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Experimental investigations on combustion, performance, and emissions characteristics of a neat biodiesel-fuelled, turbocharged, direct injection diesel engine

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Abstract: The concerns about clean environment, high oil prices, and stringent emission norms are the driving forces for the research on alternative fuels. Extensive investigations have been carried out in recent years on replacing diesel fuel by vegetable-oil-derived biodiesel fuel in compression ignition engines, mainly owing to its renewable source, reduced emissions, and direct usage without any engine modifications. The experimental work reported here has been carried out on a turbocharged, direct injection, multi-cylinder truck diesel engine using diesel fuel and neat karanja-oil-derived biodiesel under varying speed and load conditions.

The results of the experimental investigation indicate that a maximum of 2.3°CA (crank angle) advance in dynamic fuel injection timing is observed with biodiesel compared with the diesel fuel. The ignition delay is lower for biodiesel compared with diesel fuel at all engine speeds. The peak cylinder pressure is higher at most of the operating conditions and the maximum rate of pressure rise is lower at low speeds and higher at high speeds for biodiesel compared with diesel fuel. In general an earlier start of combustion and lower combustion duration are observed for biodiesel relative to diesel fuel. The maximum thermal efficiency occurring at the maximum torque speed of 1400 r/min is observed to be 40.7 per cent and 40.5 per cent for diesel and neat biodiesel respectively. A significant reduction in unburnt hydrocarbon and smoke emissions, comparable carbon monoxide emissions, and higher nitric oxide emissions are observed with biodiesel compared with diesel fuel operation. In general, an improvement in combustion characteristics and exhaust emissions, except nitric oxide, are observed for biodiesel compared with diesel fuel, with a slight penalty in brake thermal efficiency.

Keywords: combustion duration, emissions, engine speed, neat biodiesel, ignition delay

1 INTRODUCTION

Owing to continually growing applications of the compression ignition engine in agricultural, goods transport, and automotive sectors, along with its extensive use in power generation, the demand for diesel fuel is increasing at a significant rate of around six times higher than gasoline in India compared with the rest of the world [**1**]. The fast-depleting reserves of petroleum fuels and the consequent escalation in their prices are a matter of serious concern the world over. The search for an environment-friendly and economical alternative fuel is intensifying with a sense of urgency. The full replacement or even a part substitution of a small fraction of diesel fuel by any suitable alternative fuel will have a significant impact on the economy and the environment. In the recent past, the processed form of vegetable oil known as biodiesel has emerged as a potential fuel to substitute petroleum diesel in compression ignition engines. Depending on the

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availability and cost in various countries, different feed stocks are being used to produce biodiesel fuels. More than 350 oil-bearing crops have been identified so far as potential candidates for the production of future biofuels [**2**]. A wide variety of vegetable oil feed stocks including the waste vegetable oils are used [**3**, **4**] for producing biodiesel. Owing to high cost and shortage of edible oils in India, non-edible oils such as jatropha, karanja, mahua, neem, rubber seed, tobacco seed, rice bran, etc. are considered appropriate for producing biodiesel fuel. Karanja (*Pongamia Pinnata*) can be cultivated on any type of soil, has low moisture demand, and has high oil content (25–30 per cent) [**5**].

Ideally, the molecules comprising petroleum diesel fuel are saturated non-branched hydrocarbons with carbon number ranging from 12 to 18. In contrast to this, the vegetable oil molecules are triglycerides with non-branched chains of different lengths and different degrees of saturation [**6**]. The attractive characteristics of biodiesel include higher cetane number, non-toxic emissions, bio-degradability (degrading four times faster than petro-diesel), absence of sulphur and aromatic compounds, and excellent lubricity [**7–9**]. The objectives of the current experimental investigation include the combustion, performance, and emission evaluation of karanja-derived biodiesel fuel in a turbocharged, multi-cylinder, direct injection diesel engine in comparison to diesel fuel.

2 STATE OF THE ART

The experimental investigations carried out by several researchers [**4**, **10–26**] using biodiesel fuels derived from different feed stocks reveal that biodiesel fuel does not demonstrate any significant loss of performance during short-term engine tests. However, there is a lack of unanimity about the extent of variations in the power output, brake thermal efficiency, and exhaust emission results from different biodiesel fuels. The results reported in published literature indicate both lower [**10–12**] and higher [**13–17**] brake thermal efficiencies as well as higher [**18–22**] and lower [**4**, **23–26**] exhaust nitric oxide (NO_x) concentrations. These variations may be attributable to changes in engine type [**27**], engine operating conditions, and the type of vegetable oil used for producing the biodiesel fuel. However, it is strongly believed that the changes in fuel compositions are the major cause for variations in engine performance and emission characteristics operated on biodiesel fuels. Graboski and McCormick [**20**] reported wide variations in the measured properties of some biodiesel fuels owing to changes in their fatty acid composition.

Senatore *et al.* [**28**], during their investigation with neat rapeseed-oil-derived biodiesel in a turbocharged, direct injection diesel engine observed an injection advance, earlier heat release (3 to 5 degrees advance), higher mean gas temperatures, higher NO_x, and lower unburnt hydrocarbon (HC) and carbon monoxide (CO) emissions in comparison with diesel fuel operation.

Using neat soybean-oil-derived biodiesel and diesel fuel in a four-cylinder, naturally aspirated, direct injection diesel engine, Scholl and Sorenson [**29**] observed similar instantaneous and cumulative heat release with the two fuels; and higher peak cylinder pressure, higher rate of pressure rise, comparable NO_x, and lower CO and HC for biodiesel compared with diesel fuel.

Shaheed and Swain [**30**] conducted experiments in a single-cylinder, naturally aspirated, direct injection diesel engine with neat coconut-oil-derived biodiesel and diesel fuel and found that the rate of change in cylinder pressure is the same for both the fuels; and 3–4 per cent lower peak cylinder pressure, lower ignition delay, lower heat release, slightly higher brake thermal efficiency, and lower exhaust CO, HC, NO_x, and smoke with neat biodiesel.

Mandpe *et al.* [**31**] conducted periodic emission testing to determine on-road emissions in a modern unmodified EU3 common-rail diesel engine fuelled with neat jatropha-oil-derived biodiesel and diesel fuel operated for a cumulative 8000 km distance, and concluded comparable NO_x and HC emissions, higher CO emissions, and a 3 per cent decrease in power output with neat biodiesel compared with diesel fuel.

During their experimental investigations with neat karanja-derived biodiesel and diesel fuel in a twin-cylinder diesel engine operated at a constant speed of 1500 r/min, Srivastava and Verma [**5**] observed a 6 per cent decrease in brake thermal efficiency, higher exhaust gas temperature, and higher exhaust HC, CO, and NO_x emissions with neat biodiesel compared with diesel fuel.

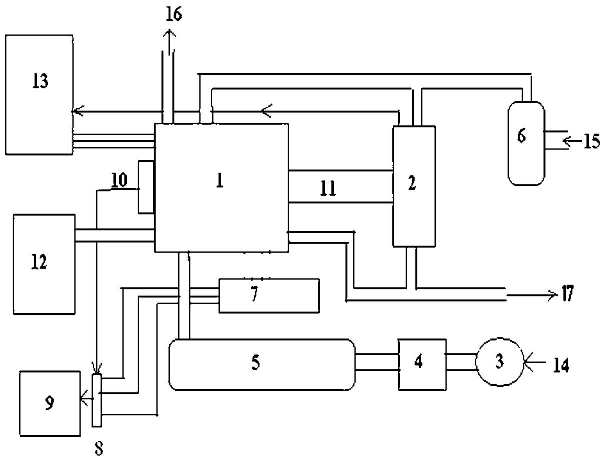
Banapurmath *et al.* [**32**] conducted experiments in a single-cylinder, direct injection diesel engine at a constant speed of 1500 r/min with neat jatropha, karanja, and sesame-oil-derived biodiesel and diesel fuels, and concluded a 2 per cent decrease in brake thermal efficiency, higher ignition delay, higher combustion duration, and higher smoke, CO, HC, and NO_x emissions with neat biodiesel fuels compared with diesel fuel.

A comprehensive literature survey on the work done by the researchers with neat biodiesel fuel reveals that in general the exhaust HC and CO emissions are lower and NO emissions are higher with biodiesel derived from edible oils (soybean oil, rapeseed oil, and coconut oil) whereas there is no unanimity in the emission results with neat biodiesel derived from non-edible oils (jatropha, karanja, and sesame) in comparison with diesel fuel. It is also observed that most of the reported experimental results are based on tests conducted at a single speed under varying load conditions.

The current objective is therefore focused on the combustion, performance, and emission evaluation of a non-edible karanja-oil-derived biodiesel fuel in a turbocharged, multi-cylinder, direct injection diesel engine in comparison to diesel fuel at different speed and load conditions.

3 TEST SET-UP AND PROCEDURE

A constant-speed and variable-load test is conducted in an Eicher E483 four-cylinder, in-line, turbocharged, direct injection diesel engine. The turbocharger system contains an intercooler but no waste gate. There is no exhaust gas recirculation (EGR) control fitted in the engine. A schematic of test engine set-up is shown in Fig. 1 and the specifications of the test engine are provided in Table 1.



1. Engine
2. Eddy current dynamometer
3. Air filter
4. Turbine meter
5. Air-box
6. Water pump
7. Signal conditioner
8. Data acquisition system
9. Display unit
10. Crank angle encoder
11. Propeller shaft
12. Fuel tank
13. Dynamometer controller
14. Air inlet
15. Cooling water inlet
16. Exhaust outlet
17. Cooling water outlet

Fig. 1 Schematic of test engine set-up

Table 1 Engine specifications

Engine type	Four-cylinder, in-line, turbocharged, direct injection diesel engine
Turbocharger	With intercooler, no waste gate
Combustion system	Re-entrant type, bowl dia: 10 mm, bowl depth: 20.9 mm
Bore	100 mm
Stroke	105 mm
Displacement	3298 cm^3
Compression ratio	17.5:1
Max. torque condition	285 N m @ 1400 r/min
Max. power condition	70 kW @ 3200 r/min
Cooling method	Forced circulation
Fuel pump type	Distributor type
Nozzle opening pressure	230 bar
Injector : hole × diameter	5 × 0.209 mm
Injection timing (static)	12°CA BTDC

BTDC: before top dead centre.

