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An Experimental and Numerical Investigation of the Performance, Combustion and Emission Characteristics of a Diesel Engine fueled with Jatropha Biodiesel

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Abstract

The effect of addition of jatropha biodiesel to mineral diesel on the performance and emission characteristics of a conventional compression ignition engine have been experimentally investigated and compared with simulated data using Diesel-RK software. The experiments were carried out using pure diesel (B0) and pure jatropha biodiesel (JB100) as fuels. The performance characteristics shows that brake specific fuel consumption (BSFC) increases and brake thermal efficiency decreases with the use of jatropha biodiesel. Experimentally, pure diesel has maximum efficiency 29.6%, where as pure biodiesel has maximum efficiency of 21.2%. In the simulation, the pure diesel has maximum efficiency 30.3% where as pure jatropha biodiesel has the maximum efficiency of 27.5%. In respect of emission characteristics, NO_x emission is found to increase with load as well as use of biodiesel in both experimental and simulation study. After the successful validation of the numerical study with the experimental, another simulation was done, where the performance, combustion and emission characteristics of the same engine fueled with pure diesel (B0), pure jatropha biodiesel (JB100) and 50% jatropha blend (JB50) were derived. In the numerical study it is found that, with the use of jatropha biodiesel the BSFC increases whereas brake thermal efficiency decreases. Combustion characteristics show an increase in peak cylinder pressure and a decrease in ignition delay period with the increase in biodiesel share in the blends; whereas the emission of NO_x and CO_2 increases; smoke and PM emission decreases for the same.

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Keywords: compression ignition engine, jatropha biodiesel, BSFC, thermal efficiency, NOx.

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1. Introduction

The constant increase in the rate of consumption of the fossil fuels, consequent upon the ever increasing population and the urbanization in the present day world, has made the depletion of these conventional fuel resources in the near future a quite inevitable fact. Also, the Greenhouse Gas emissions from these fossil fuels are constantly degrading the planet and causing global warming and other pollutant emission related problem. As such, the situation demands for an alternate source of energy that can be used to overcome the forecasted future energy crisis. In addition to this, if the energy source is clean and renewable, it will reduce the environmental issues as well. In this quest for an alternate and renewable energy resource, scientists have come up with a variety of options among which biodiesel-diesel blends as alternative fuels has become a popular option and is gaining the attention of many researchers. This is because scientists have seen that the properties of biodiesel prepared from vegetable oils are very close to commercial diesel and thus it has a promising future as an alternative fuel for diesel engine [1, 2]. Biodiesel being renewable, biodegradable and green fuel can reduce our dependence on conventional/ non-renewable fossil fuels as well as improve environmental quality in metro cities, urban and rural sectors by reducing obnoxious automotive/vehicular emissions [3, 4]. As such biodiesel has the potential to replace petroleum diesel in near future.

Generally, biodiesels are fatty acid esters produced from vegetable oils or animal fats [5] through a chemical process known as transesterification. The differences in the composition and properties of biodiesels produced from soyabean oil, rapeseed oil, karanja oil, jatropha oil or animal fats, from pure diesel will influence the engine performance, combustion and also the emission characteristics. Dwivedi *et al.* [6] experimentally observed that the increase in the content of biodiesel in diesel-biodiesel blend decreases engine power. As reported by Xue *et al.* [7], this loss in engine power with the use of biodiesel is mainly due to the reduction in heating value of biodiesel compared to diesel. The same reason can be accounted for the increase in the brake specific fuel consumption. On the other hand, Canaki and Van Garpen [1] stated that compared to the fossil diesel fuel, biodiesel improves thermal efficiency as it gets injected earlier, resulting in an earlier start of combustion. Also, the shorter delay time of fuel combustion due to the higher cetane number of biodiesel provides more time for complete combustion [8, 9]. However, the low calorific value and high viscosity of bio-fuels again tend to decrease the thermal efficiency [6]. Biodiesel and its blend have larger cetane number than that of diesel, resulting in earlier combustion [10]. Due to this difference in cetane number, the use of biodiesels decreases the ignition delay period compared to pure diesel. The higher cetane number and the reduced ignition delay for the biodiesels tend to increase the in cylinder pressure [11]. The higher oxygen content in biodiesels, leading to improved combustion may be another reason for this [12, 13]. In comparison with conventional diesel fuels, biodiesels promote more complete combustion and thus effectively reduce emissions of particulate matter (PM), carbon monoxide (CO) and smoke [14, 15]. However, the use of biodiesel increases the content of NO_x in the combustion products [16, 17]. This higher NO_x emission is due to the comparatively high temperature inside the cylinder owing to the combustion of biodiesel and higher oxygen content of the fuel as explained by Mustapic [18].

The literature is rich in experimental works on biodiesel. Numerical predictions are sometimes preferred due to difficulties involved in experimental works. But the works available in the literature is limited. The authors have attempted to simulate a biodiesel-diesel fuelled compression ignition engine using commercially available software, Diesel-RK. The simulations predict performance as well as combustion and emission characteristics of the biodiesel fuelled engine. Experiments have also been carried out on a double cylinder constant speed direct injection CI engine to evaluate some performance and emission parameters for comparison with the predicted results obtained from using Diesel-RK.

