Alexandria Engineering Journal (2013) 52, 141–149

Alexandria University

Alexandria Engineering Journal

www.elsevier.com/locate/aej
www.sciencedirect.com

ORIGINAL ARTICLE

Experimental investigations of ignition delay period and performance of a diesel engine operated with Jatropha oil biodiesel

Mohammed EL-Kasaby, Medhat A. Nemit-allah *

Mechanical Engineering Department, Faculty of Engineering, Alexandria University, Alexandria 21544, Egypt

Received 21 September 2012; revised 14 December 2012; accepted 18 December 2012
Available online 27 February 2013

KEYWORDS

Biodiesel;
Combustion characteristics;
Engine performance;
Delay up period

Abstract Jatropha-curcas as a non-edible methyl ester biodiesel fuel source is used to run single cylinder, variable compression ratio, and four-stroke diesel engine. Combustion characteristics as well as engine performance are measured for different biodiesel – diesel blends. It has been shown that B50 (50% of biodiesel in a mixture of biodiesel and diesel fuel) gives the highest peak pressure at 1750 rpm, while B10 gives the highest peak pressure at low speed, 1000 rpm. B50 shows upper brake torque, while B0 shows the highest volumetric efficiency. B50 shows also, the highest BSFC by about (12.5–25%) compared with diesel fuel. B10 gives the highest brake thermal efficiency. B50 to B30 show nearly the lowest CO concentration, besides CO concentration is the highest at both idle and high running speeds. Exhaust temperature and NO_x are maximum for B50. Delay period is measured and correlated for different blends. Modified empirical formulae are obtained for each blend. The delay period is found to be decreased with the increase of cylinder pressure, temperature and equivalence ratio.

1. Introduction

The combustion of fuels derived from crude petroleum oil (fossil fuel) is the culprit and has a dangerous impact on environment. In addition, irregular and fluctuated petroleum barrel price besides the rapid depletion of petroleum reserve are the main cause for searching about new alternative and clean fuel sources. Renewable energy source such as natural gas, biogas, vegetable (green) biofuels are recently gained much scientific efforts to be produced in economical, available, safe and environment friendly nature. Biodiesel as an alternative fuel derived from vegetable (green) oil or animal fats are oxygenated, biodegradable, non-toxic and environmentally safe. It consists of alkyl monester of fatty acids from tri-acyglycerols. Biodiesels are classified into two categories; namely edible and non-edible oils. Edible oils are such that sunflower, corn, rapeseed, palm, soybean and waste vegetable oils. The non-edible oils are such that Jatropha, Jojoba, Karanja, Polanga oils and likes.

* Corresponding author. Tel.: +20 966598212483.
E-mail address: Medhatahmed@kfupm.edu.sa (M.A. Nemit-allah).
Peer review under responsibility of Faculty of Engineering, Alexandria University.

ELSEVIER Production and hosting by Elsevier

1110-0168 © 2013 Faculty of Engineering, Alexandria University. Production and hosting by Elsevier B.V. All rights reserved.
http://dx.doi.org/10.1016/j.aej.2012.12.006

Nomenclatures

CA	crank angle, °	%B	percentage of biodiesel in a mixture of biodiesel and diesel fuel
HC	hydrocarbon	CN	cetane number
EXP	exponential parameter	DI	direct injection
NO_x	nitrogen oxides	N	engine speed, rpm
PM	particulate matters	P_{cyl}	cylinder pressure, bar.
SOI	start of ignition	$\overline{R}$	universal gas constant = 8.316 kJ/kg K
T_{cyl}	cylinder temperature, K	$\bar{S}_p$	mean piston speed, m/s
τ	delay period, ms	UHC	unburned hydrocarbons
Φ	equivalence ratio = $(F/A)_a/(F/A)_{th}$	ID	ignition delay, ms

