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Experimental investigation of the combustion characteristics of a biodiesel (rice-bran oil methyl ester)-fuelled direct-injection transportation diesel engine

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Abstract: Increased environmental awareness and depletion of fossil petroleum resources are driving industry to develop alternative fuels that are environmentally more acceptable. Transesterified vegetable oil derivatives called 'biodiesel' appear to be the most convenient way of utilizing bio-origin vegetable oils as substitute fuels in diesel engines. The methyl esters of vegetable oils do not require significant modification of existing engine hardware. Previous research has shown that biodiesel has comparable performance and lower brake specific fuel consumption than diesel with significant reduction in emissions of CO, hydrocarbons (HC), and smoke but slightly increased NO_x emissions. In the present experimental research work, methyl ester of rice-bran oil is derived through transesterification of rice-bran oil using methanol in the presence of sodium hydroxide (NaOH) catalyst. Experimental investigations have been carried out to examine the combustion characteristics in a direct injection transportation diesel engine running with diesel, biodiesel (rice-bran oil methyl ester), and its blends with diesel. Engine tests were performed at different engine loads ranging from no load to rated (100 per cent) load at two different engine speeds (1400 and 1800 r/min). A careful analysis of the cylinder pressure rise, heat release, and other combustion parameters such as the cylinder peak combustion pressure, rate of pressure rise, crank angle at which peak pressure occurs, rate of pressure rise, and mass burning rates was carried out. All test fuels exhibited similar combustion stages as diesel; however, biodiesel blends showed an earlier start of combustion and lower heat release during premixed combustion phase at all engine load–speed combinations. The maximum cylinder pressure reduces as the fraction of biodiesel increases in the blend and, at higher engine loads, the crank angle position of the peak cylinder pressure for biodiesel blends shifted away from the top dead centre in comparison with baseline diesel data. The maximum rate of pressure rise was found to be higher for diesel at higher engine loads; however, combustion duration was higher for biodiesel blends.

Keywords: biodiesel, rice-bran oil methyl ester, combustion characteristics, P–θ diagram, burn rate, heat release

1 INTRODUCTION

Diesel fuels have an important role in the industrial economy of any country. Because of the depletion of petroleum reserves, increasing fuel prices and uncertainties concerning petroleum availability, stringent emission standards and global warming caused by carbon dioxide (CO_2) emissions, development of alternative energy sources and fuels has become increasingly important day by day. Vegetable oils have comparable energy density (10 per cent lower) and a cetane number almost similar to diesel. The idea of using vegetable oils as fuel for diesel engine is not new. When Rudolf Diesel first invented

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the diesel engine, he demonstrated it at the 1900 world exhibition in Paris, employing peanut oil and said; 'The use of vegetable oils for engine fuels may seem insignificant today, but such oils may become in course of time as important as petroleum and the coal tar products of the present time' [**1**]. Vegetable oils can be used in diesel engines either in raw form, i.e. 'straight vegetable oil' (SVO), which requires many engine modifications, or connected into biodiesel. However, several properties of SVO such as the high viscosity, high molecular weight, and low volatility cause poor fuel atomization, leading to incomplete combustion, which results in problems such as severe engine deposits, injector coking, and piston ring sticking [**2**, **3**]. Biodiesel is a monoalkyl-ester-based oxygenated fuel made from natural renewable sources such as vegetable oils and animal fats. The blends of biodiesel are referred to as Bxx where xx refers to the percentage of ester in the blend. Biodiesel is prepared by chemically modifying vegetable oil using the process of transesterification. Transesterification is a chemical process in which triglycerides in vegetable oils (from agriculture and forest resources available locally) are converted to monoalkyl esters of the fatty acids, called biodiesel, using primary alcohols in the presence of a catalyst [**4**, **5**]. The chemical scheme of transesterification is shown in Fig. 1. The resultant product, i.e. biodiesel, has a comparatively higher cetane number and heating value close to diesel (higher than SVOs and marginally lower than diesel). Additionally, the esters of vegetable oils are non-toxic, biodegradable, sulphur-free, and renewable fuels that reduce exhaust emissions. The physical and chemical characteristics of the raw vegetable oil can be improved by transesterification [**6–9**], and a reduction in the fuel viscosity (ASTM D445) may be achieved which otherwise causes problems such as fuel pumping system failure, injector deposits and coking, and lubricating oil dilution. Biodiesel usage may help to reduce greenhouse gas emissions since the carbon cycle time for fixation of CO_2 from biodiesel is quite small compared with diesel and very little energy is required for fuel production [**10**]. Depending upon the climate and soil conditions, several nations are looking into different vegetable oils as substitute for diesel fuels, e.g. soybean oil in the USA and Brazil, rapeseed and sunflower oils in Europe, palm oil in Malaysia, linseed and olive oil in Spain, cottonseed oil in Greece, beef tallow in Ireland, used frying oils in Austria, jatropha in Nicargua and South Americas, and several other oils in different countries. Many studies have shown that diesel engines can run with biodiesel successfully, and the performance of the engine is comparable with that of diesel.

Some researchers reported improved performance, particularly thermal efficiency for biodiesel-fuelled engines [**6–9**, **11–17**]. With the use of biodiesel as a fuel in the engine, there is considerable reduction in emissions such as CO, HC, and smoke, but NO_x emissions increase [**6–8**, **11**, **13–18**]. A host of non-edible vegetable oils such as linseed, mahua, karanja, rice-bran, and jatropha are available in abundance in developing countries and are underutilized. Some of these oils already have been evaluated as substitutes for diesel fuel. Agarwal [**6**] and Agarwal and Das [**7**] used linseed oil methyl ester as a fuel in a stationary engine. Other researchers used oils such as jatropha and karanja in a stationary diesel engine [**17**, **19**]. Agarwal and Das [**7**] and Raheman and Phadatare [**17**] found that 20 per cent blend of biodiesel is the optimum blend; B20 gave the maximum thermal efficiency of all biodiesel blends. Similar results were also reported in an earlier study on rice-bran biodiesel used in a transportation diesel engine by the present authors [**8**].

Boehman *et al.* [**19**] investigated the benefits of moving from low-sulphur diesel to ultralow-sulphur diesel, Fischer–Tropsch diesel, and blending with biodiesel and B100 (soybean oil methyl ester). The impact of fuel formulation on the fuel injection timing, bulk modulus of compressibility, in-cylinder combustion process, gaseous and particulate emission, and the diesel particulate filter regeneration temperature was

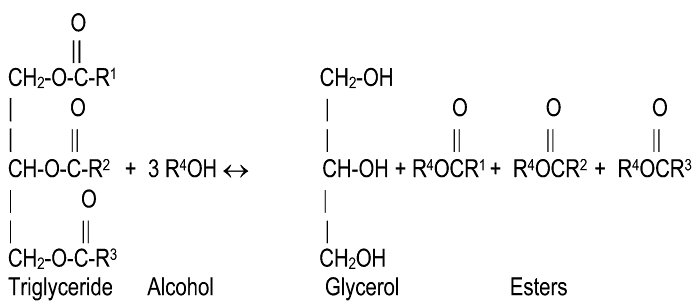

Fig. 1 Basic transesterification reaction

examined. It was reported that the fuel injection timing is affected by the bulk modulus of the fuel. The bulk modulus for soy-based biodiesel was lower than that of raw soy oil but higher than that of diesel; hence, in a mechanical fuel injection system, injection timing was advanced by 0.3° crank angle and 1° crank angle for B20 and B100 respectively. Combustion was reported to start earlier in the case of B20 and B100 fuel in comparison with diesel.

