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COMPARISON AND OPTIMIZATION OF COMBUSTION PERFORMANCE AND EMISSIONS OF A SINGLE-CYLINDER DIESEL ENGINE FUELED WITH SOY BIODIESEL-DIESEL BLENDS

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ABSTRACT

This paper investigates numerically the performance and emission characteristics of Soy Methyl Esther (SME) in a diesel engine. A single-cylinder, four strokes, naturally aspirated and direct injection compression ignition engine was fueled by a mixture of SME and pure diesel fuel, forming 4 blends: pure Diesel (D100), 20%SME+80%, diesel(B20), 40%SME+60%diesel(B40) and 100%SME(B100). The computation was made via the full-cycle engine simulation software Diesel-KR. The model set was first validated against experimental results from literature with pure Diesel. Performance and emission characteristics at different loads and constant speed were then compared to conclude which percentage is more suitable for the engine, in term of efficiency and pollution emissions. In this study, it is found that B20 offers similar results in term of consumption and pollutants emission. Thus, the engine could run with the B20 blend, giving similar performance than pure diesel. The investigation goes on to see the effect of an engine modification, namely retarding the start of injection on the engine performance when the B20 is used. It was found that retarding the start of injection leads to lower fuel consumption, ignition delay and combustion duration, which improves the thermal efficiency of the engine. For the NOx emissions, the retarded start of injection lowers the NOx emissions, yet increases the PM emissions by as much as 62%.

Keywords: biodiesel, diesel engine, thermal efficiency, start of injection, NOx, PM emissions.

INTRODUCTION

The need for alternative energy sources is getting more consideration among researchers and industrials. The increasing stringent government's regulations to maintain a certain quality of air as well as the crisis of fossil fuels have pushed to investigate and experiment new kinds of fuels as surrogate for the conventional fuel. Several sources have shown their capability to insure the close performances while meeting the environmental and governmental restrictions. Among these alternative fuels, blended alcohols [1, 2] and Biodiesel [3, 4, 5, 6] with diesel emerge as promising sources that could reduce the use of mineral fuels. The interest to biodiesel came from different points: first it is an environmentally friendly alternative fuel. It can be derived from renewable sources: vegetable oils, animal fats and even waste restaurant greases [7]. It also has similar properties to mineral diesel, which makes it suitable with only minor changes on the engine.

Several studies have reported the potential of biodiesel as a serious option of energy source while highlighting its drawbacks. Aydin and Bayindir [8] studied the effect of cottonseed oil methyl ester (CSOME) blended at 5%, 20%, 50% and 75% and 100% with Diesel in a single cylinder, direct injection, air cooled diesel engine at different speeds and full load. They observed that the utilization of 5% mixture makes torque higher at medium and higher speeds. The medium blends showed no difference in performance compared to diesel. Also, the higher the biodiesel percentage, the lower the exhaust emissions are. They concluded that blends with small amounts of CSOME can be substituted for diesel without any modification in the engine.

Rakopoulos *et al.* [9] studied the performance of a bus engine using blends of diesel fuel with bio-diesel (10% and 20%) of sunflower in a six-cylinder, turbocharged and after-cooled, direct injection, mini-bus diesel engine at two speeds and three loads. Their results showed that all biodiesel blends, smoke density and CO emissions were reduced, NOx and HC emissions increased while engine performance was similar. They concluded that all tested blends can be safely used and advantageously in small blending ratio.

Habibullah *et al.* [10] studied the performance of mixtures of two biodiesels, namely Palm and coconut in single-cylinder, four-stroke, and direct injection diesel engine at full load and with different speeds. The results showed that all biodiesel-diesel mixtures yield lower torque and higher BSFC. Except for NOx, the emissions are reduced. The mixture with 15% Palm biodiesel (PB), 15% coconut biodiesel (CB) and 70% diesel (PB15CB15) leads to the best brake torque while lowering BSFC and NOx emissions compared to the mixture with 30% CB and 70% diesel. It also gives lower CO and HC emissions and at the same time increases BTE. They concluded that the use of a different blend of palm and coconut oil leads to better performance and emission than coconut and palm biodiesel blends.