Nomenclature

$\dot{m}_j$	mass flow rate of j^{th} species
m	total mass within the control cylinder
Y_i^j	stoichiometric coefficients on the product side
Y_i^{cyl}	stoichiometric coefficients on the reactant side
Ω_i	dimensionless integral of order unity dependent on the force interaction during a collision of i^{th} species
P	cylinder pressure

T_z	temperature in a burnt gas zone
R	gas constant
ω	angular crank velocity
m_a	intake air flow rate
m_f	intake fuel flow rate
T	enginw torque
V_c	piston displaced volume per cylinder
m_f	fuel mass flow rate into the engine
ρ_a	air density
N	engine speed
n	no of revolutions of crankshaft per cycle
$[NO]_e$	equilibrium concentrations of oxide of nitrogen
$[N_2]_e$	equilibrium concentrations of molecular nitrogen
$[O]_e$	equilibrium concentrations of atomic oxygen
$[O_2]_e$	equilibrium concentrations of molecular oxygen
V	current volume of cylinder
q_c	cycle fuel mass
dx/dt	heat release rate

2. Simulation model

The numerical simulation is carried using the commercial software Diesel-RK. The basic governing equations needed for the simulation of the software are taken from the previous work of Hamdan and Khalil [19] and are given below.

Conservation of mass: The rate of change of mass within any open system is the net flux of mass across the system boundaries. Mathematically, it can be written as

$$\frac{dm}{dt} = \sum_j \dot{m}_j \tag{1}$$

Conservation of species: Equations tracking the evolution of species within the combustion chamber will be developed on a mass basis. The mass balance of individual species can be expressed as

$$\dot{Y}_j = \sum_j \left(\frac{\dot{m}_j}{m}\right)\left(Y_i^j - Y_i^{cyl}\right) + \frac{\Omega_i W_{mw}}{\rho} \tag{2}$$

Conservation of energy: The generalized energy equation for an open thermodynamic system may be written as

$$\underbrace{\frac{d(mu)}{dt}}_{Internal\,Energy} = \underbrace{-p\frac{dv}{dt}}_{Displacement\,Work} + \underbrace{\frac{dQ_{ht}}{dt}}_{Heat\,Transfer} + \underbrace{\sum_j \dot{m}_j h_j}_{Enthalpy\,Flux} \tag{3}$$

In the following, the equations for the cylinder pressure and temperature(s) are derived. This will show where additional information, in the form of sub-models, is necessary in order to close these equations. Before conservation of energy is written out for the cylinder volume, from inlet valve closing time to exhaust valve opening time, some assumptions are generally adopted to simplify the equations. During compression and expansion, pressure is invariably assumed uniform throughout the cylinder, with fixed unburned and burned gas regions in chemical equilibrium. During flame propagation, burned and unburned zones are assumed to be separated by an infinitely thin flame front, with no heat exchange between the two zones. All gases are considered ideal gases;

possible invalidity of the ideal gas law at high pressures is countered by the associated high temperatures under engine combustion conditions.

The parameters, which were calculated to find the performance of the engine, are by the Diesel-RK: brake power, brake mean effective pressure, brake torque, specific fuel consumption volumetric efficiency. These parameters were calculated for each blend ratio and at different engine speeds. Air/fuel mixture equivalence ratio, λ, is defined by the relationship between the actual air/fuel ratio and the stoichiometric air/ fuel ratio:

$$\lambda = \frac{(A/F)}{(A/F)_S} = \frac{(m_a/m_f)}{(\dot{m}_a/\dot{m}_f)_S} \tag{4}$$

The stoichiometric air/fuel mixture contains the necessary air amount to fully burn the fuel. Brake power (P_b) is given by the product of engine torque (T) and rotational speed (ω):

$$P_b = T.\omega \tag{5}$$

The torque is obtained by the vector product of the load applied and the dynamometer arm length. Brake mean effective pressure is a measure of the power produced per cycle as a function of engine size:

$$BMEP = \frac{T}{V_C} = \frac{P_b}{V_C\omega} \tag{6}$$

Specific fuel consumption, SFC, is the fuel amount consumed per unit of power produced, that is:

$$SFC = \frac{\dot{m}_f}{P_b} \tag{7}$$

The engine volumetric efficiency η_V refers to the efficiency with which the engine can move the charge into and out of the cylinders. More specifically, volumetric efficiency is a ratio (or percentage) of what quantity of fuel and air actually enters the cylinder during induction to the actual capacity of the cylinder under static conditions. Volumetric efficiency is defined as:

$$\eta_V = \frac{nm_a}{\rho_a V_C N} \tag{8}$$

The emission parameters like NO_x formation, soot emission, PM and CO_2 emissions are also simulated by this software using different models and empirical relations available in the literature.

