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Full Length Research Paper

Experimental and theoretical evaluation of the performance parameters and emission characteristics of bio diesel using C.I engine for various injection pressure

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Cost and limited reserves of conventional fossil fuels have intensified the search for alternative fuels for use in internal combustion engines. A possible alternative engine fuel is vegetable oil because it is a clean burning, renewable, non-toxic, biodegradable and environmentally friendly transportation fuel. It can be used in a neat form without any modification of the engine. Vegetable oils are produced from crops such as soybean, peanut, sunflower, cotton, jatropha, mahua, neem, coconut, linseed, mustard, *Millettia pinnata*, rapeseed and castor oil plant. A theoretical model was developed to evaluate the performance characteristics and combustion parameters of vegetable oil esters like Jatropha, Mahua and Neem for various injection pressures and compared to diesel fuel. From the investigation it was concluded that the performance of vegetable oil esters such as Jatropha, Mahua, and Neem were much better. Thus the developed model was highly capable for simulation work with bio-diesel as a suitable alternative fuel for diesel engines.

Key word: IC engines, injection pressure, tranesterification, emission, performance, brake power.

INTRODUCTION

An experimental investigation was carried out to assess the performance parameters and emission levels of the three different vegetable oil esters in a single cylinder with four strokes constant speed computerized diesel engine test rig. The performance characteristics and emission levels such as NO_x, CO, HC, and smoke were measured using eddy current dynamometer, crypton computerized emission analyzer instruments (Model-EN2-390) and Bosch Smoke meter. From the experimental results, it is concluded that in terms of performance characteristics and emission levels, vegetable oil esters can be regarded as the best alternative fuel instead of diesel fuel.

THEORETICAL CONSIDERATION

Description of the four zone model

The present four-zone model is developed without deviating much from the basic concepts of the two zone model like the jet penetration, volume of spray, preparation rate, reaction rate for the purpose of heat release, the effect of impingement of the spray on the

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cylinder walls etc. In essence the burning zone of the two-zone model is further subdivided to provide a total of four distinct zones, namely:

(1) Fuel zone
(2) Stoichiometric burning zone
(3) Product plus air zone
(4) Air zone- unburnt zone, Figure 1.

The main advantage of this model is that it can truly represent the temporal and spatial variations of the fuel-air ratio and temperature (Venkatraman and Devaradjane, 2010).

Where $\frac{dM_{fi}}{d\theta}$ =Fuel injection rate

$\frac{dM_{fr}}{d\theta}$ = Fuel reaction rate

$\frac{dM_{ae}}{d\theta}$ = Air Entrainment Rate

$\frac{dM_{ac}}{d\theta}$ = Air consumption rate for stoichio burning

$\frac{dM_{sp}}{d\theta}$ = stoichiometric product movement rate

Theory of spray formation and combustion heat transfer

The theory used to simulate the combustion process in the combustion chamber of a diesel engine mainly involves the jet formation, calculation of jet penetration and spray volume, estimation of fuel burning rate and finally the heat transfer between the cylinder contents and the surroundings (Heywood,1989).

Fuel jet penetration

The development of the fuel spray or jet in the diesel combustion chamber is based upon the theory of a steady flow semi-infinite free jet, and finally modified by the use of empirical factors of transient (real) jets. Assuming that fuel jet penetration model is developed by modifying the transient (real) jet equation, the fuel jet penetration is X_{max}.

$$XU_{\max} = 7.414\sqrt{K} \tag{1}$$

Where x = jet penetration. (m)
U_{max} = centre line velocity (m/sec)
K = kinematics momentum flux (m^4/sec^2)

$$XdX = 7.414\sqrt{K}dt \tag{2}$$

$$X_{max} = 0.685 \text{ X } 2.420 \left(\left[\frac{\Delta p}{\rho_a} \right]^{0.5} dn.t \right)^{0.5} \tag{3}$$