While biodiesel is already used in many combustion engines, especially common transportation means, its understanding in terms of the effects on the environment and also engine durability still is in progress. In fact, the complex composition of biodiesel and the difference between its physical and chemical properties and those of conventional Diesel make its direct usage almost unpredictable, therefore often damaging for the engine and more polluting the environment. Experimental work allows accurate understanding of the combustion process of biodiesel in different modes. With its available properties and the results of the test bench, one can state the circumstances under which this biofuel is competing with conventional Diesel. While experience gives actual results, its economic cost is also an issue since it requires too many tests to represent all the operating modes. That's where simulation turns out to be an efficient tool for the prediction of performance and emissions of engines. The thermodynamic models, unlike CFD models, enable to get satisfactory results in a minimum amount of time. The accuracy of the results depends hugely on the number of zones considered and the correlations used to model the phenomena in action.

SCOPE AND NUMERICAL MODEL

The point of this paper is to numerically examine the impact of a specific biodiesel, namely Soy Methyl Ester (SME), on the performance and combustion characteristics of a compression ignition engine, under different loads and constant speed. A further study on the impact of retarding the start of injection on the same performance will then take place. A single cylinder, naturally aspirated, air cooled, direct injection diesel engine is used. The numerical model should behave in the same way as the real engine in terms of the parameters available on the output of the engine. The numerical simulation is conducted with the thermodynamic code *Diesel-RK*. The computation of thermodynamic properties is processed using the first law of thermodynamics. Properties such as Pressure, temperature are computed for each crank angle. It takes advantages of empirical and semi empirical correlations which were obtained from experimental tests in the literature to compute its coefficients. Moreover, *Diesel-RK* uses the R-K model[12-14], which is a multi-zone diesel fuel spray mixture formation and combustion model.

The software uses the following governing equations [14, 15]:

- Mass conservation equation:

$$\dot{m}_{cyl} = \dot{m}_{in} + \dot{m}_{exh} + \dot{m}_{bb} + \dot{m}_{fuel} \quad (1)$$

- Species conservation equation for the species *j*:

$$\dot{Y}_j = \sum_j \left(\frac{\dot{m}_j}{m_{cyl}}\right)\left(Y_i^j - Y_i^{cyl}\right) + \dot{\omega}_j \quad (2)$$

- Energy conservation equation:

$$\dot{U}_{cyl} = \dot{Q}_w + \dot{Q}_{chem} - P_{cyl}\dot{V}_{cyl} + h_{in}\dot{m}_{in} + h_{exh}\dot{m}_{exh} \quad (3)$$

Where: m_{cyl} and $\dot{m}_{cyl}$ are the mass and the mass flow rate of the cylinder gas respectively, $\dot{m}_{in}$ is the inlet air mass flow rate, $\dot{m}_{exh}$ is the exhaust mass flow rate, $\dot{m}_{bb}$ is the blow-by rate, $\dot{m}_{fuel}$ is the fuel mass flow injected in the cylinder during the cycle, $\dot{m}_j$ is the time derivative of the mass of species j, $\dot{Y}_j$ is the time derivative of the mass fraction of the species j, Y_i^j and Y_i^{cyl} are the stoichiometric coefficients for products and reactants respectively, $\dot{\omega}_j$ is the source term, $\dot{U}_{cyl}$ is the time derivative of the internal energy, $\dot{Q}_w$ is the heat transferred from the cylinder to the wall, $\dot{Q}_{chem}$ is the heat released by combustion, P_{cyl} is the cylinder pressure, $\dot{V}_{cyl}$ is the time derivative of the cylinder volume, h_{in} and h_{exh} are the enthalpies of the inlet and exhaust gas respectively.

Models for combustion and emissions calculation are:

NO_x formation modelling

Since NO is the dominating nitrogen oxide in the exhaust gas, the software computes it as the only NO_x. The version used in the software uses Zeldovich mechanism to compute NO formation. Reactions of the Nitrogen oxidation are:

$O_2 \leftrightarrow 2O$
$N_2 + O \leftrightarrow NO + O$
$N + O_2 \leftrightarrow NO + N$

The concentration of atomic nitrogen controls the rate of the last 2 equations. Volume concentration of NO in the combustion products formed is depicted for each calculation step with the following equation [16,17]:

$$\frac{d[NO]}{d\theta} = \frac{2{,}33*10^7*p*[N_2]_e[O]_e\left\{1-\left({[NO]}/{[NO]_e}\right)^2\right\}e^{-\frac{38020}{T_z}}}{RT_z\left(1+\frac{2365}{T_z}*e^{\frac{3365}{T_z}}{[NO]}/{[O_2]_e}\right)*\omega} \quad (4)$$

Where: p is the cylinder pressure, T_z is the Temperature in the burnt gas zone (K), ω is the engine speed (rev/min).