NO_x formation Modelling: the oxides of nitrogen like nitric oxide (NO) and nitrogen dioxide (NO_2) are usually grouped together and called NO_x . Out of all the oxides of nitrogen, NO is predominant in diesel engine [20]. Therefore; only NO formation is considered. NO can also be formed through different mechanisms, but the model used here has taken care of NO formation through only thermal or Zeldovich mechanism. Similar NO formation model has been used by Kuleshov [21] for simulation of direct injection diesel engine. The model first calculates step by step equilibrium composition of combustion products for eighteen species in the burnt gas zone followed by the kinetic calculation of thermal NO following Zeldovich mechanism. The oxidizing of nitrogen is on the chain mechanism and basic reactions are as follows:

$$\begin{aligned} & O_2 \leftrightarrow 2O \\ & N_2 + O \leftrightarrow NO + N \\ & N + O_2 \leftrightarrow NO + O \end{aligned} \tag{9}$$

Rate of this reaction depends on concentration of atomic oxygen. The volume concentration of NO in combustion products formed in a current calculation step is defined using the following equation:

$$\frac{d[NO]}{d\theta} = \frac{p.2.333*10^7.e^{-\frac{38020}{T_Z}}[N_2]_e.[O]_e.\left\{1-\left([NO]/[NO]_e\right)^2\right\}}{R.T_Z.\left(1+\frac{2365}{T_z}.e^{\frac{3365}{T_z}}.\frac{[NO]}{[O_2]_e}\right)} \cdot \frac{1}{\omega} \tag{10}$$

Soot Formation Modelling: Soot is a fine dispersion of black carbon particles in a vapour carrier. It is assumed, the soot is formed mainly by two ways:

- As a result of chain destructive transformation of molecules of fuel diffusing from the surface of drops to the front of a flame.
- Owing to high-temperature thermal polymerization and dehydrogenization of a vapour-liquid core of evaporating drops.

In parallel to this, the process of burning of soot particles and reduction of their volumetric concentration owing to expansion occurs.
Soot formation rate in a burning zone is calculated as:

$$\left(\frac{d[C]}{dt}\right)_K = 0.004\frac{q_C}{V}\frac{dx}{dt} \tag{11}$$

Hartige smoke level is derived from the following relation:

$$Hartige = 100[1-0.9545\exp(-2.4226[C])] \tag{12}$$

Similar equations are used to calculate Bosch smoke number and Factor of Absolute Light Absorption (*K*).

Particulate Matter Modelling: Particulate matter emission is calculated by using an equation in terms of Bosch smoke number as given in the work of Alkidas [22]. The equation is as follows:

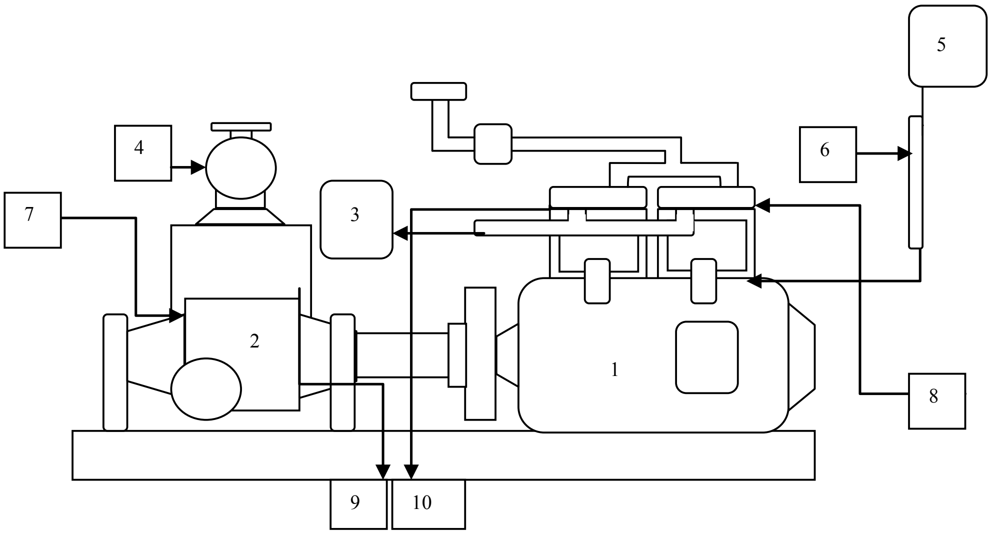
$$[PM] = 565\left(\ln\frac{10}{10-Bosch}\right)^{1.206} \tag{13}$$

3. Experimental setup

The experiment is carried out on a double cylinder, constant speed, direct injection diesel (compression ignition) engine in the IC Engine laboratory of Bengal Engineering and Science University, Shibpur. The specifications of the engine used for the experiments are given in table 1. The schematic diagram of the experimental set up is shown in figure 1. The engine is directly coupled to a hydraulic dynamometer of maximum load capacity 20 kgf. The experiment is conducted at a rated speed of 1500 rpm. The load is varied by adjusting load wheel on the top of the engine. Water pressure is kept constant at 1.5 kg/cm^2. The torque and the fuel consumption rates are measured for different loads and fuels. The exhaust gas temperatures were measured using thermocouple. The amount of different pollutants in the exhaust gases was directly measured with the help of a gas analyzer, which measured the NO_x emission.

Table 1 Engine Specifications

Ignition type	CI (4-stroke)
No of cylinder	2
Manufacture	Kirloskar
Type	AV2
Engine No	11.1001/81801
RPM	1500
BHP	10 (7.35KW)
SFC	199 G/bhp-hr
Bore Diameter	80 mm.
Stroke Length	110 mm.



