Clean Techn Environ Policy (2017) 19:1297–1312
DOI 10.1007/s10098-016-1328-3

ORIGINAL PAPER

An experimental investigation on the performance, combustion and emission characteristics of a variable compression ratio diesel engine using diesel and palm stearin methyl ester

Ambarish Datta[1] · Bijan Kumar Mandal[1]

Received: 15 August 2016 / Accepted: 29 December 2016 / Published online: 9 January 2017

Abstract An attempt has been made to use biodiesel prepared from non-edible portion of palm oil as fuel of a conventional mono-cylinder compression ignition engine. The present experimental investigation takes into account the combined effect of using blends of diesel–palm stearin biodiesel as fuels and the compression ratio on different performance, combustion and emission characteristics of the said engine. The experiments have been carried out on a single-cylinder, direct injection diesel engine at varying compression ratio of 16:1–18:1 in four steps. It is observed that the brake thermal efficiency reduces by 7.9% when neat biodiesel is used instead of diesel. But, it increases with the increase in compression ratio for all the blends. Brake specific fuel consumption and exhaust gas temperature increase with the addition of biodiesel to diesel and also with the increase in compression ratio. Heat release rate decreases with biodiesel, and it is minimum at the rated compression ratio of 17.5:1 for all the fuels considered here. On the other hand, ignition delay is found to be more with neat diesel, and it increases with the decrease in compression ratio. Significant reductions in emissions of carbon monoxide (CO), hydrocarbon (HC) and smoke are observed with biodiesel, while the emissions of oxides of nitrogen (NO_x) and carbon dioxide (CO_2) increase. The decrease in compression ratio increases the emissions of CO, HC and smoke, but the emissions of NO_x and CO_2 decrease with the decrease in compression ratio.

✉ Bijan Kumar Mandal
bkm375@yahoo.co.in

[1] Department of Mechanical Engineering, Indian Institute of Engineering Science and Technology, Shibpur, Howrah, West Bengal 711103, India

Keywords Biodiesel · Compression ratio · Performance · Combustion · Emission

Introduction

The depletion of the petroleum reserve, continuous unpredictability in fuel prices and the environmental hazards due to the burning of petro-diesel are some of the most alarming problems in the automotive sector. The applications of diesel powered vehicles are increasing day by day due to their high thermal efficiency, longer life service and low maintenance cost. Therefore, in order to meet the continuous supply of energy, protect the environment and maintain the stability in domestic economy, an alternative fuel for diesel engine is needed to be selected (Sonar et al. 2015). Biodiesel is probably a solution to this dilemma for most of the developing countries. It can be blended at any level with petroleum diesel to create a diesel–biodiesel blend. It can be used in compression ignition (diesel) engines without any major modification. Biodiesel has higher cetane number and lower heating value than mineral diesel. Generally, the emission characteristics of biodiesel are better than those of conventional diesel except higher NO_x emissions. The short term use of biodiesel was studied by many researchers, and its application in diesel engines was found to be promising (Datta and Mandal 2016).

Amarnath and Prabhakaran (2012) observed that brake specific fuel consumption decreases by 15–20% with the increase in compression ratio with karanja biodiesel and its blends with diesel. It was noted that the increase in compression ratio from 14:1 to 18:1 increased the thermal efficiency by 7%. They reported that exhaust gas temperature, CO and HC emissions and smoke density decreased with the increase in compression ratio. It was also observed that NO_x emission increased with the increase in biodiesel in the blends and also with the compression ratio. Amarnath et al. (2014) reported 1.42, 5.31 and 6.34% increase in brake thermal efficiency (BTE) with diesel, jatropha biodiesel and karanja biodiesel, respectively, for the increase in compression ratio from 14:1 to 18:1. CO and HC emissions were found to be less with both the biodiesels than with diesel. They observed a reduction in smoke opacity by 40–45% with the increase in compression ratio from 14:1 to 18:1 for different fuels.

Jindal et al. (2010) found an increase in brake specific fuel consumption (BSFC) by 25–34% and a decrease in brake thermal efficiency by 16.74% for jatropha biodiesel compared to diesel. It was also reported by them that the brake thermal efficiency improved by 5.5% with the increase in compression ratio from 16:1 to 18:1. It was also observed that with the increase in compression ratio, CO emission and smoke opacity decreased, whereas the CO_2 emission increased. On the other hand, NO_x emission and exhaust gas temperature increased with the increase in compression ratio. It was also reported by Jindal (2011) that an increase in brake thermal efficiency for neat karanja biodiesel by 5.45% with the increase in compression ratio from 16:1 to 18:1.

Hirkude and Padalkar (2014) found an increase in brake thermal efficiency of 8.51% with the increase in compression ratio from 16.5:1 to 17.5:1 using waste frying oil methyl ester blended with diesel. They found reductions in CO and PM emissions and increase in NO_x emission at higher compression ratios for all the tested fuels. Sayin and Gumus (2011) observed an enhancement in NO_x emission by 15.56% with B50 blend for an increase in compression ratio from 17:1 to 19:1. It was also noted that increase in compression ratio reduced smoke opacity and HC emission by 15.56 and 4.39%, respectively, for B50 blend with the increase in compression ratio. Brake specific fuel consumption increased by 1.78%, and brake thermal efficiency (BTE) increased by 0.55% for B50 blend with the increase in compression ratio (CR). Mohanraj and Kumar (2013) found a reduction in BSFC with an increase in compression ratio with esterified tamanu oil. The maximum brake thermal efficiency was recorded at compression ratio 18:1, and the corresponding value was 30.57%. Sharon et al. (2012) performed an experimental study on a single cylinder, four stroke, diesel engine using palm biodiesel and its blends with diesel as fuel. It was observed that the peak cylinder pressure increased with the use of palm biodiesel and its blends compared to that with diesel. Peak pressures for all the tested fuel increased with the increase in load applied to the engine. Biodiesel and its blends exhibited shorter ignition delay and lower heat release rate compared to diesel.

Ramalingam et al. (2014) experimented with diesel–annona methyl ester blends as fuels and reported reductions in brake specific fuel consumption, exhaust gas temperature, emissions of CO, HC and smoke with the increase in compression ratio. It was also reported that at higher compression ratio the ignition delay period reduced and peak pressure increased resulting in higher combustion temperature and eventually higher NO_x emission. Raheman and Ghadge (2008) noted an enhancement of brake thermal efficiency at higher compression ratio. However, the effect of increase in compression ratio (varied from 18:1 to 20:1) was noted to be more prominent in the case of blends having higher percentages of biodiesel. Sharma and Murugan (2015) observed that the combustion started slightly earlier for tire pyrolysis oil-jatropha biodiesel blends compared to diesel at the rated compression ratio of 17.5:1. With the increase in compression ratio, the ignition delay was found to be shorter by 2.3°CA (degree crank angle). Also, the higher compression ratio enhanced the heat release rate for the blended fuels. Babu et al. (2015, 2016) observed an enhancement of NO_x emission maximum by 17.51% and reduction of CO, HC and smoke emission maximum by 12.78, 39.62 and 4.79% at higher compression ratio (19:1) with neat palm stearin methyl ester compared to mineral diesel. It was also observed by them that the peak pressures were also higher for neat palm stearin biodiesel and its blends with diesel compared to that with diesel.

Palm oil, the basic feedstock of palm stearin biodiesel, is available in India and Southeast Asia. The non-edible fraction of palm oil is called palm stearin oil. One of the most important points in favor of using palm stearin biodiesel is that it will not create any food versus fuel debate. Another point is that the raw material used for the production of this fuel is otherwise a waste product. The cetane number of palm stearin biodiesel is less than that of palm biodiesel, but is more than that of petro-diesel. In the same way, heating value of this biodiesel is slightly less than that of biodiesel prepared from edible palm oil. But the viscosity of palm stearin biodiesel is less than that of palm biodiesel. Although some studies with biodiesel prepared from palm oil have been carried out by several researchers, not many studies on biodiesel prepared from palm stearin oils are reported in the literature. Moreover, the effect of compression ratio on the engine characteristics with palm stearin biodiesel and diesel blends is not investigated properly. Keeping this in mind, the authors have performed experimental studies on a single cylinder, four stroke, variable compression ratio diesel engine having rated compression ratio of 17.5:1. No such major modification of the engine done during this study, only the compression ratio is varied using the tilting block arrangement. The performance, combustion and emission characteristics of the test engine are investigated by using pure palm stearin biodiesel, pure diesel and different blends of palm stearin biodiesel with diesel.

Properties of fuels

It has been found that both high and low melting point triacylglycerols are present in palm oil. Palm oil can be separated into two fractions, namely olein (liquid fraction) and stearin (solid fraction) using a simple dry fractionation process under controlled conditions. The high melting point of the solid fraction (palm stearin, melting point ranging from 45 to 55 °C) is not normally used in producing edible fats due to its low plasticity. The maximal rate of palm stearin that is usually added to a standard table margarine formulation is 10% only (Lai et al. 2000). Palm stearin oil is converted to its methyl esters via transesterification process in the presence of a catalyst. Generally, the purpose of the transesterification process is to lower the viscosity of the oil. Ideally, it is a less expensive way of transforming the large, branched molecular structure of the straight vegetable oils into smaller, straight chain molecules of the type required as fuel in diesel engines.

