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Experimental Analysis and Energetic Evaluation for Mahua Oil Methyl Ester with Diethyl Ether

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Abstract

A theoretical method based on various existing process models has been developed for the performance evaluation of a compression ignition engine by using diesel and Mahua oil as an input fuel. Experimental evaluations of Mahua oil methyl ester/diesel along with diethyl ether were tested on a single cylinder naturally-aspirated indirect-injection diesel engine. The purpose of using diethyl ether with diesel/biodiesel is to control the emission from the diesel engine and to improve its performance. The effect of test fuels at different volumetric proportion and loads are also analyzed on the engine performance and emission. The comparisons of theoretical and experimental results are discussed. Engine performance characteristics predicted by this model are in closer approximation to that of experimental results. Engineering equation solver has been used to validate the model with the experimental results.

Keywords: Diesel engine, thermodynamic modeling, Mahua oil methyl ester, engine performance and emissions, engineering equation solver

Introduction

The theoretical models used in the case of internal combustion engines are mainly described as thermodynamic and fluid dynamic models. Models based on thermodynamics can be further classified as single zone heat release, phenomenological jet based and multi-zone models. Single zone models assume that the cylinder content is uniform in composition and temperature and are suitable for prediction of engine performance. Phenomenological combustion models are based on each individual processes occurring in the engine cycle such as fuel injection, mixture formation, heat release, heat transfer and emission formation. Multi-zone models incorporate the development of the fuel spray with time and simplified quasi steady equations are used to describe processes like fuel injection, atomization, air entrainment, droplet formation, evaporation, wall impingement, ignition, heat release, heat transfer and so on. Fluid dynamic based models, often called multi-dimensional or computational fluid dynamics models are based on solving the governing equations for conservation of mass, momentum and energy and species concentration through a definite discretization procedure was carried out by Heywood [1].

In a low heat rejection engine, during the tests it was observed that more fuel coming from injectors heated up the fuel in the fuel tank. This reduced the fuel viscosity, so the need for preheating was eliminated as explained by Hasimoglu *et al.* [2]. Biodiesels from palm and rapeseed oils show shorter ignition delay than diesel fuel as discussed by Rodriguez *et al*. [3]. The new proposed correlations for biodiesels were compared against the fossil diesel. Watson *et al.* [4] presented comparisons with ignition delay measurements show that the new correlations predict ignition delay for bio-fuels better than the available correlations for diesel fuel. The ignition delay data were correlated as a function of the equivalence ratio, the mean cylinder pressure and mean temperature over the ignition delay interval.

Computational fluid dynamics (CFD) models are extensively used for flow visualization because these models are based on the numerical calculations of mass, momentum, energy, and species conservation equations in either 1, 2, or 3 dimensions, can produce a great amount of estimated data of flow, temperature, and concentration fields in operating engines.

The fuel economy and combustion control of the diesel engine have been improved by adopting several techniques like fuels modification, modification of combustion chamber design and exhaust after treatment. Some oxygenated fuels like ethanol, ether etc can be used with biodiesel to improve its performance, emissions and combustion quality. Diethyl ether (DEE) is a renewable one having a very higher cetane number (> 125), reasonable energy density and high volatility as carried out by Erwin and Moulton [5]. Antonini [6] produced an acid clay catalyst with 90 % conversion of ethanol at a reasonable cost. Utilization of DEE with biodiesel can improve the combustion quality as it has very high burning velocity which may increase the flame speed of biodiesel as discussed by Geo *et al*. [7]. The important properties of DEE and Mahua oil Methyl Ester (MOME) compared with diesel are given in **Table 1**.

Table 1 Test fuel properties.

Properties	Mahua oil methyl ester (MOME)	Diesel	Diethyl ether (DEE)
Viscosity at 30 °C (cSt)	6.2	2.6 - 4.1	0.23
Specific gravity	0.875	0.84	0.713
Flash point (°C)	184	70	–40
Cloud point (°C)	15	–1	-
Pour point (°C)	13	–6	-
Carbon residue (%)	0.9	0.7	-
Calorific value (MJ/Kg)	38	43	34

Theoretical modeling

A mono zone thermodynamic model is used for analyzing the performance characteristics of 4 stroke indirect injection compression ignition engines. A spatial uniformity of pressure, temperature and composition in each combustion chamber, at each instant of time, is assumed. The gas medium is assumed to obey the perfect gas law as discussed by Ramadhas *et al.* [8]. The fuels considered are diesel and Mahua oil. The molecular formula of diesel is approximated as $C_{12}H_{22}$ and for Mahua oil is considered as $C_{18}H_{32}O_2$ as explained by Balat and Balat [9]. Polynomial expressions are used for each species (O_2, N_2, CO_2, H_2O) considered in the calculation of specific heats, internal energy and enthalpy as a function of temperature as explained by Heywood, and Ganesan [1,10].