The test engine is attached with an AVL eddy current dynamometer of type ECB 150 HS. The air and fuel flowrates are measured using turbine flow meter and burette stop-watch arrangements respectively. The engine exhaust gas temperatures are obtained using a K-type thermocouple. Kistler uncooled piezo-electric type 6055 sensors having a pressure-measuring range of 250 bar are used for measuring cylinder pressures. The sensitivity of the pressure sensor ranges between ± 0.5 per cent over a temperature range of 200 ± 50 °C. Another Kistler pressure sensor type 4067 is used for measuring fuel line pressures. The crank angle measurement is carried out using a Kistler type 2613B crank angle encoder mounted at the front end of the crankshaft having a speed range of 20 000 r/min with a resolution of 1 CAD. A signal conditioner Kistler type 2853A having eight slots for input/output channels and a capability to communicate with the pressure sensors is attached to the set-up for converting pressure signals from sensors to the analogue voltage signals, which are then fed to the data acquisition system through a signal mixer. The data sampling and an online combustion analysis are done using Dewetron software. The combustion parameters such as energy release rates, cylinder pressures, manifold pressures, and pressure–volume diagrams are obtained from this software.

The gaseous emission measurements are made using a Rosemount chemiluminescent analyser (model 951A) having a precision of ± 5 per cent of full-scale range for NO_x; a Rosemount flame ionization detector (model 400A) with a repeatability of 1 per cent of full-scale range for unburned hydrocarbons; and a Horiba NDIR exhaust gas analyser (MEXA 324F_B) with an accuracy of ±0.5 per cent vol of full-scale range for carbon monoxides. The exhaust smoke is measured using a Bosch smoke meter.

4 EXPERIMENTATION METHODOLOGY

The experimental tests are conducted for various loads at four different speeds, namely 1000 r/min, 1400 r/min, 2000 r/min, and 2500 r/min, using base diesel and 100 per cent karanja-oil-derived biodiesel. The fatty acid composition and the important fuel properties of karanja biodiesel are measured according to their ASTM standards. The fatty acid composition of karanja biodiesel is provided in Table 2, and the fuel properties of interest in combustion and performance estimation for diesel [**33**] and karanja methyl ester fuels are given in Table 3.

At different engine operating conditions the loads, speed, fuel flowrate, exhaust gas temperature, and cylinder pressure histories are recorded at steady-state condition of the engine. The average engine cooling water temperatures observed during the experiments with both diesel and karanja biodiesel varied from 40 to 80 °C with increasing load and speed; the difference of the temperatures between diesel and biodiesel cases ranged within ±3 °C.

5 RESULTS AND DISCUSSION

To understand engine combustion characteristics with base diesel and neat karanja-derived biodiesel, the fuel line and engine cylinder pressures are acquired during engine tests carried out at various speed and load conditions.

5.1 Fuel injection

Figure 2 shows variations of typical fuel line pressures at no load and full load conditions for neat karanja biodiesel and diesel fuels at engine speeds of 1000, 1400, 2000, and 2500 r/min. The start of dynamic fuel injection timings in terms of crank angle are inferred from these fuel line pressure histories. A designed nozzle opening pressure of 230 bar is taken as a criterion for the start of fuel injection. It is clearly evident from the fuel line pressure histories that the injection characteristics of biodiesel are different to those of diesel fuel primarily owing to differences in their physical properties namely density and bulk modulus, which affect the injection process. It is important to note that the fuel injection pump used in the present engine is of rotary type and thus has a fixed end injection timing [**37**]. An increase in fuel line pressure is observed with neat biodiesel compared with diesel fuel, and the increase is significant at higher speeds.

For the two fuels, namely neat karanja biodiesel and diesel fuels, Fig. 3 shows the variations of the dynamic start of injection with load at different speeds. The results suggest that the start of injection advances for the neat biodiesel in comparison with the diesel fuel, and this advance is relatively greater at higher speeds where the fuel line pressures are higher. Typically, a maximum of 2.3°CA (crank angle) advance is noted for neat biodiesel relative to diesel at 2500 r/min engine speed. This injection advance in case of biodiesel is attributed to its higher density resulting in faster travel of acoustic pressure waves [**38**]. Further, in the case of the rotary injection pump used in the set-up, the timings for the end of injection remain fixed and hence, to accommodate an additional quantity of biodiesel (owing to lower heating value) required to maintain the power level obtained in case of diesel fuel, the start of injection

Table 2 Fatty acid composition of karanja biodiesel

Composition (% wt)	C 16:0	C 18:0	C 18:1	C 18:2	C 20:0	C 22:0	C 24:0
Karanja	11.1	2.9	60.6	16.9	0.8	4.1	0.3

Table 3 Properties of diesel and karanja methyl ester

S. No.	Properties	Diesel [**33**]	Karanja methyl ester
1	Density (kg/m^3)	840	889
2	Calorific value (kJ/kg)	42 490	36 658
3	Viscosity (cSt)	4.59	5.71
4	Flash point (°C)	50	181
5	Cetane number	45–55	57.8*
6	Distillation temperature (°C)	188–343‡	356†

*Estimated from fatty acid composition following reference [**34**]; †taken from reference [**35**]; ‡taken from reference [**36**].

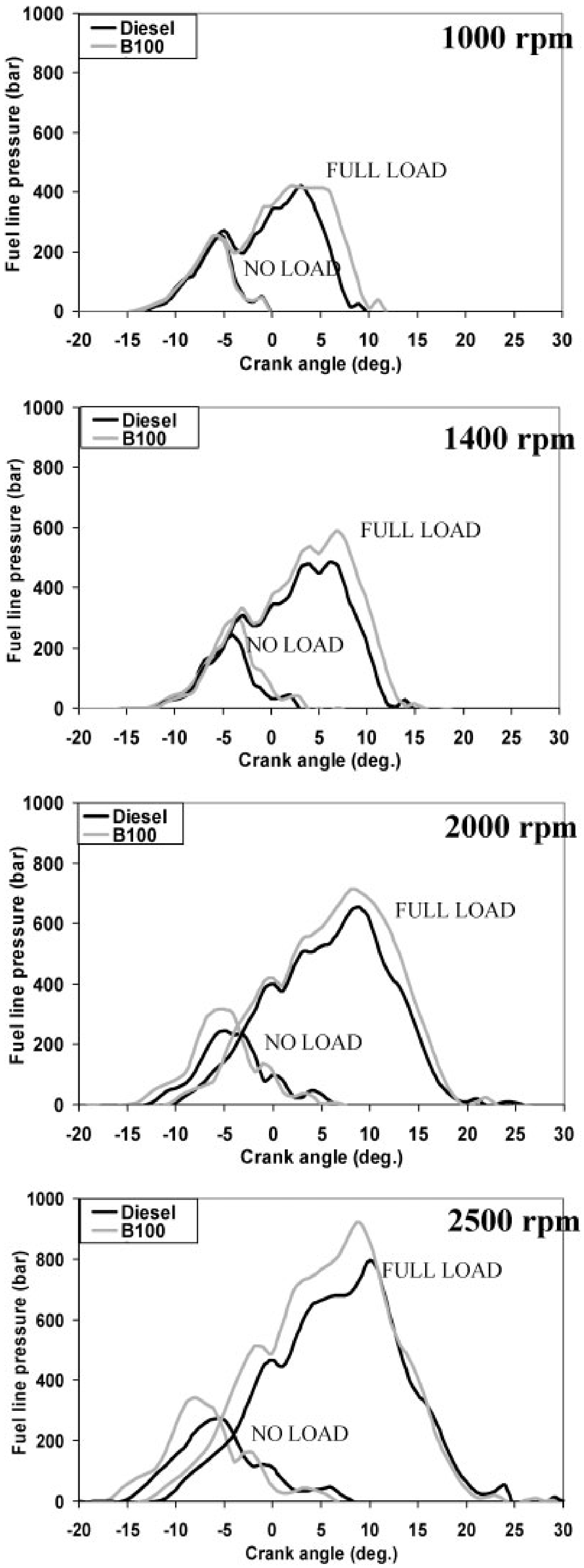


Fig. 2 Variations of fuel line pressure for neat biodiesel in comparison with diesel

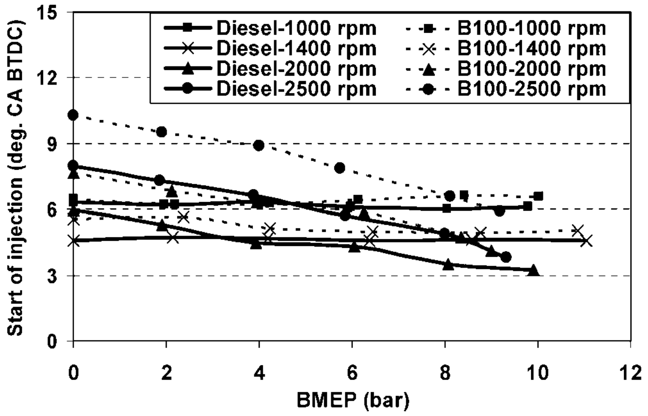


Fig. 3 Variations of start of injection with load for neat biodiesel in comparison with diesel

will advance [**37**]. Szybist *et al.* [**39**] observed 1.2°CA advance in fuel injection timing for soy methyl ester compared with diesel fuel and attributed it to 11 per cent higher bulk modulus of biodiesel.

5.2 Measured cylinder pressure and apparent net energy release

Figure 4 shows typical cylinder pressure histories at no load and full load conditions for neat karanja biodiesel and diesel fuels at engine speeds of 1000, 1400, 2000, and 2500 r/min. The comparisons of the cylinder pressure histories of the two fuels at different speeds suggest that there is generally a rise in cylinder pressure with neat biodiesel in the early phase of combustion (see Fig. 4). At no load and various engine speeds, the peak cylinder pressure occurrence is slightly advanced for the neat biodiesel, owing to an earlier start of dynamic injection, and a lower ignition delay owing to its higher cetane number relative to diesel fuel. At full load condition, the magnitude of peak cylinder pressure is higher for the neat biodiesel while its instance of occurrence is more or less the same as that of diesel fuel. As the full load condition is dominated by the diffusion phase of combustion, the neat biodiesel results in improved burning rate and higher peak cylinder pressure compared with diesel fuel owing to the oxygen available in the biodiesel fuel molecule. To understand the variations in combustion characteristics of the two fuels, the parameters such as instantaneous energy release rate, ignition delay, peak cylinder pressure, maximum rate of pressure rise, cumulative energy release, and combustion duration are obtained from these cylinder pressure histories at different operating conditions, and the effects are discussed in subsequent sections.