Palm stearin biodiesel exhibits excellent lubricating properties. It contains very small amount of phosphorus and sulfur, and thus, the emission of oxides of sulfur (SO_x) is practically negligible. The density and the cetane number of palm stearin biodiesel (density: 0.86 g cm^{-3} and cetane number: 49.4) are higher than those of commercial petro-diesel (density: 0.83 g cm^{-3} and cetane number: 48). It is a much safer fuel than diesel because of its higher flash and fire points. The fuel used for this work has been procured directly from Emami Biotech Pvt. Ltd. The important fuel properties are tested in the National Test House, Saltlake, Kolkata, and in the Thermal Power Laboratory of the Department of Mechanical Engineering, IIEST, Shibpur, India. Some of the important fuel properties of palm stearin biodiesel and mineral diesel are listed in Table 1 for comparison. Some of the important fuel properties of blended fuels are also shown in Table 2.

Experimental setup

A Kirloskar made single cylinder, four stroke, compression ignition engine has been used for this experimental investigation. The compression ratio of the test engine can be altered in running condition without changing the combustion chamber geometry by means of tilting block arrangement. The test setup is well equipped with the necessary devices for measurement of in-cylinder pressure with respect to crank angles. The stand-alone control unit of the setup is consisting of air box, two separate fuel tanks (for multi-fuel operation), a manometer for the measurement of suction air, fuel measuring buret and air as well as fuel flow transmitters. Two separate rotameters are provided at the bottom of the stand-alone control unit of the setup for measuring the flow rates of cooling water and calorimeter water. The engine is connected to an eddy current water cooled dynamometer using a flexible coupling and a stub shaft assembly for loading on crankshaft with the help of electromagnetic force. The output of the eddy current dynamometer is fixed to a strain gauge load cell of electronic data acquisition system for measuring load applied to the engine. The fuel measurements are taken by differential pressure transmitter (Yokogawa make, model no: EJA110A-EMS-5A-92NN). It is connected through a fuel line, and the signal of flow rate is transferred to the data acquisition system. A piezoelectric pressure sensor (Model: HSM111A22, Make: PCB Piezotronics, INC.) is fitted to the cylinder head for measuring the cylinder pressure. The piezoelectric pressure transducer is a polystable quartzcrystal which produces an electric charge that is proportional to the pressure developed in the cylinder. All the performance and combustion related data are collected through a laptop which is directly connected to the electronic data acquisition system of the test engine setup. This generates the entire test results using LabVIEW based software "EnginesoftLV." The instantaneous experimental data are acquired over several cycles. For averaging, pressure data of approximately 10 thermodynamic cycles are chosen by the software itself. A Chromel-Alumel (K-Type) thermocouple and a temperature indicator are connected to the exhaust pipe for the measurement of exhaust gas temperature.

The emission study is carried out using a Testo 350 flue gas analyzer. During steady engine operation, the exhaust gas is allowed to rush forward through the probe inserted at the tail pipe of the engine. Thereafter, the exhaust gas passes through a condensation trap, where the moisture is removed. The dry gas is then analyzed by the individual sensors for measuring the respective pollutants, and the corresponding values are displayed on the main screen of the control panel. The emissions of CO and CO_2 are measured in terms of percentage volume. CO, CO_2 and HC emissions are measured on the basis of non-dispersive infrared (NDIR) detection principle, while NO_x is measured by means of pre-calibrated electrochemical sensor. The opacity of the exhaust smoke is measured by Indus made OMS 103 smoke meter. The measurement is taken by the extinction of a light beam by scattering and absorption principle. The schematic diagram of the setup is shown in Fig. 1a, and the specifications of the test engine used for this investigation are given in Table 3. The details of the

Table 1 Fuel properties of palm stearin methyl ester (PSME) and diesel along with the standards

Properties	Diesel (B0)	PSME (B100)	ASTM D6751	EN 14214	IS 15607
API gravity	40.24	42.82	–	–	–
Ash content (%)	0.06	0.02	–	–	–
Water content (%)	0.07	0.08	0.05 max	0.05 max	0.05 max
Carbon residue (%)	0.00	0.12	0.05 max	0.3 max	0.05 max
Flash point (°C)	57	170.0	93 min	101 min	120 min
Pour point (°C)	−19	17.8	–	–	–
Fire point (°C)	65	186.0	–	–	–
Viscosity @ 30 °C (cSt)	2.66	4.10	1.9–6.0 @ 40 °C	3.5–5.0 @ 40 °C	3.5–6.0 @ 40 °C
Cetane number	48	49.40	47 min	51 min	51 min
Density @ 20 °C (g cm^{-3})	0.83	0.86	–	0.86–0.9 @ 15 °C	0.86 @ 15 °C
Calorific value (kJ/kg)	42,000	38,602	–	–	–

Table 2 Some important fuel properties of palm stearin biodiesel and diesel blends

Properties	B0	B20	B40	B100
Density @ 20 °C (g cm^{-3})	0.830	0.835	0.840	0.860
Calorific value (kJ/kg)	42,000	41,320	40,641	38,602

gas analyzer and the smoke meter are given in Tables 4 and 5, respectively.

Methodology

The experiments have been conducted using neat diesel, neat biodiesel and two diesel–biodiesel blends under different loading conditions. The load is varied by adjusting load regulator on the control panel of the setup. Each set of engine data is taken few minutes after the load is applied to get steady state condition. The engine has been run for four different compression ratios of 16:1, 17:1, 17.5:1 (rated) and 18:1. It has already been mentioned that the compression ratio has been altered using tilting block arrangement. The actual diagram of tilting block arrangement is given in Fig. 1b. For varying the compression ratio during running condition of the engine, the 6 Allen bolts provided for clamping the tilting block are slightly loosened first. Next, the lock nut is loosened on the adjuster and the adjuster is rotated so that the compression ratio is set to the desired one by referring the marking on the CR (compression ratio) indicator. After setting the desired compression ratio, the lock nut and all the 6 Allen bolts are tightened. The load is varied up to a maximum value of 12 kg, whereas the rated speed of 1500 rpm is kept constant for all the cases. The injection timing and injection pressure are also kept constant at 23°bTDC and 210 bar, respectively.

Uncertainty analysis

The errors in the measurement of different independent parameters before experimentation are noted and presented in Table 6 which are obtained from the manual of the equipment. The overall uncertainty is then calculated based on the individual errors following Holman (2012) as:

Percentage of uncertainty = [(liquid fuel flow rate)^2 + (LHV of the fuel)^2 + (height of the liquid column)^2 + $(CO)^2$ + $(CO_2)^2$ + $(HC)^2$ + $(NO_x)^2$ + (Smoke opacity)^2 + (Crank angle encoder)^2 + (Load cell)^2 + (Pressure transducer)^2 + (Speed sensor)^2 + (Temperature indicator)^2]$^{1/2}$. The obtained value of overall uncertainty is found to be ±2.39% which lies within the acceptable range.

Results and discussion

The analysis of engine performance gives an idea about the utilization of the fuel energy by the engine, and the combustion analysis gives a qualitative idea about the pattern of the heat energy released during the burning of the fuel–air mixture inside the cylinder. A direct measure of the environmental impact of the fuel used in an internal combustion engine can be judged from the emission study. The variations of different performance, combustion and emission characteristics parameters with load (in some cases with crank angle) are presented graphically and discussed below under different subsection.

