ester (FAME), having a lower heating value of 36,000 kJkg. Its physical-chemical parameters meet the requirements of the main EU standard concerning biodiesel fuel for automotive traction (EN 14214:2008; Table 3). The diesel oil (B0) used during the tests is a commercial fuel, having a lower heating value of 43,500 kJ/kg.

Table 2 - Test equipment used in this study.

Test equipment, manufacturer, model	**Technical specifications**	**Other specifications**
Hydraulic mobile test stand, M&W Gear (Gibson City, IL, USA), P-400M hydra-gauge dynamometer	• Full scale values: gauge pressure 14,000 kPa (140 bar) PTO shaft speed 1400 rpm • Resolution: 200kPa (2 bar) 10 rpm • Oil operative temperature: 140-180°F (60-82°C)	• Manually-operated through a hand-wheel acting on a valve which increases the counter pressure on avolumetric pump driven by the tractor PTO (operative fluid: oil) and, hence, the breaking load on the tractor • Equipped with an internal water-oil radiator for cooling (needs a temporary connection with the water mains) • Provided with a pressure-power (kPa-kW) calculator
Fuel consumption rate measurement	• Full scale: 20 000 g • Resolution: 1 g	• Diesel oil/biodiesel tank on a precision balance • Functioning principle: chrono-gravimetric

Table 3 - Main characteristics of the used biodiesel

Property	**Unit**	**Value**	**Requir.**	**Test method**	**Standard**
FAME content	%	98.0	≥96.5	EN 14103	EN 14214:2008
Density at 15°C	kg m^{-3}	882	860-900	EN ISO 3675 EN ISO 12185	EN 14214:2008
Kinematic viscosity at 40°C	mm^2 s^{-1}	4.5	3.5-5.0	EN ISO 3104	EN 14214:2008
Flash point	°C	107.0	≥101.0	EN ISO 2719 EN ISO 3679	EN 14214:2008
Pour point	°C	-14.0	0	ISO 3016	EN 14213:2003
Carbon residue (on 10% distillation residue)	%	<0.30	≤0.30	EN ISO 10370	EN 14214:2008
Cetane number	-	53	≥51	EN ISO 5165	EN 14214:2008
Iodine value	g/(100g)	118	≤120	EN 14111	EN 14214:2008

3 Results
3.1 Fitting the model to the engine fuelled with B0

The method of multiple regression was used to determine the values of C_0 through C_8 (Table 4) which give the best fit of equation (5) to the Brake Thermal Efficiency (*BTE*) data of engine fuelled with diesel oil (B0).
The *BTE* (%) data were obtained from the experimental triplets: speed ω (s^{-1}), torque T (Nm) and fuel consumption rate G (kg/h) through the brake specific fuel consumption *BSFC* (g/kWh):

$$BSFC = \frac{G \cdot 10^6}{T \cdot \omega} \tag{6}$$

And combining (6) with (1):

$$BTE = \frac{3.6 \cdot T \cdot \omega}{G \cdot H} \cdot 100 \tag{7}$$

Table 4 – Coefficients of fitting the model to the engine fuelled with B0.

C_0	C_1	C_2	C_3	C_4	C_5	C_6	C_7	C_8
10.806	-0.01677	$-4.350 \cdot 10^{-05}$	0.06115	$2.566 \cdot 10^{-03}$	$-5.69 \cdot 10^{-06}$	$1.734 \cdot 10^{-04}$	$-9.648 \cdot 10^{-06}$	$2.031 \cdot 10^{-08}$

Figure 2, shows the calculated brake thermal efficiency (*BTE*) contours for the engine fuelled with diesel oil (B0). The comparison between the calculated values and the measured values of *BTE* gives a mean relative error ± standard deviation $MRE \pm SD = 2.1 \pm 2.2\%$, representing a good accuracy.
Using equation (1) was also possible to produce the graph of *BSFC* contours (Figure 3), with the same mean relative error.