Energy conservation

By applying the first law of thermodynamics for the main and pre combustion chamber as an open system, the net heat release rate is determined by Heywood [1];

$$\frac{dQ_{n,1}}{dt} - p_1\frac{dV_1}{dt} + h_{2,1}\frac{dm_{2,1}}{dt} = \frac{dU_1}{dt} \quad (1)$$

$$\frac{dQ_{n,2}}{dt} - p_2\frac{dV_2}{dt} - h_{2,1}\frac{dm_{2,1}}{dt} = \frac{dU_2}{dt} \quad (2)$$

where the volume of the pre-chamber V_2 is constant so, Eq. (2) becomes;

$$\frac{dQ_{n,2}}{dt} - h_{2,1}\frac{dm_{2,1}}{dt} = \frac{dU_2}{dt} \tag{2a}$$

where U_1 and U_2 are gas sensible internal energies, $h_{2,1}$ as the gas sensible specific enthalpies, $m_{2,1}$ is the mass flow between the 2 chambers , $Q_{n,1}$ and $Q_{n,2}$ are net heat releases which are the difference between the combustion energy release (gross) and the heat transferred (loss) to the chamber walls.

The perfect gas state equations for the 2 chambers areas;

$p_1V_1 = m_1RT_1 \qquad p_2V_2 = m_2RT_2$

Therefore, the corresponding gross heat release rates, which are the energies released from the combustion of fuel, are given by;

$$\frac{dQ_{g,1}}{dt} = \frac{dQ_{n,1}}{dt} + \frac{dQ_{w,1}}{dt} \tag{3}$$

$$\frac{dQ_{g,2}}{dt} = \frac{dQ_{n,2}}{dt} + \frac{dQ_{w,2}}{dt} \tag{4}$$

where $\left(\frac{dQ_{w,1}}{dt}\right)$ and $\left(\frac{dQ_{w,2}}{dt}\right)$ are the rates of heat transferred to the walls of the main chamber and the pre-chamber.

By knowing the lower heating value Θ of the fuel, the fuel burned mass rate in each chamber is computed from;

$$\frac{dm_{b,1}}{dt} = \frac{1}{\Theta}\frac{dQ_{g,1}}{dt} \tag{5}$$

$$\frac{dm_{b,2}}{dt} = \frac{1}{\Theta}\frac{dQ_{g,2}}{dt} \tag{6}$$

Considering each combustion chamber as an open system, gas mass balance in differential form gives;

$$\frac{dm_1}{dt} = \frac{dm_{2,1}}{dt} + \frac{dm_{f,1}}{dt} \tag{7}$$

$$\frac{dm_2}{dt} = -\frac{dm_{2,1}}{dt} + \frac{dm_{f,2}}{dt} \tag{8}$$

where m_1 and m_2 are the gas mass in each chamber consisting of the corresponding air plus fuel, $m_{f,1}$ and $m_{f,2}$ are the fuel mass existing in each chamber. The existing fuel mass in each chamber is the quantity due to combustion and the fuel transported with the mass flow rate (dm/dt) exchanged through the connecting throat. This is expressed mathematically as;

$$\frac{dm_{f,1}}{dt} = \frac{1}{\Theta}\frac{dQ_{g,1}}{dt} + \frac{m_{f2,1}}{m_{2,1}}\frac{dm_{2,1}}{dt} \tag{9}$$

$$\frac{dm_{f,2}}{dt} = \frac{1}{\Theta}\frac{dQ_{g,2}}{dt} - \frac{m_{f2,1}}{m_{2,1}}\frac{dm_{2,1}}{dt} \tag{10}$$

where, if $dm_{2,1}/dt > 0$ then $m_{f2,1}/m_{2,1} = m_{f,2}/m_2$, and if $dm_{2,1}/dt < 0$ then $mf_{2,1}/m_{2,1} = m_{f,1}/m_1$.

It is obvious, that all the previous equation from (1) to (10) are in the form of differentiations with respect to time t .These can be converted to differentiations with respect to degrees crank angle θ, by multiplying by a factor $dt/d\theta = 1/6N$, where N is the engine speed.