Volume of fuel spray

The volume of fuel before impingement consists of conical part of the half cone angle θ and the bell shaped part. The volume of the conical part of the spray is calculated analytically and for the bell shaped part of the jet, numerical integration is used. The volume flow rate at any section along the axis of the spray can be computed by integrating the product of velocity and the area.

$$\text{Volume flow rate} = 2\pi r_o u_{\max} \int_0^{r_o} \frac{r}{r_o} \frac{u}{u_{\max}} dr \tag{4}$$

$$\frac{r}{r_0} = \varepsilon \tag{5}$$

$$r_0 = X \tan\theta \tag{6}$$

$$U_{\max} = 7.414 \frac{\sqrt{k}}{x} \tag{7}$$

From the free jet theory.

$$\text{Volume flow rate} = 5.9902 \tan^2\theta\sqrt{K}X.KF \tag{8}$$

Combustion and heat release

In this model the combustion period is assumed to consist of two periods. They are pre-mixed period and diffusion period. The combustion of the reacted fuel is assumed to be ideal. It produces only H_2O and CO_2 for the purpose of calculating the cylinder pressure and the energy level of the spray as a whole. Whitehouse model incorporating the rate of preparation of the fuel, surface area of fuel droplets and partial pressure of oxygen in the cylinder was used for this work. The preparation rate equation is as follows:

$$P = K'M_i^{1-x}M_u^{x}PO_2^{L} \tag{9}$$

$$P = K'M_i^{1/3}M_u^{2/3}PO_2^{0.4} \quad \text{(Kg/°CA)} \tag{10}$$

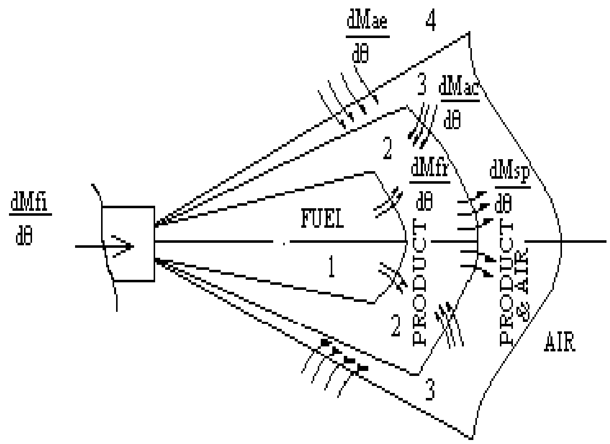

Figure 1. Schematic representation of combustion zones.

Heat transfer

$$M_i = n \times \rho \frac{1}{6} \pi {D_0}^3 \tag{11}$$

$$M_u = n \times \rho \frac{1}{6} \pi D^3 \tag{12}$$

Total surface area at any instant $= n \times \pi \times D^2$

Arrhenius type of equation is used for reaction rate of the prepared but unburnt fuel, which is as follows:

$$R = K'' \frac{PO_2}{N\sqrt{t}} e^{-ACT/T} \int (P - R) dx \quad \text{(Kg/°CA)} \tag{13}$$

The heat transfer rate is calculated by using Annand's formula (1963). This formula seems more fundamental than the available alternatives. The equation considers net heat transfer as the summation of both radiative and convective heat transfer.

$$\frac{dQ_{loss}}{dt} = \left(S_c.H.\left(T - TW\right) + S_r.C.\left(T_4 - TW^4\right)\right) \tag{14}$$

where $dt = \frac{d\theta}{6N}$, $H = a\frac{K}{D}R_e^{\,b}$ $\quad R_e = \frac{\rho \times U \times D}{\mu}$, $K = \frac{C_p \mu}{0.7}$ (15)

Method of estimating the final cylinder pressure

The final cylinder pressure equation can be written as

$$P_2 = \left\{ P_1 + P_1 \left(\frac{T - t_{b1}}{t_{b1}} \right) \left(\frac{V_{b1}}{V_{u1} + V_{b1}} \right) \right\} \left(\frac{V_1}{V_2} \right)^{\frac{C_p}{C_v}} \tag{16}$$

P_2= First estimated value of the final pressure
P_1= Pressure after entrainment
T-t_{b1}=Temperature rise due to constant volume heat addition
V_1, V_2= Cylinder volume at beginning and end of the step.