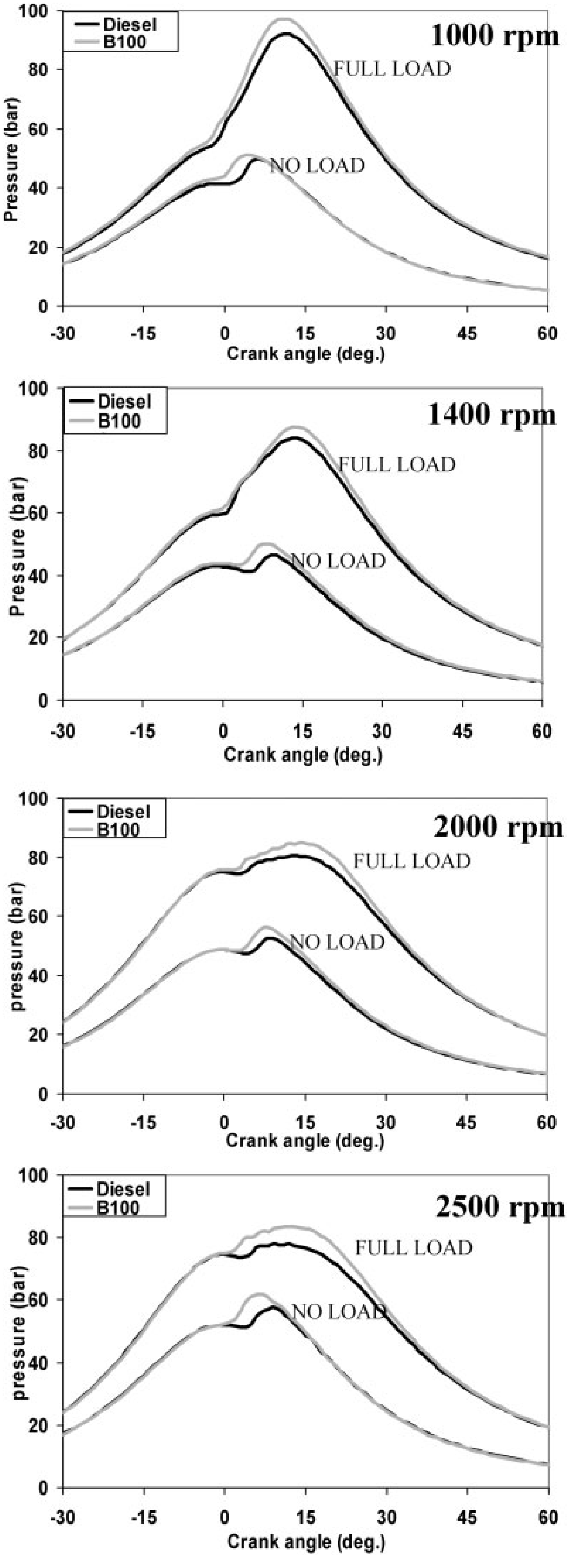


Fig. 4 Comparison of cylinder pressure histories of neat biodiesel and diesel

The net energy release estimations are obtained from the measured cylinder pressure histories by using the first law of thermodynamics applicable to the closed part of engine cycle. The equation for estimating apparent energy release rate [**40**] is expressed as

$$\frac{\mathrm{d}Q_{\mathrm{n}}}{\mathrm{d}\theta} = \frac{\gamma}{\gamma-1}P\frac{\mathrm{d}V}{\mathrm{d}\theta} + \frac{1}{\gamma-1}V\frac{\mathrm{d}P}{\mathrm{d}\theta}$$

where Q_{n} is net apparent energy release rate, θ is crank angle, P is cylinder pressure, V is cylinder volume, and γ is specific heat ratio.

Figure 5 shows the comparison of instantaneous energy release rates (Ins. ERR) of neat karanja biodiesel and diesel at no load and at full load conditions at the test speeds. From these plots it can be observed that as the load increases, there is an increase in energy release rate throughout the combustion period. Also, these energy release rate diagrams reveal that at full load there are two peaks of energy release. Also, the occurrence of the maximum energy release rate is observed to be delayed at full load and higher speeds. The energy release rate curves have two peaks depending on the relative dominance of premixed and diffusion combustion phases. At no load condition, the premixed combustion phase is more dominant than the diffusion phase and thereby the diffusion combustion phase is not distinct. This trend is reversed at full load condition wherein the diffusion combustion phase is more dominant with two peaks in the energy release rate diagrams. The two peaks at full load are a result of a significantly larger injection period of about 5 to 6 times more than the ignition delay period. This should provide a higher degree of diffusion combustion with varying fractions of fuel–air mixing to result in the second peak. At no load, the injection duration is lower than the delay period. A closer observation of energy release rate curves reveals that the magnitude of energy release during premixed phase of combustion decreases with the increase in speed at full load condition for both the fuels. This is primarily attributable to an increase in turbocharger boost with the increase in engine speed [**40**].

The general trend in the relative variations of energy release rates of neat biodiesel and diesel fuel is similar to that of the corresponding cylinder pressure histories of the two fuels. The occurrence of the peak energy release rate is earlier for neat biodiesel at no load and similar to that for diesel fuel at a full load condition. Also, the magnitude of peak energy release rate is higher for neat biodiesel at full

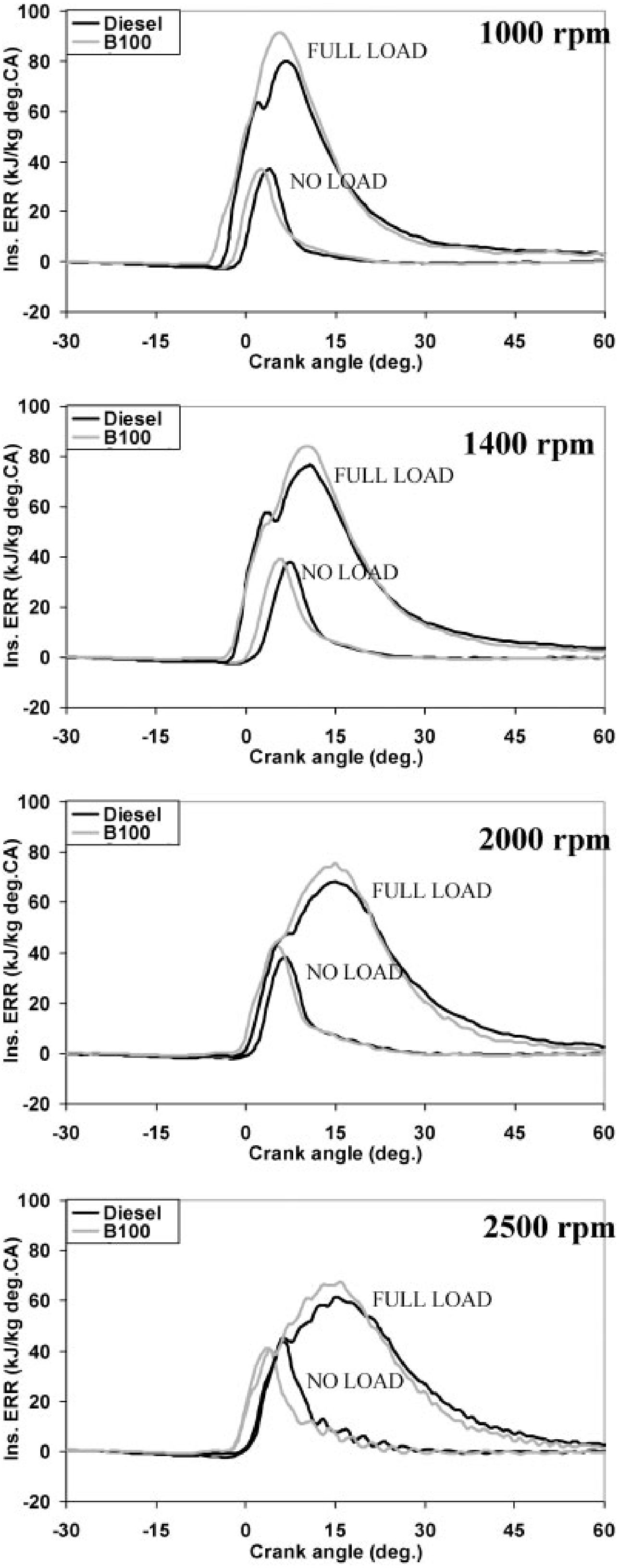


Fig. 5 Comparison of instantaneous energy release rates of neat biodiesel and diesel

load condition compared with diesel fuel, whereas the differences in peak energy release rates are negligible between the two fuels at no load condition. The higher energy release rate observed with neat biodiesel at full load condition, even though its energy content per unit mass is smaller compared with diesel fuel, signifies the higher burning rate owing to improvement in diffusion burning phase on account of fuel-bound oxygen content in biodiesel.

The crank angle position at which energy release attains a positive slope is taken as onset of combustion. It is observed that the start of combustion advances for neat biodiesel relative to diesel fuel at all the load and speed conditions. This is because of the combined effects of advancement in the dynamic injection timing due to higher compressibility of biodiesel and the changes in ignition delay period relative to diesel fuel. Similar results of advancement in start of combustion are observed by Canakci and van Gerpen [**41**] with soy methyl ester and yellow grease methyl ester biodiesel fuels compared with diesel fuel.

5.3 Ignition delay

The ignition delay values at each test condition are deduced from respective apparent energy release rate diagrams by identifying the crank angle instant for onset of combustion at which the instantaneous energy release rate attains a positive slope. The delay period is the difference between the start of dynamic injection timing and the onset of combustion. Figure 6 shows the ignition delay values at different load and speed conditions for neat karanja biodiesel and diesel fuels. It is observed that ignition delay decreases with load but increases with speed for both these fuels. A decrease in ignition delay with increase in load is a result of higher cylinder charge temperature owing to a higher residual gas temperature and a higher wall temperature at higher load conditions [**40**]. Since the time for a crank angle is lower at higher speed, the ignition delay in crank angles is found to be higher.