Particulate matter

PM emission consists of list of species and soot has the dominant fraction. The formula presented by Alkidas [18] predicts the level of particulate matter emission from obtained from the Bosch smoke number. The equation is:

$$PM = A_{PM}565\left[ln\left(\frac{10}{10-Bosch}\right)\right] \text{ (mg/m}^3\text{)} \quad (5)$$

Where: A_{PM} is a a correction coefficient (By default $A_{PM}= 1$) and $Bosch$ is the Bosch number.

Heat release rate

For the calculation of the heat release rate, Diesel R-K divides the combustion process into 4 separate stages, each stage having its own physical and chemical features. The first one is the auto-ignition, calculated with the following modified Tolstov equation[19]:

$$\tau = 3.8*10^{-6}(1-1.6*10^{-4}.\omega)\sqrt{\frac{T}{P}}\exp\left(\frac{E_a}{8.312T}-\frac{70}{CN+25}\right) \quad (6)$$

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The second stage, the premixed combustion stage, is calculated using the equation:

$$\frac{\mathrm{dx}}{\mathrm{dt}} = \varphi_0 * \left(\mathrm{A}_0\left(\frac{\mathrm{m_f}}{\mathrm{V_i}}\right)(\sigma_{\mathrm{ud}} - \mathrm{x}_0).(0.1\sigma_{\mathrm{ud}} + \mathrm{x}_0)\right) + \varphi_1\frac{\mathrm{d}\sigma_{\mathrm{u}}}{\mathrm{dt}} \quad (7)$$

The third stage, the Mixing controlled combustion phase, is calculated with:

$$\frac{dx}{dt} = \varphi_1\frac{d\sigma_u}{dt} + \varphi_2 * \left(A_2\left(\frac{m_f}{V}\right)(\sigma_u - x).(\emptyset - x)\right) \quad (8)$$

The last stage or the late combustion phase is calculated with the equation:

$$\frac{dx}{dt} = \varphi_3 A_3 K_T(1 - x)(\xi_b\alpha - x) \quad (9)$$

Where: T is cylinder temperature, A_0, A_1, A_2, A_3 are constants, $\varphi_0,\ \varphi_1,\ \varphi_2,\ \varphi_3$ are functions characterizing the achievement of combustion, m_f is the mass of fuel, σ_u is the vapor fraction composed during the ignition period,V_i and V are the Volume of the induced air and the cylinder volume respectively, x_0 is the fuel vapor portion shaped at the time of ignition delay, x is the fraction of heat release, $\emptyset$ is the equivalence ratio and ξ_b is the efficiency of the air used.

These equations assume that $\varphi_0 = \varphi_1 = \varphi_2 = \varphi_3 = \varphi$, that means the entirety of the fuel vapor has reacted. The Woschni correlation [20] was used for the heat transfer calculation.

Brake specific fuel consumption and thermal efficiency

The brake specific fuel consumption is calculated from the fuel mass flow of fuel and the brake power (obtained from the P-V graph) as follow:

$$BSFC = \frac{\dot{m}_f}{P_b} \quad (10)$$

The brake thermal efficiency is then calculated using the lower heating value of the fuel blend:

$$BTE = \frac{360}{BSFC * LHV} \quad (11)$$

Where:P_b is the brake power and LHV is the lower heating value of the fuel.

Engine specifications

The engine modeled or the validation is a single cylinder, 4 strokes and direct injection compression ignition engine. The main engine specifications are listed in Table-1.