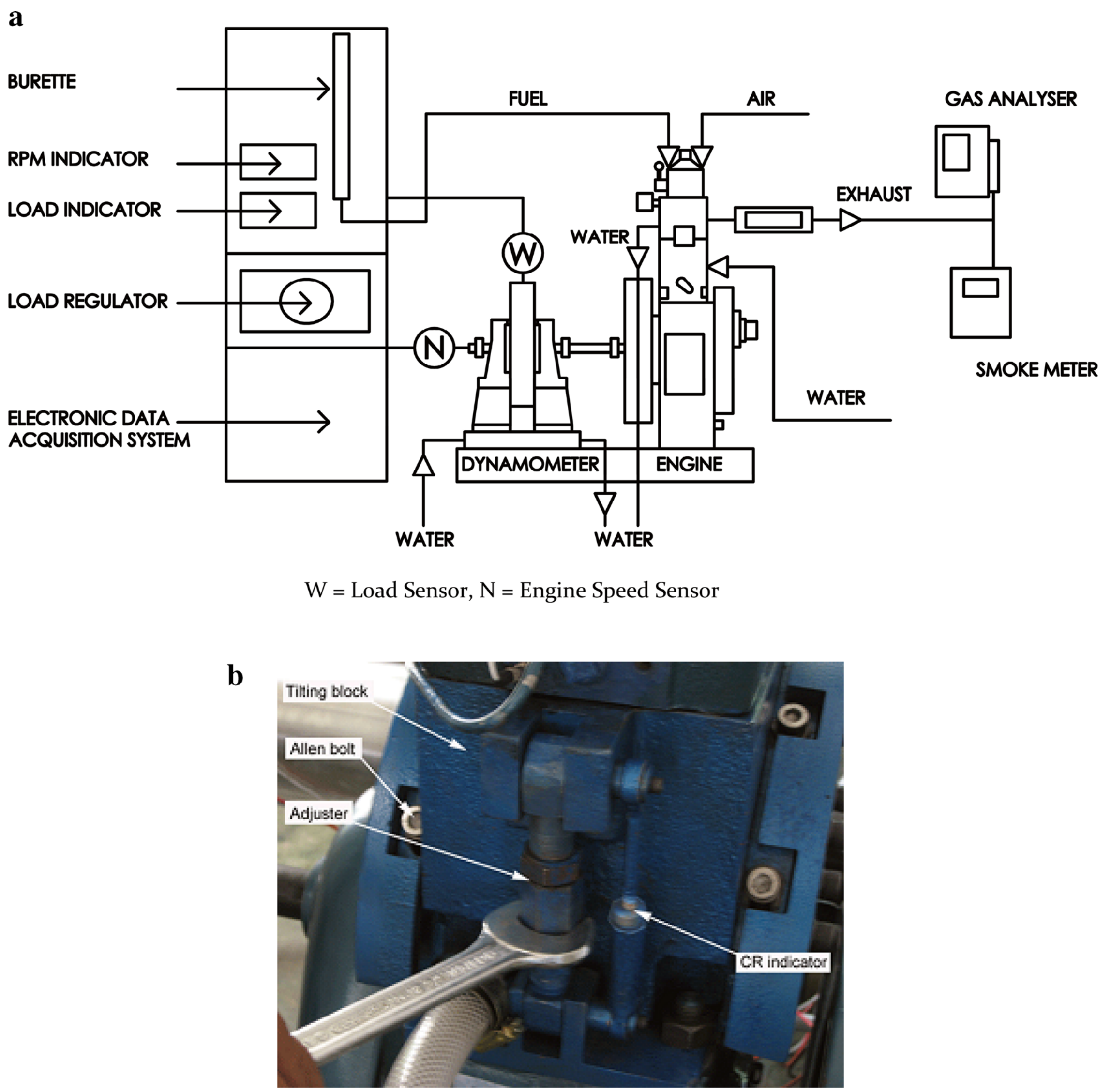

Fig. 1 **a** Schematic diagram of the experimental setup. **b** Actual diagram of tilting block arrangement

Performance analysis

Brake thermal efficiency

The brake thermal efficiency of the engine is defined as the ratio of the power obtained at the crank shaft, and the rate of energy supposed to be released from the complete combustion of fuel. Brake thermal efficiency also provides information about the performance of the combustion system and also the knowledge about the effectiveness of the fuel energy conversion to mechanical power (Sayin 2010). The variations of brake thermal efficiency of the engine with load for diesel (B0), palm stearin biodiesel (B100) and two diesel–palm stearin biodiesel blends (B20 and B40) are presented in Fig. 2a for the rated compression ratio of 17.5:1. It can be seen from the figure that the brake thermal efficiency is highest with diesel fuel and lowest with neat palm stearin biodiesel. Naturally, the brake thermal efficiency is found to decrease with the increase in percentage share of palm stearin biodiesel in the blended fuel. The higher density, higher viscosity and the lower calorific value of palm stearin biodiesel contribute to poorer atomization of the blended fuels and neat biodiesel. The corresponding values of density of B100, B40 and B20 are 0.86, 0.84 and 0.83 g cm^{-3}, respectively. The ultimate effect is reflected through the reduction of brake thermal efficiency. The thermal efficiency with diesel at full load condition is 32.72%, whereas the thermal efficiency with neat biodiesel is 30.12%. This means there is a reduction of 7.9% in brake thermal efficiency when diesel is replaced completely with biodiesel in the same engine without any modification. The reductions in BTE (brake thermal

Table 3 Engine specification

Parameter	Specification
Product	Computerized variable compression ignition engine, Code: 234
Type	Single cylinder, DI, diesel
Rated power	3.5 kW @ 1500 RPM
Bore	87.5 mm
Stroke	110 mm
Swept volume	661 cc
Compression ratio	11:1–18:1
Method of cooling	Water cooled
Injection timing	23°bTDC
Injection pressure	210 bar
Injection nozzle	3 hole
Dynamometer	Eddy current, water cooled, with loading unit
Dynamometer arm length	185 mm
Connecting rod length	234 mm
Air box	MS fabricated with orifice meter and U-tube manometer

Table 4 Gas analyzer specification

Parameters	Resolution	Accuracy	Range
CO	1 ppm	±10 < 200 ppm	0–10,000 ppm
CO_2	0.01 vol%	±0.3% < 25 vol%	0–50 vol%
NO	1 ppm	±5% reading < 2000 ppm	0–3000 ppm
NO_2	0.1 ppm	±5 < 100 ppm	0–500 ppm
O_2	0.01 vol%	±0.2 vol%	0–25 vol%
HC	1 ppm	±10% of reading	0–40,000 ppm

Table 5 Smoke meter specification

Parameters	Resolution	Accuracy	Range
HSU	0.1%	–	0–99.9
K	0.01 m^{-1}	±0.1 m^{-1}	0–∞

Table 6 Errors related to various parameters

Parameter	Errors (%)
Liquid fuel flow rate	±1
LHV of the fuel	±1
Height of the liquid column	±0.5
CO	±0.3
CO_2	±1
HC	±0.1
NO_x	±0.5
Smoke opacity	±1
Crank angle encoder	±0.2
Load cell	±0.2
Pressure transducer	±1
Speed sensor	±0.1
Temperature indicator	±0.15

efficiency) are, however, much less with B20 and B40 blends, and the corresponding values are 2.5 and 4.7%, respectively.

Figure 2b shows the variation of brake thermal efficiency for neat diesel and neat palm stearin biodiesel at three different compression ratios of 18:1, 17:1 and 16:1, respectively. The first observation is that the efficiency decreases with the decrease in compression ratio for all the fuels and accordingly, the maximum efficiency is obtained at the compression ratio of 18:1 (which is the maximum in this study) and the minimum at compression ratio of 16:1. At full load conditions, the efficiency values are noted to be 32.92, 32.52 and 32.22% corresponding to the compression ratio of 18:1, 17:1 and 16:1, respectively, for neat diesel. The corresponding values for neat palm stearin biodiesel are 31.03, 29.21 and 28.10%. It can be inferred from the above results that the lowering of brake thermal efficiency with the use of biodiesel is more at lower compression ratio. The higher air temperature due to higher compression ratio results in better air–fuel mixing and faster evaporation of charge which eventually leads to better combustion and enhances the brake thermal efficiency. The reduced ignition delay at higher compression ratio also enhances the

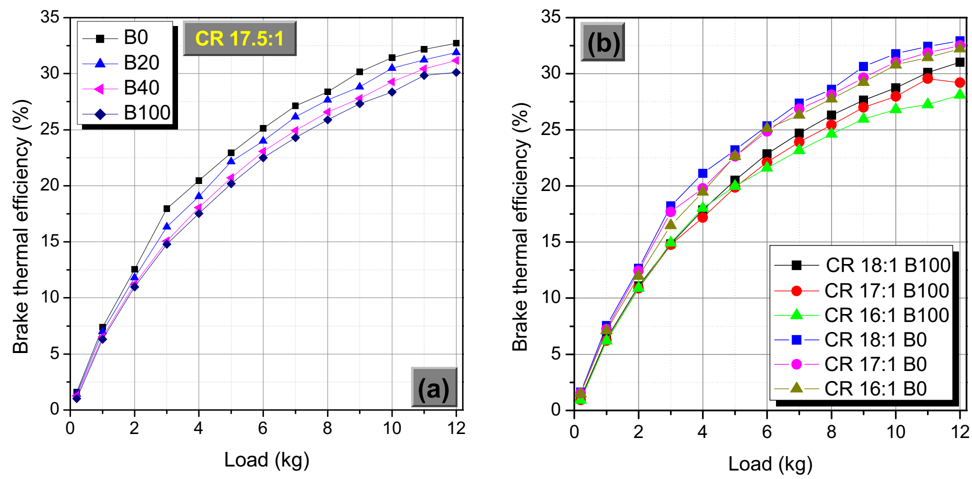

Fig. 2 Variation of brake thermal efficiency with load for diesel and biodiesel blends for different compression ratios

thermal efficiency. At higher compression ratio, the viscosity and the density of palm stearin biodiesel are reduced to some extent resulting in an improvement in thermal efficiency. The figure depicts that the differences between the values of brake thermal efficiency with neat diesel (B0) and neat biodiesel (B100) are 12.78 and 5.7%, respectively, when the compression ratio is increased from 16:1 to 18:1. This indicates that the effect of compression ratio on the brake thermal efficiency is more in the case of biodiesel-fueled engine.