Up to an engine speed of 1400 r/min, the ignition delay values for neat biodiesel are observed to be lower compared with the diesel fuel. Beyond this speed, the delay period for neat biodiesel is either the same as or higher than that of diesel fuel. The decrease in ignition delay observed with biodiesel at lower speed conditions is attributable to better ignition quality of biodiesel on account of its higher cetane number and fuel-bound oxygen content [**42**] compared with diesel. The slightly higher or similar

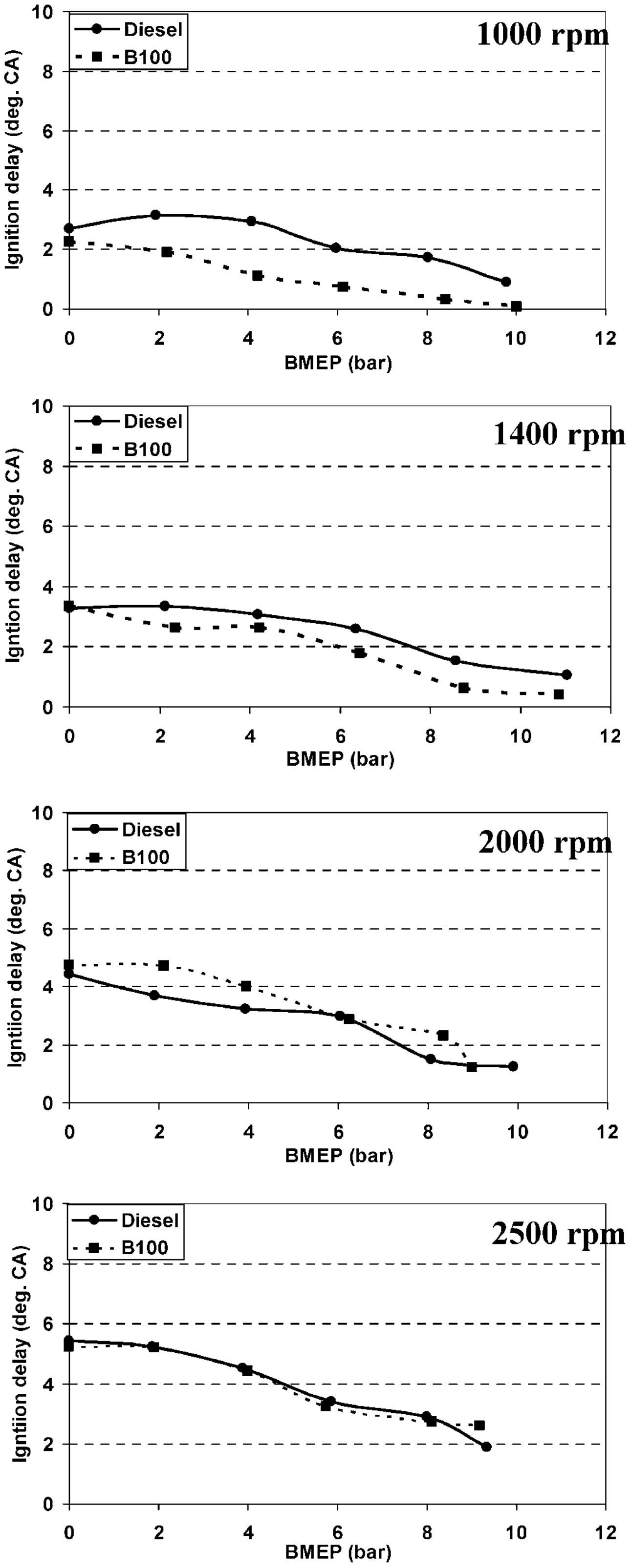


Fig. 6 Variations of ignition delay with load for neat biodiesel in comparison with diesel

ignition delay observed with neat biodiesel compared with diesel at higher speeds is attributable to a higher relative advance in the start of injection for neat biodiesel.

5.4 Peak cylinder pressures

Figure 7 shows the comparisons of peak cylinder pressure variations with load for neat karanja biodiesel and diesel fuel at various engine speeds. The magnitude of peak cylinder pressures is higher for neat biodiesel than for diesel fuel at all load and speed conditions.

The higher values of instantaneous cylinder pressures observed during early phase of combustion is attributed to the combined effect of higher density of neat biodiesel and the reduced fuel leakage losses in the fuel pump owing to higher viscosity of neat biodiesel relative to diesel fuel. In the second phase of combustion the fuel-bound oxygen content of neat biodiesel seems to accelerate the energy release and hence contribute to the higher peak cylinder pressures at all speeds under high load operations. An earlier start of combustion [**43**] and also lower combustion durations (see Fig. 10) at these operating conditions also augment this argument.

5.5 Rate of pressure rise

The rate of pressure rise is the first derivative of cylinder pressure that relates to the smoothness of engine operation. The duration of the ignition delay period, and hence the quantity of fuel injected and the extent of mixing achieved during the delay period prior to onset of combustion, contribute to the maximum rate of pressure rise.

Figure 8 shows the comparison of maximum rate of pressure rise for neat karanja biodiesel and diesel at varying speed and load conditions. The maximum rate of pressure rise is found to increase initially with load and then decrease owing to the dominance of premixed phase combustion at lower loads and the diffusion phase combustion at higher loads respectively. The maximum rate of pressure rise is lower at lower speeds up to 1400 r/min and becomes higher above this speed for neat biodiesel, as compared with that of diesel, owing to a lower ignition delay for neat biodiesel compared to diesel fuel up to 1400 r/min engine speed and a higher delay period beyond this engine speed.

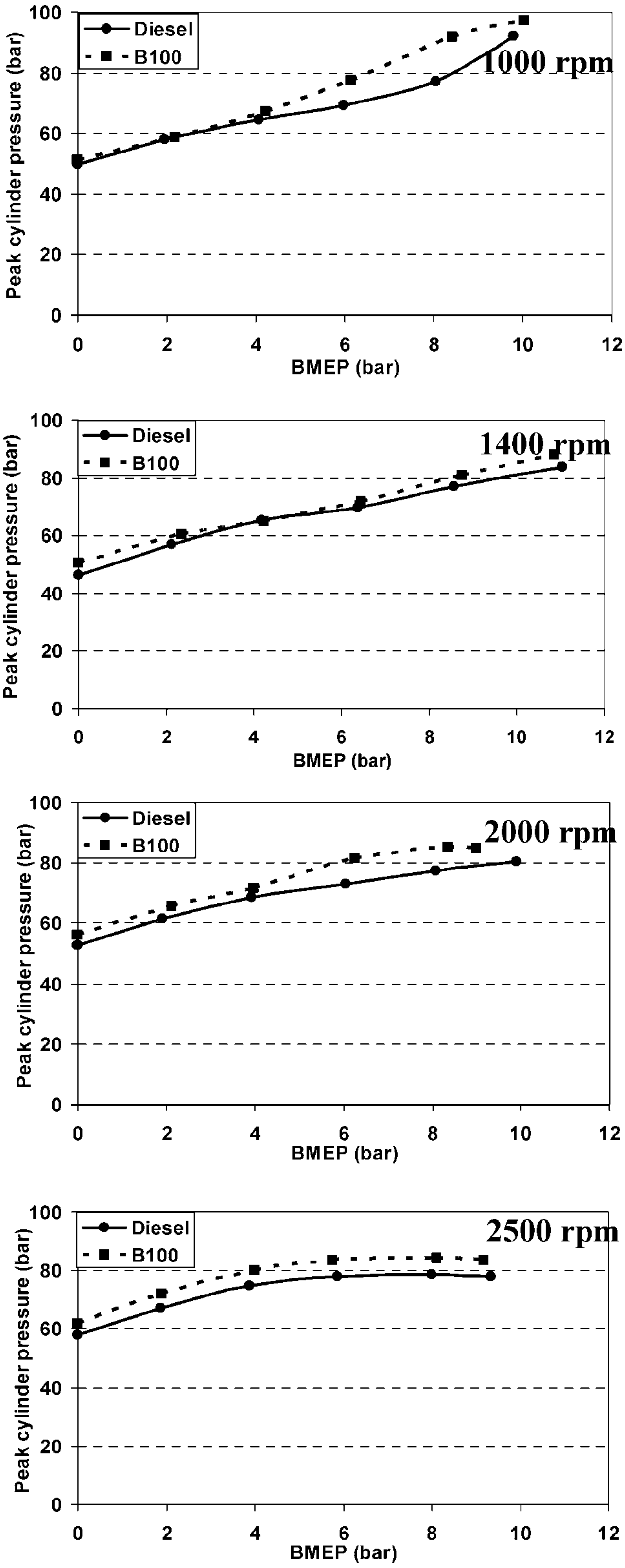


Fig. 7 Variations of peak cylinder pressure with load for neat biodiesel in comparison with diesel

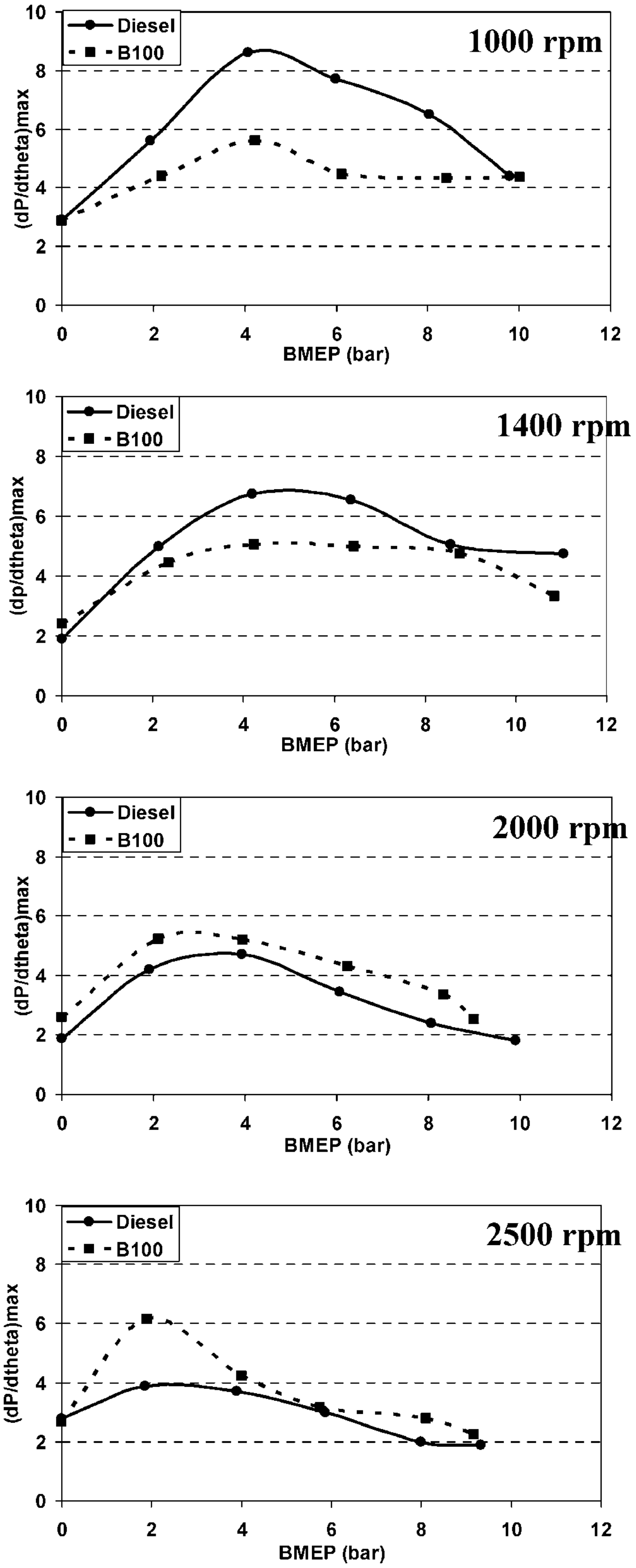


Fig. 8 Variations of maximum rate of pressure rise with load for neat biodiesel in comparison with diesel

5.6 Cumulative energy release and combustion duration

Figure 9 shows the comparison of percentage cumulative energy release of neat karanja biodiesel and diesel fuel at no load and full load conditions at the test speeds. The dip in the cumulative energy release is observed to be more at no load as compared with full load condition. This is because, as a fraction of the total energy consumed, the portion lost at low load is significant, but it is less visible at high load because there is so much more heat release. The durations of combustion are inferred from cumulative energy release diagrams by considering the time elapsed between 10 per cent and 90 per cent energy release, and are shown in Fig. 10. It is observed that the combustion duration advances for neat biodiesel as compared with the diesel fuel, and the degree of advancement is significant at higher speeds. This fact can be attributed to an earlier start of combustion and increase in diffusion phase combustion observed in the case of neat biodiesel.

