Simulation study on the effect of the ratio of biodiesel on performance and emissions of diesel engine

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Abstract

This paper presents a study on the effect of the ratio of biodiesel on performance and emissions of diesel engines. Researching engine is a cylinder engine AVL-5402 is simulated by software AVL-Boost. Simulated fuel include fossil diesel and biodiesel blended with the replacement rate from 0% to 50%, simulation mode in 1400 (rev/min) at the rate of 25%, 50% and 75% load. Parameters to be evaluated as power, fuel consumption and emissions based on the proportions of blended biodiesel. The results show that there is a relationship between the proportion of blended biodiesel and the parameters of the engine. Specifically, the ratio of biodiesel blend increases, the tendency reduces engine power, increase fuel consumption. The emissions of CO and soot reduces, while NOx increases

Keywords: engine simulation, biodiesel, emission, mixing ratio

1. INTRODUCTION

Today, with economic growth at the same time with the increasing demand for energy. Compression Ignition engines have become an indispensable part of modern life style because of their role in transportation and mechanized agriculture sector. The dwindling sources of conventional fossil fuels, their ever increasing demand and prices have prompted the scientists and researchers to find alternate fuels for diesel engines. A number of alternative fuels such as ethanol, methanol, hydrogen, Compressed Natural Gas (CNG), liquefied Natural Gas (LNG), Liquefied Petroleum Gas (LPG), Dimethyl-ether (DME) and vegetable oils have been used as alternative fuels, however biodiesel has received a considerable attention to be used as a substitute fuel for conventional petroleum.

Biodiesel has already been commercialized in the transport sector and can be used in diesel engines with little or no modification [1]. Biodiesel and its blends with conventional diesel are environment friendly and their use in diesel engine results in reduced exhaust pollutants as compared to conventional diesel fuel [2].

Some experimental investigations were cunducted on diesel engine to clarify how biodiesel affect the engine performance and exhaust emisssions [3–5]. Most of the results showed that emissions when fueled biodiesel reduced significantly, however, Nox emissions increase.

2. MODEL DESCRIPTION

2.1. Combustion model

Models with controlled combustion mixture (MCC) is used to build properties in diesel combustion. The model can be calculated by two processes: premixed combustion and mixing controlled combustion processes:

$$\frac{dQ_{toltal}}{d\alpha} = \frac{dQ_{MCC}}{d\alpha} + \frac{dQ_{PMC}}{d\alpha} \qquad (1)$$

With: Q_{total}: Total heat release over the combustion process [kJ].

Q_{PMC}: Total fuel heat input for the premixed combustion [kJ]

Q_{MCC}: Cumulative heat release for the mixture controlled combustion [kJ]

- Ignition delay model:

The ignition delay is calculated using the Andree and Pachernegg [7] model by solving the following differential equation:

$$\frac{dI_{id}}{d\alpha} = \frac{T_{UB} - T_{ref}}{Q_{ref}} \qquad (2)$$

As soon as the ignition delay integral I_{id} reaches a value of 1.0 (= at α_{id}) at the ignition delay τ_{id} is calculated from: $\tau_{id} = \alpha_{id} - \alpha_{SOI}$

With: I_{id} : ignition delay integral [-]

T_{ref}: reference temperature = 505.0 [K]

T_{UB}: unburned zone temperature [K]

Q_{ref}: reference activation energy, f(droplet, diameter, oxygen content,…) [K]

τ_{id}: ignition delay

α_{SOI}: start of injection timing [degCA]

α_{id}: ignition delay timing [degCA]

- Premixed combustion model:

A Vibe function is used to describe the actual heat release due to the premixed combustion:

$$\frac{\frac{dQ_{PMC}}{Q_{PMC}}}{d\alpha} = \frac{a}{\Delta\alpha_c}.(m+1).y^m.e^{-a.y^{(m+1)}} \qquad (3)$$

$$y = \frac{\alpha - \alpha_{id}}{\Delta\alpha_c}$$

With: Q_{PMC}: total fuel heat input for the premixed combustion = $m_{fuel,id}$. C_{PMC}