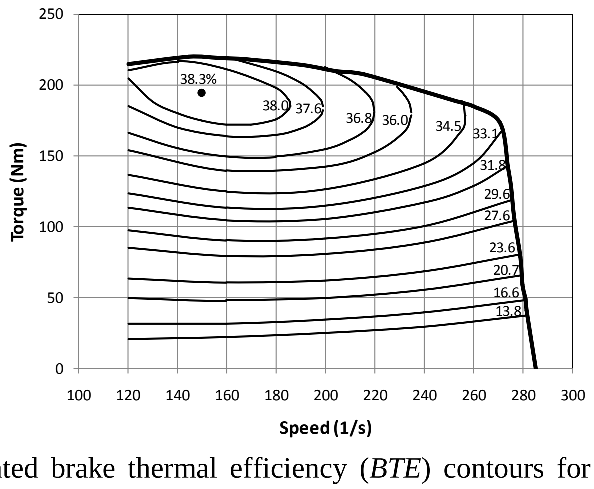

Figure 2 – Calculated brake thermal efficiency (*BTE*) contours for the diesel oil (B0) fueled engine.

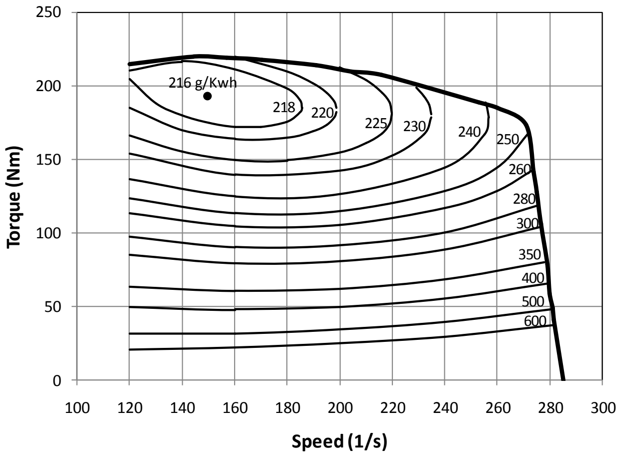

Figure 3 – Calculated brake specific fuel consumption (*BSFC*) contours for the diesel oil (B0) fueled engine.

3.2 Fitting the model to the engine fuelled with B100

Even for the engine fuelled with the B100, the method of multiple regression was applied to determine the values of C_0 through C_8 (Table 5) which give the best fit of equation (5) to the experimental data of brake thermal efficiency (*BTE*), obtained from equations (6) and (7).

Table 5 – Coefficients of fitting the model to the engine fuelled with B100.

C_0	C_1	C_2	C_3	C_4	C_5	C_6	C_7	C_8
31.333	-0.229	$4.359 \cdot 10^{-04}$	-0.1732	$5.352 \cdot 10^{-03}$	$-1.233 \cdot 10^{-05}$	$1.026 \cdot 10^{-03}$	$-2.068 \cdot 10^{-05}$	$4.743 \cdot 10^{-08}$

Figure 4, illustrates the calculated *BTE* contours for the engine fuelled with biodiesel (B100). The calculated values in comparison with those measured, provide a mean relative error ± standard deviation $MRE \pm SD = 2.0 \pm 1.5\%$, slightly better than that the diesel oil (B0) fuelled engine.

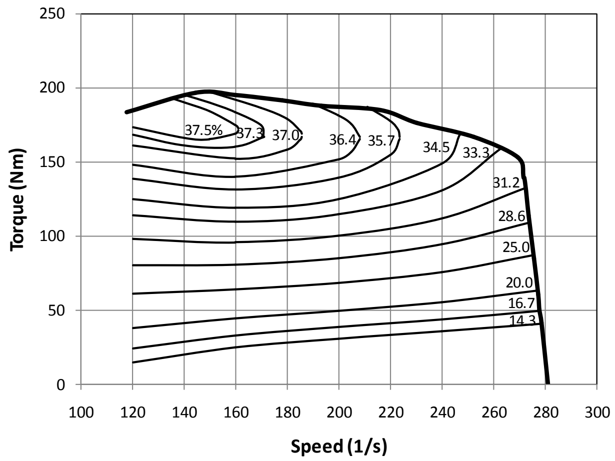

Figure 4 – Calculated brake thermal efficiency (*BTE*) contours for the biodiesel (B100) fueled engine.