The variation in combustion duration is somewhat higher at low load conditions but becomes significantly lower at high load (above half load) for neat biodiesel conditions compared with diesel fuel. A maximum decrease of 10°CA combustion duration is observed for neat biodiesel compared with diesel at full load and 1400 r/min engine speed. For neat biodiesel, the lower combustion duration at higher loads seems to be a result of reduced premixed phase and a higher rate of combustion on account of a higher cetane number and a greater amount of fuel-bound oxygen. At lower loads, the longer combustion duration with biodiesel may be attributable to its slower evaporation on account of poor spray quality.

5.7 Exhaust gas temperature

The exhaust gas temperature (EGT) provides qualitative information about the progress of combustion in the engine. Figure 11 shows the comparison of EGT of neat karanja biodiesel with diesel fuel at varying load and speed conditions. It is observed that the EGTs with the neat biodiesel do not change significantly from those of diesel fuel except at 80 per cent load and 1000 r/min engine speed. The EGTs are expected to be higher for the delayed start of combustion and its longer duration. They also depend on the peak cylinder temperature and hence the peak pressure developed in the combustion chamber. A higher EGT for neat biodiesel observed at 80 per cent

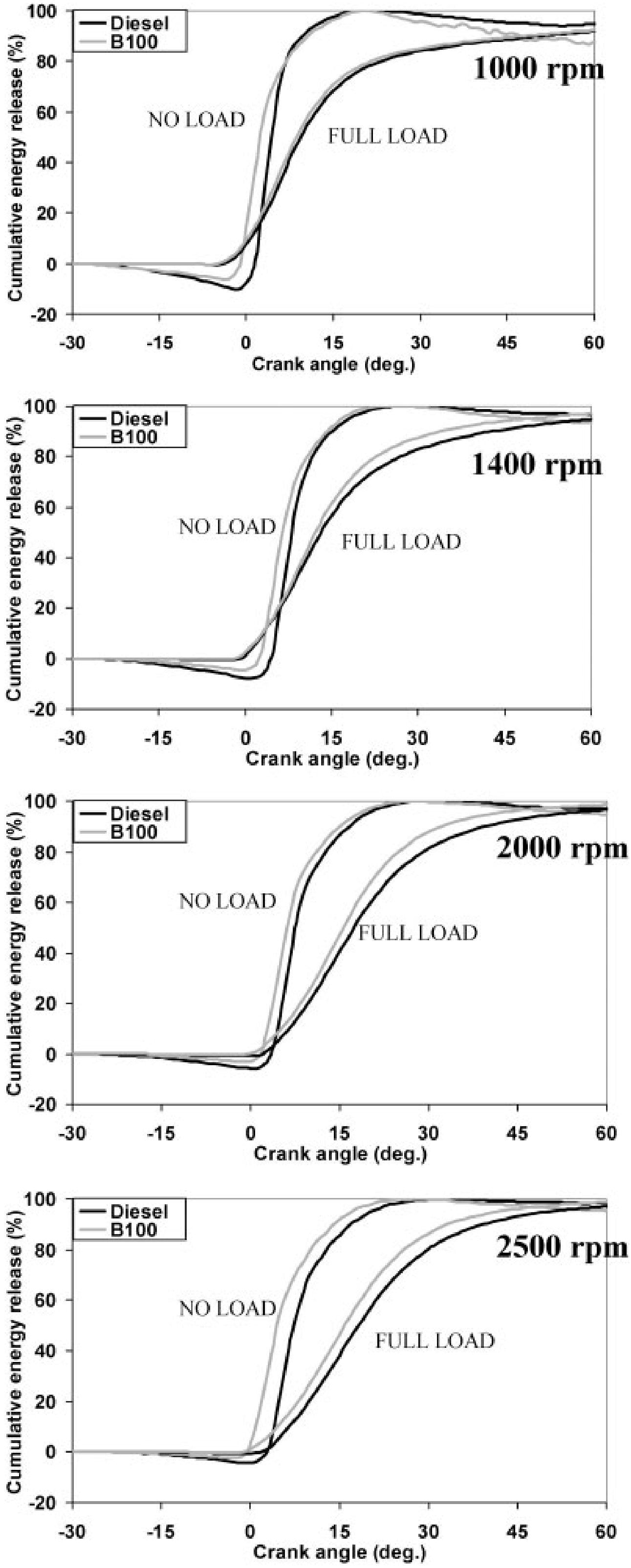


Fig. 9 Comparison of cumulative energy release rates of neat biodiesel and diesel

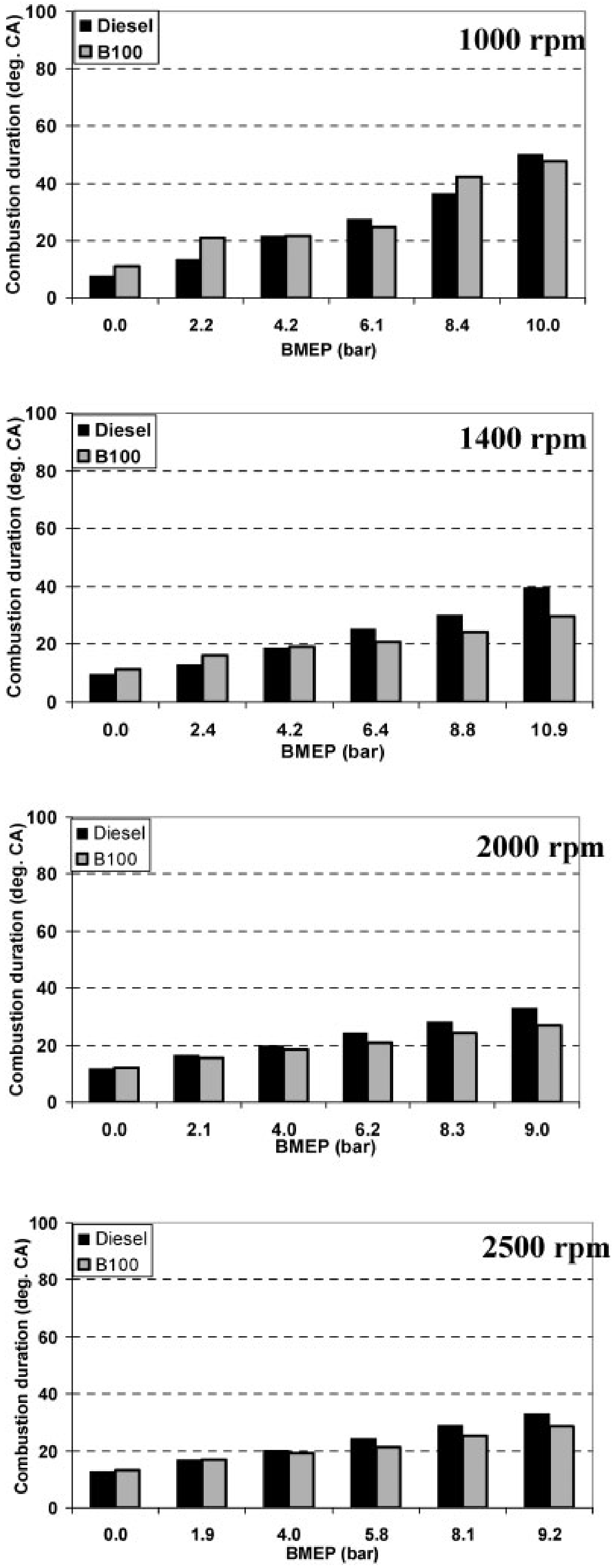


Fig. 10 Variations of combustion duration with load for neat biodiesel in comparison with diesel

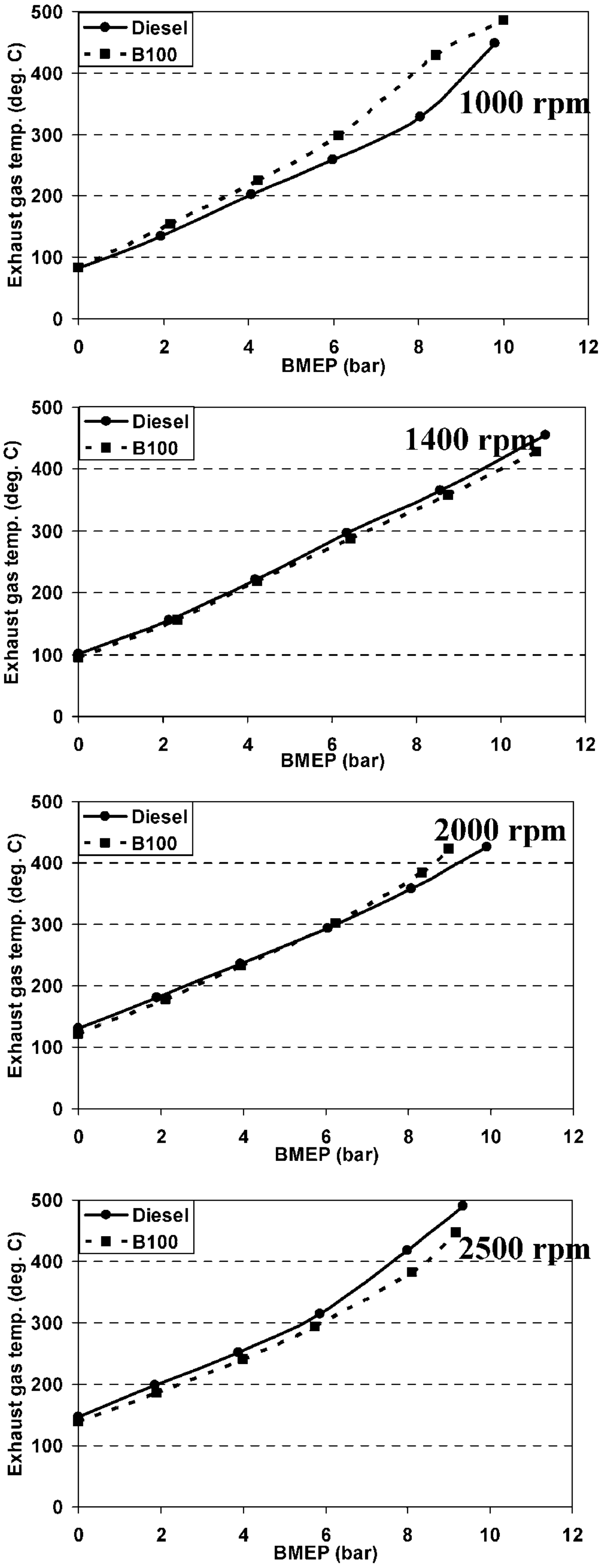


Fig. 11 Variations of exhaust gas temperature with load for neat biodiesel in comparison with diesel

load and 1000 r/min could be attributed to a combined effect of a higher peak cylinder pressure/temperature and longer combustion duration compared those other test conditions. These observations corroborate those of Puhan *et al.* [**44**], who compared the exhaust gas temperature of mahua-derived biodiesel and diesel fuel. Similarly, a decrease in EGT of neat biodiesel at higher load and speed conditions corroborates lower combustion duration compared to its diesel counterpart.