Table-1. Engine specifications [21]

Brand/model	**Lombardini 6 LD 400**
Type	Single-cylinder, 4 strokes, compression ignition
Displacement	395 cm^3
Bore	86 mm
Stroke	68 mm
Connecting rod length (mm)	112 mm
Compression ratio	18
Valves/cylinder	1
Intake valve opening	7,5 CAD bTDC
Intake valve closing	25,5 CAD aBDC
Exhaust valve opening	21 CAD bBDC
Exhaust valve closing	3 CAD aTDC
Injection timing	20 CAD bTDC
Number of nozzles	4

Fuel properties

The biodiesel used in this study is the SME (Soy Methyl Ester). Three blends were used: SME100 (pure biodiesel), SME40 (40% biodiesel, 60% diesel) and SME20 (20% Biodiesel, 80% diesel). The main properties of each blend are represented in Table-2.

Table-2. Biodiesel blends properties [22].

Blend	**D100**	**B20**	**B40**	**B100**
Lower heating value(MJ/kg)	43,1	41.2	39.9	36.12
Apparent activation energy (kJ/mol)	22	21	20	12
Cetane number	46	48.68	49.37	51.3
Density at 323K (kg/m^3)	810	841	852	885
Surface tension at 323K (N/m)	0,0275	0.03122	0.0343	0.0433
Fuel temperature (K)	380	380	380	380

Comparison between experimental data and numerical model

The engine parameters considered for comparison are: cylinder pressure, brake specific fuel consumption, brake thermal efficiency and ignition delay. Those parameters were compared to the simulation results obtained.

Table-3 shows both experimental and computational results for the 2 available loads. The numerical simulation is in good agreement with

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experimental results [21] for diesel fuel in in the two loads considered. In Figures 1 and 2, the curves of the cylinder pressure are shown for both the experimental and simulated results of the two loads considered. This allows us to go for the use of biodiesel and to investigate the impact of blending it with diesel at different loads on the combustion and performance of the engine.

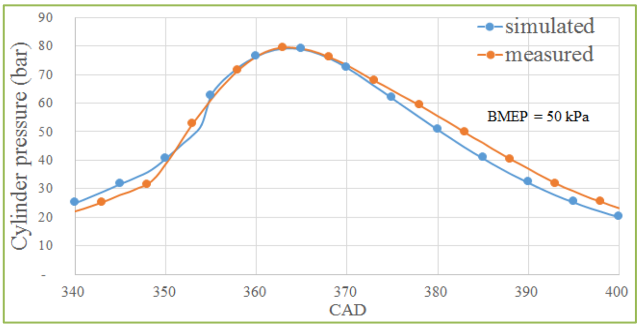

Figure-1. Comparison between simulated and experimental cylinder pressures (case BMEP=25kPa).

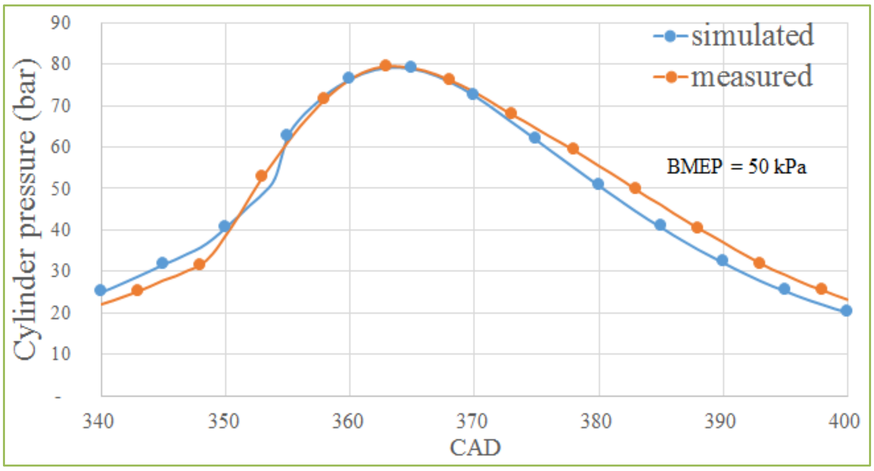

Figure-2. Comparison between simulated and experimental cylinder pressures (case BMEP=50kPa)

Table-3. Comparison of experimental and simulated results.