1. Engine, 2. Hydraulic dynamometer, 3. Exhaust Gas Analyser, 4. Loading Unit, 5. Fuel Tank, 6. Measuring Burette, 7. Inlet water for dynamometer, 8. Inlet water for engine, 9. Water outlet from dynamometer, 10. Water outlet from engine

Fig. 1. Schematic diagram of the experimental setup

4. Properties of fuels used

The performance and emission characteristics of biodiesel fuelled engine depend purely upon the thermo physical properties of the biodiesel. Since jatropha oil can be grown in wastelands, its cultivation to biodiesel production can also generate a large scale employment in a country like India. Basically the biodiesels are derived from vegetable oils via a popular process, transesterification in the presence of a catalyst and alcohol as a reactant. Due to the availability and cost factor methyl alcohol is commonly used and the derived biodiesel is also known as fatty acid methyl ester. The purpose of the transesterification process is to lower the viscosity of the oil. Ideally, transesterification is potentially a less expensive way of transforming the large, branched molecular structure of the bio-oils into smaller, straight chain molecules of the type required in regular diesel combustion engines. It contains very small amount of phosphorus and sulphur and therefore emission of oxides of sulphur (SO_x) is almost negligible. Jatropha biodiesel has higher density and cetane number compared to conventional diesel. The higher flash point and fire point of jatropha biodiesel makes it more convenient for storage and transportation and much safer than mineral diesel. The amount of carbon residue from the hot decomposition of vegetable compounds with higher molecular weight is greater than that of commercial diesel oil. The thermo-physical properties of jatropha biodiesel and conventional petro-diesel are presented in the tabular form for ready reference in table 2 for comparison. The required properties of the different fuel blends have been calculated on weighted average method for the calculation from the experimental results.

Table 2 Properties of Jatropha Biodiesel and Diesel

Property	Jatropha Biodiesel	Diesel
Mass Fractions (%)		
Carbon	0.766	0.87
Hydrogen	0.121	0.126
Oxygen	0.113	0.004
Sulfur Fraction (%)	0.0024	0
Low Heating Value (MJ/kg)	30.5	39
Apparent Activation Energy (kJ/mol)	14.73	23
Cetane Number	53	48
Density at 323K (kg/m^3)	862	830
Dynamic Viscosity coefficient (Pa.s)	0.0057	0.003
Molecular Mass	282	190

5. Results and discussions

An alternative fuel used in engines is always evaluated on the basis of both engine performances and its environmental impacts. As such, various parameters defining the performance and emissions of diesel engine which have been evaluated both experimentally and numerically in this work and have been discussed and analysed in this section.

5.1. Validation and numerically simulated result with experimental ones

In this section, the simulated results are verified with experimental results at the same operating conditions. The validation is done with two extreme fuels, namely pure jatropha biodiesel (B100) and mineral Diesel (B0). Mainly three parameters are considered for validation and the parameters are brake thermal efficiency, brake specific fuel consumption and NO_x emission.

5.1.1. Brake thermal efficiency

Brake thermal efficiency is one of the most important engine performance parameter which indicates the percentage of fuel energy converted to useful power output. Figure 2 represents the variation of brake thermal efficiency of diesel with jatropha biodiesel. From the figure, it is observed that Diesel-RK predicts higher efficiency for both the fuels, i.e. diesel (B0) and jatropha biodiesel (JB100). At low load condition (around 2 kW), efficiency values are found to be 20.27 % and 14.6 % from simulation and experiment respectively for diesel fuel. At same operating load condition, the corresponding values are found to be 18% and 10.3% respectively for jatropha biodiesel (JB100). At higher load condition the simulated value and experimental results with diesel is close enough, 29.69% and 28.65 % respectively, where as there is a variation is case of jatropha, 27.45% and 21% in simulated value and experimental result respectively.

5.1.2. Brake specific fuel consumption

Brake specific fuel consumption (BSFC) is also a vital factor for evaluating the performance of an engine fuelled by alternative or supplementary fuels. Figure 3 graphically represents the variation of brake specific fuel consumption of the engine with brake power for diesel and jatropha biodiesel. From the graph, it is noted that the experimental values of BSFC are higher than the predicted values for both the fuels. It can be the result of lower heating value of jatropha biodiesel which is much lower than mineral diesel. At low load condition, the simulated BSFC value is 0.4209 kg/kWh where as the experimental value is 0.531 kg/kWh for diesel. At same operating condition, the simulated value is 0.60 kg/kWh, where as the experimental value is 0.73 kg/kWh for jatropha biodiesel. At higher load condition the simulated value and experimental results with jatropha biodiesel and diesel are close enough, 0.4529 kg/kWh and 0.49 kg/kWh for jatropha biodiesel and 0.3229 kg/kWh and 0.313 kg/kWh respectively.

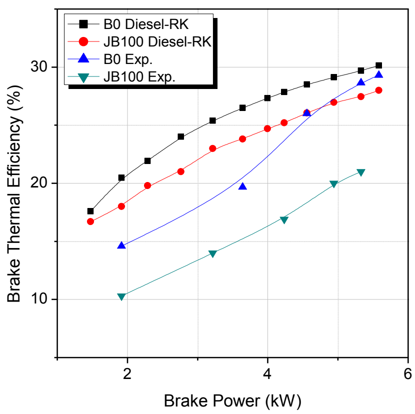


Fig. 2. Comparison of brake thermal efficiency of simulated and experimental data for B0 and JB 100.