5.8 Engine performance

The brake thermal efficiency is an important performance parameter which relates the useful power output of the engine with respect to heat input given to the engine. This parameter has been evaluated for comparing the performance of neat biodiesel and diesel fuel.

5.8.1 Brake thermal efficiency

Figure 12 shows the comparison of brake thermal efficiencies of neat karanja methyl ester biodiesel and diesel fuel at different speed and load conditions. The thermal efficiency is generally found to increase with load for both the fuels at all test speeds. A decrease in thermal efficiency at full load and 1000 r/min engine speed is attributed to a relatively higher fuel–air equivalence ratio (overall fuel–air equivalence ratio exceeds a value of 0.5) compared with any other operating conditions. The maximum thermal efficiency occurring at the maximum torque speed of 1400 r/min is observed to be 40.7 per cent and 40.5 per cent for diesel and neat biodiesel respectively. Except at 80 per cent load, 1000 r/min engine speed, the differences in thermal efficiencies between neat biodiesel and diesel fuel are not significant at all the other test speed and load conditions. For most of the operating conditions, the fuel quantity requirements of the biodiesel fuel, which has a lower heating value, increased for realizing the same power as that from the diesel fuel. On considering the energy input for respective fuels on the basis of mass of fuel delivered and lower heating value, it is observed that the thermal efficiency variations for the two fuels follow the same trends shown in Fig. 12. These observations conform to somewhat reduced efficiency for biodiesel reported by Ramadhas *et al.* [**14**] and Raheman and Phadatare [**23**] in their experiments with karanja and rubber seed biodiesel respectively.

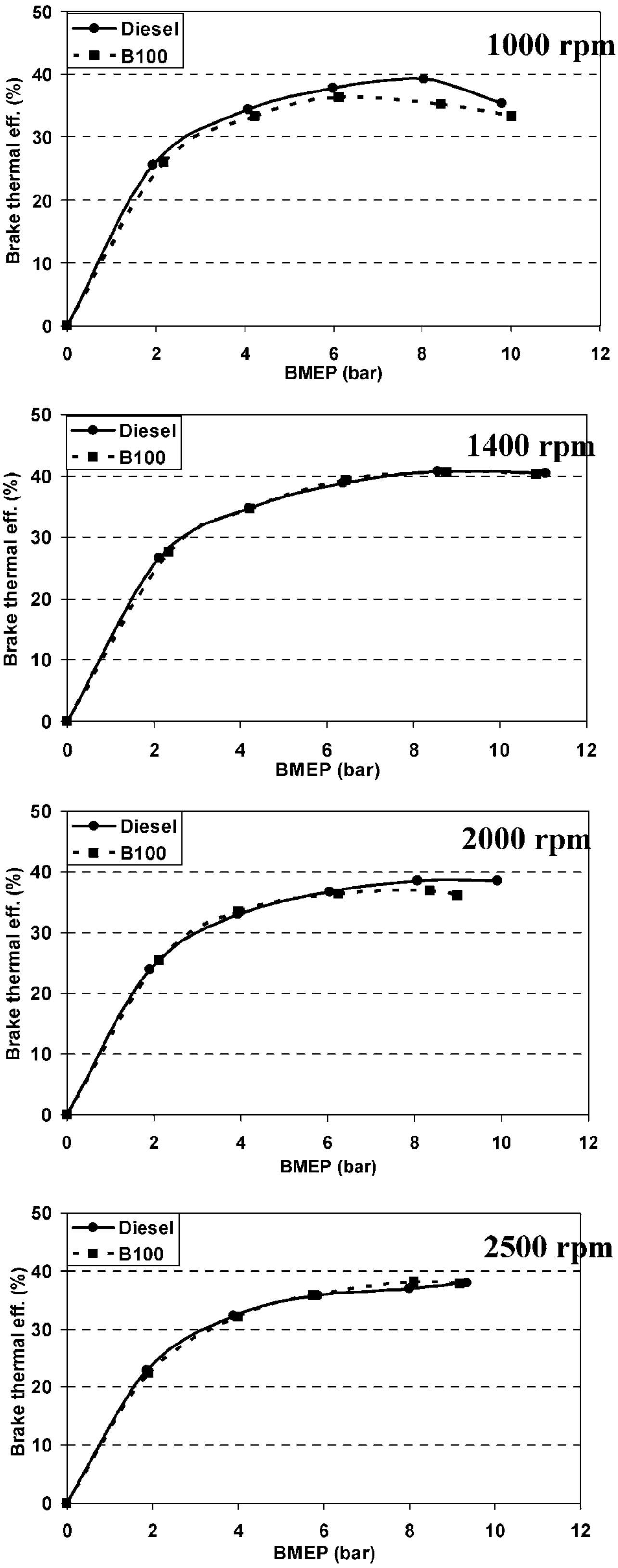


Fig. 12 Variations of brake thermal efficiency with load for neat biodiesel in comparison with diesel

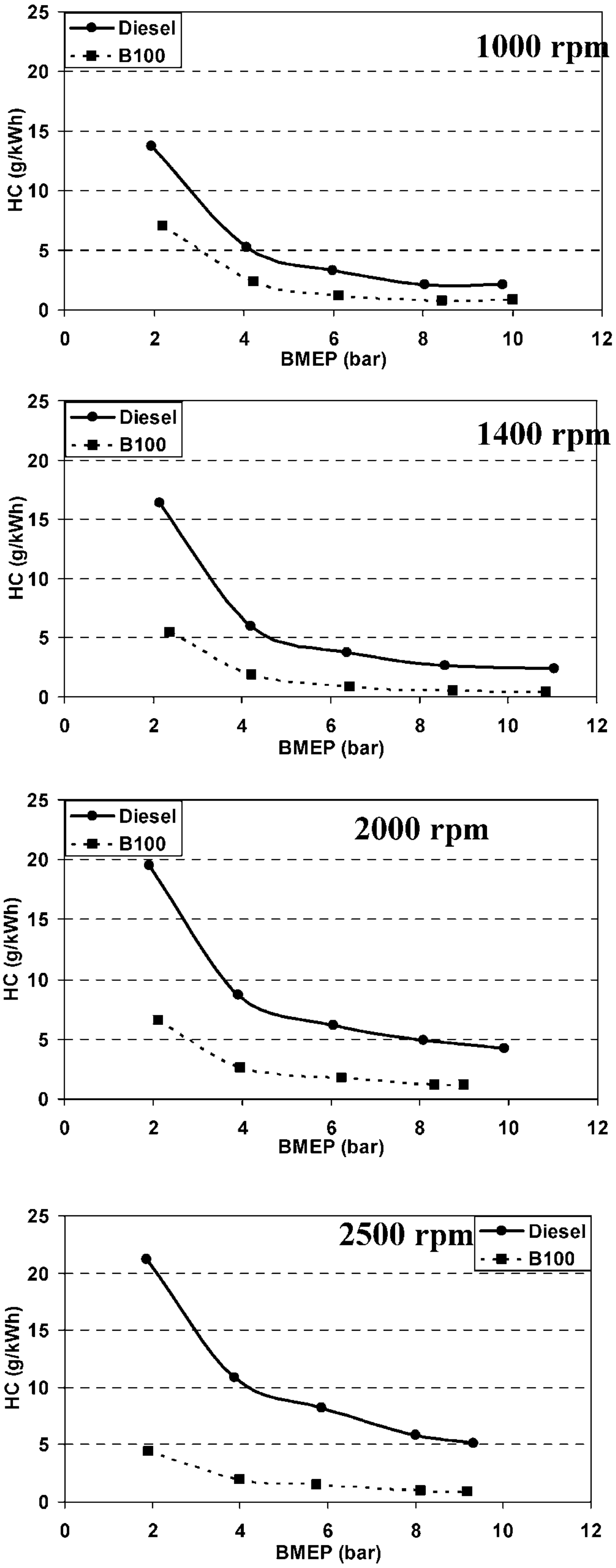


Fig. 13 Variations of hydrocarbon emissions with load for neat biodiesel in comparison with diesel

5.9 Exhaust emissions

The exhaust emissions, namely unburned HC, CO, nitric oxide (NO), and smoke (BSU (Bosch smoke units), are measured to observe the effect of using neat biodiesel in comparison to diesel fuel.

5.9.1 *Unburned hydrocarbon (HC)*

Figures 13 and 14 show the comparison of unburned HC emissions of neat karanja biodiesel and diesel fuel at varying load and speed conditions. For the neat biodiesel, there is a decrease in exhaust HC emissions compared with diesel HC over all load and speed conditions. However, this decrease is gradually increasing at higher speeds and the rate of decrease lowered with higher loads. A maximum of 4-times reduced HC emission is obtained at no load and 2500 r/min speed. The higher hydrocarbon emissions at lower speeds, and lower hydrocarbon emissions at higher loads [**21**], are attributed to over-mixing of fuel and air [**45**] and higher cylinder temperatures respectively. The increase in HC emissions at lower load and higher speed conditions is attributed to lower coolant temperatures observed at these conditions.