Gvidonas et al. [3] presented the effect of neat rapeseed biodiesel and its blends with diesel fuel on engine performance and exhaust emission. Brake specific fuel consumption at maximum torque and rated engine power varied from 35.6% to 39.8% for rapeseed and from 37.3% to 38.3% for diesel fuel. The highest fuel energy content based economy 9.36–9.6% MJ/kW h is achieved during operation on blend B10 where as the lowest one belongs to B35 and neat rapeseed. Maximum NO_X emission increases with the increase of mass percent of O_2 in biodiesel than diesel. CO_2 is higher for biodiesel while UHC is low in biodiesel than diesel fuel. Szybist et al. [4] reviewed the use of biodiesel that rapidly expanding around the world making it imperative to fully understand the impact of biodiesel on the diesel combustion process, pollutant formation and exhaust after treatment. The start of injection, SOI, timing was found to advance as the fuel bulk modulus increases biodiesel fuels have higher bulks modulus than fossil fuel. So, it is well known that NO_x emission decreases with SOI timing retarding. Therefore, it can be concluded that NO_x increases by advancing the start of injection. Nwafor [5], investigated the effect of heating the neat rapeseed vegetable oil up to 70 °C on emission characteristics and fuel delay period. The test showed longer delay of rapeseed than diesel fuel. In addition, the heated rapeseed fuel has better performance than unheated rapeseed. Recycled waste cooking oil trans-esterificated with methanol was used by Lin Ya-fen et al. [6] to produce biodiesel which was blended with diesel to operate diesel engine. Performance and neat biodiesel and its blends produced higher NO_x, lower CO and higher CO_2 than diesel. B20 and B50 were the optimum performance blends while B100, B80 and B50 are evaluated as molecular structure of $C_{15}H_{32}$ and squalence $C_{30}H_{50}$ was dominated for B20. It has been pointed out that biodiesel impact on emission varied depending on the type of biodiesel and on the type of conventional diesel fuel added to blend as well as to the engine operating conditions that makes each study specific.

Bio-diesel production from soybean trans-esterification blended with diesel was tested by Qi et al. [9]. The peak pressure and peak heat transfer rate were higher for biodiesel. The power of biodiesel was almost identical with that of diesel. The brake specific fuel consumption was also higher for biodiesel due to lower heating value. Bio-diesel provided significant reduction of CO, HC, NO_x and smoke under speed characterizing at full engine load. In spite of slightly higher viscosity and lower volatility of biodiesel, the ignition delay seemed to be lower for biodiesel than diesel due to high combustion temperature. Comparison of operational characteristics of diesel engine run with both sunflower biodiesel and diesel blends was performed by Kandasmy et al. [10]. It was found that the thermal efficiency was slightly less and the specific fuel consumption was slightly higher due to lower calorific value of sunflower biodiesel fuel. Delay period was shorter for sunflower and hence the peak pressure was slightly lower than diesel fuel for all load operating condition. Exhaust temperature of sunflower was high because of effective burning of biodiesel made by very small droplet in sizes when compared with diesel fuel. Non-edible filtered Jatropha curcas, Karanja and Polanga oil based monsters as biodiesel produced and blended with diesel were tested for their use as substitute fuel for diesel engines were dedicated by Sahoo et al. [13]. The engine combustion parameters as peak pressure, time of occurrence of peak pressure, heat release rate and ignition delay were measured. Maximum peak pressure was found for Polanga, while minimum ignition delay was found for Jatropha (5.9–4.2 CA).

2. Experimental set-up and measuring techniques

2.1. Trans-esterification of Jatropha curcas

Jatropha seeds are crushed and filtered to obtain crude vegetable Jatropha oil. The oil is methyl-estered using methanol and catalyzed by sodium hydroxide in special device and according to specific titration processes.