Where: $m_{fuel,id}$: total amount of fuel injected during the ignition delay phase

CPMC: premixed combustion parameter

$\Delta\alpha_c$: premixed combustion duration = □$_{id}$.$C_{PMC\text{-}Dur}$

where: CPMC-Dur : premixed combustion duration factor

m: shape parameter m = 2.0

a: Vibe parameter a = 6.9

- Mixing Controlled Combustion process:

In this regime the heat release is a function of the fuel quantity available (f_1) and the turbulent kinetic energy density (f_2):

$$\frac{dQ_{MCC}}{d\alpha} = C_{Comb} \cdot f_1(m_F, Q_{MCC}) \cdot f_2(k, V)$$

with:

$$f_1(m_F, Q_{MCC}) = \left(m_F - \frac{Q_{MCC}}{LVC}\right)\left(w_{Oxygen,available}\right)^{C_{EGR}}$$

$$f_2(k, V) = C_{Rate} \cdot \frac{\sqrt{k}}{\sqrt[3]{V}}$$

C_{Comb}: combustion constant [kJ/kg/degCA]

C_{Rate}: mixing rate constant [s]

m_F: vaporized fuel mass (actual) [kg]

LVC: lower heating value[kJ/kg]

V: cylinder volume [m^3]

α: crank angle [deg CA]

$w_{Oxygen,available}$: mass fraction of available Oxygen (aspirated and in EGR) at SOI [-]

$C_{EGR\ EGR}$ influent constant [-]

k: local density of turbulent kinetic energy [m^2/s^2]

$$k = \frac{E_{kin}}{m_{F,I}(1 + \lambda_{Diff}.m_{stoich})}$$

where: E_{kin} : kinetic jet energy [J]

C_{turb}: turbulent energy production constant [-]

$m_{F,I}$: injection fuel mass (actual) [kg]

λ_{Diff}: Air Excess Ratio for diffusion burning [-] mstoich :stoichiometric mass of fresh charge [kg/kg]

2.2. Heat transfer model

The heat transfer to the walls of the combustion chamber, i.e. the cylinder head, the piston, and the cylinder liner, is calculated from equation [6]

$$Q_{wi} = A_i.\alpha_w.(T_c - T_{wi}) \tag{4}$$

With: *Qwi* - wall heat flow

A_i – surface area

α_w - heat transfer coefficient

T_c - gas temperature in the cylinder

T_{wi} - wall temperature.

Heat transfer coefficient (α_w) is usually calculated by WOSCHNI Model, The Woschni model published in 1978 for the high pressure cycle is summarized as follows [8]:

$$\alpha_W = 130.D^{-0,2}.p_c^{0,8}.T_c^{-0,53}.\left[C_1.c_m + C_2.\frac{V_D.T_{c,1}}{p_{c,1}.V_{c,1}}.(p_c - p_{c,0})\right]^{0,8}$$

Where: $C_1 = 2,28 + 0,308 .c_u/c_m$

$C_2 = 0,00324$ for DI engines

D - cylinder bore

cm - mean piston speed

c_u - circumferential velocity, $c_u = \pi.D.nd/60$

VD - displacement per cylinder

$p_{c,o}$ - cylinder pressure of the motored engine (bar)

$T_{c,1}$ - temperature in the cylinder at intake valve closing (IVC)

pc,1 - pressure in the cylinder at IVC (bar)

2.3. Emission model

2.3.1. NOx formation model

NOx formed from the oxidation reaction of nitrogen in high-temperature conditions of combustion. 6 reactions introduced in Table 1, which are based on the well known Zeldovich mechanism are taken into account.