The reduced HC emissions observed with biodiesel operation are attributed to the combined effect of (a) decrease in over-mixing at lower loads [**21**] on account of reduced premixed phase (that is, lower ignition delay on account of higher cetane number) and (b) reduction in stoichiometric air requirement owing to fuel-bound oxygen in neat biodiesel, which also improves diffusion phase combustion and increases heat release/gas temperature as compared to diesel fuel. In their engine experiments with neat

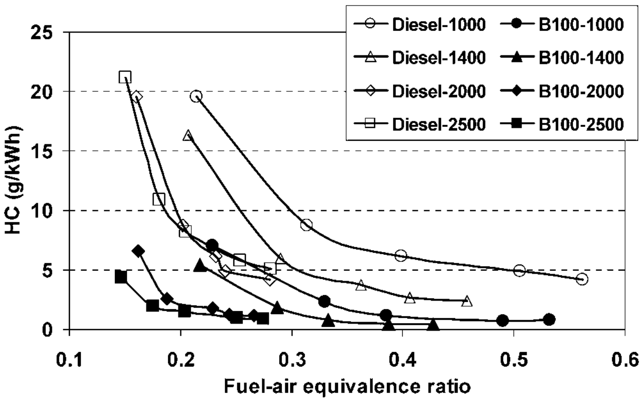


Fig. 14 Variations of hydrocarbon emissions with fuel–air equivalence ratio for neat biodiesel in comparison with diesel

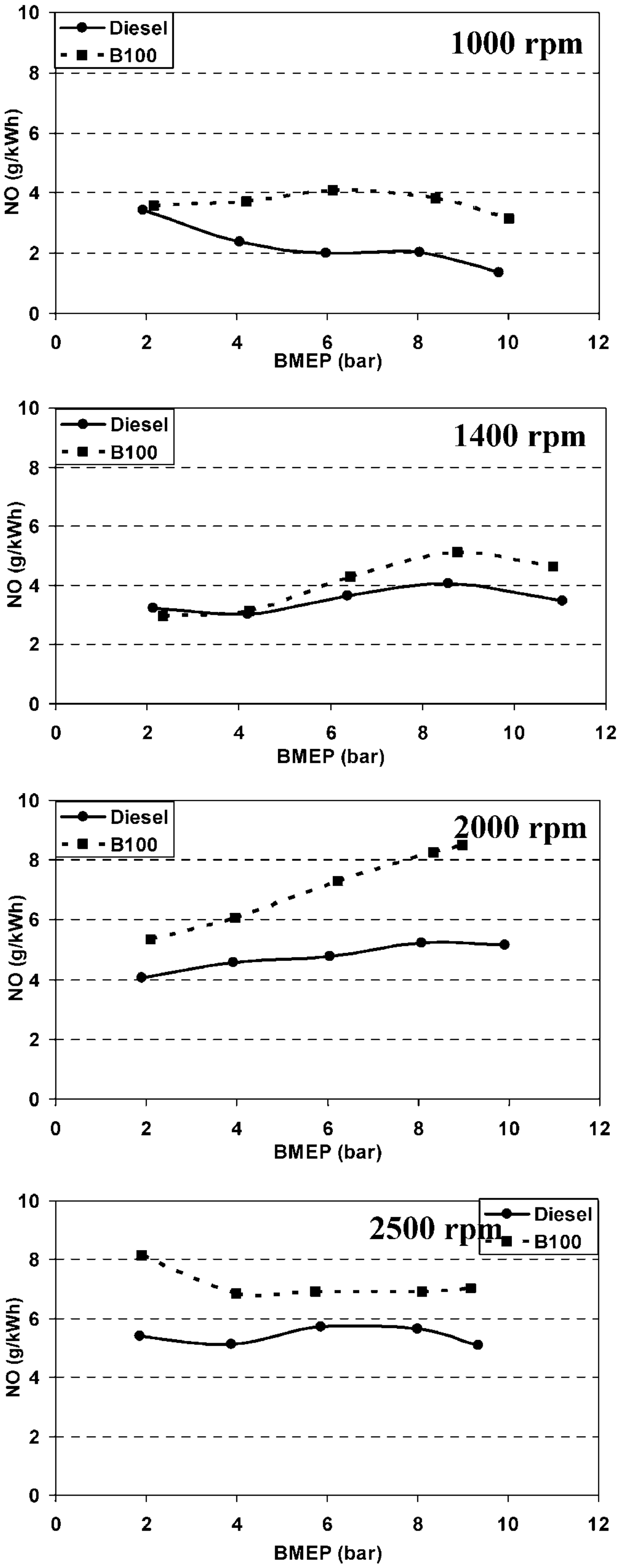


Fig. 15 Variations of nitric oxide emissions with load for neat biodiesel in comparison with diesel

biodiesel, Monyem [**21**], Scholl and Sorenson[**29**], and Suryawanshi and Deshpande [**46**] also reported reduction in HC emissions compared with diesel.

5.9.2 *Nitric oxide*

Figures 15 and 16 show the comparison of NO emissions of neat karanja biodiesel and diesel fuel at varying load and speed conditions.

It is known that the NO formation is higher at higher cylinder temperatures and advanced occurrence of its maximum value [**40**], higher oxygen availability, and the larger residence time. There is an increase in NO concentration with load at all the speeds for both the fuels because of increase in cylinder temperatures at higher loads. However, the NO concentrations with neat biodiesel are found to be higher compared with diesel fuel at all the test conditions because of higher peak cylinder pressure and somewhat advanced injection timings with increasing load. This difference in exhaust NO emissions from biodiesel is larger at higher loads at all the speeds tested. The decrease in NO observed at full load, 1000 r/min speed is attributable to relatively higher fuel–air equivalence ratio at this condition (see Fig. 16). Canakci [**18**] and Graboski *et al.* [**20**] also observed increase in NO emissions with biodiesel operation compared with diesel fuel.

5.9.3 *Carbon monoxide (CO)*

Figures 17 and 18 show the comparison of CO emissions of neat karanja biodiesel and diesel fuel at varying load and speed conditions. The CO

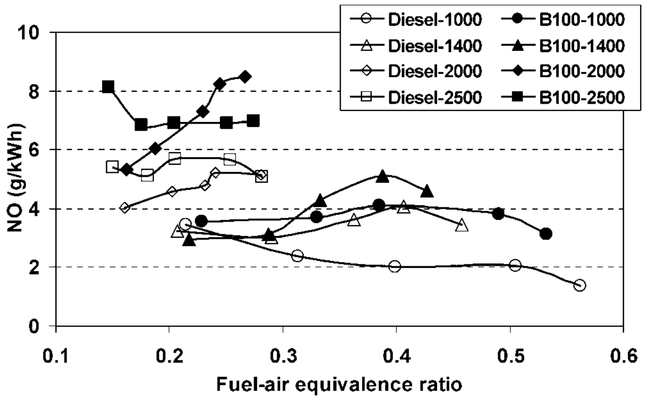


Fig. 16 Variations of nitric oxide emissions with fuel–air equivalence ratio for neat biodiesel in comparison with diesel

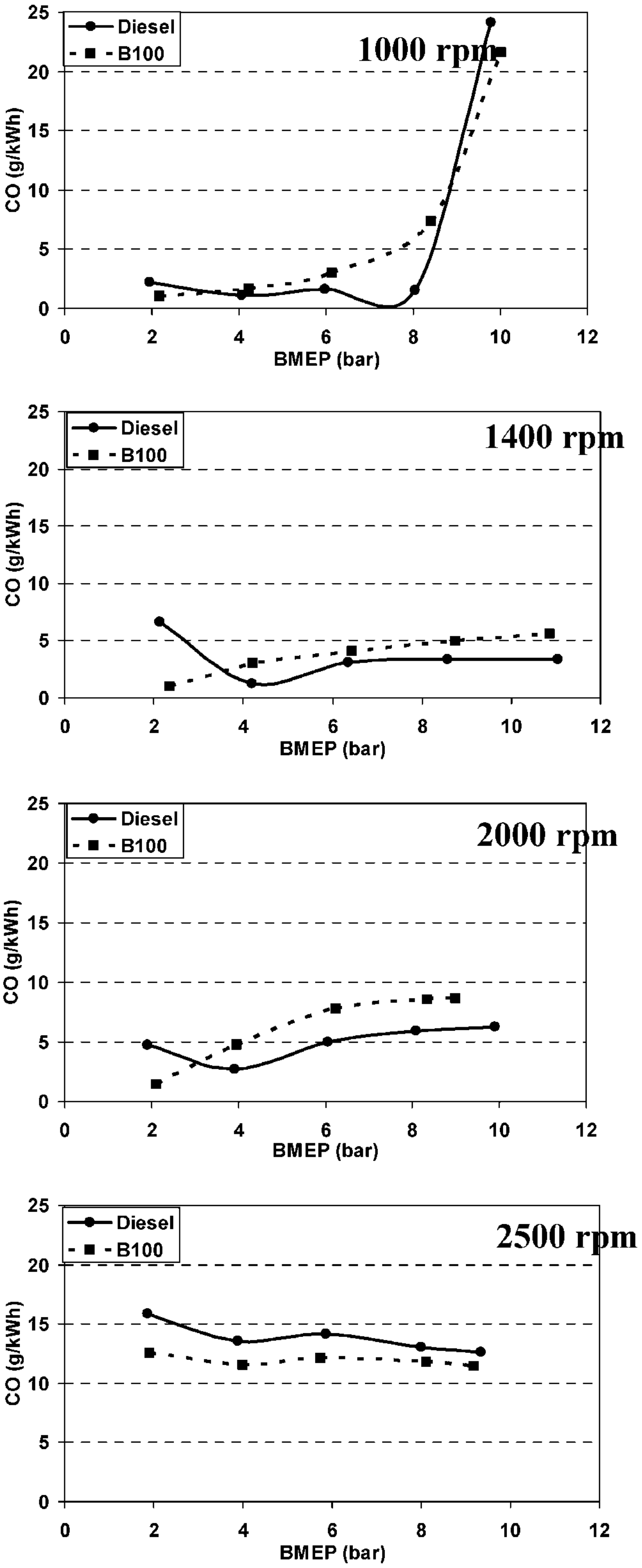


Fig. 17 Variations of carbon monoxide emissions with load for neat biodiesel in comparison with diesel

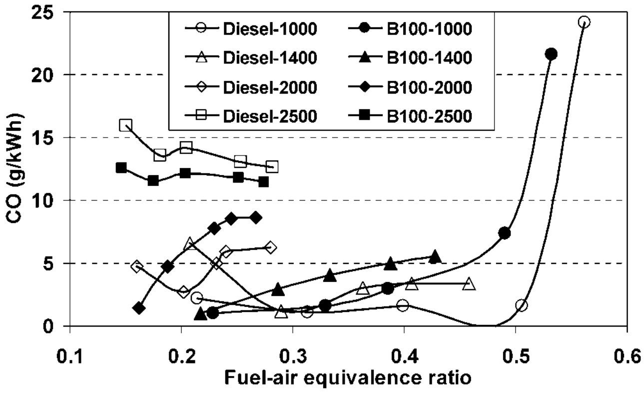


Fig. 18 Variations of carbon monoxide emissions with fuel–air equivalence ratio for neat biodiesel in comparison with diesel

emissions are controlled primarily by the fuel/air equivalence ratio [**40**]. There is no significant difference in the exhaust CO emissions for the two fuels, and the CO concentration remained within 0.1 per cent or even lower CO volume fraction as excepted for the overall lean mixture operation, except at 1000 r/min full load condition. It seems that the engine operation at 1000 r/min and full load condition experiences difficulties in the injected fuel distribution as reflected in CO and smoke results (see Fig. 19) consistently. The higher CO emissions observed at lower load and higher speed conditions are attributed to lower coolant temperatures at these conditions.