For simplicity, one can consider that Jatropha oil to be consisted of pure triolein. Triolein is a triglyceride in which all three fatty acid chains are oleic acid. This is near the actual number of carbons and hydrogens and gives a molecular weight that is near the value of Jatropha oil. If triolein is reacted with methanol, the reaction will be as shown in Fig. 1. Consequently; one can conclude that the chemical structure of the biodiesel is $C_{19}H_{36}O_2$. The physical and chemical properties of Jatropha biodiesel are listed in Table 1.

2.2. Engine specifications and test procedures

The combustion characteristics, engine performance and exhaust emission levels are measured by testing Jatropha biodiesel blend with diesel fuel, on a four-stroke, single cylinder, direct injection Tec-Quipment TD43F variable compression ratio diesel engine. The engine has a bore of 95 mm stroke of 82 mm, displacement of 582 cm^3. The percentages of biodiesel

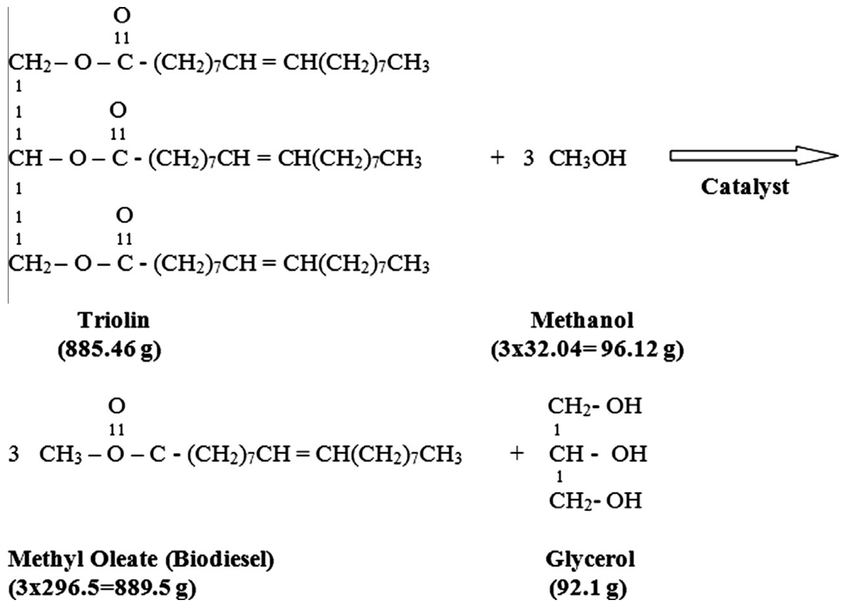

Figure 1 Biodiesel chemical reaction equation.

Table 1 Physical and chemical properties of Jatropha biodiesel.

Properties	Jatropha crude oil	Jatropha biodiesel	Diesel	Test standard
Density g/ml	0.92	0.865	0.841	
Viscosity, C. stokes at 40 °C	32.5	5.2	4.5	ASTM D445
Heating value MJ/kg	35.2	34.5	42	ASTM D240
Flash point °C	240	175	50	ASTM D93
Cloud point °C	16	13	9	ASTM D2500
Cetane number	38	51	49	ASTM D613

that were investigated are: 10% (B10), 20% (B20), and 30% (B30), 50% (B50) In addition, pure diesel fuel (B0). The emission and performance tests were performed at a compression ratio of 18 and at engine speeds ranging from 1000 to 2000 rpm, incremently increased by 250 rpm at fully opened throttle conditions for each speed (i.e at maximum load for each speed). The engine was coupled to a conventional D.C. electric dynamometer which besides loading the engine, can be used as a starter motor to start the engine or to motor the engine when measuring friction power. K type thermocouples of range (0 to +10,000 °C) were used to measure the temperature of exhaust gases, and K type thermocouples of range (0 to +1500 °C) to measure the temperature of the cooling water. These thermocouples were connected to the thermocouple card in the engine control unit which can be displayed on the temperature meter or saved on a computer disc. The cylinder pressure was measured and recorded against the crank angle for each 0.5° increment. The data from the engine under test can be collected for a given cycle, displayed, and saved on the computer connected to the unit.