Zhang and Van Gerpan [**14**] investigated the combustion characteristics of turbocharged direct-injection diesel engine using blends of methyl, isopropyl and winterized methyl ester of soybean oil with diesel as a fuel. They investigated the combustion characteristics at the maximum torque engine speed and for 20 per cent and 100 per cent engine load. They found that all fuel blends except isopropyl ester had similar combustion behaviour. Fuel injection starts earlier for high engine loads. The cetane number was higher for biodiesel and its blends in comparison with diesel. All blends had a shorter ignition delay and a lower premixed burn fraction than diesel. Senatore *et al.* [**15**] reported that with the rapeseed oil methyl ester, heat release always took place earlier than diesel, because fuel injection starts earlier for biodiesel blends owing to their higher density, leading to a higher peak cylinder temperature. McDonald *et al.* [**18**] investigated soybean oil methyl ester as a fuel on a Caterpillar indirect-injection diesel engine and found that overall combustion characteristics were quite similar to those of diesel except for a shorter ignition delay for soybean methyl ester. Kumar *et al.* [**20**] found that, for jatropha oil methyl ester, ignition delay was longer than diesel in a constant-speed diesel engine. Selim *et al.* [**9**] tested jajoba oil methyl ester (JME) as a fuel in a Ricardo compression swirl diesel engine and found that the cylinder pressures and pressure rise rates for JME are almost similar to those of gas oil. JME, however, exhibits slightly lower pressure rise rate than gas oil and seems to have slightly delayed combustion. Several researchers investigated different biodiesel fuels; however, in-depth combustion analysis for transportation engines at different speeds and engine loads is seldom reported.

The previous study on a direct-injection transportation diesel engine with rice-bran oil methyl ester as a fuel showed improved performance of the engine with a reduction in the exhaust emissions except for NO_x [**8**]. The objectives of this experimental study were to investigate the combustion behaviours of B10, B20, and B100 at different engine loads (no load, 10 per cent, 25 per cent, 50 per cent, 75 per cent, and 100 per cent rated load) and speeds (1400 r/min and 1800 r/min speed at maximum torque) and to compare them with the baseline data of diesel in an unmodified transportation diesel engine (medium duty).

2 FUEL PREPARATION AND CHARACTERIZATION

Rice-bran oil was transesterified, using methanol in the presence of sodium hydroxide (NaOH) catalyst. Process parameters such as the temperature, catalyst amount, molar ratio of alcohol to oil, and reaction time were optimized and these were as follows: temperature, 55 °C; catalyst amount, 0.75 per cent (w/w); molar ratio of alcohol to oil, 9 to 1; reaction time for transesterification of rice-bran oil, 1 h [**8**]. For transesterification, rice-bran oil was heated in a round-bottom flask; NaOH was dissolved in methanol in a separate vessel and was poured into the round-bottom flask, while stirring the mixture continuously. The mixture was stirred while being maintained at 55 °C for 1 h. The reaction products were kept in a separating funnel for 24 h. The products formed during transesterification were rice-bran oil methyl ester and glycerol. Since glycerol is heavier, it forms lower layer during gravity separation, which is separated. The ester was washed with 10 per cent (v/v) warm (70 °C) water and kept for 24 h for removal of the catalyst. The catalyst gets dissolved in the water, which is then gravity separated. This biodiesel is then blended in the required quantities to form appropriate fuel blends. Characterization of rice-bran oil, rice-bran oil methyl ester, and diesel was performed in the laboratory as per ASTM standards. Some fuel properties evaluated are shown in Table 1.

3 EXPERIMENTAL SET-UP AND PROCEDURE

3.1 Experimental set-up

A transportation direct-injection diesel engine (Make: Mahindra & Mahindra Ltd, India, model MDI-3000) was used for conducting the engine investigations. The specifications of the engine are given in Table 2.

The engine was coupled with an eddy current dynamometer and controller (Schenck Avery, India, model ASE 70). Data for the engine speed (r/min), torque (N m), inlet and outlet cooling water temperatures, exhaust gas temperature, etc., were collected. The engine speed and load were controlled by varying the excitation current to the eddy current dynamometer. A schematic diagram of the experimental

Table 1 Properties of the fuels

Characteristic	Rice-bran oil	Diesel	Rice-bran oil methyl ester
Specific gravity at 30 °C	0.928	0.839	0.877
Viscosity (cSt) at 40 °C	42	3.18	5.3
Flash point (°C)	316	68	183
Fire point (°C)	337	103	194
Cloud point (°C)	13	6	9
Pour point (°C)	1	−7	−2
Lower heating value (MJ/kg)	41.1	44.8	42.2
Cetane number	50.1	51	63.8
Conradson carbon residue (%, w/w)	0.6	0.1	0.35
Carbon: hydrogen: oxygen (%)	74 : 11 : 12	84 : 13 : 1	73 : 13 : 11.6

Table 2 Specifications of the engine

Type of engine	Four stroke, naturally aspirated, direct injection, water cooled
Number of cylinders	4
Bore (mm)	88.9
Stroke (mm)	101.6
Displacement volume (cm^3)	2520
Compression ratio	18 : 1
Rated power	40.4 kW at 3000 r/min
Maximum torque	152 N m at 1800 r/min

set-up is shown in Fig. 2. This engine was used for conducting investigations on different fuels. No engine hardware modifications or adjustments were carried out. The fuel injection pressure, fuel pump setting, etc, were the same throughout the engine experiments for all the fuel samples at different engine operating conditions.

A piezoelectric pressure transducer (AVL, Austria, model GU21C) was installed in the engine cylinder head (first cylinder) to acquire the combustion pressure–crank angle history. Signals from the pressure transducer were amplified using a charge amplifier. A high-precision shaft encoder was used for delivering signals for top dead centre (TDC) and the crank angle with a precision of 0.1° crank angle. The signals from the charge amplifier and shaft encoder were collected using a high-speed data acquisition system (AVL, Austria, model Indimeter-619) as shown in the schematic diagram in Fig. 2.

3.2 Combustion analysis

Fuels including diesel (B00), B10, B20, and biodiesel (B100) were tested at five different engine loads (no load, 25 per cent, 50 per cent, 75 per cent and 100 per cent of rated load) at engine speeds of 1400 r/min and 1800 r/min. The cylinder pressure data were recorded for 50 consecutive cycles and then averaged in order to eliminate the effect of cycle-to-cycle variations. All tests were carried out under steady state engine conditions.