Table 1. NOx formation reactions

	Stoichio metry	Rate : $k_i = k_{0,i}.T^a.e^{\left(\frac{-TA_i}{T}\right)}$
R_1	N2 + O = NO + N	r1 = k1.CN2.CO
R_2	O2 + N = NO + O	r2 = k2.CO2.CN
R_3	N + OH = NO + H	r3 = k3.COH.CN
R_4	N2O + O = NO + NO	r4 = k4.CN2O.CO
R_5	O2 + N2 = N2O + O	r5 = k5.CO2.CN2
R_6	OH + N2 = N2O + H	r6 = k6.COH.CN2

All reactions rates ri have units [mole/cm^{3}s] the concentrations c_i are molar concentrations under equilibrium conditions with units [mole/cm^3]. The concentration of N_2O is calculated according to:

$$\frac{N_2O}{N_2\sqrt{O_2}} = 1.1802.10^{-6}T_1^{0.6125}\exp\left[\frac{-18.71}{RT}\right] \tag{5}$$

NO formation rate is calculated as follows:

$$\frac{d[NO]}{dt} = 2(1-\alpha^2)\left[\frac{R_{1e}}{1+\alpha K_2} + \frac{R_{4e}}{1+K_4}\right]\frac{p}{RT} \quad (6)$$

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

$$r_{NO} = C_{Post\,ProcMult}.C_{kineticMult}.2,0.(1-\alpha_2)\frac{r_1}{1+\alpha.AK_2}\frac{r_4}{1+AK_4}$$

with: $\alpha = \frac{C_{NO,act}}{C_{NO,equ}}.\frac{1}{C_{Post\,ProMult}}$; $AK_2 = \frac{r_1}{r_2+r_3}$; $AK_4 = \frac{r_4}{r_5+r_6}$

2.3.2. CO formation model

CO formation following two reactions given in Table 2 are taken into account:

Table 2: CO formation reactions

	Stoichio metry	Rate :
R_1	CO + OH = CO2 + H	$r_1 = 6.76.10^{10}.e^{\left(\frac{T}{1102.0}\right)}.c_{CO}.c_{OH}$
R_2	CO2 + O = CO + O2	$r_2 = 2.51.10^{12}.e^{\left(\frac{-24055.0}{T}\right)}.c_{CO}.c_{O_2}$

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

$r_{CO} = C_{const}.(r_1 + r_2).(1-\alpha)$ (7)

with: $\alpha = C_{CO,\,act}/C_{CO,\,equ}$.

2.3.3. Soot formation model

Soot formation is described by two steps including formation andoxidation. The net rate of change in soot mass msoot is the difference between the rates of soot formed ms.f and oxidized ms.ox.

$$\frac{dm_s}{dt} = \frac{dm_{s,f}}{dt} - \frac{dm_{s,ox}}{dt}$$

with: $\frac{dm_{s,f}}{dt} = A_f.m_{f,v}.p^{0,5}\exp\left[\frac{-E_{s,f}}{RT}\right]$ soot formation rate

$\frac{dm_{s,ox}}{dt} = A_{ox}.m_s\frac{P_{O2}}{P}.p^{1,8}\exp\left[\frac{-E_{s,ox}}{RT}\right]$ oxidation rates

m_s: soot mass

$m_{f,v}$: fuel evaporation volume

P_{O2}: Pressure of O_2 molecules

$E_{s,f}$ = 52,335 kJ/kmol: activation energy

$E_{s,ox}$ = 58,615 kJ/kmol: oxidation energy

A_f, A_{ox}: the constant empiric selection and specific engine types.

2.4. Fuel model

First it is necessary to define fuel B100, B100 fuel is fuel 100% pure biodiesel include the chemical compound with the ratio by volume and is presented in Table 3.

Table 3. Chemical composition of fuel B100

B10, B20, B30, B40 and B50 have a percentage of volume, respectively 10%, 20%, 30%, 40% and 50% of B100.

Chemical compound	Ratio (% volume)
$C_{15}H_{30}O_2$	0,0107
$C_{17}H_{34}O_2$	0,146
$C_{19}H_{38}O_2$	0,0655
$C_{19}H_{36}O_2$	0,399
$C_{19}H_{34}O_2$	0,376
$C_{19}H_{32}O_2$	0,0028

2.5. Modeling diesel engine AVL 5402

AVL 5402 Engine is a single cylinder, four-stroke, common rail diesel engine. The engine specification is shown in Table 4, the engine is modeled by AVL Boost software (Figure 1)