The exhaust gases were analyzed by the IMR 3000P/IMR 3010P gas analyzing apparatus displayed on computer. The amounts of carbon monoxide CO (1 ppm accuracy) and carbon dioxide CO_2 (0.1 vol% accuracy), nitrogen oxides NO and NO_2 (1 ppm accuracy), as well as the amounts of sulphur dioxide SO_2 (1 ppm accuracy) emissions were recorded with the unit.

From the pressure – crank angle diagram we can specify the start of ignition point at which the pressure increases suddenly and the pressure at this point is the ignition pressure, this point was checked again using the $\log(p) - \log(v)$ curve. As the compression process is polytropic (pv^n = constant); so, by taking the logarithm for both sides and drawing the $\log(p) - \log(v)$ diagram, the compression process will be linear in this curve and the start of ignition point will be the end point of this line, also the slope of the line is the polytropic index (n). For all experiments we set the pump injection angle to be 20° BTDC (240°), so the ignition delay angle, $\Delta\theta$, is the difference between the two angles in degree. To calculate the ignition temperature, the compression process has been considered to be polytropic with the polytropic index calculated as mentioned above. Also that the pressure and temperature at the start of the compression stroke was considered to be the atmospheric conditions 1.013 bar and 25 °C respectively. Values for ignition pressure (P_{cyl}, bar), delay period (τ, ms), polytropic index (n), and ignition temperature (T_{cyl}, K) are listed in Table 2 at 1000 rpm.

The equivalence ratio (Φ) is defined to be equals $(F/A)_{actual}/(F/A)_{theoretical}$ with the actual mass flow rate of fuel and air are measured individually. The theoretical fuel to air ratio is calculated by assuming stoichoimetric combustion of the biodiesel and diesel blend with theoretical air quantity. Jatropha neat biodiesel structure is taken as $C_{19}H_{36}O_2$. On the other hand diesel fuel is taken as $C_{12}H_{26}$ as equivalent to dedocane pure fuel. Values for the equivalence ratio are listed in Table 2 at

Table 2 Measured and derived parameters for Jatropha and its blends with diesel.

Fuel	rpm	τ (ms)	P_{cyl} (bar)	T_{cyl} (K)	n exponent	$\Phi = (F/A)_a/(F/A)_{th}$	CN	E_a/R (K)
B0	1000	2.92	25.2	1037	1.47	0.71	47	1034
B10	1000	2.67	31.4	815	1.46	0.64	47.5	1027
B20	1000	2.92	28.4	1323	1.79	0.71	47.9	1021
B30	1000	2.58	34	1276	1.69	0.76	48.3	1016
B50	1000	2.83	36.2	1086	1.55	0.8	49.2	1003

1000 rpm. For Jatropha biodiesel and its blends with diesel fuel with the properties tabulated in Table 1, the apparent activation energy, E_a, is calculated for each blend using the following relation taken from Heywood [18] and listed in Table 2 for each blend at 1000 rpm.

$$E_a = 618,840/(\mathrm{CN} + 25) \tag{1}$$

3. Results and analysis

3.1. Engine performance

3.1.1. Engine torque

The results of the engine's brake power and torque, at full load, are shown in Fig. 2. It can be observed that the B50 fuel resulted in lower brake power and torque relative to the baseline fuel (B0) and all blends, because of the lower heat of combustion of the B50 fuel blend and lower fuel delivery at full load due to the higher mass flow rate required from the fixed nozzle area. The output power was reduced by 2.74%, 2.51%, 9.1%, and 8.67% for the blends B10, B20, B30, and B50 respectively at maximum engine load and speed. The maximum engine power, recorded at 2043 rpm, decreased from 4.555 kW for diesel fuel to 4.16 kW for B50. Other experiences showing similar power recoveries have been reported [20–24].