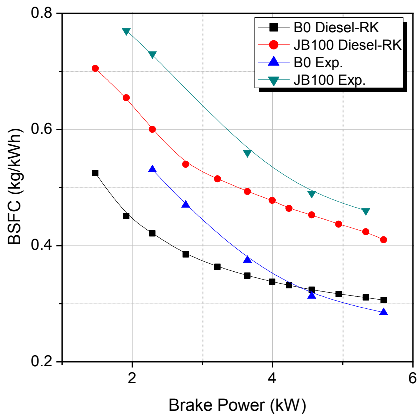


Fig. 3. Comparison of BSFC of simulated and experimental data for B0 and JB 100.

5.1.3. NO_x emission

The only environmental parameter validated in this work is NO_x emission. Figure 4 represents the variation of NO_x emission of diesel with jatropha biodiesel. A close look at the above said figure reveals that Diesel-RK predicts slightly higher NO_x emission for both the fuels. The maximum values of the predicted result and experiment result for NO_x emission are 220.77 ppm and 290 ppm respectively for diesel fuel. At same operating condition, the simulated value is 300 ppm whereas the experimental value is 470 ppm when fuelled with jatropha biodiesel. At higher load condition the simulated value and experimental results with diesel is close enough, i.e., 622.72 ppm and 610 ppm respectively, where as there is a slight variation is case of jatropha, 794.4 ppm and 840 ppm in simulated value and experimental result respectively. This clearly establishes the fact that the NO_x emission is more with jatropha biodiesel at all load conditions.

Hence form the above study it can be said that Diesel-RK gives us a realistic results. The simulated results match well qualitatively with the experimental results for the parameters compared here. The quantitative matching is not always very good, but it is able to establish the proper trends. The variation can be the results of the difference between the 3D and 1D modeling and the old engine designed for diesel fuel where experiments have been carried out. In fact, Diesel–RK uses 1D model and the experimental results are really 3D in nature, i.e., the actual real situation. Also, other environmental conditions and some uncertainties in some input values of the software can also be responsible for the deviation between the experimental and predicted results.

5.2. Numerically simulated results depicting the performance and emissions of the diesel engine

Performance and emission characteristics like brake thermal efficiency, brake torque, brake specific fuel consumption, NO_x, particulate matters, CO_2, smoke emissions have been numerically simulated and their variations have been plotted against brake power for three fuels namely B0, JB50 and JB100.

5.2.1. Brake torque

Biodiesel has lower heating value than mineral diesel and hence produces slightly less power [23].Thus, when the variation of brake torque for the three blends of jatropha biodiesel, viz. B0 or pure diesel, JB50 and JB100 are plotted against brake power, as shown in figure 5, it can be seen that the brake torque decreases with the increase in biodiesel percentage. This is due to the fact that the heat content of biofuel is lower than that of pure diesel, and hence the blends formed have lower heating values than that of the pure diesel.

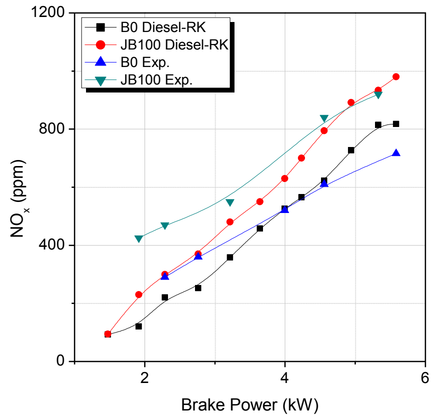


Fig. 4. Comparison of NO_x of simulated and experimental data for B0 and JB 100.

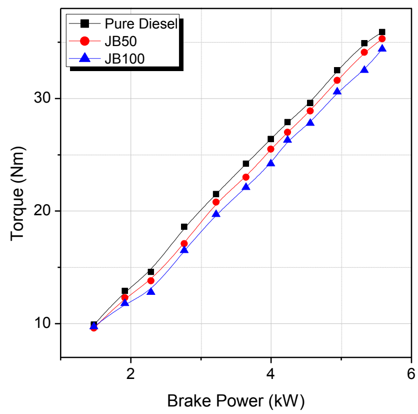


Fig. 5. Comparison of brake torque for various blends of jatropha biodiesel

5.2.2. Brake thermal efficiency

As mentioned earlier, biodiesel gets injected earlier than pure diesel causing earlier combustion of biodiesel. Moreover, the higher cetane number of biodiesel causes shorter delay time for fuel combustion, providing more time for complete combustion which improves the thermal efficiency to some extent. On the other hand, the low calorific value and high viscosity causing the poor atomization of biofuel in the engine cylinder tend to decrease thermal efficiency [6]. The simulated results, shown in figure 6, depict a decrease in the brake thermal efficiency with the increase in the biodiesel percentage in the blends. Hence, it can be concluded that the decrease in the heating value may be the dominating factor owing to the reduction in the thermal efficiency.

5.2.3. Brake specific fuel consumption

Figure 7 shows the variation of brake specific fuel consumption with brake power for the different fuels. It is seen that the brake specific fuel consumption for a diesel engine increases with the use of biodiesel. It has been stated in the earlier section that the calorific value of biofuels is lower than that of pure diesel and this fact justifies the above mentioned trend in the BSFC.