Volume at any crank angle

The combustion chamber volume (main + pre- chamber) at any crank angle position can be computed using the following equation;

$$V(\theta) = V_d\left[\frac{r}{r-1} - \frac{1-\cos\theta}{2} + \frac{1}{2}\sqrt{\left(2\frac{L}{S}\right)^2 - \sin^2\theta}\right] \tag{11}$$

where r is the compression ratio, L is the length of the connecting rod, S is the stroke length, V_d is the displacement volume and θ is the angular displacement with respect to BDC.

Combustion reaction and properties calculation

Considering the fundamental stoichiometric chemical equation for the hydrocarbon-oxygen complete combustion can be explained as [11];

$$C_nH_m+[n+(m/4)]O_2 \rightarrow nCO_2+(m/2)H_2O \tag{12}$$

where n is the number of moles of carbon and m is the number of moles of hydrogen.

Internal energy (U), enthalpy (H), specific heats at constant pressure (C_p) and constant volume (C_v) of the gaseous mixture are calculated on the basis of charge composition and temperature. These properties of the gaseous mixture are calculated by Heywood [1].

$$U(T) = A+ (B-R)*T+ C\ln(T) \tag{13}$$

$$H(T) = A+B*T+C*\ln T \tag{14}$$

$$C_p(T) = B+C/T \tag{15}$$

$$C_v(T) = (B-R) +C/T \tag{16}$$

where A, B and C are the coefficients of the polynomial equation.

Heat release analysis

Gross heat release due to combustion is calculated by using single Weibe's heat release correlation given by Heywood [1];

$$\frac{dQ_g}{d\theta} = a(m+1)\left(\frac{Q_r}{\Delta\theta}\right)\left(\frac{\theta-\theta_0}{\Delta\theta}\right)^m \exp\left[-a\left(\frac{\theta-\theta_0}{\Delta\theta}\right)^{m+1}\right] \quad (17)$$

where Q_r is the heat released per cycle, θ_0 is the start of combustion, $\Delta\theta$ is the combustion duration. The value of m for all the fuel is taken as 2.0 and value of a is taken as 5.0.

Heat transfer analysis

The heat transfer between each chamber trapped mass and surrounding walls is calculated by using the formula explained by Annand [12].

$$\frac{dQ_w/d\theta}{A} = a\frac{k}{D}(\mathrm{Re})^b (T_g - T_w) + c(T_g^4 - T_w^4) \quad (18)$$

$$\mathrm{Re} = \rho D u_p/\mu$$
$$u_p = 2NS/60$$

where A is the heat transfer area, T_w is the temperature of the combustion chamber walls, T_g is the gas temperature, Re is the Reynolds number, k is the thermal conductivity, D is the cylinder bore, u_p is the mean piston speed, S is the piston stroke, ρ is the gas density and μ is the dynamic viscosity.

For both the combustion chamber constant b = 0.75 and c = 0 for the compression period and otherwise $c = 3.3\times10^{-8}$ [14] and a = 0.30 for the main chamber [13].

Mass of fuel injected

Considering that the nozzle open area is constant during the injection period, the total mass of the fuel injected for each crank angle is calculated as follows;

$$m_f = C_d A_n \sqrt{2\rho_f \Delta P}\left(\frac{\Delta\theta_f}{360N}\right)n \quad (19)$$

where n is the number of injector nozzle holes, C_d is the coefficient of discharge of injector nozzle, A_n is the cross sectional area of nozzle, ΔP is the pressure drop in the nozzle, N is the engine speed and $\Delta\theta_f$ is the fuel injection period.

Pressure drop in the nozzle

The pressure drop in the nozzle is calculated as explained by Rakopoulos *et al.* [14];

$$\Delta P = 0.5\rho_f\left(\frac{u_{inj}}{C_d}\right)^2 \quad (20)$$

where u_{inj} the spray velocity from the nozzle hole is given as;

$$u_{inj} = \left(\frac{dm_f}{d\theta}\right)\left(\frac{6N}{\rho_f A_n}\right) \quad (21)$$

where $\left(\frac{dm_f}{d\theta}\right)$ is the fuel injection rate (kg/°CA);

$$\left(\frac{dm_f}{d\theta}\right)=\left(\frac{m_f}{n\Delta\theta_f}\right) \quad (22)$$

Nitric oxide formation

The chemical equilibrium cannot predict the actual NO concentration. The generally accepted kinetics formation scheme proposed by Lavoie *et al.* [15] is used. The equations which describe the model together with their forward reaction rate constants k_{if} (m^3/kmol/s) are as follows;