Load (kPa)	Max. cylinder pressure (bar)		Difference (%)	BSFC (g/kWh)		Difference (%)	BTE (%)		Difference (%)	Ignition delay (CAD)		Difference (%)
	Exp	Sim		Exp	Sim		Exp	Sim		Exp	Sim	
25	71,9	71.555	-0,48	376	383,31	1,9	22,30	21,974	-1,5	14,89	12.551	-16
50	79,6	79,45	-0,19	338	337,17	-0,2	24,80	24,982	0,7	10,93	11,956	9

PERFORMANCE AND EMISSIONS OF THE ENGINE UNDER DIFFERENT FUEL BLENDS

Ignition delay

Ignition delay (ID) is defined as the time lag separating the start of fuel injection and the start of combustion. [23]. In the curves of pressure, it can be visually calculated, as the Start of combustion leads to a higher pressure gradient. The result of ID for each case is represented in Figure-3. For all fuels, ID decreases with the increase of load. Moreover, the addition of biodiesel to the blend also decreases the ignition delay. Pure biodiesel is found to give the smallest delays, between 2 and 3 CAD. This decrease can be attributed to the higher cetane number of the blends as the percentage of biodiesel grows. in parallel, Labecki and Ganippa [24], when running an experiment with Rapeseed biodiesel observed that the decrease of IG with load was due to the oxygen content in biodiesel.

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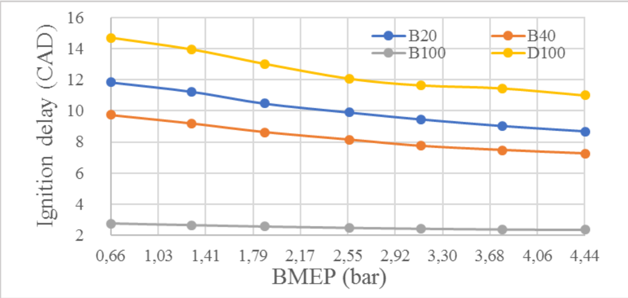

Figure-3. Ignition delay for the 4 blends.

Brake specific fuel consumption

The Brake specific fuel consumption (BSFC) is the amount of fuel consumed to produce 1kW of power output. The results obtained for BSFC are shown in Figure-4. As a general trend, the increase of load leads to a decrease in BSFC. The effect of biodiesel is an increase of consumption. In fact, the more biodiesel is added to the blend the more the consumption is, for all fuels and loads. This tendency is explained with the lower calorific value of biodiesel than that of diesel by 16, 2%. In some experiments [25, 26], increase in BSFC is imputed to both the lower heating value and the higher density of the biodiesel blends.

The engine consumes then more fuel to produce the same power output.

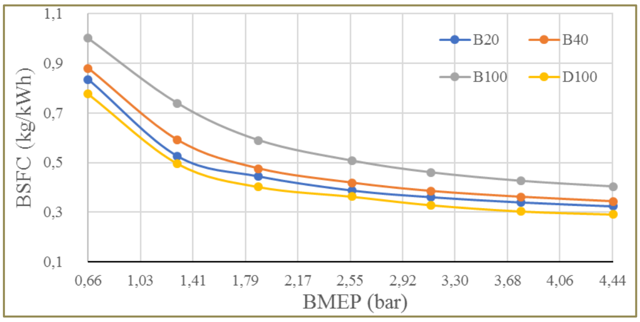

Figure-4. Brake specific fuel consumption for the 4 blends.

Brake thermal efficiency

The brake thermal efficiency (BTE) represents the amount of chemical energy used to produce the brake power. Results for the different fuels are presented in Figure-5. For all the fuels, the BTE increases with load. The biodiesel blends show a decrease of BTE, in comparison with pure diesel. Since its calculation is based on the BSFC and the lower heating value of the blend, it is clear that adding biodiesel percentage decreases the BTE. We note that B20 is the blend that is giving the closer results to those of diesel.

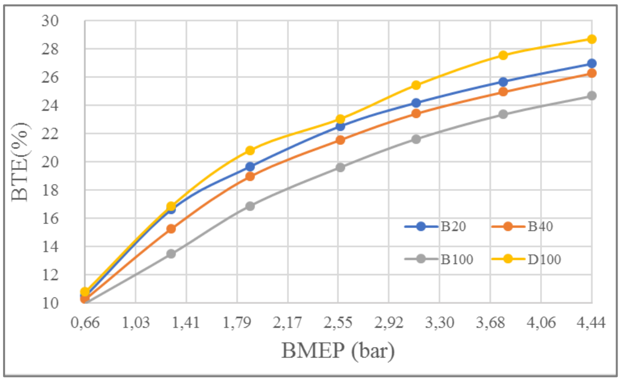

Figure-5. Brake specific thermal efficiency for the 4 blends.