5.2.4. Cylinder pressure vs Crank angle

From figure 8 it can be seen that the use of jatropha biodiesel and its blend increases the in-cylinder peak pressure. Also, the point of peak pressure and the start of combustion shifts towards left compared to pure diesel. As such it can be said that the combustion starts earlier and the ignition delay period decreases with the use of biodiesel. Thus, the lowest peak pressure and highest ignition delay period is that of pure diesel followed by the blend (JB50) and finally pure jatropha (JB100) which shows highest peak pressure rise and the lowest delay period. Lower delay period results in earlier combustion, which gives rise to higher temperature and pressure rise in the cylinder. Also, the higher oxygen content of biodiesels resulting in better combustion may also be another reason for biodiesel showing higher temperature and pressure rise than diesel.

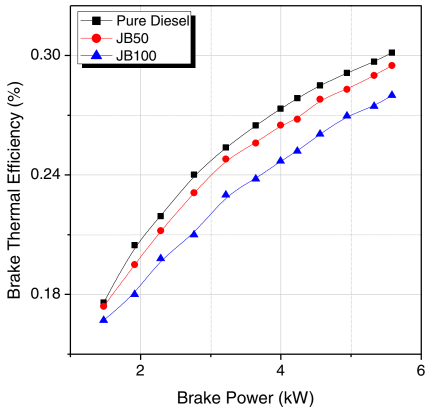


Fig. 6. Comparison of brake thermal efficiency for various blends of jatropha biodiesel

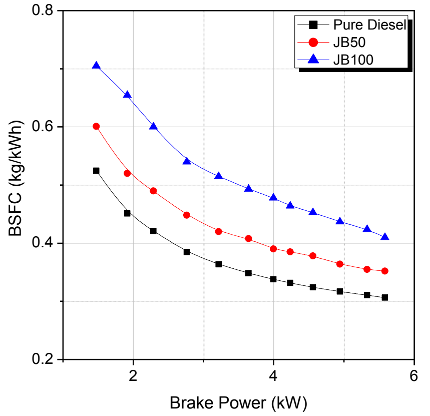


Fig. 7. Comparison of BSFC for various blends of jatropha biodiesel

5.2.5. Cylinder pressure vs volume

Figure 9 depicts the variation of cylinder pressure with cylinder volume for the different fuels. It can be seen that the pressure increases with the use of biodiesel compared to pure diesel. Again, between the fuels, pure jatropha (JB100) shows the highest pressure rise and pure diesel shows the lowest, with 50% blend (JB50) lying in between.

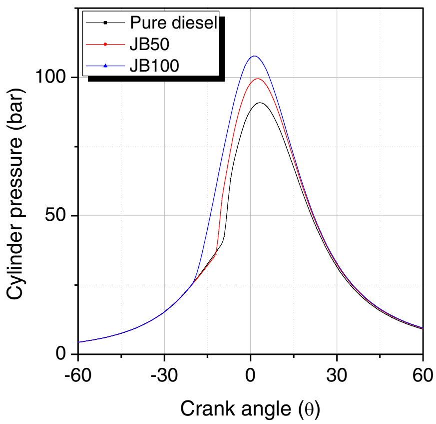


Fig. 8. Comparison of in cylinder peak pressure for various blends of jatropha biodiesel

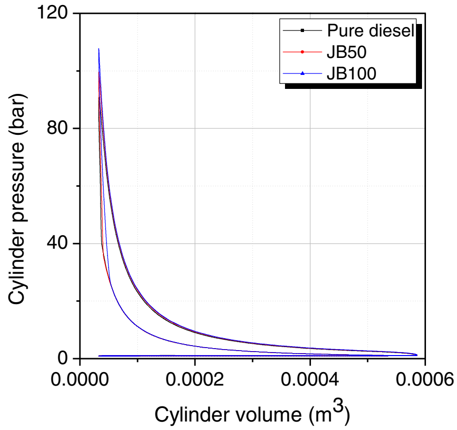


Fig. 9. Comparison of in cylinder pressure Vs volume for various blends of jatropha biodiesel

5.2.6. NO_x emission

It has been observed that the use of biodiesel in the diesel engine increases the NO_x emissions, compared to conventional diesel. Biodiesel contains higher amount of oxygen than pure diesel. Moreover, the complete combustion of biodiesels results in a higher temperature, causing the formation of valance oxygen form dissociation, which eventually increases the NO_x. As expected, the simulated results shown in figure 10, show an increase in the NO_x emission with the increase in the biodiesel content of the blended fuel and pure biodiesel.

5.2.7. CO_2 emission

It is clearly seen from the figure 11 that with the addition of biodiesel into diesel engine cause higher amount of carbon dioxide at tailpipe. The higher carbon-hydrogen ratio and presence of oxygen molecule in biodiesel causes higher CO_2 emission. It can be said that with biodiesel the peak temperature and pressure rise is higher than diesel. The higher temperature causes almost complete oxidation of carbon mono-oxide to carbon dioxide resulting in more CO_2 in the tailpipe exhaust for biodiesel. It may be noted that the net CO_2 emission or life cycle CO_2 emission is less with biodiesel despite of its higher CO_2 emission, as the bio-fuel crops absorbed CO_2 during their photosynthesis process.