The details about combustion stages and events can often be determined by analysing the heat release rates. The trend of heat release (instantaneous heat release rate and cumulative heat release) can be obtained by processing in-cylinder pressure data. The analysis for the heat release rate is based on the application of the first law of thermodynamics for an open system [**21**]. It is assumed that the cylinder

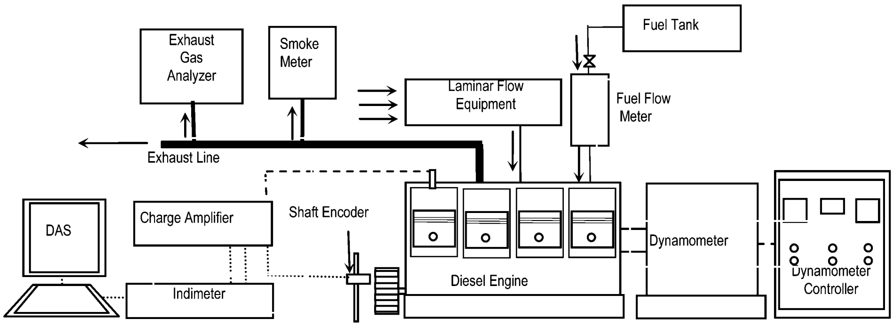

Fig. 2 Schematic diagram of the experimental set-up

contents is a homogeneous mixture of air and combustion products and is at uniform temperature and pressure at each instant in time during the combustion process. The first law for such a system can be written as

$$\frac{\mathrm{d}U}{\mathrm{d}t}=\frac{\mathrm{d}Q}{\mathrm{d}t}+\sum m_{\mathrm{i}}h_{\mathrm{i}}-p\frac{\mathrm{d}V}{\mathrm{d}t} \tag{1}$$

where $\mathrm{d}q/\mathrm{d}t$ is the heat transfer rate across the system boundary, $p(\mathrm{d}V/\mathrm{d}t)$ is the rate of work transfer done by the system due to system boundary displacement, and m_{i} and h_{i} are the mass and enthalpy respectively of the flow into the system. p and V are the pressure and volume respectively of the cylinder and U is the internal energy of the cylinder contents. If the crevice effect is neglected, the above equation can be reduced to

$$\frac{\mathrm{d}U}{\mathrm{d}t}=\frac{\mathrm{d}Q}{\mathrm{d}t}+m_{\mathrm{f}}h_{\mathrm{f}}-p\frac{\mathrm{d}V}{\mathrm{d}t} \tag{2}$$

where m_{f} and h_{f} are the fuel mass flowrate and enthalpy respectively of the fuel entering the engine cylinder.

For an ideal gas this equation can be further reduced to

$$\frac{\mathrm{d}Q_{\mathrm{n}}}{\mathrm{d}t}=\frac{\mathrm{d}Q_{\mathrm{ch}}}{\mathrm{d}t}-\frac{\mathrm{d}Q_{\mathrm{hw}}}{\mathrm{d}t}=p\frac{\mathrm{d}V}{\mathrm{d}t}+mc_V\frac{\mathrm{d}T}{\mathrm{d}t} \tag{3}$$

where the net heat release rate $\mathrm{d}Q_{\mathrm{n}}/\mathrm{d}t$ is the difference between the gross heat release rate $\mathrm{d}Q_{\mathrm{ch}}/\mathrm{d}t$ and the heat transfer rate $\mathrm{d}Q_{\mathrm{hw}}/\mathrm{d}t$ to the walls; after eliminating T from the ideal gas relation ($pV=mRT$), this equation can be further reduced to

$$\frac{\mathrm{d}Q_{\mathrm{n}}}{\mathrm{d}t}=\frac{\mathrm{d}Q_{\mathrm{ch}}}{\mathrm{d}t}-\frac{\mathrm{d}Q_{\mathrm{hw}}}{\mathrm{d}t}=\frac{\gamma}{\gamma-1}p\frac{\mathrm{d}V}{\mathrm{d}t}+\frac{1}{\gamma-1}V\frac{\mathrm{d}p}{\mathrm{d}t} \tag{4}$$

This relation makes it possible to calculate the heat release rate; all the quantities on the right-hand side are known or can be easily derived once the pressure–time history has been recorded. The crank angle for the start of combustion can be determined as the start of the measurable heat release. Owing to the fuel vaporization immediately after fuel injection in hot compressed air, the heat release curve will typically adopt negative values before the start of combustion for a direct-injection diesel engine. The start of combustion is taken as the crank angle degree where the heat release curve changes the direction of the slope, i.e. it reflects the start of measurable heat release. Other combustion parameters such as the peak cylinder pressure, rate of pressure rise, crank angle at peak pressure, and mass burning rates can be determined from the p–θ diagram. The software used for the analysis of the processing of raw acquired data is AVL Indicom (version 1.2). For the analysis of combustion parameters such as the peak pressure, crank angle for the peak pressure, mass burning rates, and rate of pressure rise, the average of 50 consecutive engine cycles has been taken. The data are analysed statistically for 95 per cent confidence level and the error bars are reported in all the figures.

4 RESULTS AND DISCUSSION

In a compression-ignition engine, the peak cylinder pressure depends on the burned fuel fraction during the premixed burning phase, i.e. the initial stage of combustion. The cylinder pressure characterizes the ability of the fuel to mix well with air and burn. The variations in the cylinder pressure with crank angle for all the blends at different engine operating conditions are shown in Figs 3 to 5. From these figures, it is clear that the peak pressure increases as the engine load increases. The peak pressure does not

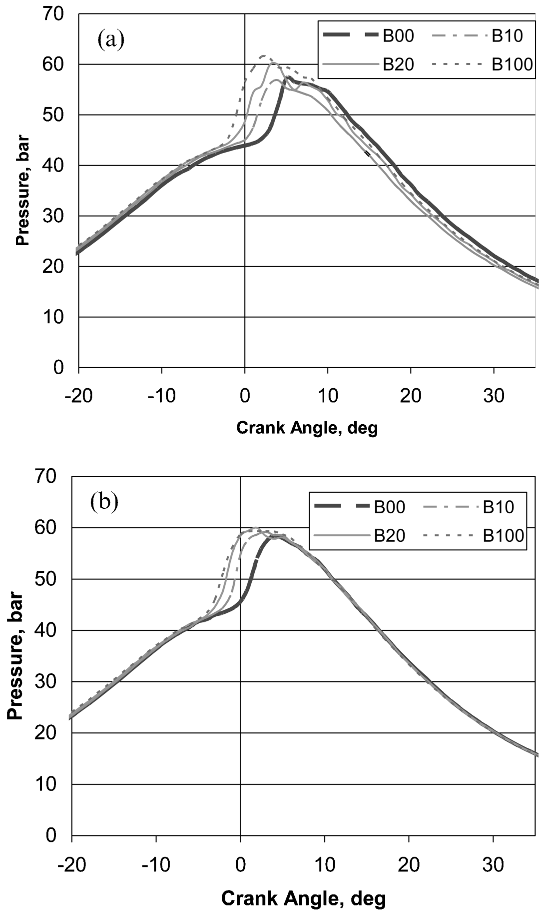

Fig. 3 Pressure–crank angle diagram for no engine load at (a) 1800 r/min and (b) 1400 r/min