The graph (Figure 5) of *BSFC* contours, with the same mean relative error was obtained using equation (1) similarly to B0, even for engine fueled with B100.

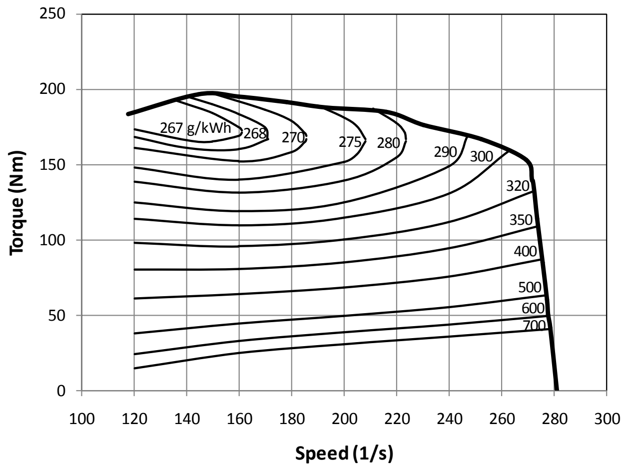

Figure 5 – Calculated brake specific fuel consumption (*BSFC*) contours for the biodiesel (B100) fueled engine.

3.3 Discussion

By comparing the engine power with diesel oil (B0) and biodiesel (B100), if the flow rate of fuel supplied by the injection pump remained the same, according to the equation (7) there would be a decline in the power equal to the lower heating value *H* (from 43,500 to 36,000 kJ/kg: -17.2%). This is true if we assume the same efficiency for the two fueling.

Instead, it was found experimentally a decrease of the maximum torque (from 220 to 197 Nm at engine speed of 144 s^{-1}: -10.5%) and therefore also a reduction of maximum engine power (from 48.1 to 42 kW: -12%). On average -11%, instead of -17.2%. This fact was explained because the fuel consumption rate was not the same when the engine is fuelled with the two different fuels.

In fact, it was also experimentally found an increase of the maximum flow rate of biodiesel (B100) compared to diesel oil (B0), in an attempt by the governor, to reach the prescribed maximum speed. This was reflected in an increased flow rate of B100, compared to B0, in the entire speed range. For example at a speed of 144 s^{-1}, at maximum torque, the B100 fuel consumption was G = 7.52 kg/h, against the 6.98 kg/h of B0. On average, the B100 flow rate resulted +6.5% higher than that of B0.

Therefore, the numerator of equation (7), which contains the engine power, is reduced by an average of -11%. The denominator is reduced by -17.2% due to the lower heating value H and increases of +6.5% due to the greater amount of fuel flow rate G. That's: $(G \cdot H)_{B100} = (1.065 \cdot G_{B0} \cdot 0.828 \cdot H_{B0}) = 0.882 (G \cdot H)_{B0}$ and then -11.8%.

By observing equation (7), this means that the average *BTE* remains almost the same when the engine is fuelled with the two different fuels.

In fact, looking at Figures 2 and 4 of the *BTE* contours, drawn through the mathematical model from the experimental data, this substantial equality of the two *BTE* was confirmed. The absolute difference between the two average values of *BTE* was not significant (30.3% vs. 30.5%).

On the one hand, the relative difference between the two *BTE* resulted of just 0.7%, lower than the mean relative error of the mathematical model (*MRE* about 2%), therefore not significant. On the other side, it was seen that this relative difference showed significant values depending on the operating point in the speed-torque plane.

Figure 6, illustrates the relative differences with positive sign, localized at low values of speed and torque. This means that B100 burns better at reduced loads and speeds, while the combustion of B100 tends to worsen with the speed increase. Similar results were obtained by [26].

This is, probably, the result of B100 reduced inclination to evaporate compared to diesel oil [14 and 15], which is critical when the speed increases and the time available for combustion is reduced.