Maximum cylinder pressure

The simulation results for maximum cylinder pressure are presented in figure 6. For all the fuels, the increase in load leads to an increased peak pressure. As the load increases, the fuel quantity injected in the cylinder also increases, which yields higher peak pressures. What's interesting is that peak pressure increases with the increase in biodiesel percentage. This increase ca be explained by the early start of combustion, which enables longer combustion and hence higher pressures. Also, the higher oxygen content of biodiesel improves the combustion process, which also gives higher pressures. Similarly, Golimowski *et al*. [27] Reported that the higher density of biodiesel leads to more burning of fuel during premixed combustion stage, and hence a higher pressure. In contrast, some experiments reported no increase or even less peak pressure for the biodiesel blends [28]. Ibrahim [29] noted that the increase in peak pressure and in combustion duration was more attributed to load increase than the biodiesel percentage itself.

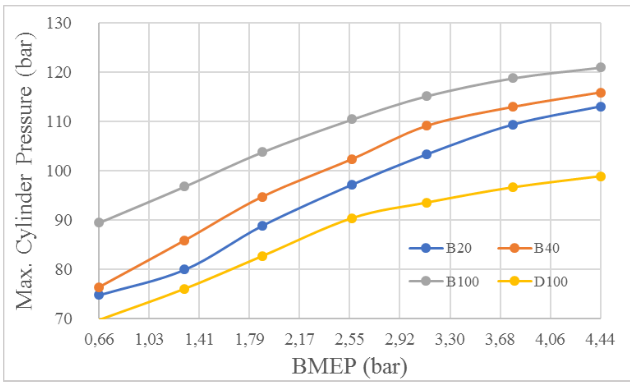

Figure-6. Maximum cylinder pressure for the 4 blends.

NOx emissions

Simulation results for the emissions of NOx are shown in Figure-7. It can be noted that NOx Emissions increase with load and biodiesel percentage. The NOx emissions are one of the main issues with the use of biodiesel as the figure shows. Emissions level is far higher than those of diesel. The short ignition delay, the high peak pressure and the oxygen content of biodiesel raise the

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cylinder temperature; thus, the thermal NOx also increases.

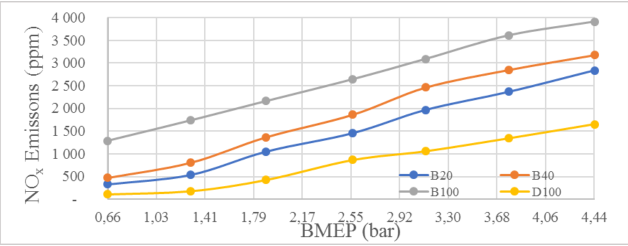

Figure-7. NOx emissions for the 4 blends.

CO_2 emissions

Figure-8 shows the results of the emissions for CO_2. For all fuels, the CO_2 emissions decrease with the increase in load but increase with the percentage of biodiesel in the blend. The raise is attributed to the oxygen contained in biodiesel which enhances the combustion, making it more complete. In the results, B100 exhibits up to 34% higher emissions than pure diesel (at BMEP=1, 3 bar). The emissions of CO_2 contribute to global warming; however, the production and growing of biodiesel plants absorb CO_2 emissions, hence reducing the excess of the emissions.

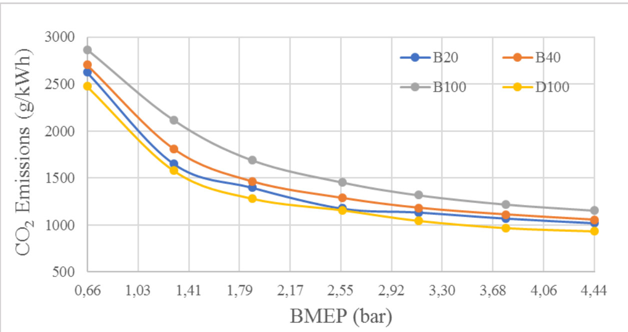

Figure-8. CO_2 emissions for the 4 blends.