Brake specific fuel consumption

Brake specific fuel consumption is defined as the fuel consumption rate to produce unit brake power. Figure 3a shows the variation of brake specific fuel consumption with load for diesel (B0), palm stearin biodiesel (B100) and its blends with diesel (B20 and B40) at the rated compression ratio of 17.5:1. The graphs have been plotted for load 1 kg onwards for getting a clear picture. At full load condition, BSFC increases by about 17.30% when diesel is replaced completely by palm stearin biodiesel. With the decrease in load, the BSFC increases in general for all the tested fuels, but the increase is maximum for neat biodiesel and in between for the blends. The values of the BSFC obtained at 1 kg load are 1.37, 1.55, 1.65 and 1.8 kg/kWh for B0, B20, B40 and B100, respectively. Even much higher values are observed when the load reaches near to the no-load condition. The same trend is also observed at other compression ratios considered in this study. It is also clear from the figure that with the increase in biodiesel proportion in the blended fuel, brake specific fuel consumption increases. As the calorific value of palm stearin biodiesel is lower than that of mineral diesel, more amount of fuel is needed to maintain the same power output and hence, the specific fuel consumption rate is increased. The corresponding calorific values of B100, B40 and B20 are 38,602, 40,641 and 41,320 kJ/kg, respectively. During the combustion of biodiesel, the oxygen present (11% by mass) in its molecule does not contribute to heat generation. This results in lower calorific value of biodiesel and eventually

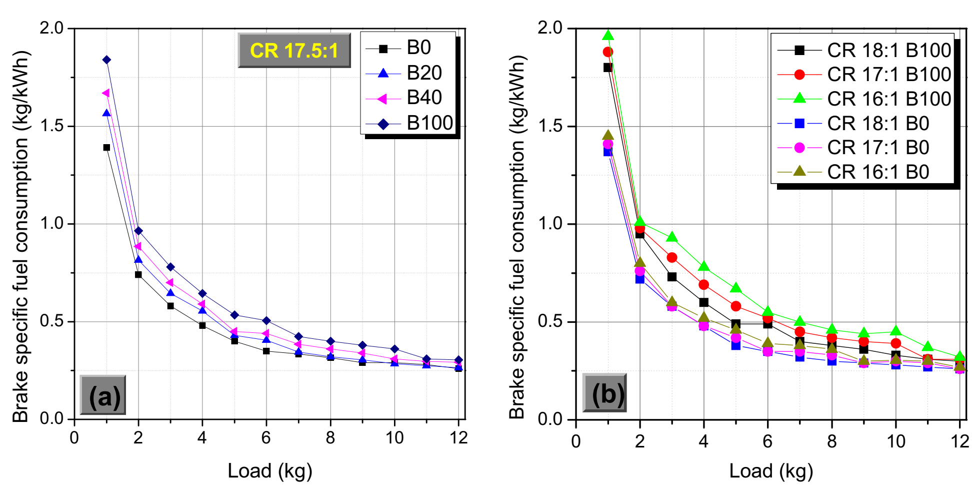

Fig. 3 Variation of brake specific fuel consumption with load for diesel and biodiesel blends for different compression ratios

increase the fuel consumption rate for neat palm stearin biodiesel and its blends with diesel.

It can also be seen from Fig. 3b that the brake specific fuel consumption increases for all the fuels when the compression ratio is decreased. The minimum BSFC values are found to be 0.26 and 0.27 kg/kWh for neat diesel fuel at compression ratios of 18:1 and 16:1, respectively. For neat palm stearin biodiesel, the corresponding values are 0.3 and 0.32 kg/kWh. At higher compression ratio, the pressure and the temperature inside the cylinder increase and for this fact the fuel is injected into a relatively hotter combustion chamber which leads to an improvement of power output. Consequently, the fuel consumption rate for generating unit brake power decreases at higher compression ratio (Sayin and Gumus 2011). On the other hand, the poorer combustion may be held responsible for higher BSFC at lower compression ratio.

Exhaust gas temperature

Exhaust gas temperature is also an important parameter as it provides some qualitative information about the combustion process (Anand et al. 2011). Exhaust gas temperature is an indicator of the heat release rate of the fuels tested during combustion and its effective utilization to produce power (Behçet 2011). It depends on the combustion characteristics of the fuel and also on the heat loss to exhaust (Raheman and Phadatare 2004). Figure 4a shows the variation of exhaust gas temperature (EGT) with load for diesel, palm stearin biodiesel and its blends with diesel for the rated compression ratio of 17.5:1. It can be clearly seen from the figure that exhaust gas temperature increases with the increase in load for all the tested fuels. This is due to the fact that temperature after combustion becomes more as more fuel is required to be burnt at higher load condition. It can be also seen from the above-said figure that the addition of biodiesel to diesel increases the exhaust gas temperature and it becomes maximum for neat biodiesel at any load condition. The higher flame temperature, as well as the increase in combustion temperature due to the oxygen enrichment in the case of biodiesel-blended fuels, leads to a higher exhaust gas temperature. It is noted that at full load condition, the exhaust gas temperatures of diesel and neat palm stearin biodiesel are 298.5 and 320 °C, respectively.

It can also be noted from Fig. 4b that the exhaust gas temperature increases with the decrease in compression ratio for all the fuels considered in this work. From the above figure, it can also be seen that the exhaust gas temperatures are nearly 300 and 305 °C for diesel and neat biodiesel, respectively, at compression ratio of 18:1. The corresponding values are 334 and 348 °C at compression ratio of 16:1 for neat diesel and neat palm stearin biodiesel, respectively. At higher compression ratio, the ignition delay of the fuel decreases in general and it provides more time for combustion. The heat release takes place for a longer period although the peak heat release rate becomes less. This provides an opportunity for the heat to be utilized in a better manner for the production of mechanical power. Consequently, the energy lost with exhaust gas decreases and eventually results in a lower exhaust gas temperature. On the other hand, at lower compression ratio, more amount of heat is released during premixed combustion phase due to delay in ignition. As a result of this, more amount of heat is carried by the exhaust gas and, thus, the exhaust gas temperature is enhanced (Sharma and Murugan 2015).

Combustion analysis

In-cylinder pressure

The smoothness of engine operation can be evaluated from the variation of in-cylinder pressure. The peak cylinder

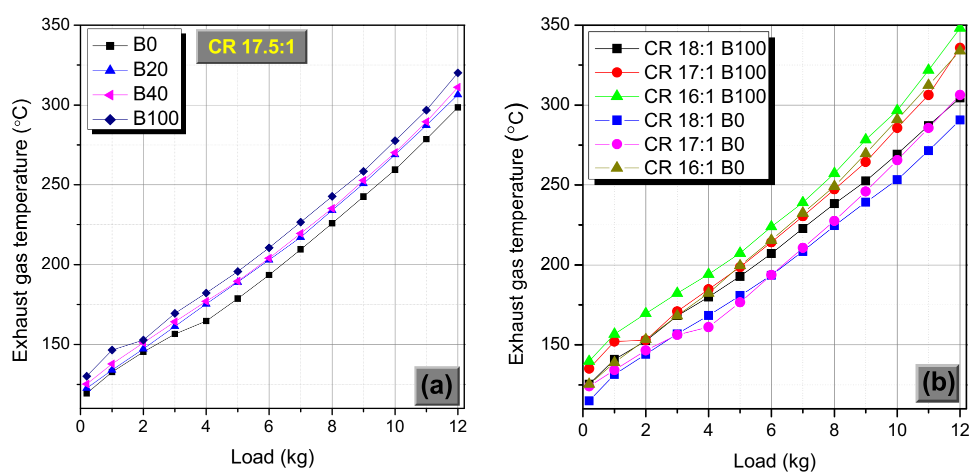

Fig. 4 Variation of exhaust gas temperature with load for diesel and biodiesel blends for different compression ratios

pressure in a compression ignition (CI) engine depends on the fraction of fuel burnt in premixed mode of combustion (Huang et al. 2004). The variations of in-cylinder pressure with crank angle for the diesel, palm stearin biodiesel and diesel–biodiesel blends are shown in Fig. 5a for full load condition at rated compression ratio of 17.5:1. It can be interpreted from the figure that the peak pressures with biodiesel and its blends are higher than with diesel. The figure also shows that the combustion starts earlier for neat biodiesel and biodiesel blends compared to neat diesel. This tendency of an early start of ignition is due to the higher cetane number of biodiesel and its blends with diesel. More time available for combustion and the oxygen present in biodiesel molecules create favorable condition for more complete combustion. As a result, the pressure and the temperature inside the combustion chamber are increased. The maximum pressure for diesel is noted to be 75.92 bar at 12°aTDC (degree crank angle after top dead center), and for neat palm stearin biodiesel, it is noted to be 78.32 bar at 11°aTDC.

