The influence of the biofuel blends on the energetic and ecological performances of the Diesel engine

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Abstract. This study presents the influence of the diesel fuel blended with biodiesel fuel obtained from sunflower oil, corn oil and peanut oil on the energetic performances, combustion process and pollutant emissions. This research was done virtually and experimentally. In this study pure diesel fuel and two concentrations (6% and 10%) of blends with biofuels were used for experimentally tests on a Renault K9K diesel engine. Five parameters were observed during experimental tests: engine power, fuel consumption, cylinder pressure, and the amount of CO and NOx emissions. The same five parameters were simulated using AVL Boost program. The variations of effective power and maximal cylinder pressure are caused due to the lower calorific value of the tested fuels. Better oxidation of the biofuels induces a better combustion in cylinder and less CO and NOx emissions. The CO emissions are either influence by the lower carbon content of biofuels. The results of this study sustain that using 6% and 10% of blended biofuels with diesel fuel decrease the pollutant emissions of the diesel engine. Deviations between experimental and the simulation results confirm the validity of the mathematical model adopted for the simulation.

1. Introduction

Internal combustion engines will continue to dominate transport technology, using liquid fuels produced from fossil sources. Since the time of production of fossil fuels is millions of years, the reserves are limited. The only renewable carbon source that can be used as a substitute for fossil fuels is biomass. Biomass is formed from vegetation, bodies, municipal solid waste, waste derived from wastewater treatment, animal waste, manure, waste from agriculture, forestry and certain industrial waste. As opposed to fossil fuels, biomass is renewable, requiring a short time to replacing the raw materials consumed.

The usage of fuel increase annually due to the lower impact on the environment and their contribution to the reducing of greenhouse gases emissions. In Europe about 20% of total pollutant emissions are due the transport sector, which indicate a high potential to reduce those emissions [1]. Diesel engines are often used in road transport due to the higher efficiency compared to spark ignition engines and lower price of diesel. First generation of biofuels (bioethanol and biodiesel) have the highest use due to the low price and ease of use and because it can be used in current engines without significant changes. In Europe, consumption of biofuels is regulated by Directive 2009/28 /EC which provides a 10% share of energy from renewable sources in transport until 2020.

In this study pure diesel fuel and two concentrations (6% and 10%) of blends with biofuels were studied experimentally and numerically for the determination of engine power, fuel consumption,
cylinder pressure and CO and NOx emissions. Numerical simulation was made in AVL Boost program, using mixing controlled combustion model of burning.

2. Test description

Tests were performed on a Renault diesel engine using neat diesel fuel and blends of biodiesel produced from sunflower oil, corn oil and peanut oil. The blends was made in two different volume of biodiesel blended with diesel fuel. The volume ratios of biodiesel fuel were 6% for B6 and 10% for B10. The selected ratios of blends consider the current level and the proposed level of biodiesel use until 2020.

Some of the fuel properties are presented in table 1.

| Fuel       | Density at 20°C (kg/m³) | Viscosity at 20°C (mm²/s) | Cetane number | Flash point (°C) | Caloric value (MJ/kg) |
|------------|-------------------------|---------------------------|---------------|-----------------|----------------------|
| D          | 840.2                   | 5.34                      | 51.1          | 67              | 43.16                |
| B6SF       | 841.9                   | 5.27                      | 54.5          | 67.2            | 42.58                |
| B10SF      | 843.1                   | 5.10                      | 57.6          | 67.8            | 42.19                |
| B6C        | 841.7                   | 5.04                      | 57.6          | 71.4            | 42.63                |
| B10C       | 842.4                   | 4.99                      | 62.1          | 67.3            | 42.27                |
| B6P        | 842.1                   | 4.52                      | 57.8          | 70.6            | 42.27                |
| B10P       | 843.4                   | 5.94                      | 60.9          | 70.2            | 42.20                |

D – diesel fuel, B6SF – 6% biodiesel from sunflower oil, B10SF – 10% biodiesel from sunflower, B6C – 6% biodiesel from corn oil, B10C – 10% biodiesel from corn oil, B6P – 6% biodiesel form peanut oil, B10C – 10% biodiesel from peanut oil.