Energy equations

The energy equation can be written as:

$$\frac{dE_{cyl}}{d\theta} = DQF - DW - DQC - DQR \tag{18}$$

DQC – DQR = Heat transfer to the system
DQF = Energy associated with the external flow to the system
DW = External work done by the system

EXPERIMENTAL WORK

The performance tests were carried out on a single cylinder, four stroke naturally aspirated and water cooled kirloskar computerized diesel engine test rig. Diesel engine was directly coupled to an eddy current dynamometer. The engine and dynamometer were interfaced with a control panel which was connected to a computer. This computerized test rig was used for recording the test parameters such as fuel flow rate, air flow rate, temperature and load for calculating the engine performance, brake thermal efficiency and emissions like CO, HC, NOx and smoke (Yaman et al., 2001), shown in Figure 2 and Table 1.

RESULTS AND DISCUSSION

The performance and emission characteristics curves were carried out experimentally like predicting various performance characteristics such as thermal efficiency, specific fuel consumption for different vegetable oil esters such as jatropha, mahua and neem. The experimental results of the vegetable oil esters were compared with diesel fuel. From the results it was concluded that the performances of the vegetable oil esters are more or less equal to diesel. But pollutants like HC, CO, Nox and Smoke are reduced nearly 18% when compared to diesel. The results are shown in Figures 3 to 8.

Experimental brake thermal efficiency for various injection pressures

The brake thermal efficiency is also predicted with respect to various injection pressures for different

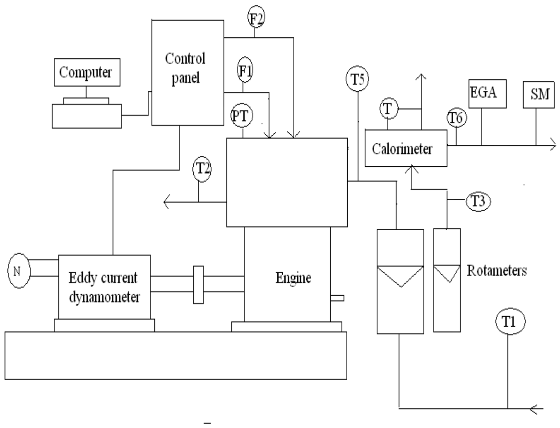

Figure 2. Schematic of experimental set-up.

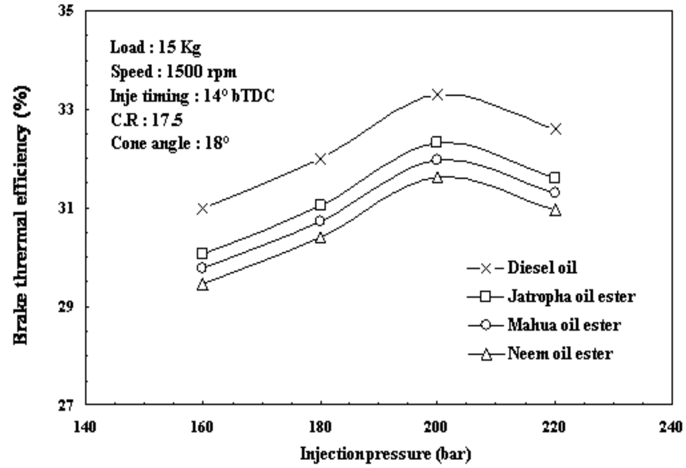

Figure 3. Comparision of brake thermal efficiency of three different biodiesel with respect to injection pressure.

Table 1. Engine Specifications.