Table 4. Specifications of the engine

N^o	Parameter	Value
1	Cylinder diameter (D)	85 mm
2	Stroke (S)	90 mm
3	Displacement volume	510,7 cm^3
4	Compression ratio	17 : 1
5	Rate power/speed	9/3200 kW/rpm

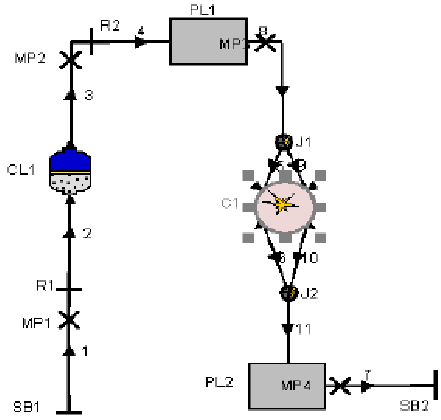

Fig.1 Diesel engine AVL 5402 model

3. RESULTS AND DISCUSSION

3.1. Model validation

To evaluate the model, conducted a comparison between simulation results and experimental. Figure 2 presents the results of comparisons of power (Fig 2a) and fuel consumption (Fig 2b) between simulation and experimental fuels B0, B10, B20, B30 to keep the fuel supply corresponding to 75% load.

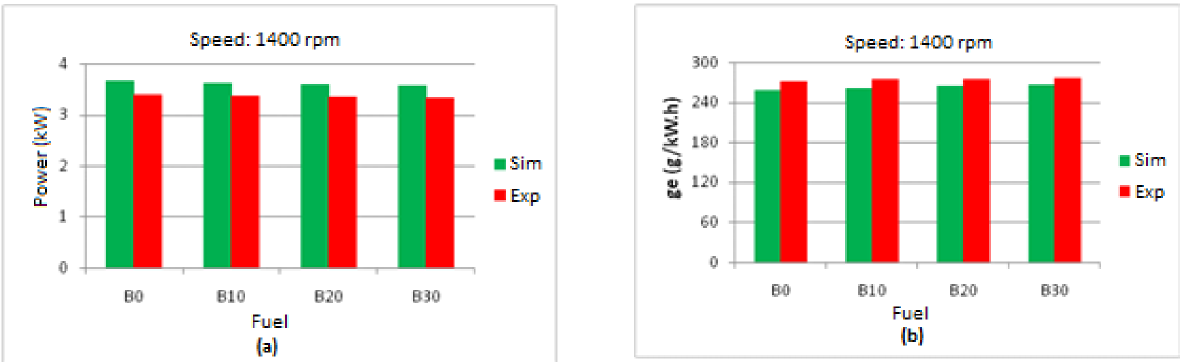

Fig.2. Comparisons between simulation results and experimental

Results showed that the power as well as fuel consumption between simulation and experiment matched quite well: the difference in power and fuel consumption was about 3.7% and 2.3%, respectively. Thus, it is possible to use this model to simulate the engine with biodiesel fuel.

3.2. Combustion characteristics

Fig 3 compares the pressure in engine cylinder when using 6 kinds of fuel: B0, B10, B20, B30, B40 and B50. The results show that increasing proportion of blended biodiesel, the time of increasing pressure correspond the time start of rapid fire soon appear gradually, beside the speed of increasing pressure was decreased. Biodiesel fuel has a higher Cetane rating value help mixed catching fire easily, results of the fire starting time is earlier.

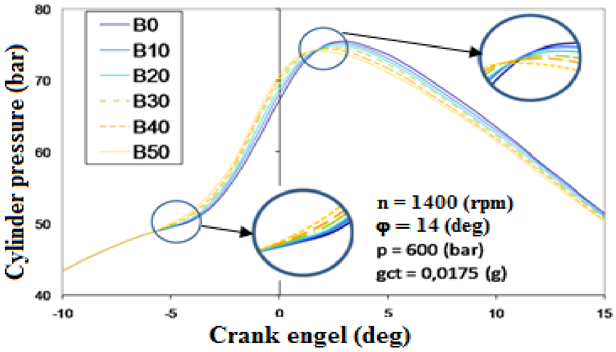

Fig 3. Evolution of cylinder pressure of fuel B0, B10, B20, B30, B40 and B50.