The experimental tests were made on a 4 cylinders Renault diesel engine. This engine is equipped with a common-rail injection system. The engine is mounted on a Horiba Titan 250 engine test bed. The engine test bed is equipped with an electric Dynas, LI250 dynamometer, which is designed for operated within a range of 0-8000 rotations per minute. It can measure engine power up to 250 kW with an accuracy of ±2%. Pressure in combustion chamber was measured with a Kistler 6005 piezoelectric pressure transducer. This pressure transducer can measure up to 1000 bar and it has an accuracy of ±0.8%.

For the measurement of CO and NOx emissions was used the HGA 400 Pierburg chemiluminescence analyser. This analyser can measure CO emission between 0 and 10% vol, with an accuracy of 0.01%; for the NOx emission the range is between 0 and 5000 ppm, with an accuracy of 1 ppm.

The engine tests were made for full throttle position for two different speeds: 2700 rpm (maximum torque speed) and 3700 rpm (maximum power speed).

The engine specifications are presented in table 2.

| Engine type            | Renault K9K four stroke |
|------------------------|-------------------------|
| Number of cylinders    | 4                       |
| Bore (mm)              | 76                      |
| Stroke (mm)            | 80.5                    |
| Total displacement (cm³)| 1451                    |
| Compression ratio      | 15.3                    |
| Fueling                | Common-rail direct injection |
3. Simulation model

The AVL Boost program was used to simulate the combustion process of the engine. It was used the mixing controlled combustion model developed by Chmela and Orthaber [2],[3] to determine the engine’s performances.

The model considers the effect of the premixed (PMC) and diffusion (MCC) controlled combustion processes according to:

\[
\frac{dQ_{total}}{d\alpha} = \frac{dQ_{MCC}}{d\alpha} + \frac{dQ_{PMC}}{d\alpha} \quad (1)
\]

The fuel injected after the start of combustion is burned after the MCC model. In this regime the heat release is a function of the fuel quantity available \((f_1)\) and the turbulent kinetic energy density \((f_2)\):

\[
\frac{dQ_{MCC}}{d\alpha} = C_{comb} \cdot f_1(m_F, Q_{MCC}) \cdot f_2(k, V) \quad (2)
\]

with

\[
f_1(m_F, Q_{MCC}) = \left( m_F - \frac{Q_{MCC}}{LCV} \right) \cdot \left( w_{O_{av}} \right) C_{EGR} \quad (3)
\]

\[
f_2(k, V) = C_R \cdot \left( \frac{k}{V} \right)^{1/2} \quad (4)
\]

where \(Q_{MCC}\) – cumulative heat release for the mixture controlled combustion \([kJ]\), \(C_{comb}\) – combustion constant \([kJ/kg^{\circ}CA]\), \(C_R\) – mixing rate constant \([s]\), \(k\) – local density of turbulent kinetic energy \([m^2/s^2]\), \(m_F\) – vaporized fuel mass (actual) \([kg]\), \(LCV\) – lower heating value \([kJ/kg]\), \(V\) – cylinder volume \([m^3]\), \(\alpha\) – crank angle \([^\circCA]\), \(w_{O_{av}}\) – mass fraction of available oxygen \([-\)], \(C_{EGR}\) – EGR influence constant \([-\]).

Since the distribution of squish and swirl to the kinetic energy are relatively small, only the kinetic energy input from the fuel spray is taken into account. The amount of kinetic energy imparted to the cylinder charge is determined by the injection rate. The dissipation is considered as proportional to the kinetic energy giving:

\[
\frac{dE_{kin}}{dt} = 0.5 \cdot C_{turb} \cdot m_F \cdot v_F^2 - C_{Diss} \cdot E_{kin}^{1.5} \quad (5)
\]

\[
k = \frac{E_{kin}}{m_F \cdot \left( 1 + \lambda_{Diff} m_{stoich} \right)} \quad (6)
\]

where

\(E_{kin}\) – kinetic jet energy \([J]\), \(C_{turb}\) – turbulent energy production constant \([-\)], \(C_{Diss}\) – dissipation constant \([1/s]\), \(m_F\) – injected fuel mass (actual) \([kg]\), \(v\) – injection velocity \([m/s]\), \(m_{stoich}\) – stoichiometric mass of fresh charge \([kg/kg]\), \(\lambda_{Diff}\) – Air Excess Ratio for diffusion burning.

Premixed combustion appears at the end of the ignition delay and fuel vapors ignite.