It can also be observed from the Fig. 5b that the in-cylinder pressure decreases with the decrease in compression ratio. The in-cylinder pressures are found to be 82.15 bar at 10°aTDC and 83.71 bar at 10°aTDC for diesel and neat biodiesel, respectively, at compression ratio of 18:1. The corresponding values are 67.49 bar at 14°aTDC and 69.42 bar at 11°aTDC for neat diesel and neat palm stearin biodiesel, respectively, at compression ratio of 16:1. At higher compression ratio, the temperature of air increases which leads to better fuel atomization and enhances the combustion process. At lower compression ratio, the relatively slower premixed combustion phase decreases in-cylinder pressure. Also, it was stated by Rao and Kaleemuddin (2011) that in a compression ignition engine, increase in compression ratio increases the peak cylinder pressure because the density of fuel–air mixture is increased at higher compression ratio and there is a better mixing of the burnt and unburnt charges. On the other hand, cylinder pressure decreases due to slow combustion because of weak swirl and improper mixing of burnt and unburnt charges at low compression ratios.

Heat release rate

The graph showing the heat release rate at different crank angles may be used to identify the start of combustion, the fraction of fuel burned in the premixed combustion phase and the differences in combustion rates of different fuels (Canakci et al. 2008). The variations of heat release rate for diesel, palm stearin biodiesel and diesel–biodiesel blends investigated in this work are shown in Fig. 6a for full load condition at rated compression ratio of 17.5:1. The figure shows that the heat release rate for neat diesel is highest and that for neat palm stearin biodiesel is lowest among the tested fuels. The longer ignition delay of diesel provides more mixing time to the intake charge. Initial combustion of the accumulated fuel takes place in the premixed mode and results in higher peak heat release rate. The similar trend is followed by all other tested fuels consisting of premixed combustion phase and diffusion combustion phase. It can also be observed from the figure that the position of the maximum heat release rate has shifted toward the left for biodiesel. This happens because combustion starts earlier for biodiesel due to its higher cetane number compared to diesel. At the rated compression ratio of 17.5:1, the maximum heat release rates for diesel and biodiesel are observed at 6°aTDC and 3°aTDC and the corresponding values are 34.7 and 26.81 J/CA, respectively. However, a negative heat release is observed for all the tested fuels under all test conditions and it is due to the vaporization of accumulated fuel before the start of ignition. After the initiation of combustion in premixed phase, the heat release rate becomes gradually positive and afterward, it burns rapidly. This phase is followed by the

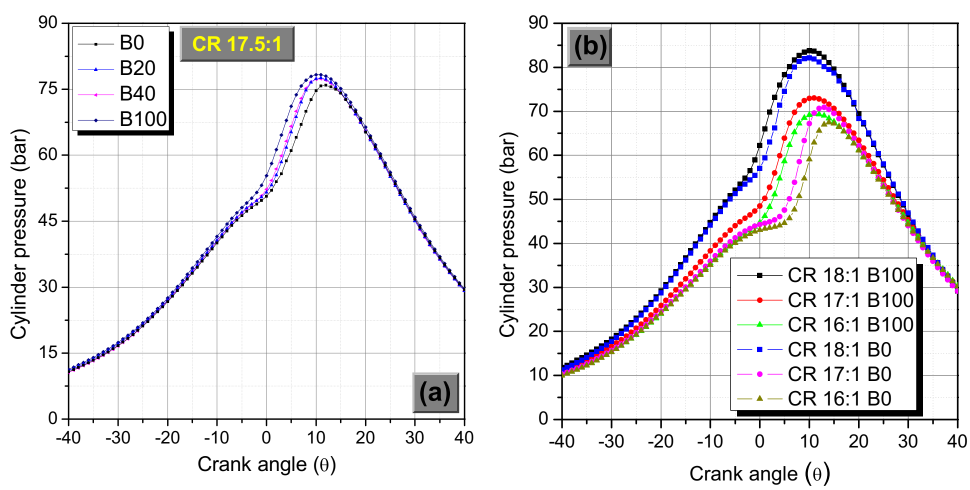

Fig. 5 Variation of cylinder pressure with crank angle for diesel and biodiesel blends for different compression ratios

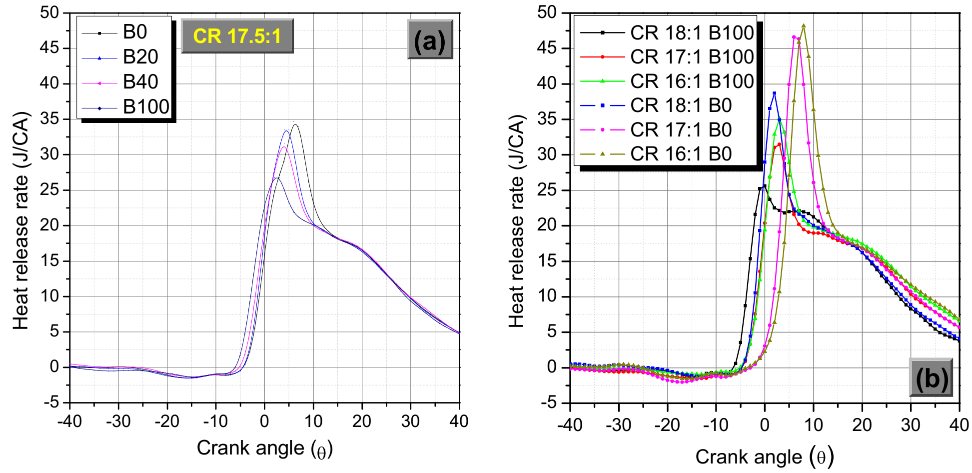

Fig. 6 Variation of heat release rate with crank angle for diesel and biodiesel blends for different compression ratios

diffusion phase, where the reaction rates are controlled by the fuel–air mixing rate (Qi et al. 2009).

The heat release rate increases for all the fuels when compression ratio is decreased from 18:1 to 16:1 as shown in Fig. 6b. It can also be noted from Fig. 6a, b that heat release rate for diesel slightly decreases when the compression ratio of the engine is changed from 18:1 to 17.5:1. This may be due to the fact that compression ratio of 18:1 is higher than the rated compression ratio of 17.5:1. The peak heat release rate is found to be 38.71 J/CA at 2°aTDC and 25.63 J/CA at TDC for diesel and neat biodiesel, respectively, at compression ratio of 18:1. The corresponding values are 48.16 J/CA at 8°aTDC and 35.15 J/CA at 3°aTDC for neat diesel and neat palm stearin biodiesel, respectively, at compression ratio of 16:1. Lower compression ratio provides longer ignition delay period during which more fuel is injected, resulting in more time for premixing of air–fuel mixture, and this, in turn, tends to increase the heat release rate. However, at higher compression ratio, the reduction in viscosity and better spray formation could be the possible reasons behind lower peak heat release rate (Muralidharan and Vasudevan 2011). The slower air–fuel mixing, weak air entrainment and poorer combustion are responsible for higher peak heat release rate at lower compression ratio as pointed out by Sharma and Murugan (2015).

Ignition delay

The ignition delay or ignition lag of a fuel may be defined as the gap in time or crank angle between the fuel injection and start of ignition (Sahoo and Das 2009). It also represents the time duration between physical and chemical reactions. The results available from the data acquisition system of the setup in graphical and tabular forms have been used in this present study to identify the start of fuel injection and start of fuel ignition. Figure 7a shows the variation of ignition delay with load for diesel, palm stearin biodiesel and its blends with diesel for rated compression ratio of 17.5:1. The figure depicts that ignition delay decreases with the increase in load for all the tested fuels. This is due to the heat loss during compression which decreases with the increase in load. Temperature and pressure of the compressed air become more, and this eventually decreases ignition delay period (Gumus 2010). Ignition delay periods as observed from the above-said figure are 18°CA and 10°CA for diesel and neat biodiesel, respectively, at the rated compression ratio under very low load (0.2 kg) condition. However, at full load (12 kg) condition, the ignition delay decreases for both diesel and palm stearin biodiesel and the corresponding values are 15°CA and 9°CA. The decrease in ignition delay with load is less in the case of biodiesel compared to diesel as the delay period is already much less in the case of biodiesel. The behavior of the blended fuels lies in between diesel and neat biodiesel as seen in the figure. The higher cetane number of biodiesel compared to diesel causes shorter ignition delay period for the blended fuels.

It can also be noted from Fig. 7b that ignition delay is less at higher compression ratio, whereas at lower compression ratio the delay period is more for all the investigated fuels. The ignition delay period at very low load condition is found to be 17°CA and 9°CA for diesel and neat biodiesel, respectively, at compression ratio of 18:1. The corresponding values are 21°CA and 15°CA for neat diesel and neat palm stearin biodiesel, respectively, at compression ratio of 16:1. The increased air temperature at higher compression ratio reduces the viscosity of the fuels by breaking the intermolecular bonds and lowers the self-ignition temperature of the fuel. This causes early ignition and reduces the ignition delay period.