Specific parameters of combustion process shown in table 5.

Table 5. Comparisons parameters of combustion process 6 type of fule

Combustion parameters	**B0**	**B10**	**B20**	**B30**	**B40**	**B50**
Cylinder pressure max (MPa)	75,46	75,25	74,91	74,52	74,31	74,08
Pressure angle max after TDC (^{0}TK)	3,29	3,02	2,80	2,67	2,23	2,01
Speed of increasing pressure max (MPa/^{0}TK)	5,68	5,67	5,60	5,58	5,53	5,48
Combustion starting angle before TDC (^{0}TK)	5,20	5,28	5,35	5,48	5,50	5,58
Calorific speeding max (J/^{0}TK)	167,5	165,1	163,2	159,6	157,5	156,3
Calorific angle max before TDC (^{0}TK)	0,5	0,6	0,7	0,85	1,0	1,1

Evolution of calorific speeding is shown in figure 4. Amount of fuel supply to a cycle is the same for all fuel, beside thermal power of biological fuel is lower than diesel fuel, so calorific speeding of biological diesel fuel is lower.

Rise time of calorific speeding for biodiesel fuel will be earlier. It was explained by Cetane rating value of biodiesel fuel is higher

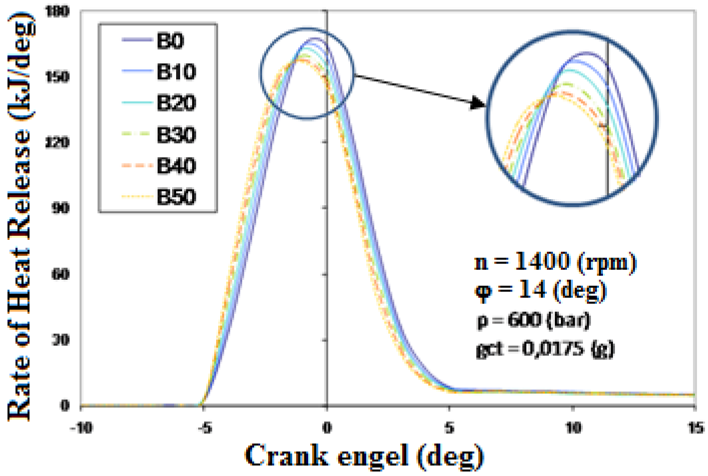

Fig 4. Calorific speeding of types of fuel

3.3. Engine performance

The power of engine is lower than using diesel fuel (B0) and decreasing when mix rate of biodiesel increase. The average power in different load decrease: 1,06%;1,81%; 2,74%; 3,81%; 4,79% with respective fuel B10, B20, B30, B40, B50. With the amount of fuel supply to a cycle is the same for al fuels, so power is reduced by thermal value of biodiesel fuel is lower. Beside, cause of time fire delay reduces so have phenomenon of double combustion and compression for using biodiesel fuel, the results also make reduce power.

While fuel consumption rate increases with increasing biodiesel fuel mixed rate. The average increasing rate: 1,32%; 2,02%; 2,98%; 4,08%; 5,55% with respective fuel B10, B20, B30, B40, B50. Because the amount of fuel supply to constant cycle for fuel and reduce engine power to increasing fuel consumption rate of engine.

The trend of power change and fuel consumption rate is shown figure 5 and figure 6.

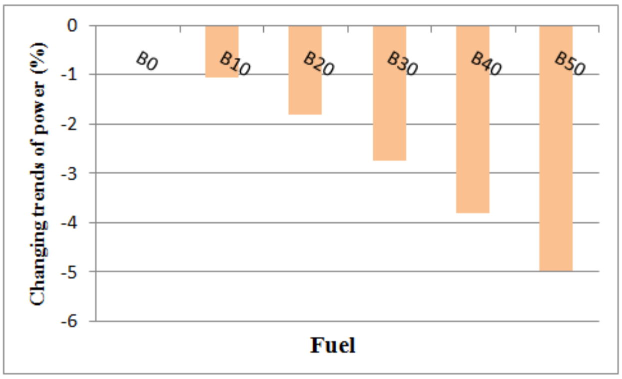

Fig 5. The trend of power change.