The ignition delay is calculated using the Andree and Pachernegg [4] model by solving the following differential equation:

\[
\frac{dI_d}{d\alpha} = \frac{T_{UB} - T_{ref}}{Q_{ref}} \quad (7)
\]
where \( I_{id} \) – ignition delay integral [-], \( T_{\text{ref}} \) – reference temperature = 505.0 [K], \( T_{\text{ub}} \) – unburned zone temperature [K], \( Q_{\text{ref}} \) – reference activation energy [K].

The main source of nitrogen oxides is the thermal mechanism of forming oxides in the flue gas, the high temperature (> 1800K) being an element that influences the formation of nitrogen oxides.

The NOx formation is based on Pattas and Häfner [5].

The final rate of NO production/destruction in [mole/cm³s] is calculated as:

\[
r_{\text{NO}} = C_{KM} \cdot 2 \cdot \left(1 - \alpha_{\text{NO}}^2\right) \cdot \frac{r_{\text{1,NO}}}{1 + \alpha_{\text{NO}} R_2} - \frac{r_{\text{4,NO}}}{1 + R_4}
\]

where \( C_{KM} \) – NOx formation kinetic multiplayer, \( \alpha_{\text{NO}} \) – the ratio between the actual (calculated) and the equilibrium NO molar mass, \( r_{\text{1,NO}} \) – the reaction rates of the Zeldovich mechanism.

The CO formation model is based on Onorati et al. [6].

The final rate of CO production/destruction in [mole/cm³s] is calculated as:

\[
r_{\text{CO}} = C_{\text{const}} \cdot \left(\alpha_{\text{CO}} + r_{\text{2,CO}}\right) \cdot (1 - \alpha_{\text{CO}})
\]

where \( C_{\text{const}} \) – CO formation kinetic multiplayer, \( r_{\text{1,CO}} \) – the reaction rates, \( \alpha_{\text{CO}} \) – the ratio between the actual (calculated) and CO molar masses.

The equilibrium NO and CO molar masses of each tested fuel were automatically determined from the program’s data base. The program determines the equilibrium molar masses based on fuel properties, air/fuel ratio, in-cylinder pressure and temperature.

4. Results

Figure 1 shows the experimentally and numerically results of engine’s power. The results presented in figure 1 indicate that the highest experimentally obtained power was for B6 from peanut oil. In the case of simulation the highest power was obtained also for B6 peanut too. In all cases of experimental tests, the power of engine fuelled with biofuel blends was higher than for the engine fuelled with neat diesel. The lower caloric value of biodiesel blends are compensated by the higher density, that influences on the greater mass of injected fuel.

The values obtained by simulation are higher than the experimental result except for diesel fuel, B6 sunflower and B10 sunflower where the experimental results are higher than the numerical ones. The maximum difference between experimental and numerical results of engine’s power is 4.1%.

Figure 2 shows the numerical and experimental results of maximum cylinder pressure.

The maximum pressures were obtained for B6 sunflower both for numerically and experimentally tests. The maximal difference between numerical and experimental results of cylinder pressure is 4.6%.

The chemical composition of fuel and the engine speed influence the amount of CO and NOx emissions, as presented in figure 3 and figure 4.

The results indicate that when engine speed increases the amount of CO emissions is decreased and the amount of NOx is increased.

For all biodiesel blends the amount of experimental NOx emissions are lower than for diesel fuel. The lowest concentration of NOx was obtaining for B6 corn at 2700 rpm. The numerical results are lower than experimental ones. The difference between experimental and numerical results was 2%.

The CO emissions are lower at higher speed. The highest amount of CO is obtained for diesel fuel at 2700 rpm.
Figure 1. Numerical and experimental engine power.

Figure 2. Numerical and experimental maximum pressure.
**Figure 3.** Numerical and experimental CO emissions.

**Figure 4.** Numerical and experimental NOx emissions.
5. Conclusions
The paper studied the influence of diesel fuel and six blends of diesel with biodiesel fuel on diesel engine performances and emission formation. The tests were made on a Renault diesel engine and computer simulation were done using the AVL Boost program.

The following conclusions can be made:
1. Although the calorific value of mixture is lower than that of the diesel, due to higher density is injected into the cylinder a greater mass of mixtures.
2. The greater amount of fuel mass injected into cylinder had led to increase the engine power.
3. The chemical composition of biodiesel influences the NOx and CO emissions. Higher oxygen content and higher cetane number contribute to the combustion process, producing a more complete combustion, which reduces the formation of CO.
4. Because the deviations between simulation results and experimental results are small these mathematical models can be used for studying the effects of other fuels without conducting experimental tests, with results close to the real ones.

6. References
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