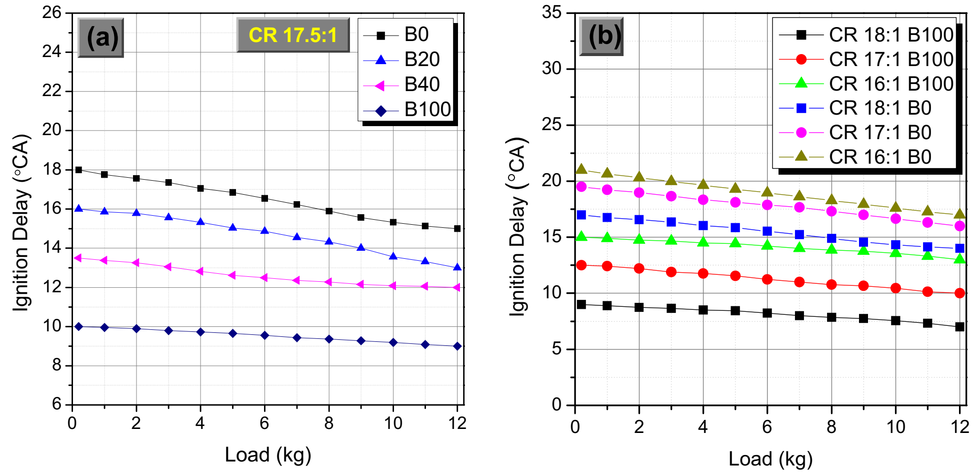

Fig. 7 Variation of ignition delay with load for diesel and biodiesel blends for different compression ratios

Emission analysis

CO emission

Carbon monoxide (CO) is a product of incomplete combustion or partial oxidation of carbon present in the fuel molecules. Carbon present in any fuel is oxidized to CO_2 and partially to CO in the presence of oxygen during combustion. In general, CO emission will decrease with biodiesel fuel due to better combustion by taking advantage of the inherent oxygen of the fuel molecules. The variation of CO emission for the diesel, biodiesel and its blends is shown in Fig. 8a for the rated compression ratio of 17.5:1. It is observed from the figure that the CO emission increases with load which can be attributed to the higher amount of fuel consumption at higher load conditions. It can also be seen from the above-said figure that CO emissions for biodiesel and its blends are lower than for diesel. CO emission with neat palm stearin biodiesel is found to be around 12% lower compared to that with neat diesel under full load condition at the rated compression ratio. It is already stated earlier that CO emission is the result of incomplete combustion and the inherent oxygen content of biodiesel makes the combustion more complete. Consequently, the formation of CO is found to be less with palm stearin biodiesel and its blends with diesel than that with neat mineral diesel.

The effect of compression ratio on CO emission can be studied from Fig. 8b. It is observed from the figure that the increase in compression ratio decreases the CO emission for all the tested fuels. The CO emission at full load condition is found to be 0.11% by vol. and 0.09% by vol. for diesel and neat palm stearin biodiesel, respectively, at compression ratio of 18:1. The corresponding values are 0.115% by vol. and 0.10% by vol. for neat diesel and neat palm stearin biodiesel, respectively, at compression ratio of 16:1. At higher compression ratio, the higher temperature developed inside the cylinder creates favorable environment for further oxidation of the charge. It has already been mentioned that ignition delay period decreases with the increase in compression ratio and this provides more time for the combustion in diffusion mode. These eventually

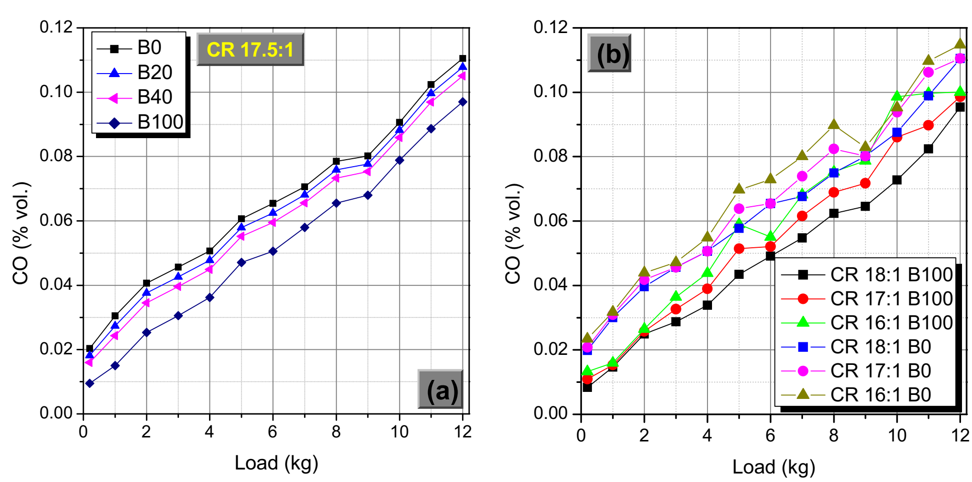

Fig. 8 Variation of CO emission with load for diesel and biodiesel blends for different compression ratios

improve the overall combustion process and decrease the CO emission at higher compression ratio. The decrease in CO emission may not be in the same ratio as the percentage of biodiesel in the blends at all compression ratios. The negative effect of higher viscosity and poor atomization of biodiesel dominates over the positive effect of the inherent oxygen content of biodiesel for its higher blends, and this may have an adverse effect on CO emission particularly when the engine is operated at a compression ratio other than the rated one (Amarnath and Prabhakaran 2012).

CO_2 emission

It has already been discussed in the previous section that the addition of biodiesel to mineral diesel improves the quality of combustion. This eventually reduces CO emission and increases CO_2 emission. The variations of CO_2 emission for the diesel, biodiesel and its blends are shown in Fig. 9a for the rated compression ratio of 17.5:1. The figure clearly shows the increase in CO_2 emission with the increase in biodiesel percentage in the blended fuels. The oxygen content of biodiesel and its blends enhances the combustion, and thus, the emission of CO_2 is increased. CO_2 emission with neat palm stearin biodiesel is about 50% higher than that with neat diesel at full load condition. It can be mentioned that CO_2 is a greenhouse gas and its emission into the atmosphere is detrimental to the environment. But the oil crops from which biodiesel is produced absorbed CO_2 through the process of photosynthesis during their cultivation. Hence, the life cycle CO_2 addition to the atmosphere is substantially reduced when biodiesels are used as fuels.

The effect of compression ratio on CO_2 emission associated with different fuel blends can be interpreted from Fig. 9b. One can note that CO_2 emission increases for all the tested fuels with the increase in compression ratio. CO_2 emission at full load condition is found to be 6.30% by vol. and 9.67% by vol. for diesel and neat palm stearin biodiesel, respectively, at compression ratio of 18:1. The corresponding values are 6.07% by vol. and 9.00% by vol. for neat diesel and neat palm stearin biodiesel, respectively, at compression ratio of 16:1. The percentage increase in CO_2 emission with neat biodiesel as fuel is almost double the corresponding value with neat diesel when the compression ratio is increased from 16:1 to 18:1. At higher compression ratio, the combustion is improved and it leads to more amount of CO_2 formation. On the other hand, poorer atomization of fuels at lower compression ratio reduces the amount of CO_2 emission. The effect of increase in compression ratio is more in the case of biodiesel, and this may be attributed to the inherent oxygen available in the molecules of palm stearin biodiesel.

NO_x emission

All the oxides of nitrogen formed during combustion due to the oxidation of nitrogen are collectively known as NO_x. The amount of NO_x formed and its emission depend mainly on the combustion temperature, oxygen concentration and the residence time of the combustion products within the combustion zone. The variations of NO_x emission with brake power for the diesel, palm stearin biodiesel and their different blends are shown in Fig. 10a for the rated compression ratio of 17.5:1. From the figure, it can be seen that the NO_x emission is enhanced with the increase in the percentage of biodiesel in the diesel–biodiesel blended fuel. The highest NO_x emission is found with neat biodiesel and least with diesel for any load condition and compression ratio. At the rated compression ratio, NO_x emission with neat biodiesel is about 48% higher than that with neat diesel at full load condition. The inherent oxygen molecules of biodiesel contribute to better combustion, and thus, the combustion temperature is increased which creates a favorable environment for NO_x formation. It was also

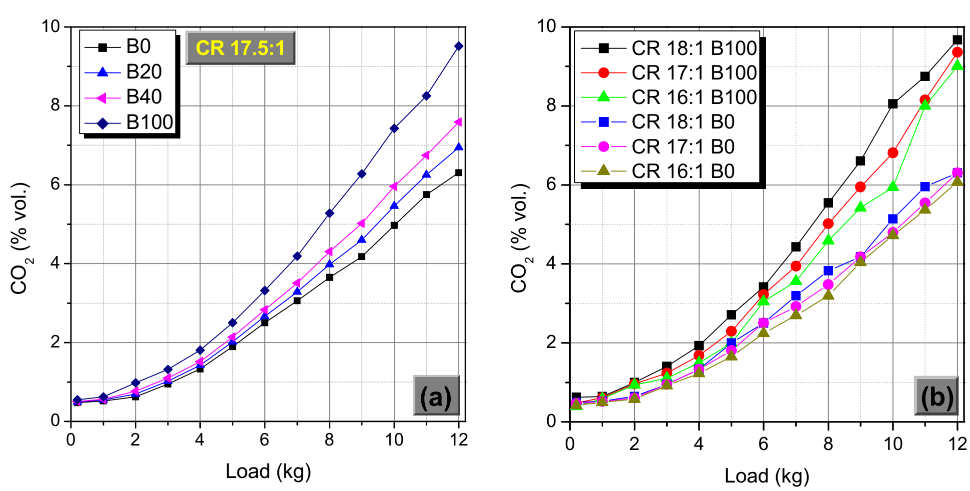

Fig. 9 Variation of CO_2 emission with load for diesel and biodiesel blends for different compression ratios