Particulate matter

Emissions of particulate matter are shown in Figure-9. For all fuels, the PM emissions decrease with load. The biodiesel blends clearly show a decrease in particulate matter emissions compared to diesel. Many studies reported that this decrease was because of the oxygen content of biodiesel. Labeckas *et al.* [30] concluded that the decrease is related to the last stage of combustion. When the load increases, the oxygen content of biodiesel enables better diffusion combustion, and that leads to higher oxidation of particles formed, so the PM traces in the burnt gases are lowered.

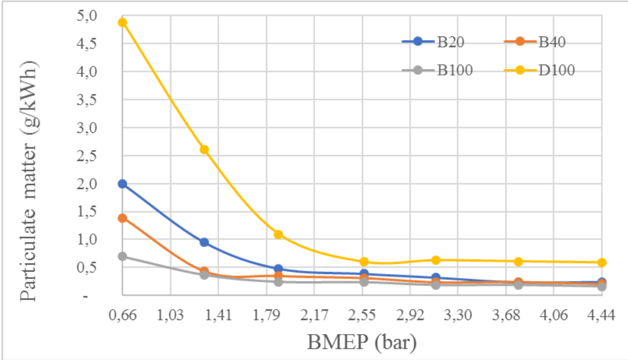

Figure-9. Particulate emissions for the 4 blends.

The simulations show that closer results to those of diesel are obtained with B20. For the next section, constant load (BMEP=3bar) and speed (2200 rev/min) are considered. The injection timing of the engine is retarded from 19 CAD bTDC to 7 CAD bTDC to see the effect a modification on the engine can have on the performance and emissions characteristics when B20 is used as fuel.

EFFECT OF RETARDING THE START OF INJECTION ON THE COMBUSTION AND EMISSION PERFORMANCES

Effect on brake specific fuel consumption

The effect of retarded injection timing on BSFC is shown in Figure-10. There is a slight increase in BSFC as the injection is set earlier. Advanced injection leads to longer ignition delay (Figure-11). This means that more fuel has to be injected and therefore the BSFC increases. However, the consumption remains in an interval of 0, 32 Kg/kWh to 0, 37 kg/kWh. Experimental and numerical works reported similar trends. In his experimental work, Jindal [31] noted a rise of 9% in BSFC from when injection timing was set to 20 CAD BTDC instead of standard 23 CAD bTDC. Datta and Mandal [32] numerically noted a boost of 2% in BSFC when injection was set 3 CAD before TDC.

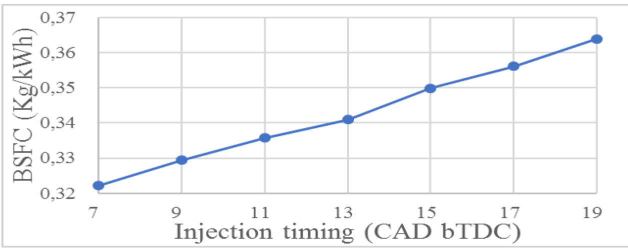

Figure-10. Variations of BSFC with injection timing.

Effect on ignition delay

Figure-11 shows the variations of ignition delay with respect to injection timing. The ignition delay can be defined as the crank angle gap separating the time of injection and the start of combustion. According to Figure-11, it is reduced as the injection is set closer to TDC. The IG is reduced from 9, 1 ° at 20° bTDC to 5, 7 ° at 7° bTDC. When injection is retarded, the temperature and

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pressure in the cylinder make the atomization of the fuel faster and the ignition starts earlier.

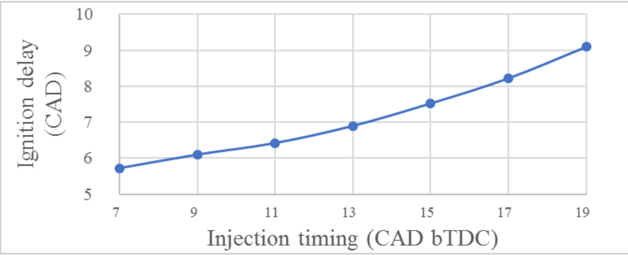

Figure-11. Variations of ignition delay with injection timing.