(a) $N+NO \leftrightarrow N_2+O$ $\quad k_{if} = 3.1\times10^{10}\exp^{(-160/T)}$

(b) $N+O_2 \leftrightarrow NO+O$ $\quad k_{2f} = 6.4\times10^{6}\exp^{(-3125/T)}$

(c) $N+OH \leftrightarrow NO+H$ $\quad k_{3f} = 4.2\times10^{10}$

The change in NO concentration (in kmol/m^3) is expressed as follows;

$$\frac{1}{V}\left(\frac{d((NO)V}{dt}\right)=2(1-\alpha^2)\left(\frac{R_1}{1+\alpha R_1/(R_2+R_3)}\right) \quad (23)$$

where R_i is the one way equilibrium rate for reaction i, defined as;

$R_1 = k_{1f}(N)_e(NO)_e$, $\quad R_2 = k_{2f}(N)_e(O_2)_e$, $\quad R_3 = k_{3f}(N)_e(OH)_e$

with the index e denoting equilibrium concentration and term $\alpha = (NO)/(NO)_e$

Procedure for theoretical solution

The equations of the model adopted in the previous section are suitable for any hydrocarbon fuel such as diesel, vegetable oil and biodiesel etc. These equations are solved numerically using with time step size of 2° crank angles. The engine geometrical parameters, molecular weight of gaseous products and various constants used in the modeling are defined. The input parameters used in modeling are engine load and the molecular formula of the diesel and Mahua oil. The properties of the gaseous constituents such as enthalpy, internal energy and specific heats are calculated as a function of temperature. The pressure and temperature of the gases in the combustion chamber are calculated for every 2 degree crank angle. From the typical design of the engine, the combustion chamber volume at every degree crank angle is calculated with the help of Eq. (11). The outputs of the modeling program are instantaneous pressure, temperature, volume and the performance parameters that include Brake thermal efficiency (BTE), Brake specific fuel consumption (BSFC) and NO_x emission.

Table 2 Engine specifications.

Manufacturer	Kirlosker Engine Ltd., India
Rated power	3.5 kW@1500 rpm
No of Strokes	4
Mode of injection and injection pressure	Indirect injection, 145 bar
No of cylinders	1
Stroke	110 mm
Bore	87.5 mm
Compression ratio	12 to 18:1

Experimental setup and test procedure

A single cylinder 4 stroke water cooled diesel engine developing 3.5 kW at 1500 rpm was used. The engine details are given in **Table 2**. A high-speed digital data acquisition system was used to record the output signal of pressure. A piezoelectric transducer and crank shaft position optical encoder were used for the measurement of cylinder pressure as a function of crank angle. Emission analysis was carried out for exhaust smoke opacity, unburned HC, CO, CO_2 and NO_x. For measuring NO_x emissions, a chemiluminescent NO/NO_x (rose mount analytical-951 A) analyzer was used. For measuring the smoke opacity, an AVL437 smoke analyzer was utilized which gave readings in terms of percentage opacity. A diagram of the experimental test rig is shown in **Figure 1**.

Figure 1 Experimental set up.

Preparation and characterization of test fuels

Transesterification of vegetable oil

The most popular method of producing biodiesel is transesterification. In this process as shown in Eq. (24) the methyl esters and glycerin are formed through the reaction of fatty acids (or triglycerides) with methanol in the presence of a catalyst such as sodium hydroxide (NaOH) or potassium hydroxide (KOH). The vegetable oil is transesterified. The parameter involved in the processing such as catalyst amount, molar ratio of alcohol to oil, reaction temperature and reaction time were optimized.

RCOOR' + R" OH ↔ RCOOR" + R' OH (24)
Vegetable oil + Methanol → Methyl Ester + Glycerol

Diethyl Ether as Oxygenated fuel

Diethyl ether can be easily be converted by dehydration of ethanol. It is oxygenated (21 % by weight) and has a very low self-ignition temperature (160 °C). Therefore it is a compatible fuel for use in the CI engine.