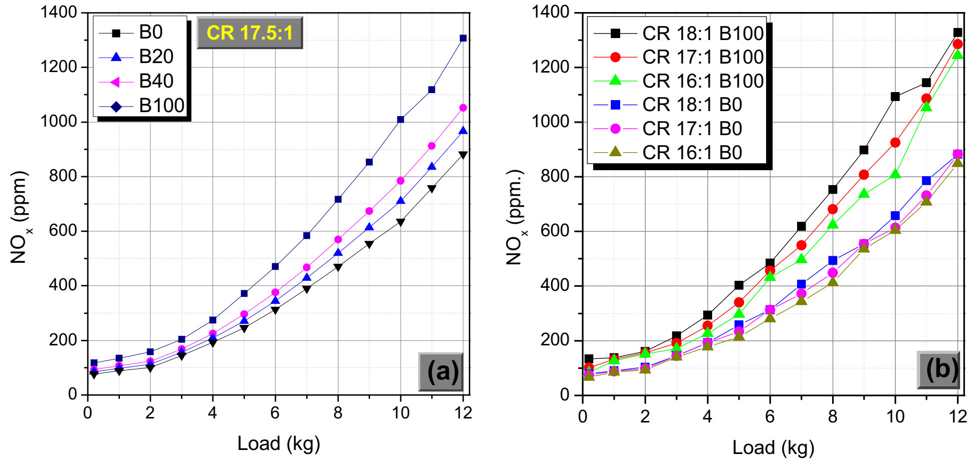

Fig. 10 Variation of NO_x emission with load for diesel and biodiesel blends for different compression ratios

reported in the literature that as biodiesel contains double bonded molecules, the adiabatic flame temperature is higher and it promotes more NO_x formation (Ramalingam et al. 2014). The physical properties of the fuel such as viscosity, density, sound velocity and compressibility were found to be responsible for the advanced injection of biodiesel. As a consequence, the start of combustion is advanced and higher peak temperature is achieved in the cycle. The pressure rise produced by the fuel pump becomes faster due to higher bulk modulus (lower compressibility) of biodiesel. The pressure waves propagate rapidly toward the injector as a consequence of its higher sound velocity and cause advanced injection of fuel (Palit et al. 2011).

The effect of compression ratio on NO_x emission can be observed from Fig. 10b. At higher compression ratio, the NO_x emission is higher. The higher temperature inside the cylinder creates the suitable environment for NO_x emission. It has also been reported by (Ramalingam et al. 2014) that at lower compression ratio, the premixed combustion phase is lengthened due to the longer ignition delay and eventually decreases the NO_x emission. NO_x emission at full load condition is found to be 882 and 1328 ppm for diesel and neat biodiesel, respectively, at compression ratio of 18:1. The corresponding values are 849 and 1245 ppm for neat diesel and neat biodiesel, respectively, at compression ratio of 16:1.

HC emission

The amount of unburned hydrocarbon (HC) from any engine depends on the composition and also the combustion characteristics of the fuel used. It has already been observed that the combustion is improved with biodiesel or biodiesel blended mineral diesel. This clearly indicates the possibility of lower HC emissions with biodiesel fuel. The variation of HC emissions with load for diesel, palm stearin biodiesel and their blends is shown in Fig. 11a for the rated compression ratio of 17.5:1. It can be seen from the figure that the HC emission decreases with the increase in biodiesel share in the blended fuel. The highest HC emission is observed with neat diesel, and the lowest one is noted in the case of neat biodiesel. This is due to the presence of oxygen in biodiesel molecules which results in smooth and more complete combustion. HC is a product of incomplete combustion, and hence, its formation and emission are reduced with the addition of biodiesel to diesel fuel. HC emission from the engine is found to be around 26% less with neat biodiesel compared to that with diesel at full load condition operating at the rated compression ratio.

The effect of compression ratio on HC emission using neat diesel (B0) and neat biodiesel (B100) as fuels has been shown in Fig. 11b. The figure clearly indicates the decrease in the HC emission with the increase in compression ratio for diesel and neat palm stearin biodiesel. The HC emission at full load condition is found to be 87 ppm and 64 ppm for diesel and neat biodiesel, respectively, at compression ratio of 18:1. The corresponding values are 90 ppm and 68 ppm for neat diesel and biodiesel, respectively, at compression ratio of 16:1. At higher compression ratio, the increase in air temperature at the end of the compression stroke enhanced the combustion temperature. This improves the combustion and reduces HC emission. On the other hand, the longer ignition delay and higher accumulation of fuel during the delay period may be a possible reason for higher HC emission at low compression ratio.

Smoke emission

The occurrence of smoke in the engine exhaust is considered to be one of the major environmental problems with diesel fuel. Smoke is formed when the fuel is not fully oxidized due to the lack of sufficient amount of

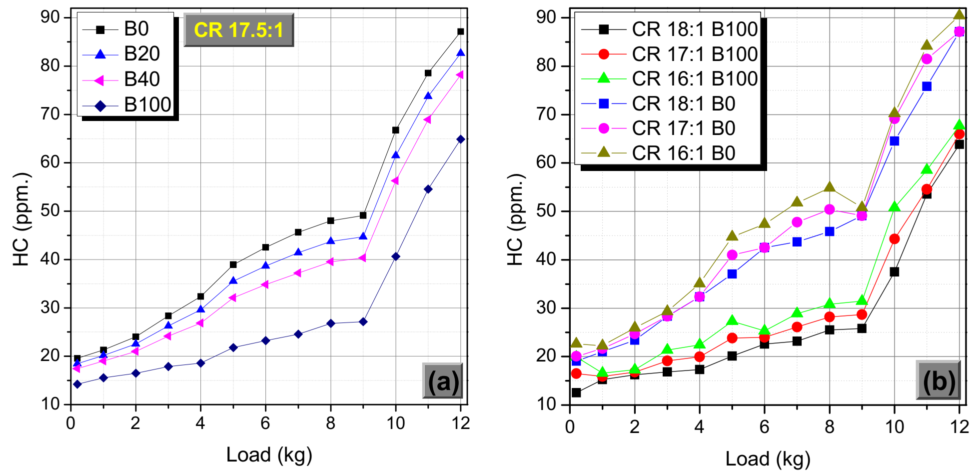

Fig. 11 Variation of HC emission with load for diesel and biodiesel blends for different compression ratios

oxygen in the combustion zone. The oxygen present in the biodiesel molecules is likely to improve the situation. The variation of smoke opacity (in terms of Hartridge smoke unit) for diesel, biodiesel and its blends is shown in Fig. 12a for the rated compression ratio of 17.5:1. As expected, the figure shows lowering of smoke emission with the use of biodiesel fuels. In general, smoke formation is dependent upon the incompleteness of combustion and the presence of partially reacted carbon mass fraction of the fuel (Heywood 1988). The more complete combustion with biodiesel results in less smoke formation in comparison with diesel. It was also reported by (Ramalingam et al. 2014) that the oxygen content of biodiesel led to the oxidation of soot in CI engine and thus reduced the soot emission. However, the smoke opacity increases with the increase in load for all the tested fuels, which may be attributed to the burning of richer mixture at higher load conditions. At the rated compression ratio and full load condition, the smoke opacity with biodiesel is noted to be nearly 16% less compared to that with mineral diesel.

The effect of compression ratio on smoke opacity can be analyzed from Fig. 12b. The figure shows that the decrease in compression ratio increases the smoke opacity to some extent and the increase is slightly more with biodiesel. The smoke emission at full load condition is found to be 73 and 61 HSU for diesel and neat palm stearin biodiesel, respectively, at compression ratio of 18:1. The corresponding values are 76 and 69 HSU for neat diesel and biodiesel, respectively, at compression ratio of 16:1. The smoke opacity at higher compression ratio is less because of higher combustion temperature which improves the atomization of fuel. On the other hand, the poorer combustion results in more soot formation on the decrease in compression ratio.

Conclusion

The following conclusions can be drawn from this experimental study using different diesel-palm stearin biodiesel blends as fuels in a variable compression ratio CI engine.