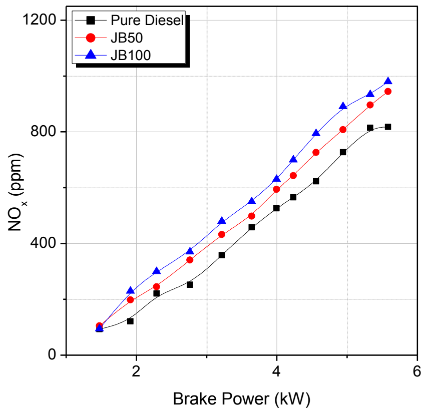


Fig. 10. Comparison of NO_x emission for various blends of jatropha biodiesel

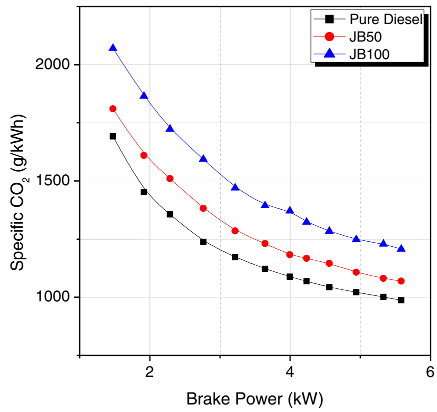


Fig.11. Comparison of specific CO_2 emission for various blends of jatropha biodiesel

5.2.8. Particulate matter and smoke emission

The variation of specific PM emission and smoke emission with brake power for the different fuels is shown in figure 12 and figure 13. The primary reason of the particulate emission (PM) from CI engine is improper combustion and combustion of heavy lubricating oil and smoke formation occurs primarily in the fuel-rich zone of the cylinder, at high temperatures and pressures [24]. The results depicted by the simulation, for the three different blends are plotted with respect to engine load. Both the PM and smoke emissions for all the blends show a significant decreasing trend initially which is due to the incomplete combustion at low loads. When compared between the blends, it can be observed that an increase in biodiesel share in the blends reduce the PM and smoke emissions which is primarily due to the complete combustion of the biodiesel, owing to the higher oxygen content in it. The higher cetane number of biodiesel can also be a reason for this.

6. Conclusion

Both experimental and numerical investigations on the performance and emission characteristics of a diesel engine, powered with various blends of biodiesel have been carried out. The following conclusions can be drawn from the analysis of the results obtained during the investigation.

- The use of jatropha biodiesel in a conventional diesel engine decreases its torque and brake thermal efficiency, the decrease being more with increase in the biodiesel share in the blends.
- BSFC increases with the percentage of biodiesel in the blended fuels.
- Cylinder peak pressure increases and ignition delay period decreases with the increase in biodiesel share in the blended fuels.
- Use of jatropha biodiesel increases the NO_x emission compared to pure diesel. This is due to the higher oxygen content of jatropha and the higher temperature obtained due to complete combustion of the biodiesel.

- An increase in the jatropha biodiesel share in the blends reduces the PM and smoke emissions which are primarily due to the complete combustion of the biodiesel, owing to the higher oxygen content in it.
- The addition of jatropha biodiesel into diesel engine causes higher amount of carbon dioxide at tailpipe. This can be explained by the higher carbon-hydrogen ratio and presence of oxygen molecule in the biodiesel.

This study shows that biodiesel can be used as alternative fuel in diesel engine. This may lead to a slight decrease in performance but improves emission significantly which is the call of the day. This study also establish the fact that the engine can be simulated using commercial software like Diesel-RK before going for actual experiments under optimized conditions which saves fuel as well as overall cost.

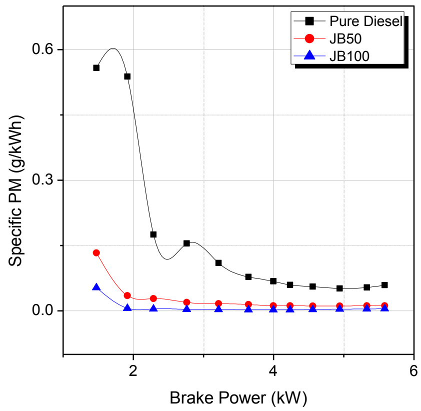


Fig. 12. Comparison of specific PM emission for various blends of jatropha biodiesel

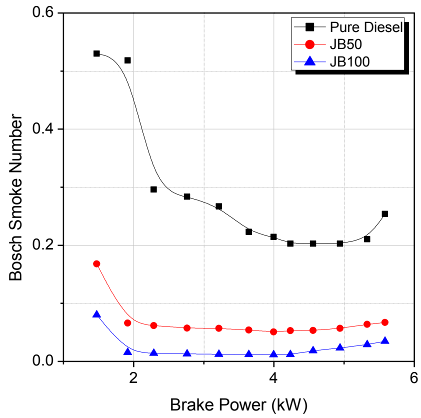


Fig. 13. Comparison of bosch smoke number for various blends of jatropha biodiesel

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References

[1] Canaki M., Van Garpen J. H., “Comparison of engine performance and emissions for petroleum diesel fuel, yellow-grease biodiesel and soybean-oil biodiesel”, Transactions of the ASAE, Vol. 46, Issue. 4, pp. 937-944, 2003.