Results and discussion

Brake specific fuel consumption

The variation in BSFC with load for different blends is presented in **Figure 2**. The BSFC, in general, was found to increase with increasing proportions of DEE in the fuel blends, whereas it decreased sharply with an increase in load for all blends. The main reason for this could be that the percent increase in fuel required to operate the engine is less than the percent increase in brake power due to relatively less portion of heat losses at higher loads or more specifically an increase in mechanical efficiency. Also the higher densities of biodiesel blends caused higher mass injection for the same volume at the same injection pressure as compared to diesel. However, the rate of decrease in brake specific fuel consumption is more during lower loads up to 40 % than that of higher loads (40 to 100 %). Similar trends of BSFC with increasing load in different biodiesel blends were also reported by Raheman and Ghadge [16].

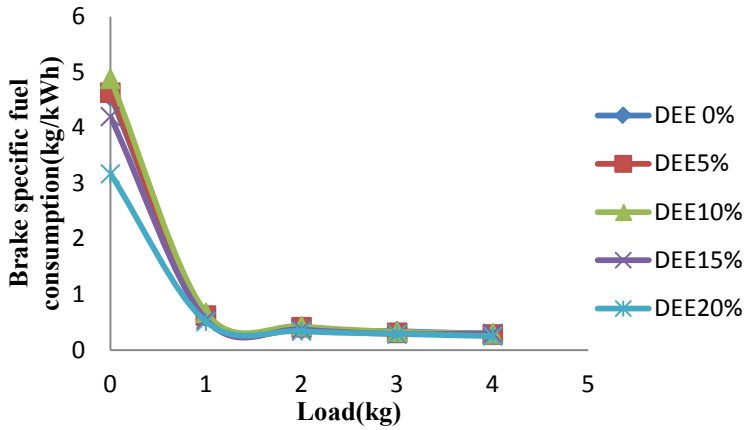

Figure 2 Variation of brake specific fuel consumption with load for Mahua biodiesel and its blends.

Brake thermal efficiency

It is evident from **Figure 3** that the overall trends of BTE characteristics of Mahua biodiesel and their blends are almost similar in nature. The maximum brake thermal efficiency of the engine obtained with biodiesel at full load is about 30 % when using 20 % DEE blend. But it remains almost constant for all blends of DEE at full load. At medium loads, 15 % DEE and blend gives higher brake thermal efficiency. This lower BTE of Mahua biodiesel other than diesel operation is due to the combined effect of higher viscosity, higher density and lower calorific value of Mahua biodiesel. Based on these results it can be concluded that the performance of the engine with biodiesel blends is comparable to that with diesel in terms of brake thermal efficiency. This is in line with the findings reported by Puhan *et al.* [17] while fueling diesel engines with biodiesels obtained from Mahua.

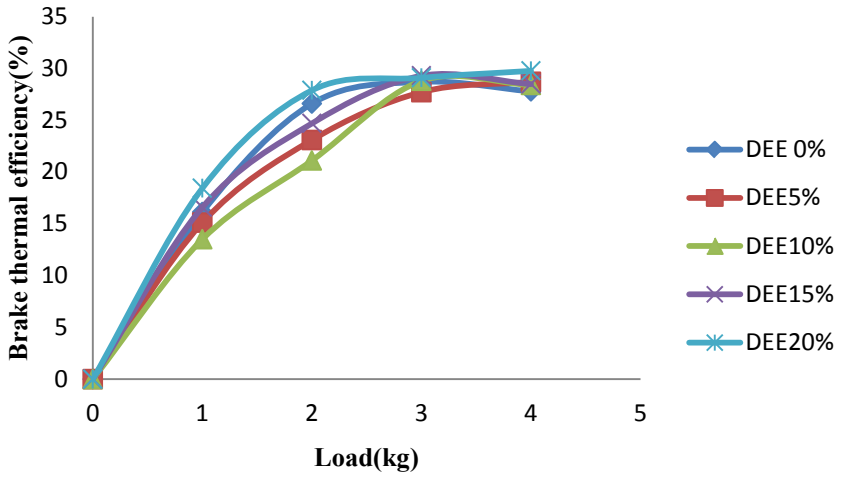

Figure 3 Variation of brake thermal efficiency with load for Mahua biodiesel and its blends.

Exhaust gas temperature

An exhaust gas temperature increase for all the test fuels with increase in load is shown in **Figure 4**. It is seen that MOME indicates higher exhaust gas temperature at all power output. The maximum exhaust gas temperature at peak power output is 186 °C with neat MOME. The higher viscosity is the reason for poor atomization of MOME and the result is a poor mixture formation of neat MOME which leads to poor combustion due to higher ignition delay. As a result, fuel is burned continuously even during exhaust, due to its heavier molecules, in later stages and causes higher exhaust gas temperature. The blending of DEE with diesel in different volumetric proportions reduces the exhaust gas temperature as diethyl ether has a higher latent heat of evaporation value than diesel fuels. The exhaust gas temperature is reduced to 178 °C with 15 %DEE and biodiesel blend. It was also observed that there was no significant reduction in exhaust gas temperature operated on MOME with DEE beyond 15 %.