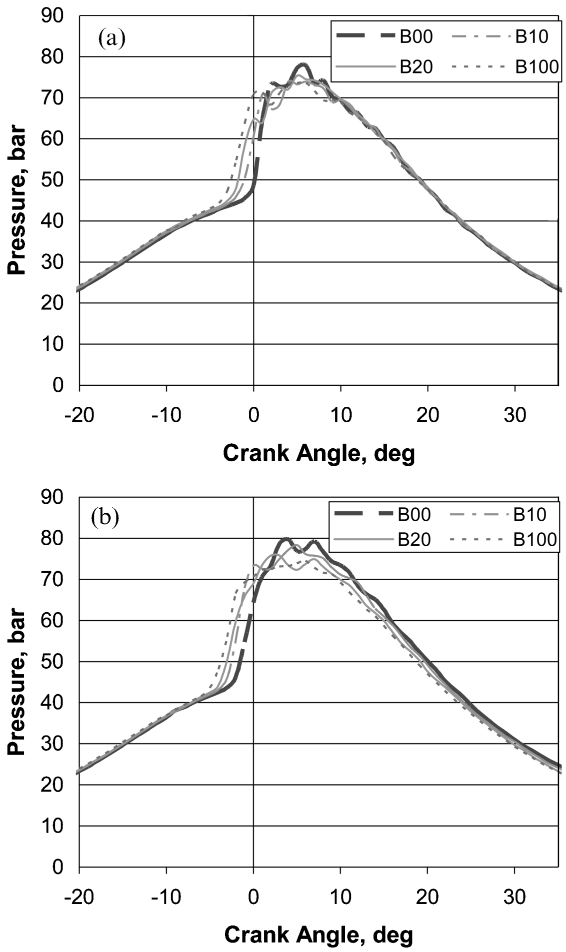

Fig. 4 Pressure–crank angle diagram for 50 per cent of the rated engine load at (a) 1800 r/min and (b) 1400 r/min

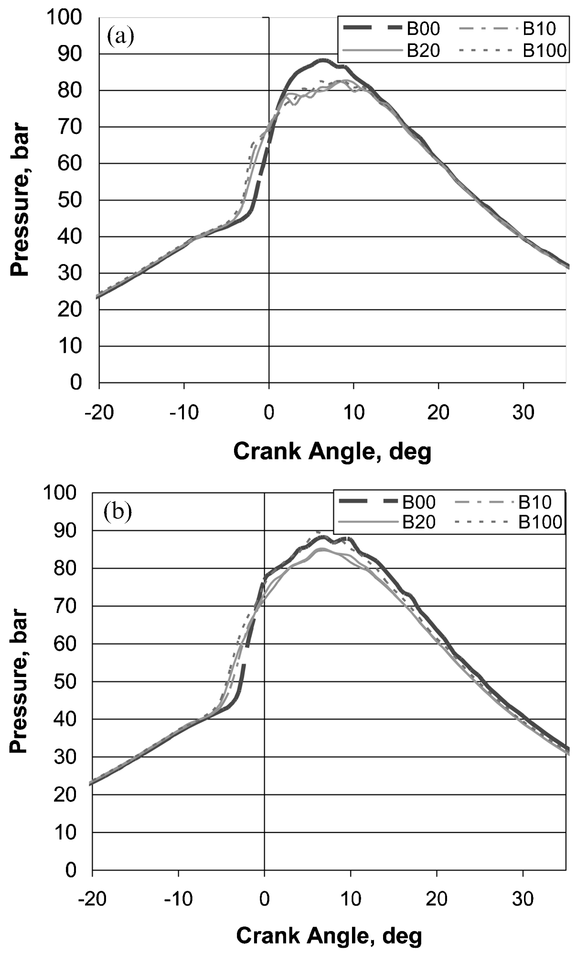

Fig. 5 Pressure–crank angle diagram for rated engine load at (a) 1800 r/min and (b) 1400 r/min

vary significantly with increasing engine speed. The peak pressure is higher for biodiesel blends at low loads (Fig. 3) but, at higher engine loads, the peak pressure for diesel is higher (Fig. 5). The peak cylinder pressure increases as the proportion of biodiesel in the blends is increased but, for all biodiesel blends at higher loads, the peak pressure is lower than for diesel (Figs 3 to 5). This may be because, at low engine loads, the peak pressure for blends occurs near the TDC. At all engine loads, combustion starts earlier for biodiesel blends than for diesel. As the engine load is increased, the combustion start point comes closer for all the fuels. Ignition delay for all fuels decreases as the engine load increases because the gas temperature inside the cylinder is higher at high engine loads, which reduces the physical ignition delay. Ignition delay represents the time taken in physical and chemical preflame reactions and does not vary much on a timescale of milliseconds. However, it will increase in terms of crank angle degrees with increasing engine speed as a high engine speed will correspond to a larger crank angle for the same time duration. In this study, ignition delay was not measured; however, the start of combustion may reflect the variation in ignition delay because fuel pump and injector settings were kept identical for all fuel samples. Combustion starts earlier for biodiesel (Figs 3 to 5) partially owing to a shorter ignition delay and partially owing to advanced injection timing (because of a higher bulk modulus and higher density of biodiesel). In spite of the slightly higher viscosity and lower volatility of the biodiesel, the ignition delay seems to be lower than for diesel. This may possibly be because a complex and rapid preflame chemical reaction takes place at high temperatures. As a result of the high in-cylinder temperature existing during fuel injection, biodiesel may undergo thermal cracking; as a result of this, lighter compounds are produced, which might have ignited earlier, resulting in a shorter ignition delay

[22]. Biodiesel also has a higher cetane number than diesel (Table 1) and thus has a shorter ignition delay than diesel.

Figures 6 to 8 show the heat release rate diagrams for all fuels at different engine operating conditions. Because of the vaporization of the fuel accumulated during ignition delay, at the beginning a negative heat release is observed and, after combustion is initiated, this becomes positive. All biodiesel blends experience identical combustion stages as diesel. After the ignition delay, premixed fuel air mixture burns rapidly, followed by diffusion combustion, where the burn rate is controlled by fuel–air mixing. It can be observed that combustion starts earlier for biodiesel blends under all engine operating conditions, as found by other researchers [**14–16**], and it becomes more prominent with higher biodiesel blends. The premixed combustion heat release is higher for diesel owing to higher volatility and better mixing of diesel with air. Another reason possibly