Engine parameters	Specifications
Engine type	Kirloskar, Four stroke
No of cylinder	Single
Bore	87.5 mm
Stroke	110 mm
Cubic capacity	661 cc
Compression ratio	17.5
Rated speed	1500 rpm
Dynamometer	Eddycurrent,Water cooling

vegetable oil esters and compared with diesel fuel. Injection pressure is one of the very important operating variables, which affects the brake thermal efficiency (Nagarhalli and Nandedkar, 2011). The brake thermal efficiency is predicted for various injection pressures, and the injection pressure tried was 140 bars to 240 bars. When the injection pressure rose to 200 bar, it was found that the brake thermal efficiency has reached its maximum.

Fuel particles were uniformly mixed with air particles at an injection pressure of 200bar. When the injection pressure was increased above 200bar, the wall-wetting problem was created. This problem leads to a decrease in the performance of the engine. When the cone angle is decreased below 200 bar the fuel penetration and velocity were is also decreased and it generates more unburnt hydrocarbon. So injection pressure of 200 bar is found as the optimum value for maximum efficiency in the engine.

Experimental brake specific fuel consumption for various injection pressures

The brake specific fuel consumption found for diesel is 0.257 kg/kw-h; whereas for jatropha oil ester it is 0.277 kg/kwhr; for mahua oil ester, 0.286 kg/kw-h; and for neem oil ester, 0.291 kg/kw-h for optimum engine conditions such as 75% of load, injection pressure of 200bar.

Because at optimum engine condition fuels burn completely. Misfiring does not occur at this time. This increases the temperature and the number of moles of the burned gases in the cylinder. This effect increases the pressure to give increased thermal efficiency and decreases the specific fuel consumption. The specific fuel consumption is predicted with respect to injection pressure for various vegetable oil esters and compared with diesel fuel.

Experimental carbon monoxide for various injection pressures

Figure 5 shows the comparison of predicted emission result of carbon monoxide emission with various injection pressures for different vegetable oil esters and compared with diesel fuel. The carbon monoxide for vegetable oil esters are nearly 18 % reduced than diesel. The main difference in ester-based fuel compared to diesel is the oxygen content and cetane number. As the ester based fuel contains some oxygen which acts as a combustion promoter inside the cylinder, results in better combustion than diesel fuel. Hence carbon monoxide, which is present in the exhaust due to incomplete combustion, reduces drastically. The reduction of carbon monoxide in case of ester is lowered when compared to diesel.

Experimental hydrocarbon for various injection pressures

Figure 6 shows the comparison of predicted result of

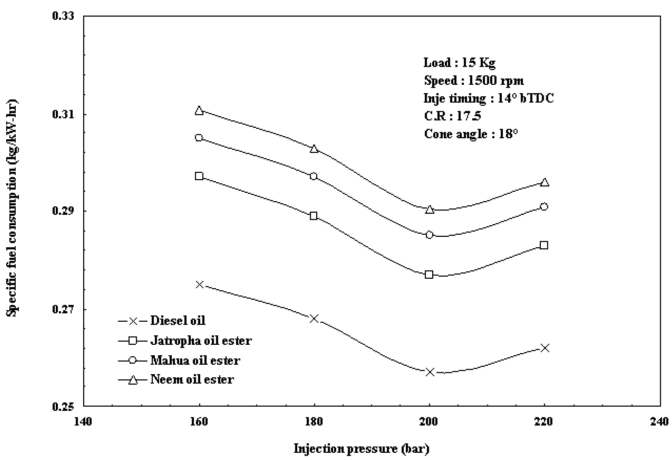

Figure 4. Comparision of specific consumption of three different biodiesel with respect to injection pressure.