Effect on combustion duration

The variation of combustion duration as a function of injection timing is given in Figure-12. It is common to set the combustion duration as the angle difference between 90% and 10% of the mass burnt fraction. The figure demonstrates that the combustion duration goes from 34, 2 to 22, 2 CAD as the injection is set from 19 to 7 CAD bTDC. Agarwal *et al* [33] found similar trend for the combustion of Karanja biodiesel-diesel blends. It is noted also that Retarded starts of injection deferred both begin and end of ignition, however the delay in SOC timing was higher compared to End of combustion. This higher ignition leaded to shorter combustion duration with retarded start of injection.

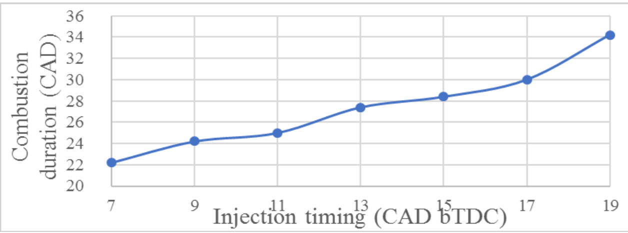

Figure-12. Variations of combustion duration with injection timing.

Effect on NOx emissions

Figure-13 shows the variation of NOx emissions as a function of injection timing. It is clear that NOx emissions are significantly reduced when the start of injection is closer to TDC. With retarded injection, the quantity fueled to the engine contains lesser oxygen percentage; since the ignition delay becomes shorter with retarded start of injection, the reached temperature doesn't cause an increased emission of NOx. Same behavior is reported in the literature. Agarwal *et al.* [33] noted a decrease in NOx emissions with retarded injection timing for Karanja Biodiesel. It is to remember however that the NOx emissions remain higher than that of diesel, but with decreased injection timing, the emissions fall to lower values than those of diesel.

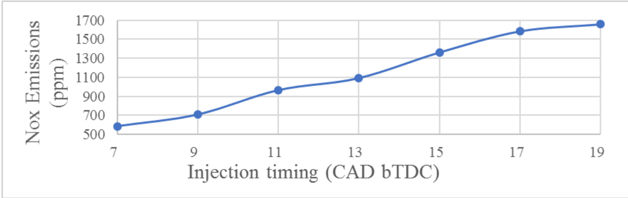

Figure-13. Variations of NOx emissions with injection timing.

Effect on PM emissions

Figure-14 shows the variation of particulate matter with the advancement of injection. The retarded start of injection leads to an increase from 0, 22 to 0, 53 kg/kWh. As with Figure-12, the shorter ignition delay combined to the retarded injection results in shorter combustion time, leading to a poorer reaction of the particles and thus increased emission in PM by 62%. We note that only these emissions are increased in the case of retarded injection. In general, the behavior of NOx and PM emissions are quite opposite with hydrocarbons combustion.

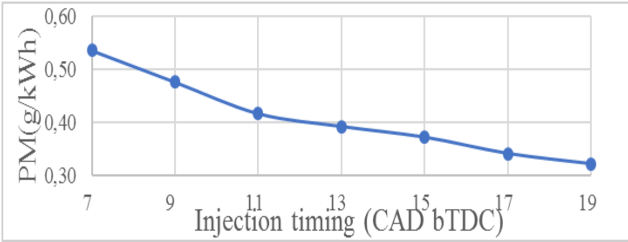

Figure-14. Variations of PM with injection timing.

CONCLUSIONS

In this study, a numerical model for Soy methyl ester biodiesel blended with Diesel was established in order to predict performance and emission characteristics of a single cylinder compression ignition engine. The model was first validated against experimental results, then comparative study of different biodiesel-diesel blends was performed in order to choose the optimal one. The latter was then used with a retarding in the injection timing to see the effects on the combustion and emission performances. The results showed that:

- B20 is the blend which offers similar performances to those of pure diesel, and relatively similar emissions compared to higher biodiesel percentage.
- Retarding the start of injection results in lower fuel consumption, and therefore better thermal efficiency, and lower ignition delay and combustion duration.
- In terms of emissions, retarding the start of injection reduces the NOx emissions to lower values than those for pure diesel at standard injection. However, the PM emissions increase by as much as 62, due to the

combustion that's not given enough time so burn all the particles.

In general, the use of a biodiesel-diesel blend in a compression ignition engine requires a change in the engine parameters. The injection parameters are to be modified because of the different physical properties of the biodiesel to ensure a better combustion and a lasting engine.

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