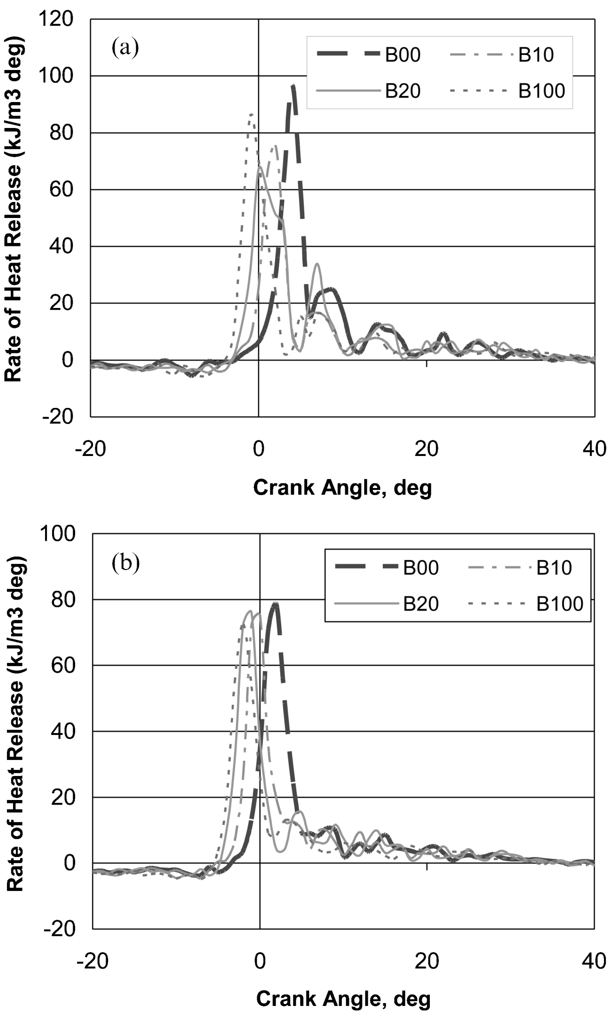

Fig. 6 Instantaneous rate of heat release for no engine load at (a) 1800 r/min and (b) 1400 r/min

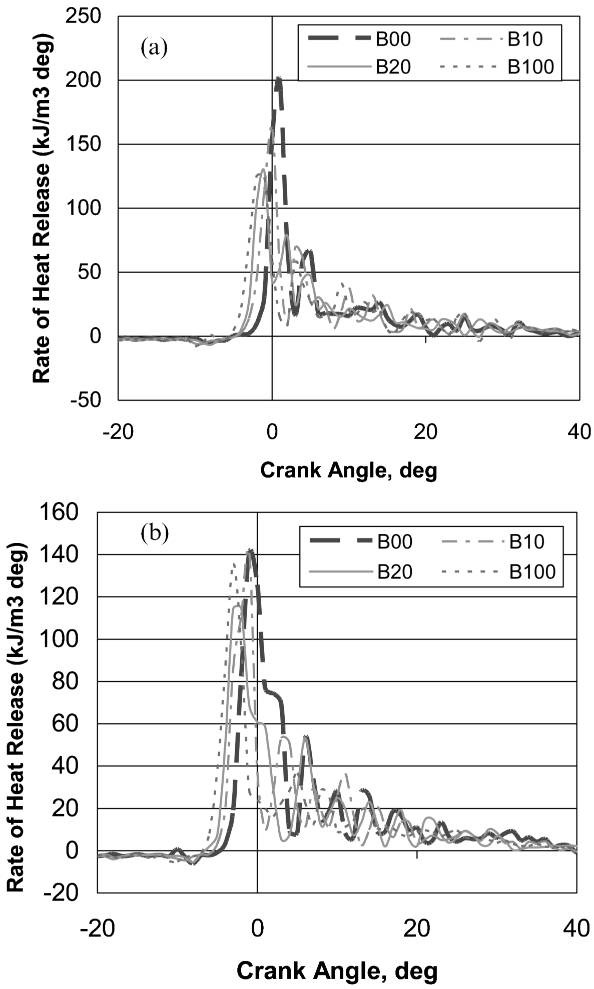

Fig. 7 Instantaneous rate of heat release for 50 per cent of rated engine load at (a) 1800 r/min and (b) 1400 r/min

may be the longer ignition delay of diesel, which leads to a larger amount of fuel accumulation in the combustion chamber at the time of the premixed combustion stage, leading to a higher rate of heat release.

Figures 9 to 11 show the cumulative heat release for all fuels at different engine operating conditions. These figures show the tendency of earlier release of fuel energy for biodiesel blends, which becomes less prominent at higher engine loads. Combustion for diesel starts later but quickly it exceeds the cumulative heat released for biodiesel blends, suggesting a faster burn rate of diesel. Cumulative heat release decreases as the proportion of biodiesel increases in the blend, owing to the lower heating value of the biodiesel.

Figure 12 shows the crank angle for 5 per cent mass fraction burned. This figure shows that 5 per cent fuel burns earlier for biodiesel blends and it burns

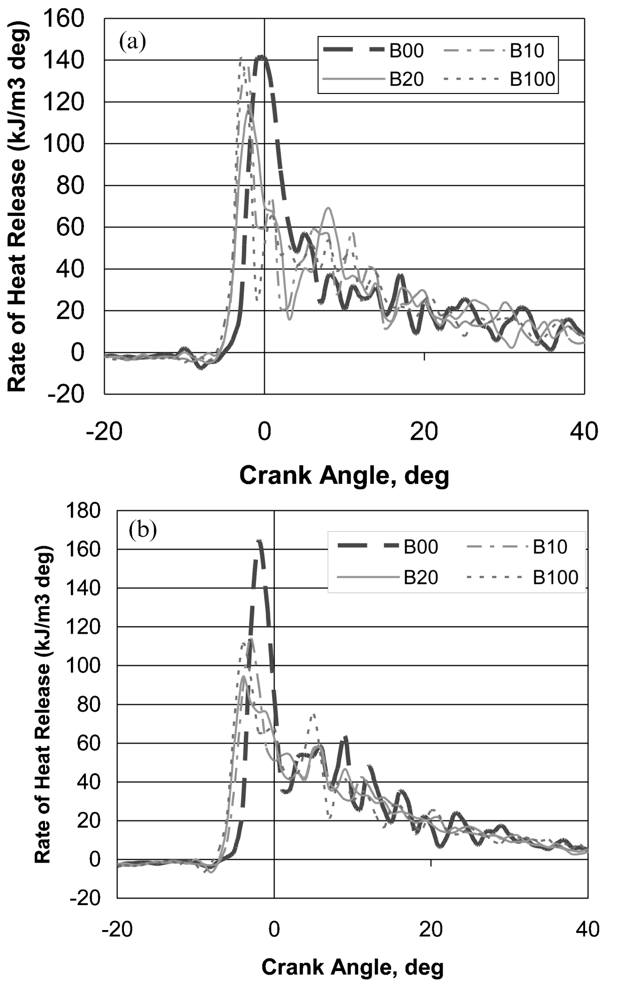

Fig. 8 Instantaneous rate of heat release for rated engine load at (a) 1800 r/min and (b) 1400 r/min