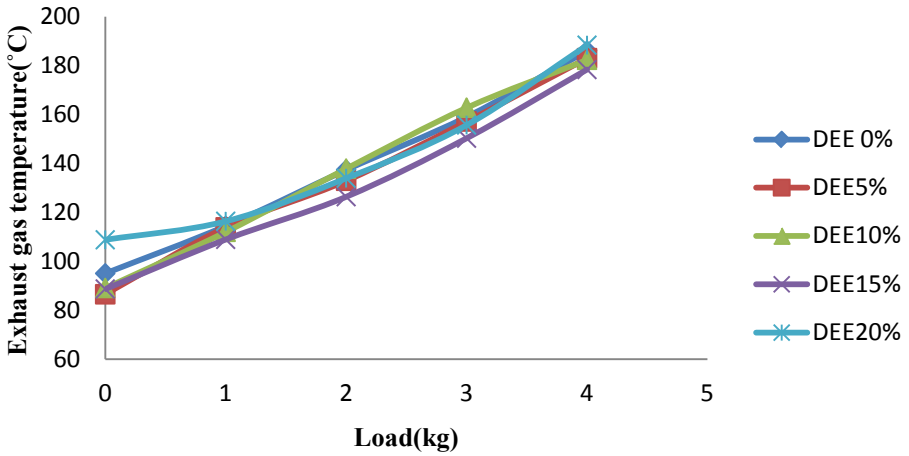

Figure 4 Variation of exhaust gas temperature with load for Mahua biodiesel and its blends.

Smoke emissions

Figure 5 shows that the smoke opacity increases with respect to load of biodiesel blends with diethyl ether. This may be due to heavier molecular structure, double bonds in vegetable oil chemical structure and higher viscosity of Mahua biodiesel and their blends. The smoke level at the maximum power output is 4.2 BSU (Bosch Smoke Units) with the pure Mahua oil. The smoke level with the diesel is 3.7 BSU at maximum power. At full load, for biodiesel blends it is suppressed for 20 % Diethyl ether-biodiesel (DEE-BD) blends. This may be due to the high cetane number of DEE which results in shorter ignition delay. Shorter ignition delay reduces the accumulation of fuel in the combustion chamber and better flame propagation and the result is a reduction in smoke emission.

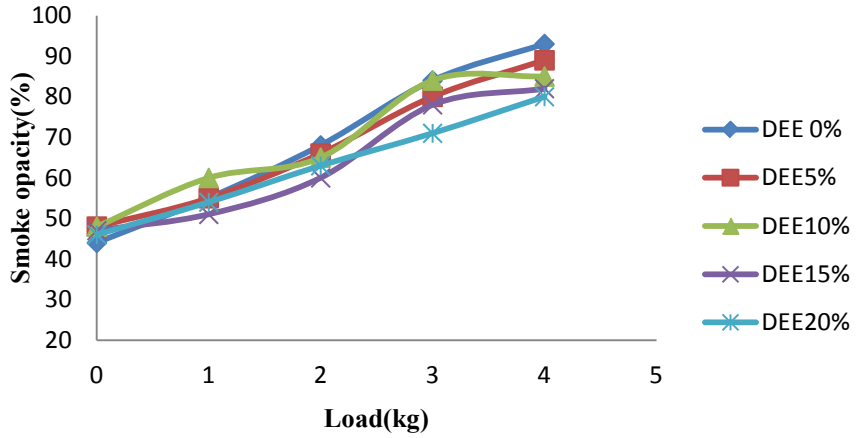

Figure 5 Variation of smoke levels with load for Mahua biodiesel and its blends.

NO_x emissions

As shown in **Figure 6** at full load on the engine, the temperature of the combustion chamber is higher so the NO_x emission is highest. DEE blending into biodiesel reduces the temperature both by increased latent heat of vaporization and reduced flame temperature and the result is a reduction in NO_x emissions. It has been observed that DEE addition to all of the blended fuels results in a reduction in NO_x emissions. The NO_x emission are 713 ppm (at least) for 20 %DEE-BD blends at full load. However, the higher viscosity and density of biodiesel caused delayed the combustion phase which results in slower combustion characteristics of Mahaua biodiesel.