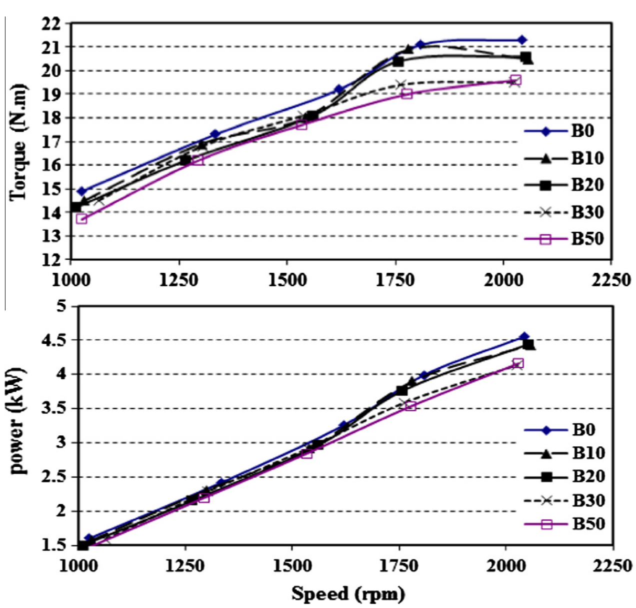

Figure 2 Engine torque and power as a function of engine speed at full load condition for all blends of Jatropha.

3.1.2. Brake specific fuel consumption

Brake specific fuel consumption (bsfc) is the ratio between mass fuel consumption and brake effective power, the loss of heating value of biodiesel must be compensated with higher fuel consumption. Fig. 3 compares the bsfc of pure diesel and its blends with Jatropha oil biodiesel fuel. There was about 8.62%, 6.6%, 5.57%, and 16.82% higher specific fuel consumption when running on B10, B20, B30, and B50 than diesel fuel at 2000 rpm respectively. This was due to the higher relative density and lower energy density of Jatropha oil biodiesel. The net calorific value of the Jatropha oil biodiesel is about 14.88% lower than that of diesel fuel. This may explain the increase in fuel consumption. The trends of the blends showed an increase in fuel consumption approximately proportional to the amount of Jatropha oil biodiesel added to the diesel fuel. The higher density of Jatropha oil biodiesel and its blends would cause higher mass injection for the same volume because of high injection pressures. It is normal to have intentional leakage past the injector needle for lubrication purposes. The high viscosity of Jatropha oil biodiesel was noticed to reduce this leakage, again resulting in more fuel charge entering the combustion chamber and so higher bsfc. Some studies have reported results which support our general trends. Lin et al. [25] observed 3.3% and 16.7% increases in bsfc when palm-oil biodiesel was used in 20% blends and pure diesel fuel respectively with respect to that obtained with diesel fuel. Similarly, Haas et al. [26] found 18% increases when they used pure biodiesel from soybean oil and soap stock.

3.1.3. Brake thermal efficiency

The variation of brake thermal efficiency with speed for different blends is presented in Fig. 4. The maximum brake thermal efficiencies were obtained to be 21.1% for B0, which was higher than that of all blends. The maximum brake thermal efficiencies obtained for B10, B20, B30, and B50 were

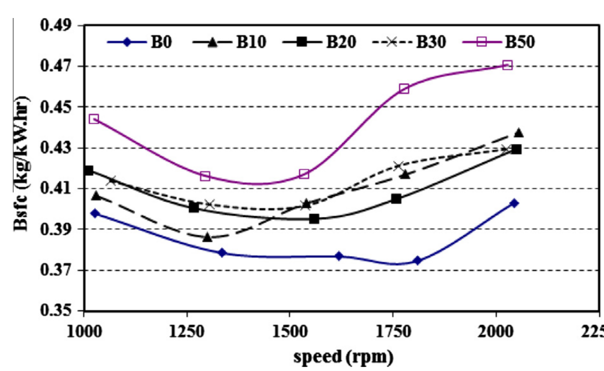

Figure 3 Comparisons of brake specific fuel consumption as a function of speed at full load condition for all blends of Jatropha.