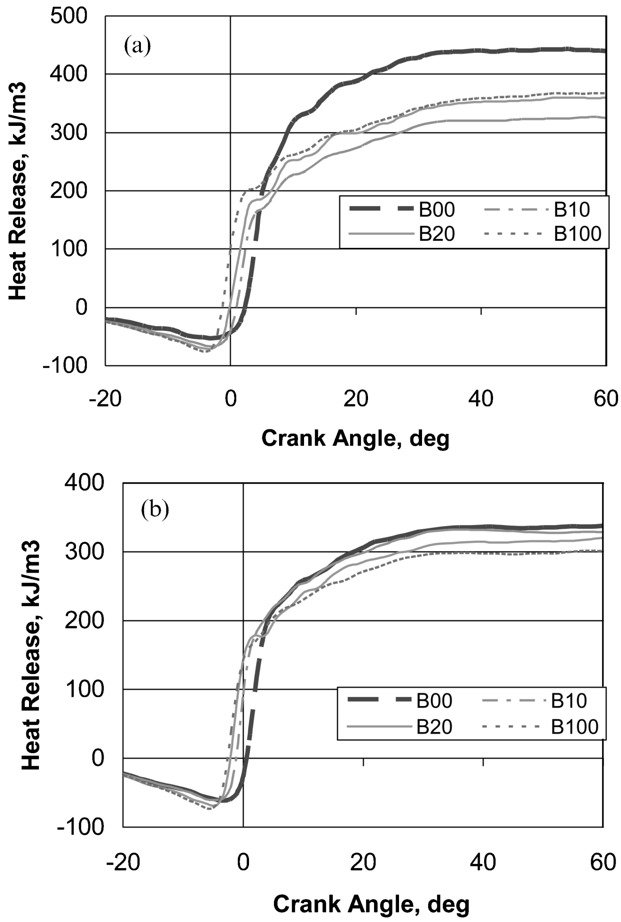

Fig. 9 Cumulative heat release for no engine load at (a) 1800 r/min and (b) 1400 r/min

successively earlier for an increasing proportion of biodiesel in blend. This is due to the earlier start in combustion for biodiesel blends, as suggested earlier. As the engine load is increased, this deviation decreases because, at higher loads, the difference in the combustion start crank angle decreases (Figs 3 to 5). Figure 13 shows the crank angle degree for 90 per cent mass fraction burned at different engine operating conditions. This increases with increasing engine load because a larger fuel quantity needs to be injected for higher loads. This figure suggests that 90 per cent mass fraction burns earlier for diesel, because of the faster rate of heat release. Biodiesel has a higher flash point (183 °C (Table 1)) and it has a lower volatility, which cause slower burning of biodiesel in the first place. More fuel is required in the case of biodiesel blends because the calorific value of these blends is lower than that of diesel. These factors lead to a longer combustion duration for biodiesel blends.

Figure 14 shows the maximum cylinder pressure at different loads for different fuels. It shows that, at higher engine loads, the peak pressure for diesel is higher than for biodiesel blends; however, in the case of different biodiesel blends, the change in the peak pressure is not significant. The peak pressure for diesel is higher because of the longer ignition delay, during which more fuel is accumulated in the combustion chamber to release higher heat during the premixed combustion phase. The possible reason for the opposite trends in the peak pressure at low and high engine loads is because the ignition delay increases with decrease in engine load. As the engine load decreases, the residual gas temperature and wall temperatures decrease, leading to lower charge temperature at injection, thus lengthening the ignition delay [**21**]. At very low engine loads (particularly idling and 10 per cent rated load) because of the longer ignition delay, combustion starts later for diesel than for biodiesel blends. As evident from the P–θ diagram at the no-load condition for diesel

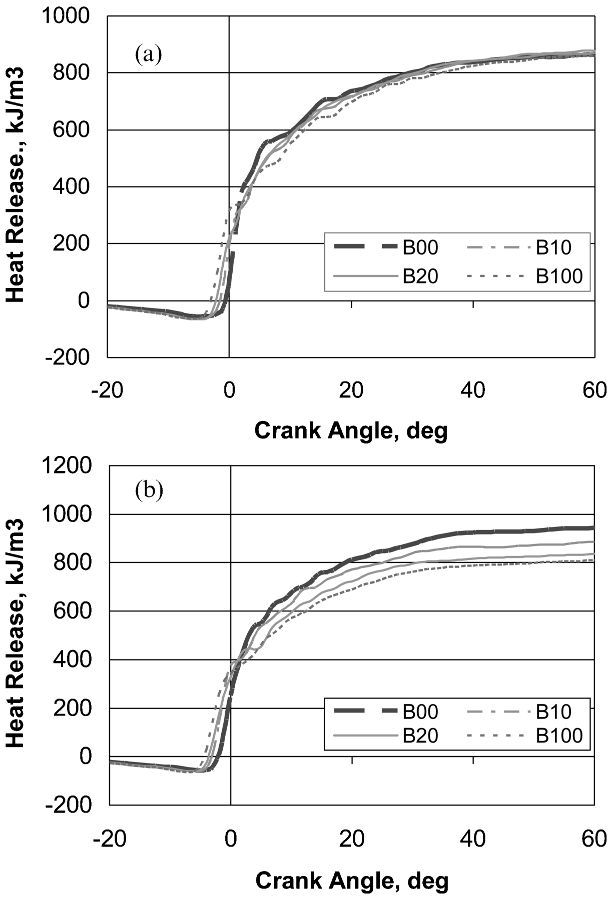

Fig. 10 Cumulative heat release for 50 per cent of rated engine load at (a) 1800 r/min and (b) 1400 r/min

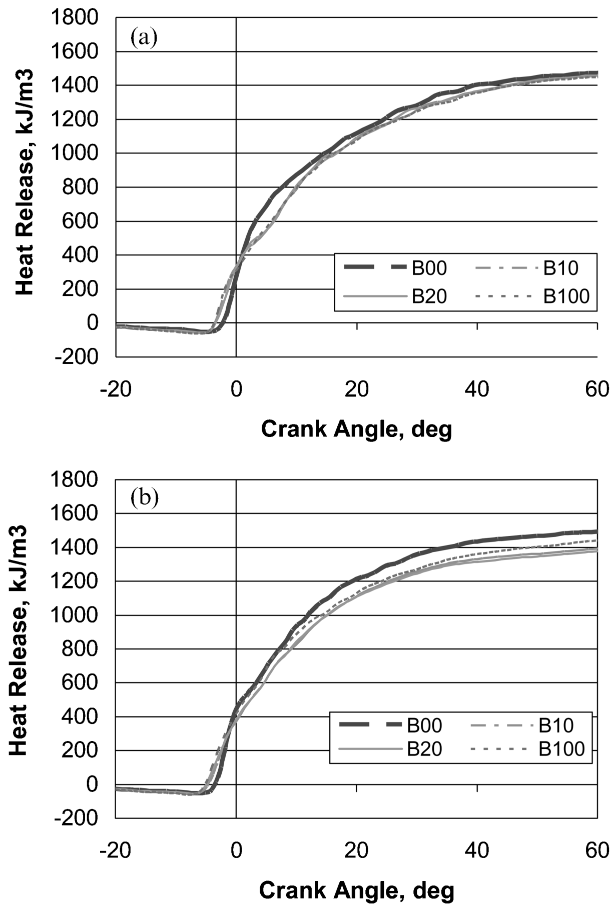

Fig. 11 Cumulative heat release for 100 per cent of rated engine load at (a) 1800 r/min and (b) 1400 r/min