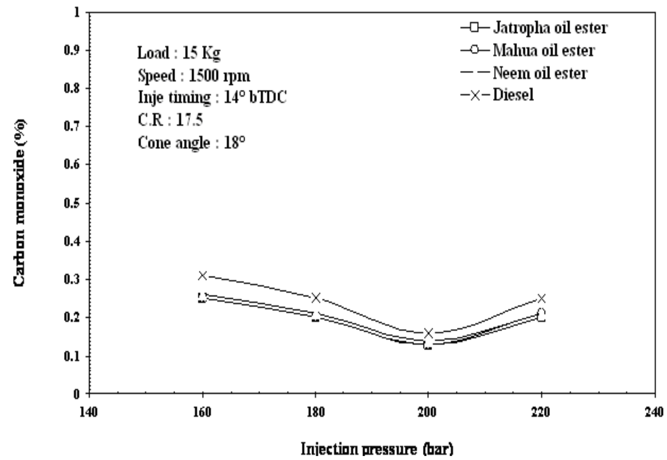

Figure 6. Comparison of carbonmonoxide for three different biodiesel with respect to injection pressure.

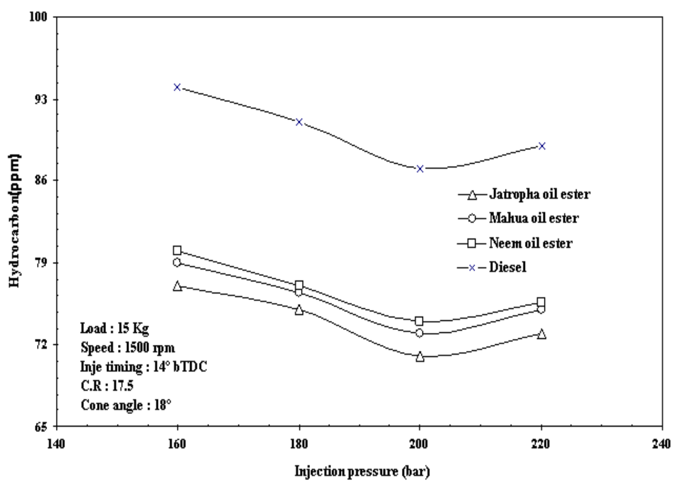

Figure 5. Comparision of hydrocarbon for three different biodiesel with respect to injection pressure.

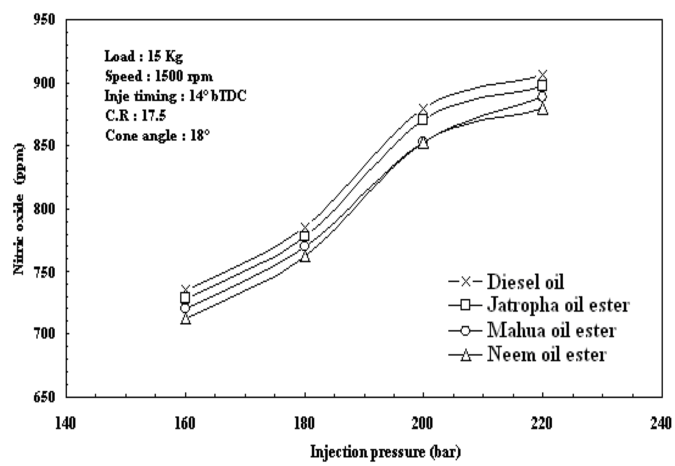

Figure 7. Comparision of predicted nitric oxide of three different biodiesel with respect to injection pressure.

hydrocarbon emission with various injection pressures for different vegetable oil esters and compared with diesel fuel. The hydrocarbon emission for diesel is about 119 ppm; while for jatropha oil ester, mahua oil ester and for neem oil ester it is 98, 100, and 101 ppm respectively. Cetane number of the fuel plays a vital role in ignition process.

As cetane number of ester-based fuel is higher than diesel, it exhibits a shorter delay period and the fuel undergoes better combustion. Here, the oxygen content of the fuel comes into picture as it enhances the combustion process. Therefore overall result of oxygen content and cetane number of the fuel leads to low CO and HC emission. Thus it is very clear from the graph that esters emit lower Hydrocarbon emission than that of diesel.