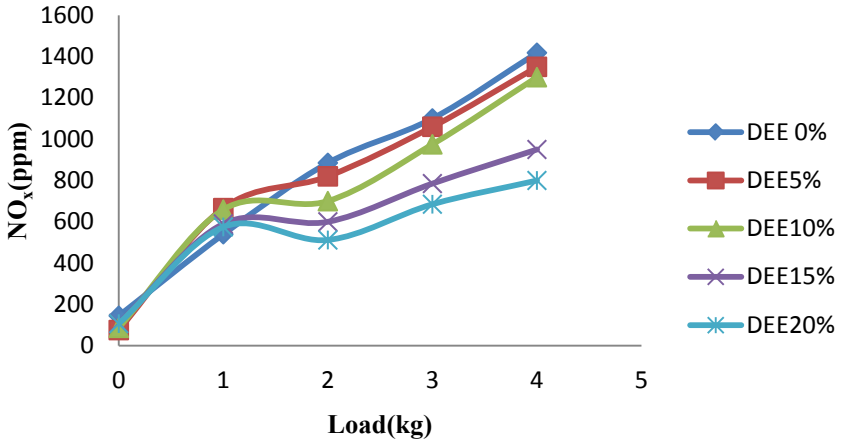

Figure 6 Variation of NO_x emission with load for Mahua biodiesel and its blends.

HC emissions

The effect of load on hydro-carbon (HC) emissions for Mahua biodiesel and their blends is shown in **Figure 7**. The HC emission is 100 ppm with diesel and 135 ppm with neat Mahua oil. It can be observed from the figure that lower HC emissions were obtained with blends of Mahua biodiesel. Lower HC emissions in the exhaust gas of the engine may be attributed to the efficient combustion of Mahua biodiesel and blends due to the presence of fuel bound oxygen and warmed-up conditions at higher loads. Whereas at lower loads conditions, higher HC emissions were observed with Mahua biodiesel blends. This is due to the fact that at lower loads the lower cylinder pressure, lower temperature and higher heat of evaporation of DEE blends tends to produce slow vaporization, flame quenching and poorer fuel-air mixing which results in incomplete combustion of the mixture. This feature results in higher HC emissions at lower load conditions.

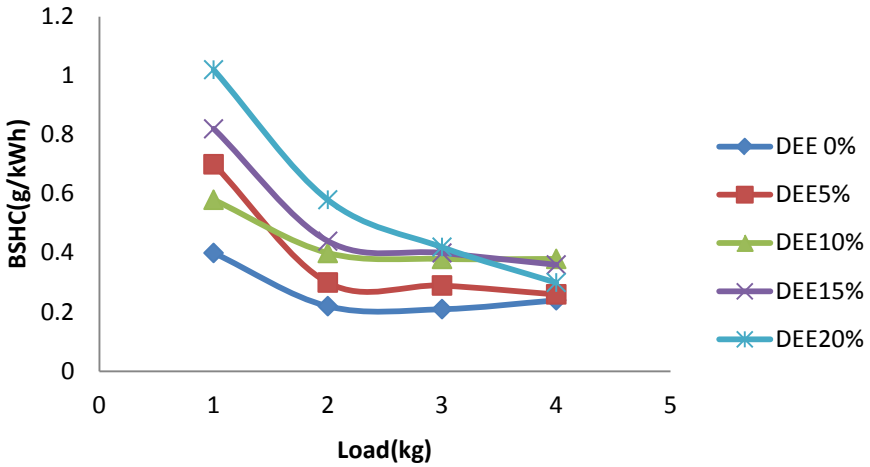

Figure 7 Variation of HC emission with load for Mahua biodiesel and its blends.

CO emissions

From **Figure** 8 it can be observed that CO emissions are reduced slightly with increasing load but at higher loads it increases for all the blended fuels. At lower load conditions the cylinder temperatures might be too low, which increase with loading due to more fuel injected inside the cylinder. As the temperature increases, performance of the engine improved with relatively better burning of the fuel resulting in decreased CO reduction. At full load, for the 20 % DEE-BD blend, CO emissions are higher. The reason behind this type of behavior is due to low volatility, which affects the atomization process and mixing of air and fuel causing locally rich mixture, which leads to difficulties in atomization and vaporization of Mahua biodiesel, so an improper spray pattern is produced. This feature increases incomplete combustion and hence higher CO emission.