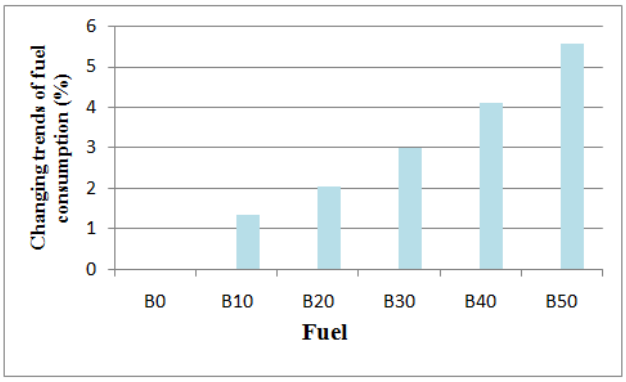

Fig 6. The trend of fuel consumption rate change.

3.4. Exhaust Emission

CO emissions is the burning in low oxygen. When the engine use biodiesel fuel, cause of biodiesel fuel have O_2 molecules, leading to decrease area rich-mixture and the results decrease CO emissions. CO emissions reduce while biodiesel mixed rate decrease. On average for the load mode, then turn reducing: 4,70%; 8,63%; 14,00%; 18,90%, 26,37% with respectively B10, B20, B30, B40, B50.

For NO_x emissions, when we increase biodiesel mixed rate with respective emissions NO_x also increase. This change cause of residual air coefficient of biodiesel fuel is higher so make advantage for NO_x formation process, beside mixed biodiesel fuel is fire fast to make chamber temperature higher. Average NO_x increase: 2,30%; 3,80%; 5,37%; 8,07%; 10,53% with respectively B10, B20, B30, B40, B50.

Soot is a special pollutant in diesel engine exhaust. Diffusion combustion in diesel engines are are very favorable for the formation of soot. However, with engine using biodiesel fuel, is has reduced emissions of soot cause of fuel have oxygen elements enables soot oxidation process more thoroughly. Results showed that, average soot for regimes load in reducing: 6,30%; 12,17%; 18,60%; 24,13%; 30,03% with respectively B10, B20, B30, B40, B50.

Trends change the toxic emissions of the engine when using B10, B20, B30, B40, B50 compared to B0, It was shown in figure 7.

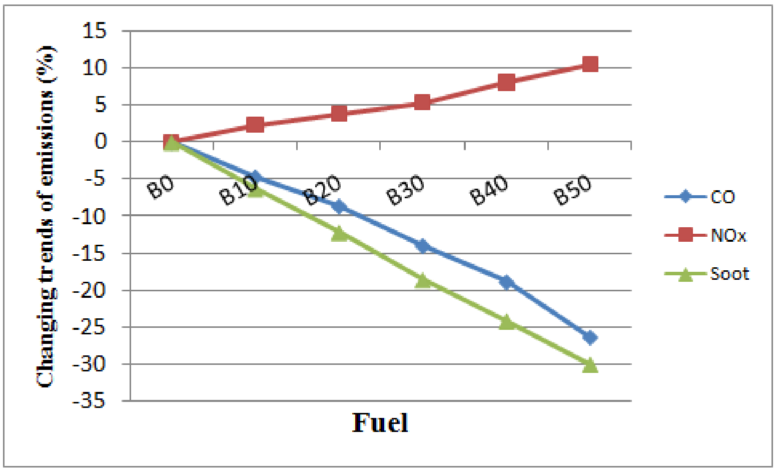

Fig 7. Trends change of engine emissions.

4. CONCLUCSIONS

- Time of starting fire is earlier when we increase mixed biodiesel rate. Because biodiesel fuel has a higher Cetane rating of mixture ignite better.

- Power of engine tends to decrease while fuel consumption increase that using biodiesel. Power reduce because thermal value of biodiesel fuel is lower.

- CO emissions and soot are decreased while NO_x increase mixed biodiesel rate. Cause by oxygen ingredients help mixture fire better.

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