Experimental Nitric oxide for various injection pressures

Figure 7 shows the comparison of predicted result of NO_x formation with various injection pressures for different vegetable oil esters and compared with diesel fuel. The NO_x for diesel is about 985 ppm; while for jatropha oil ester, mahua oil ester and for neem oil ester it is 967, 960 and 955 ppm respectively.

In a direct injection naturally aspirated four-stroke diesel engine NO_x emission is sensitive to oxygen content, adiabatic flame temperature and spray properties. A change in any of those properties may change the NO_x production. Further, more fuel chemistry effects in the flame region could account for a change in NO_x production. NO_x formation is increased with increase in temperature (Sathiyagnanam and Saravanan, 2011).

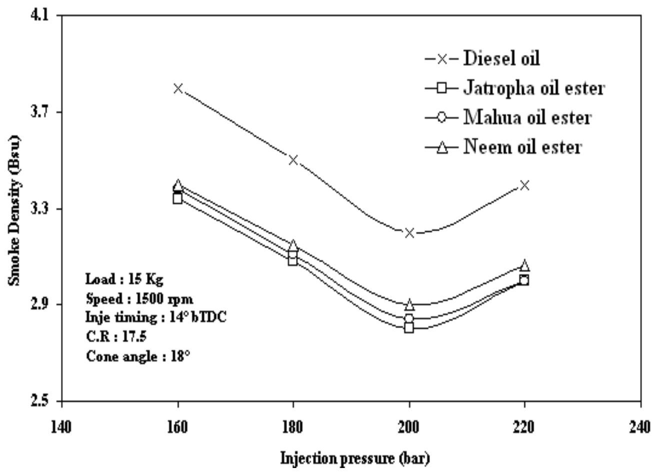

Figure 8. Comparision of predicted smoke density of three different biodiesel with respect to injection pressure.

EXPERIMENTAL SMOKE DENSITY FOR VARIOUS INJECTION PRESSURES

Figure 8 shows the comparison of experimental results of smoke with various engine injection pressure for different vegetable oil esters and compared with diesel fuel. The smoke for diesel is 4 bsu; for jatropha oil ester, 3.5bsu; for mahua oil ester, 3.6bsu; and for neem oil ester, 3.6bsu. The smoke formed due to incomplete combustion was much lower for esters compared to diesel. This is because of better combustion of esters. The main difference in ester-based fuel compared to diesel is the oxygen content and cetane number. As the ester based fuel contains some oxygen which acts as a combustion promoter inside the cylinder, thus resulting in better combustion than diesel fuel.

Conclusion

1. The brake thermal efficiency is reduced to about 3% for Jatropha, 4% for Mahua and 5% for Neem oil ester when compared to diesel. It is concluded that the brake thermal efficiency for vegetable oil ester slightly decreased when compared to diesel.
2. The specific fuel consumption for vegetable oil esters increased to about 8, 11 and 13% respectively for jatropha, mahua and neem oil ester when compared to diesel. It is concluded that the specific fuel consumption for vegetable oil ester is slightly increased than diesel.
3. The Carbon monoxide reduced by 18% for jatropha, 17 % for mahua, and 16 % for neem oil ester when compared to that of diesel. It is concluded that the carbon monoxide for vegetable oil ester is less when compared to diesel fuel.
4. The concentration of Hydro carbon decreased by 18 % for jatropha oil ester, 16 % for mahua oil ester and 15 % for neem oil ester when compared to diesel fuel.
5. The formation of Nitric oxides decreased by 1.8% for jatropha oil ester, 2.5% for mahua oil ester, and 3 % for neem oil ester when compared to diesel fuel.
6. The Smoke level decreased by 12 % for jatropha oil ester, 11 % for mahua oil ester, and 10 % for neem oil ester when compared to diesel fuel. Hence, it is concluded that in terms of performance characteristics, vegetable oil esters serve as a potential substitute for diesel fuel, (Babu and Devaradjane, 2003).

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