The equation (6) showed an average *BSFC* of B100 compared to B0, influenced by an average increase of +6.5% of the fuel flow rate G in the numerator and by an average decrease of -11% of the engine power $T \cdot \omega$ in the denominator, namely:

$$(BSFC)_{B100} = \frac{1.065}{0.89} (BSFC)_{B0} = 1.197 (BSFC)_{B0}.$$

This is an increase of +19.7% similar to that shown by comparison of Figures 3 and 5. For example, the minimum *BSFC*, in the two cases, resulted in 216 and 267 g/kWh, with an increase of +19%.

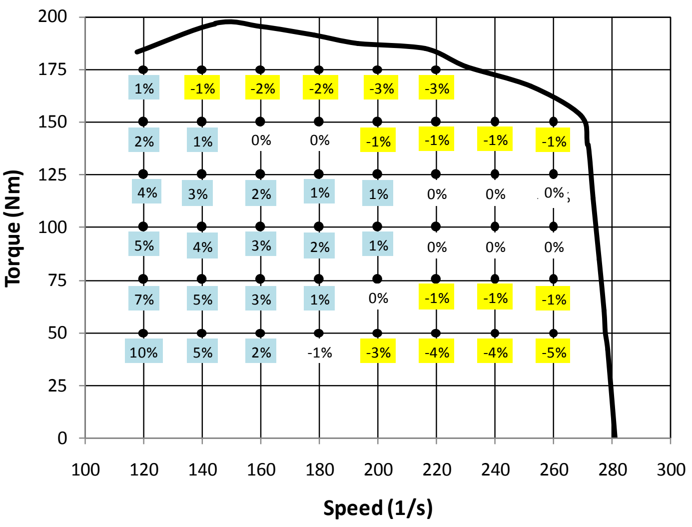

Figure 6 – Relative differences between the *BTE* of the two fuels in various parts of the speed-torque plane. The negative sign shows a worsening of the *BTE* of B100 fuelled engine.

4 Conclusions

To investigate the Brake Specific Fuel Consumption (*BSFC*) of diesel engine fuelled with diesel oil (B0) and then with biodiesel (B100), experimental tests on the engine of an agricultural tractor, were carried-out. This was done either vs. the engine speed and vs. engine load.

The use of biodiesel in a diesel engine causes an expectable decrease in the engine performance compared with the same system fuelled by diesel oil due to the differences in the molecular structure of these two fuels.

To understand the influence of these fuel differences represented by different heating values and the different evaporation rates and, hence, the brake thermal efficiency (*BTE*), a mathematical modeling of the *BTE* vs. engine speed and engine torque was developed.

From the experimental tests on the engine, using the equation (7), it was found that the average value of the *BTE* of the engine fueled with B100 was the same of the engine fueled with B0. Instead, the average value of *BSFC* of the engine fueled with B100 was +19% higher than that of the engine fueled with B0, consistent with the lower heating value of B100 (-17.2 %) . However, the engine power with the B100 was not reduced by the same amount, but with an average value of -11%, as there was a partial recovery due to the slight increase in fuel consumption rate (about 6.5%).

The mathematical modeling of the *BTE* of the engine produced an equation with nine coefficients to be determined for each engine and for each fuel. Thus, the fitting of the mathematical model to the engine, fuelled with the two fuels (B0 and

B100), allowed to draw the charts of the calculated *BTE* and *BSFC* contours for both fuels, with a good accuracy represented by a mean relative error of 2.1±2.2% (B0) and 2.0±1.5% (B100).
In conclusion, it was true that the average value of the *BTE* of the engine fueled with B100 was the same as that of the engine fueled with B0, but it was seen, through the comparison of *BTE* contours diagrams, that the *BTE* relative difference showed significant values depending on the operating point in the speed-torque plane.
It emerged that the engine fuelled with B100 had a *BTE* higher at low load and speed, but lower than that of B0 at high speed. This means that B100 burns better at reduced loads and speeds, while the combustion of B100 tends to worsen with the speed increase, due, probably, to reduced inclination to evaporate compared to diesel oil, which is critical when the speed increases and the time available for combustion is reduced.

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