(Figs 3(a) and 3(b)), combustion starts near the TDC at 1400 r/min and immediately after the TDC at 1800 r/min owing to the increase in ignition delay (as explained earlier). As a result, the peak cylinder pressure attains a lower value as it is further away from the TDC in the expansion stroke at low engine loads. Figure 15 shows the variation in the rate or pressure rise $dP/d\theta$ with engine loads for all fuels. The rate of pressure rise varies from 5 bar/deg at lower engine loads to 12 bar/deg at higher engine loads. $dP/d\theta$ decreases as the fraction of biodiesel increases in the blend. This is possibly because biodiesel contains heavier hydrocarbon molecules which have a higher boiling point and lower volatility. At no load, the rate of pressure rise for diesel is slightly lower than for biodiesel blends because, at this engine condition, a very small quantity of fuel is injected into the combustion chamber and combustion starts after the TDC (Fig. 3) for diesel, having a slightly higher delay period as mentioned earlier. However, the rate of pressure rise is higher for diesel at higher engine loads (Fig. 15) because of higher rate of heat release during premixed combustion as evident from the discussion in the later part of the paper. Figure 16 shows the crank angle, at which the peak cylinder pressure is attained for all fuels at different engine operating conditions. Figure 16 shows that maximum cylinder pressure is attained within 2–8° crank angle after the TDC for all fuels under different operating conditions. This figure shows that the peak cylinder pressure occurs earlier in the cycle for biodiesel blends at lighter engine loads (up to 25 per cent rated load). However, at higher engine loads, the peak pressure occurs at nearly the same crank angle position for all fuels. As the engine load increases, the ignition delay of diesel decreases, resulting in initiation of combustion before the TDC and the pressure rises more quickly because of higher premixed burning. This again reaffirms slower burning characteristics of biodiesel. The maximum rate of pressure rise for biodiesel and its blends are lower than for diesel, which shows

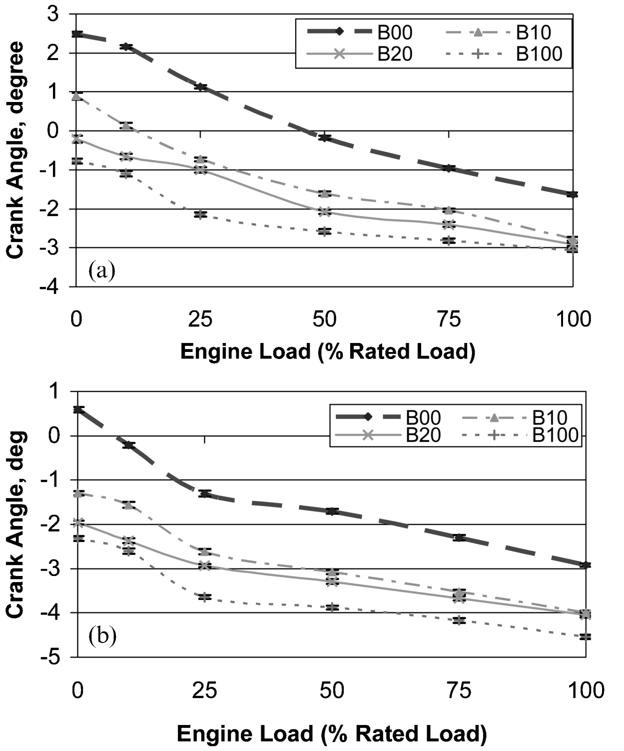

Fig. 12 Crank angle for 5 per cent fuel mass burning at (a) 1800 r/min and (b) 1400 r/min

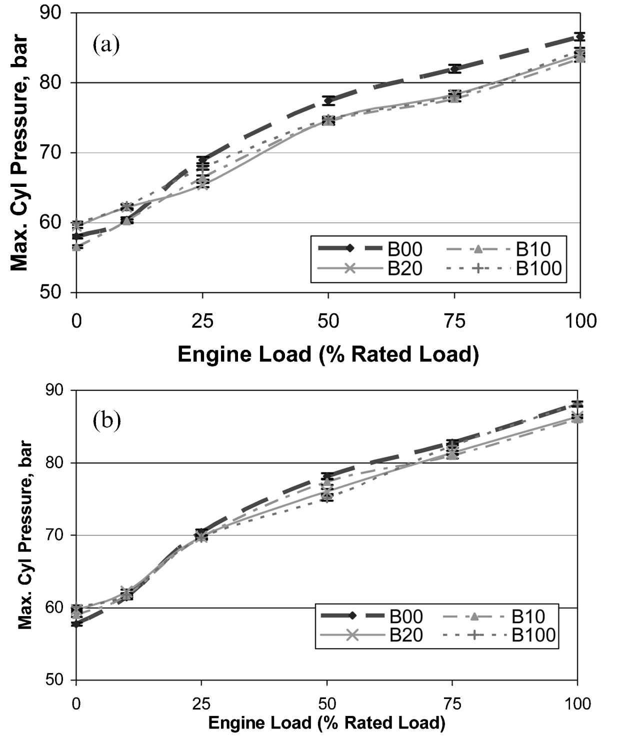

Fig. 14 Maximum cylinder pressure at (a) 1800 r/min and (b) 1400 r/min

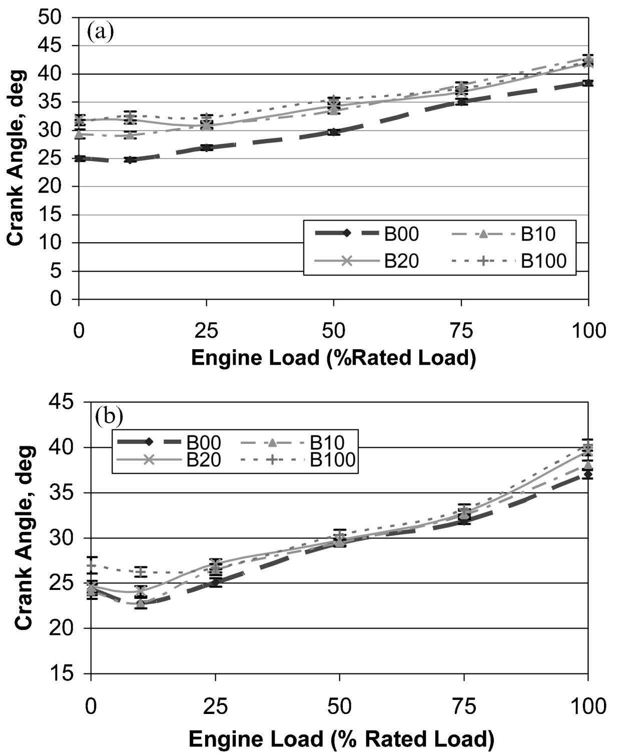

Fig. 13 Crank angle for 90 per cent fuel mass burning at (a) 1800 r/min and (b) 1400 r/min