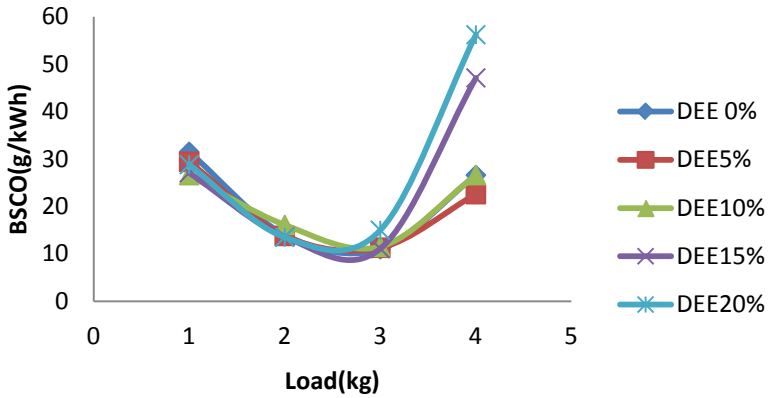

Figure 8 Variation of CO emission with load for Mahua biodiesel and its blends.

Comparison of theoretical and experimental results

The brake thermal efficiency of the compression ignition engine fueled by Mahua biodiesel and their blends is compared with that obtained from the theoretical model and experimental as shown in **Figure 9**. The brake thermal efficiency of the diesel engine is slightly higher in the Mahua biodiesel fueled engine. The brake thermal efficiency of the engine decreases when using a lower percentage of DEE. The calorific value of Mahua biodiesel is lower than that of diesel (by about 11 %) because of the presence of oxygen in the molecule. Hence, the brake thermal efficiency of Mahua biodiesel is lower as compared to that of a diesel fueled engine. The variation in experimental and theoretical results may be due to the fact that in the theoretical model a homogeneous mixture with complete combustion is assumed. But in general, it is difficult to attain complete combustion. Despite the simplification resulting from the assumed hypothesis and empirical relations, the developed simulation proved to be reliable and adequate for the proposed objectives.

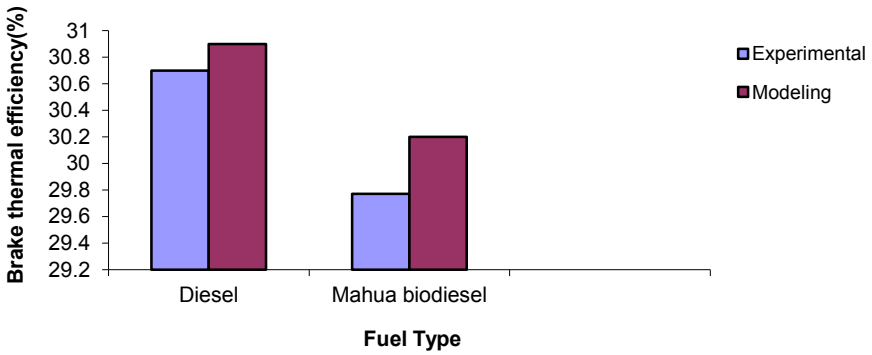

Figure 9 Comparison of brake thermal efficiency.

Conclusions

A thermodynamics existing model was modified for analyzing the performance characteristics of the compression ignition engine. The model was modified in such a way that it can be used for characterizing any hydrocarbon-fueled engines, viz., diesel or biodiesel or their blends. The performance characteristics of the engine follow the same trend for all fuels. The predicted results are compared with the experimental results of the engine fueled by diesel, DEE 0 %, DEE 5 %, DEE 10 %, DEE 15 % and DEE 20 %. This model predicted the engine performance characteristics in closer approximation to that of experimental results. The maximum brake thermal efficiency of the engine is 30 % with 20 % DEE biodiesel blend but it almost remained constant for all DEE-diesel blends at full load. At medium loads, 15 % DEE blend gives higher brake thermal efficiency. Smoke emissions of the engine are suppressed with 5 % DEE at full load while for biodiesel blends it is reduced for 20 %DEE-BD blends. DEE blends for diesel as well as for biodiesel reduce carbon monoxide emissions. The HC emissions have a moderate increase for all DEE addition to base fuels at lower loads but it increases marginally at higher loads. DEE blending into biodiesel reduces NO_x emissions. It is concluded that DEE can be blended with biodiesel to improve the combustion process. Hence, it is concluded that this model can be used for the prediction of the performance characteristics of the compression ignition engine fueled by any type of hydrocarbon fuel.

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