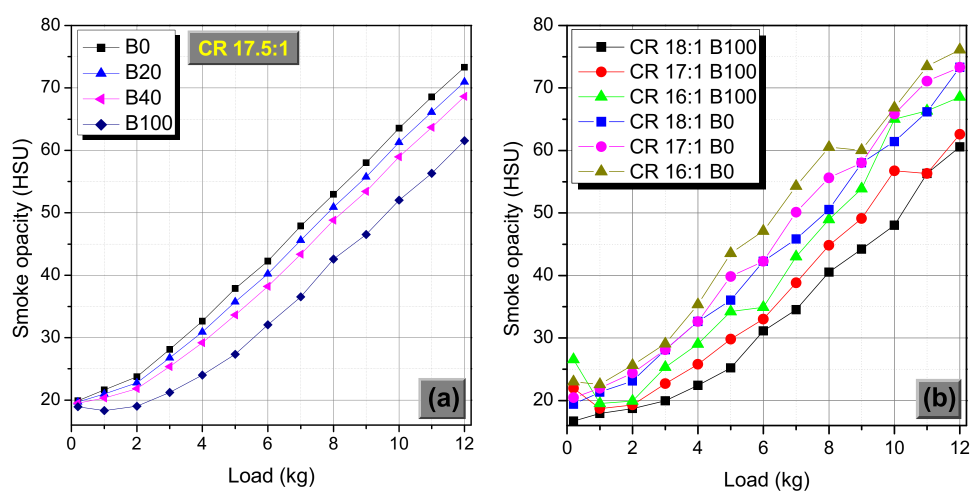

Fig. 12 Variation of smoke opacity with load for diesel and biodiesel blends for different compression ratios

- Brake thermal efficiency (BTE) decreases with the increase in biodiesel percentage in the blends of diesel and biodiesel, and the reduction is found to be 7.9% when diesel is completely replaced by biodiesel. BTE increases with increase in compression ratio for all the tested fuels; however, the increase is more with neat diesel.
- Brake specific fuel consumption for pure biodiesel (B100) is higher than for diesel fuel (B0) for any load condition. The maximum increase in BSFC is noted to be 17.30% at the rated compression ratio of 17.5:1.
- Exhaust gas temperature increases with the increase in biodiesel fraction in the blend and also with the decrease in compression ratio.
- The in-cylinder pressure and temperature are higher for biodiesel fuels. The highest pressure recorded is 83.71 bar at 10°aTDC for neat biodiesel at compression ratio of 18:1.
- The highest ignition delay is noted to be 21°CA for B0 at compression ratio of 16:1.
- The peak heat release rate increases with the lowering of compression ratio, and its maximum value is found to be 48.16 J/CA at 8°aTDC for B0 at compression ratio of 16:1.
- CO, HC and smoke emissions are significantly reduced when pure biodiesel or its blends with diesel are used as fuels. The emissions of HC, CO and smoke with neat biodiesel are found to be less by 26, 12 and 16%, respectively, compared to diesel at rated operating condition. HC, CO and smoke emissions increase with the decrease in compression ratio.
- However, NO_x and CO_2 emissions increase with biodiesel concentration in the blended fuels. The increment of NO_x and CO_2 emission is 48 and 50%, respectively, with neat biodiesel compared to mineral diesel at rated operating condition. Also NO_x and CO_2 emissions increase with the increase in compression ratio for all the tested fuels.

At higher compression ratio, the performance and combustion characteristics are improved as well as the emissions of CO, HC and smoke are reduced, which can be beneficial for CI engines running with neat or blended biodiesel. The higher NO_x emission with biodiesel can be reduced by implementing some after treatment method such as exhaust gas recirculation (EGR) technique. The future research work may be carried out with more parametric variations such as the effect of injection pressure and injection timing, exhaust gas recirculation and intake air preheating.

References

Amarnath HK, Prabhakaran P (2012) A study on the thermal performance and emissions of a variable compression ratio diesel engine fuelled with karanja biodiesel and the optimization of parameters based on experimental data. Int J Green Energy 9:841–863

Amarnath HK, Prabhakaran P, Bhat SA, Paatil R (2014) A comparative analysis of thermal performance and emission characteristics of methyl esters of karanja and jatropha oils based on a variable compression ratio diesel engine. Int J Green Energy 11:675–694

Anand K, Sharma RP, Mehta PS (2011) Experimental investigation on combustion, performance and emission characteristics of neat karanji biodiesel and its methanol blend in a diesel engine. Biomass Bioenergy 35:533–541

Babu AR, Rao GAP, Prasad TH (2015) Performance and emission characteristics of CI engine with methyl esters of palm stearin under varying compression ratios and fuel injection pressures. Int J Eng Manag Res 5:61–69

Babu AR, Rao GAP, Prasad TH (2016) Experimental investigations on a variable compression ratio (VCR) CIDI engine with a blend of methyl esters palm stearin-diesel for performance and emissions. Int J Ambient Energy. doi:10.1080/01430750.2015.1132768

Behçet R (2011) Performance and emission study of waste anchovy fish biodiesel in a diesel engine. Fuel Process Technol 92:1187–1194

Canakci M, Sayin C, Gumus M (2008) Exhaust emissions and combustion characteristics of a direct injection (DI) diesel engine fueled with methanol-diesel fuel blends at different injection timings. Energy Fuels 22:3709–3723

Datta A, Mandal BK (2016) A comprehensive review of biodiesel as an alternative fuel for compression ignition engine. Renew Sustain Energy Rev 57:799–821

Gumus MA (2010) Comprehensive experimental investigation of combustion and heat release characteristics of a biodiesel (hazelnut kernel oil methyl ester) fueled direct injection compression ignition engine. Fuel 89:2802–2814

Heywood JB (1988) Internal combustion engine fundamentals. McGraw-Hill Co, New York

Hirkude J, Padalkar AS (2014) Experimental investigation of the effect of compression ratio on performance and emissions of CI engine operated with waste fried oil methyl ester blend. Fuel Process Technol 128:367–375

Holman JP (2012) Experimental methods for engineers, 4th edn. McGraw-Hill Co, New York

Huang ZH, Lu HB, Jiang DM, Zeng K, Liu B, Zhang JQ, Wang XB (2004) Combustion characteristics and heat release analysis of a compression ignition engine operating on a diesel/methanol blend. Proc Inst Mech Eng D—J Autmob 218:1011–1024

Jindal S (2011) Experimental investigation of the effect of compression ratio and injection pressure in a direct injection diesel engine running on karanj methyl ester. Int J Sustain Energy 30:S91–S105

Jindal S, Nandwana BP, Rathore NS, Vashistha V (2010) Experimental investigation of the effect of compression ratio and injection pressure in a direct injection diesel engine running on jatropha methyl ester. Appl Therm Eng 30:442–448

Lai OM, Ghazali HM, Cho F, Chong CL (2000) Physical and textural properties of an experimental table margarine prepared from lipase-catalysed transesterified palm stearin: palm kernel olein mixture during storage. Food Chem 71:173–179

Mohanraj T, Kumar KMM (2013) Operating characteristics of a variable compression ratio engine using esterified tamanu oil. Int J Green Energy 10:285–301

Muralidharan K, Vasudevan D (2011) Performance, emission and combustion characteristics of a variable compression ratio engine using methyl esters of waste cooking oil and diesel blends. Appl Energy 88:3959–3968

Palit S, Chowdhuri AK, Mandal BK (2011) Environmental impact of using biodiesel as fuel in transportation: a review. Int J Glob Warm 3:232–256

Qi DH, Geng LM, Chen H, Bian YZH, Liu J, Ren XCH (2009) Combustion and performance evaluation of a diesel engine fueled with biodiesel produced from soybean crude oil. Renew Energy 34:2706–2713

Raheman H, Ghadge SV (2008) Performance of diesel engine with biodiesel at varying compression ratio and ignition timing. Fuel 87:2659–2666

Raheman H, Phadatare AG (2004) Diesel engine emissions and performance from blends of karanja methyl ester and diesel. Biomass Bioenergy 27:393–397

Ramalingam S, Chinnaia P, Rajendran S (2014) Influence of compression ratio on the performance and emission characteristics of annona methyl ester operated DI diesel engine. Adv Mech Eng. doi:10.1155/2014/832470

Rao GAP, Kaleemuddin S (2011) Development of variable timing fuel injection cam for effective abatement of diesel engine emissions. Appl Energy 88:2653–2662

Sahoo PF, Das LM (2009) Combustion analysis of jatropha, karanja and polanga based biodiesel as fuel in a diesel engine. Fuel 88:994–999

Sayin C (2010) Engine performance and exhaust gas emissions of methanol and ethanol–diesel blends. Fuel 89:3410–3415

Sayin C, Gumus M (2011) Impact of compression ratio and injection parameters on the performance and emissions of a DI diesel engine fueled with biodiesel-blended diesel fuel. Appl Therm Eng 31:3182–3188

Sharma A, Murugan S (2015) Potential for using a tyre pyrolysis oil-biodiesel blend in a diesel engine at different compression ratios. Energy Convers Manag 93:289–297

Sharon H, Karuppasamy K, Kumar DRS, Sundaresan A (2012) A test on DI diesel engine fueled with methyl esters of used palm oil. Renew Energy 47:160–166

Sonar D, Soni SL, Sharma D, Srivastava A, Goyal R (2015) Performance and emission characteristics of a diesel engine with varying injection pressure and fuelled with raw mahua oil (preheated and blends) and mahua oil methyl ester. Clean Technol Environ Policy 17:1499–1511