[2] Freedman B., Pryde E. H., “Fatty esters from vegetable oils for use as a diesel fuel”, Transactions of the ASAE, pp. 117-122, 1982.

[3] Akintayo E. T., “Characteristics and composition of Parkia biglobbossa and Jatropha curcas oils and cakes”, Bioresource Technology, Vol. 92, pp. 307-310, 2004.

[4] Geyer S. M., Jacobus M. J., Lestz S. S., “Comparison of diesel engine performance and emissions from neat and transesterified vegetable oils”, Transactions of the ASAE, Vol. 27, Issue. 2, pp. 375-381, 1984.

[5] Krishnakumar J., “Technical Aspects of Biodiesel Production from Vegetable Oils”, Thermal Science, Vol. 12, Issue. 2, pp. 159-169, 2008.

[6] Dwivedi G., Jain S., Sharma M. P., “Impact of Biodiesel and its Blends with Diesel and Methanol on Engine Performance”, International Journal of Energy Science, Vol. 1, Issue. 2, pp. 105-109, 2011.

[7] Xue J., Grift T. E., Hansena A. C., “Effect of biodiesel on engine performances and emissions”, Renewable and Sustainable Energy Reviews, Vol. 15, pp. 1098–1116, 2011.

[8] Raheman H., Phadatare A. G., “Diesel Engine Emissions and Performance from Blends of Karanja Methyl Ester and Diesel”, Biomass and Bioenergy, Vol. 27, Issue. 39, pp. 393-397, 2004.

[9] Kim H., Choi B., “The Effect of Biodiesel and Bioethanol Blended Diesel Fuel on Nanoparticles and Exhaust Emission from CRDI Diesel Engine”, Renewable Energy, Vol. 35, Issue. 1, pp. 157-163, 2010.

[10] Shah A. N., Yun-shan G. E., Chao He., Baluch A. H., “Effect of Biodiesel on the Performance and Combustion Parameters of a Turbocharged Compression Ignition Engine”, Pakistan Journal of Engineering & Applied Science, Vol. 4, pp. 34-42, Janurary 2009.

[11] Abdullah N. R., Liu D., Park T., Xu H. M., Wyszynski M. L., Tsolakis A., "Effect of Variations of Biodiesel Blends on Engine Combustion and Emissions", available: http://www.birmingham.ac.uk/Documents/college-eps/mechanical/staff/xu/xu-biodesiel-blends.pdf, accessed on: 16-7-2013.
[12] Tsolakis A., Megaritis A., "Partially premixed charge compression ignition engine with onboard H_2 production by exhaust gas fuel reforming of diesel and biodiesel", International Journal of Hydrogen Energy, Vol. 30, Issue. 7, pp. 731-745, 2005.
[13] Chuepeng S., Xu H. M., Tsolakis A., Wyszynski M. L., Price P., Stone R., Hartlandand J. C., Qiao J., "Particulate Emissions from a Common Rail Fuel Injection Diesel Engine with RME based Biodiesel Blended Fuelling Using Thermogravimetric Analysis", SAE Paper 2008-01-0074, 2008, doi:10.4271/2008-01-0074.
[14] Zheng M., Mulenga M. C., Reader G. T., Wang M., Ting D.Sk., Tjong J., "Biodiesel engine performance and emissions in low temperature combustion", Fuel, Vol. 87, pp. 714–722, 2008.
[15] Çelikten I., Koca A., Arslan M. A., "Comparison of performance and emissions of diesel fuel, rapeseed and soybean oil methyl esters injected at different pressures", Renewable Energy, Vol. 35, pp. 814–820, 2010.
[16] Imdat T., Mucahit S., "Performance and Emission Characteristics of a Diesel Engine Using Esters of Palm Olein/Soybean Oil Blends", International Journal of Vehicle Design, Vol. 54, Issue. 2, pp. 177-189, 2010.
[17] Hazar H., "Effects of Biodiesel on a Low Heat Loss Diesel Engine", Renewable Energy, Vol. 24, Issue. 6, pp. 1533-1537, 2009.
[18] Mustapic Z., "Biodiesel as Alternative Engine Fuel", Energy, Vol.55, Issue. 6, pp. 634-657, 2006.
[19] Hamdan M. A., Khalil R. H., "Simulation of compression engine powered by Biofuels", Energy Conversion and Management, Vol. 51, Issue. 8, pp. 1714–1718, 2010.
[20] Heywood J.B., "Internal Combustion Engine Fundamentals", US: McGraw-Hill Co, 1988.
[21] Kuleshov A. S., "Use of Multi-Zone DI Diesel Spray Combustion Model for Simulation and Optimization of Performance and Emissions of Engines with Multiple Injection", SAE Technical Paper 2006-01-1385,2006, doi:10.4271/2006-01-1385.
[22] Alkidas A. C., "Relationship between smoke measurements and particular measurements", SAE Technical Paper, 840412, 1984, doi:10.4271/840412.
[23] Kegl B., "Influence of biodiesel on engine combustion and emission characteristics", Applied Energy, Vol. 88, pp. 1803–1812, 2011.
[24] Nabi Md. N., Rahman Md. M., Akhter Md. S., "Biodiesel from cotton seed oil and its effect on engine performance and exhaust emissions", Applied Thermal Engineering, Vol.29, pp. 2265–2270, 2009.