The CO emissions for neat biodiesel are slightly higher than for diesel at most of the test conditions. The trends of CO and HC emissions reported here conform with those observed by Mandpe *et al.* [**31**] in their experiment with jatropha-oil-derived biodiesel on a modern unmodified EU3 common-rail diesel engine.

5.9.4 Exhaust smoke

Figures 19 and 20 show the comparison of smoke emissions of neat karanja biodiesel and diesel fuel at varying load and speed conditions. The engine-out smoke emissions are sensitive to changes in fuel–air equivalence ratio, extent of fuel–air mixing, and the quantity of heat release during diffusion phase combustion. The smoke numbers measured in Bosch units are found to be very low (within 1 BSU) for both the fuels at all operating conditions except at near full load and 1000 r/min. These variations of smoke number almost follow the trend observed in

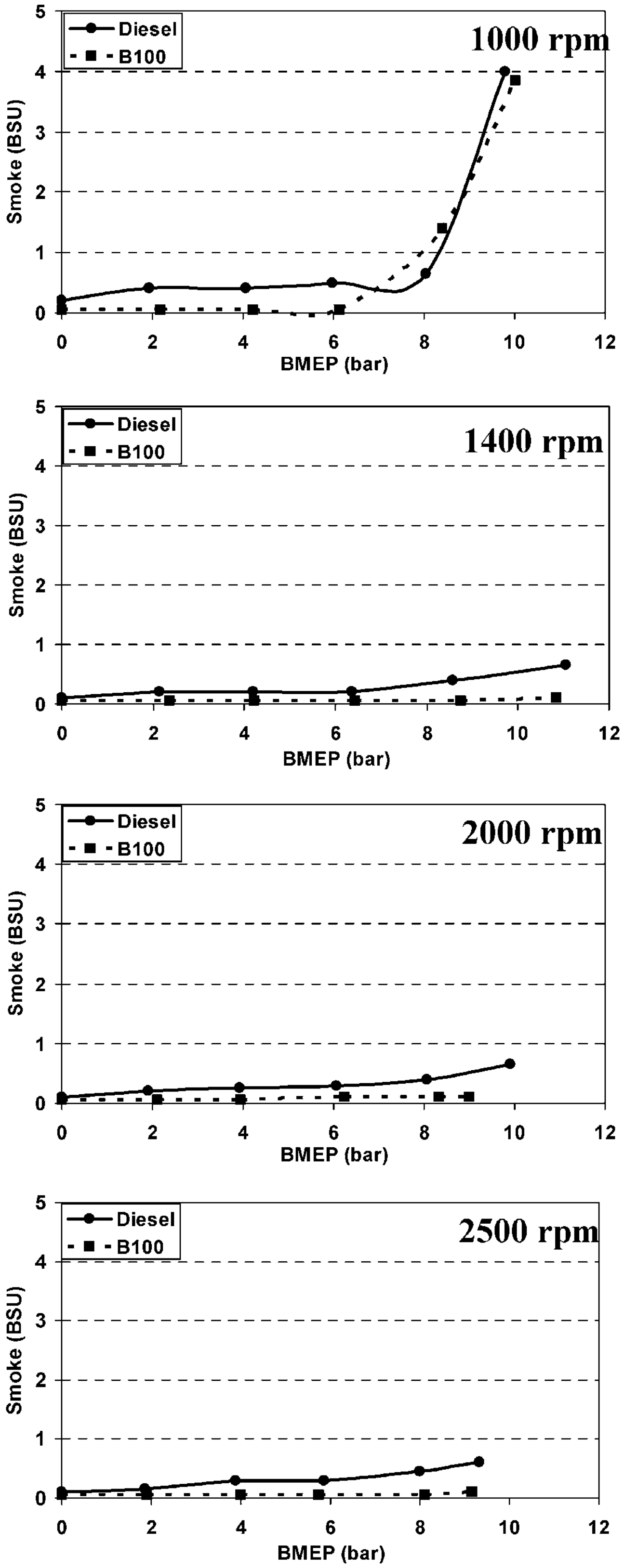


Fig. 19 Variations of smoke emissions with load for neat biodiesel in comparison with diesel

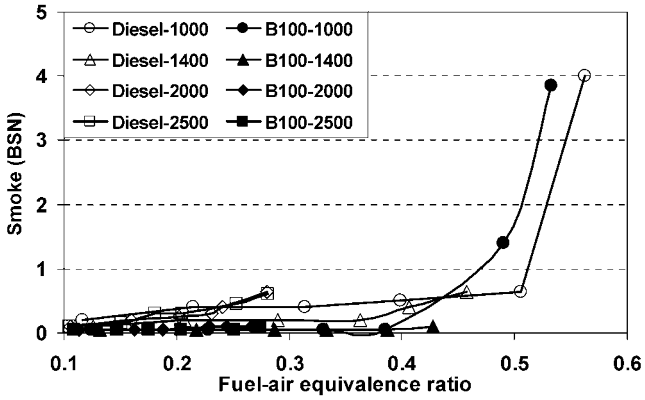


Fig. 20 Variations of smoke emissions with fuel–air equivalence ratio for neat biodiesel in comparison with diesel

CO concentrations at different test conditions except that the reduction in smoke number with neat biodiesel fuel over diesel operations decreases in a significant proportion. A maximum decrease of 6.5 times is observed at full load and engine speed of 1400 r/min. From these observations, it is evident that there is a significant improvement in diffusion phase combustion owing to the fuel-bound oxygen available in the neat biodiesel fuel molecule. The similar observations of lower smoke density are reported by Ramadhas *et al.* [**14**] in their engine experiments with neat biodiesel derived from rubber seed oil compared with diesel fuel.

5.9.5 Nitric oxide versus smoke trade-off

The smoke and NO are the major constituents of diesel exhaust, which are primarily formed owing to the heterogeneous nature of the in-cylinder charge and higher cylinder charge temperatures respectively. The NO emissions are controllable through the retarded injection timing, exhaust gas recirculation, and water injection, but any reduction in the cylinder charge temperature tends to increase smoke emissions. Hence, NO and smoke trade-off is a classical problem for investigation on diesel emission aspects. Figure 21 shows a trade-off between NO and smoke emissions for neat biodiesel and diesel fuel at full load with varying engine speeds. These results suggest that there is a slight increase in NO concentration but smoke emissions are lowered in the case of neat biodiesel fuel as compared with the diesel fuel. The NO concentrations and the smoke emissions at all loads and the engine speed exceeding 1000 r/min remained within 600 ppm and

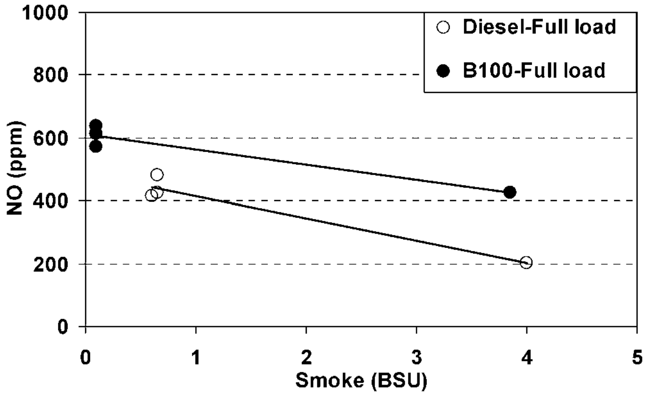


Fig. 21 Trade-off between nitric oxide and smoke emissions with neat biodiesel and diesel at full load under varying speed conditions

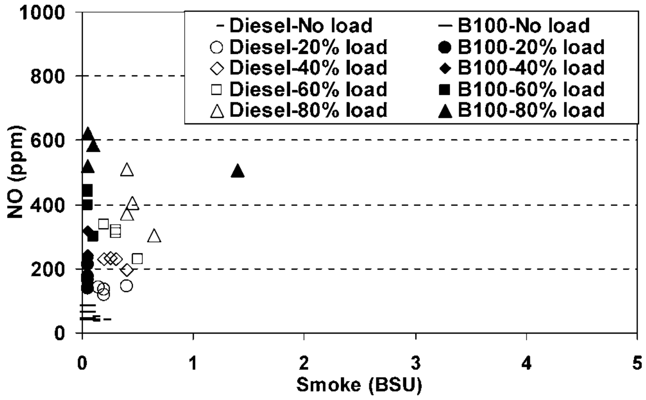


Fig. 22 Trade-off between nitric oxide and smoke emissions with neat biodiesel and diesel from no load to 80 per cent load under varying speed conditions

1 BSU respectively. At an engine speed of 1000 r/min and all other loads except full load, the nitric oxide and smoke emissions are found to remain in the same range, as shown in Fig. 22. Nevertheless, the increase in smoke emissions observed at full load and 1000 r/min engine speed provide significantly higher emissions than any other operating conditions investigated.

6 CONCLUSIONS

The present investigation evaluates the combustion characteristics, performance, and emissions of karanja-derived biodiesel fuel compared with diesel fuel in a turbocharged, direct injection diesel engine under varying load and speed conditions of engine operations. This experimental investigation comparing karanja biodiesel fuel with diesel reveals the following:

1. The use of biodiesel provides an advance injection (dynamic) timing and an early onset of combustion as compared with the diesel fuel. This injection advance is greater at higher speeds. A maximum injection advance of 2.3°CA is observed at an engine speed of 2500 r/min.
2. The use of biodiesel shortens the ignition delay compared with the diesel fuel up to an engine speed corresponding to the maximum torque condition, i.e. 1400 r/min. The difference in ignition delay between the fuels reduces at higher speeds. The maximum rate of pressure rise also at these operating conditions follows a trend of the ignition delay variations. However, the peak cylinder pressure and peak energy release rate increase for biodiesel fuel compared with diesel fuel at all operating conditions because of an early start of combustion and better combustion in diffusion phase owing to fuel bound oxygen in the biodiesel fuel.
3. The use of biodiesel provides a higher exhaust gas temperature difference than the diesel fuel values at the lower (1000 r/min) end of the engine speed, which tends to minimize at intermediate engine speeds of 1400 and 2000 r/min but lowers at the higher engine speed of 2500 r/min.
4. The differences in thermal efficiencies between neat biodiesel and diesel fuel are not significant at most of the operating conditions.
5. There is a significant reduction in smoke and unburnt HC emissions with use of biodiesel. However, there is an increase in NO concentration without significant changes in CO emissions.
6. The exhaust concentrations of NO and smoke for biodiesel remain within 600 ppm and 1 BSU respectively at almost all engine operating conditions except at full load, 1000 r/min operation.

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