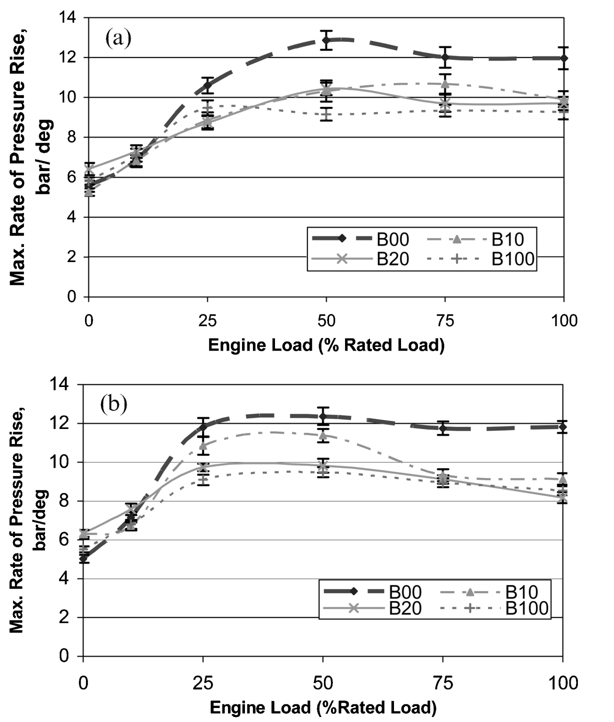

Fig. 15 Maximum rate of pressure rise at (a) 1800 r/min and (b) 1400 r/min

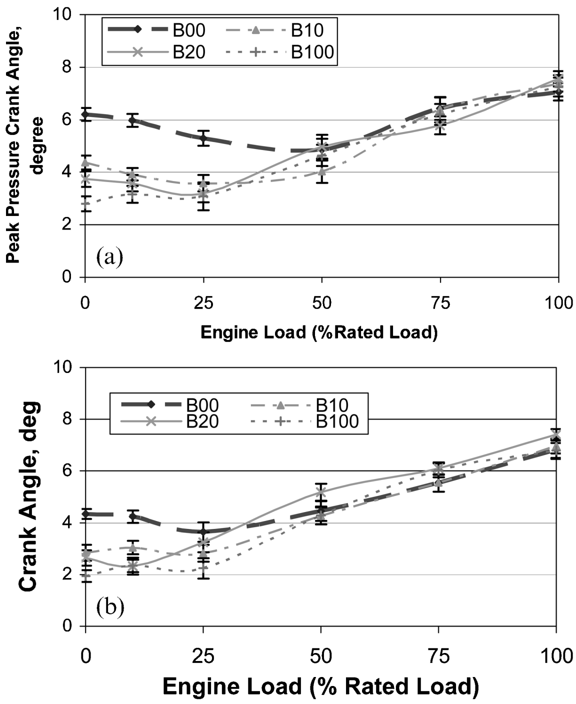

Fig. 16 Crank angle for the maximum cylinder pressure at (a) 1800 r/min and (b) 1400 r/min

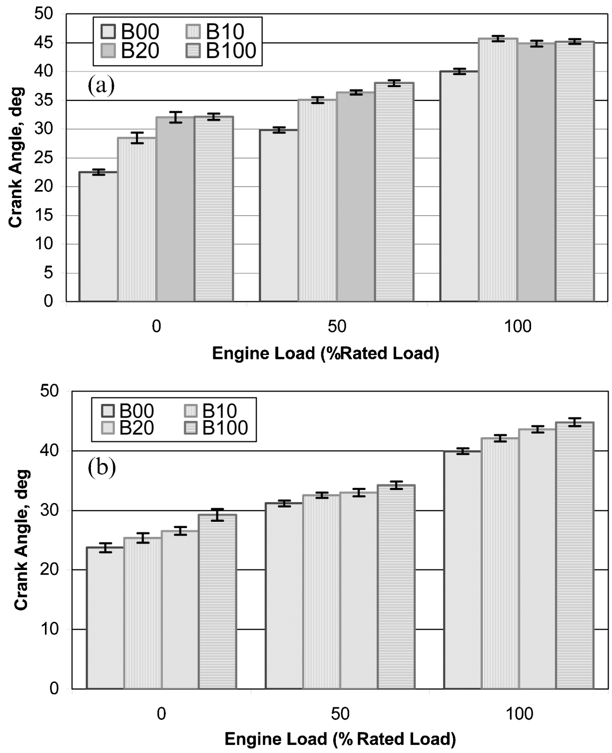

Fig. 17 Variation in the combustion duration at (a) 1800 r/min and (b) 1400 r/min

satisfactory operation of compression-ignition engine with these fuels (Fig. 15).

Figure 17 shows the variation in combustion duration for different blends at different engine loads. Crank angle duration from 5 per cent mass burn to 90 per cent mass burn has been taken as the combustion duration for comparing the different fuels. Combustion duration increases with increases in the engine speed and engine load owing to the increase in the quantity of fuel injected. Combustion duration was observed to be higher for biodiesel blends than for diesel. It was found to increase with the increase in the proportion of biodiesel in the blend. This may be possibly due to the slower rate of combustion of biodiesel blends as shown in Figs 9 to 11.

Further 100 h endurance tests on the same engine using B20 and diesel as fuels respectively in two phases were conducted as per the Indian Standards IS: 10000 [**23**, **24**]. The engines were dismantled at the end of the tests and it was found that the engine operating with both the fuels operated successfully without any major repairs during the test duration. An important observation was that, in the case of B20, lower wear of the engine components together with lower carbon deposits on the piston, cylinder head, and injector tip were observed.

5 CONCLUSIONS

A direct-injection transportation diesel engine was operated under steady state, at different engine loads at 1400 and 1800 r/min to investigate the combustion characteristics of biodiesel (rice-bran oil methyl ester) blends with diesel. Experiments suggest that the combustion starts earlier in the case of biodiesel blends than for diesel. Analysis of the pressure–time history and heat release analysis indicate that all fuel blends exhibit similar combustion stages as diesel and no undesirable combustion features such as knocking were observed. At all engine operating conditions, biodiesel blends had lower heat release rates than diesel during the premixed combustion phase. It also showed that, under all engine operating conditions, heat release always takes place earlier for biodiesel blends than for diesel. Combustion starts even earlier as the concentration of biodiesel in the blend is increased. The combustion duration is observed to be higher for biodiesel blends than for diesel; however, the maximum rate of pressure rise for all biodiesel blends was found to be lower than for diesel, and satisfactory engine operation was observed with biodiesel blend fuels. Hence it can be concluded that biodiesel (rice-bran methyl ester) and its blends can be used in the engine without any hardware modification, and undesirable combustion features such as pre-ignition and knocking were not observed.

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