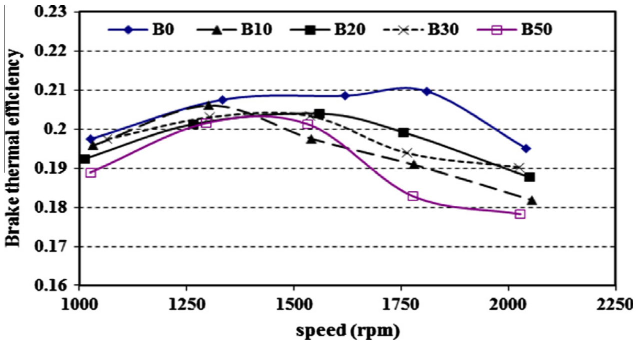

Figure 4 Comparisons of brake thermal efficiency as a function of speed at full load condition for all blends of Jatropha.

20.51%, 20.4%, 20.4% and 20.3%, respectively. This lower brake thermal efficiency obtained for these blends than diesel could be due to a reduction in the calorific value and an increase in fuel consumption as compared to B0. This behavior was found in many studies, Murillo et al. [27] found similar synergies. These authors tested different blends of conventional diesel fuel and *d* biodiesel from used cooking oil, at full load, in a marine outboard three-cylinder naturally aspirated engine. With blends of 10%, 30% and 50% of biodiesel, efficiency was lower than that obtained with diesel fuel, but the highest efficiency was found with pure biodiesel.

3.1.4. Exhaust gas temperature

The combustion temperature is an indicator of the amount of energy released during combustion. The exhaust temperature measured about 30 mm away from the combustion chamber served as an indicator of the combustion temperature relating to heat release. As shown in Fig. 5 the blend B20 showed the nearest exhaust temperature to the pure diesel operation and the diesel fuel operation recorded lower exhaust temperatures than all blends at low speeds. This behavior may be related to the oxygenated nature of biodiesel which will lead more complete combustion and so higher exhaust gas temperature at low speeds. At higher speeds the exhaust temperature of all blends decrease than diesel fuel as shown in Fig. 5. This may be related to the higher relative density and lower energy density of biodiesel, where the oxygen content loss its positive effect at high speeds, and the effect of heating value is the dominant. Note that these results are matched with the results obtained for the six cylinder engine with waste oil biodiesel operation.

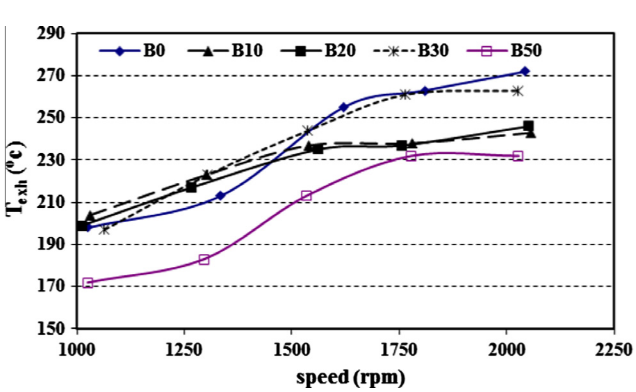

Figure 5 Comparisons of exhaust temperature as a function of speed at full load condition for all blends of Jatropha.

3.2. Emissions

3.2.1. CO emission

With regard to most of the literature reviewed, a decrease in CO emissions when substituting diesel fuel with biodiesel can be considered as the general trend [28–32]. Nevertheless, a few authors found no differences between diesel and biodiesel [33], and even noticeable increases when using biodiesel [34]. As shown in Fig. 6, CO formation decreased with increasing engine speeds for all blends till 1700 rpm, then increase with increasing the speed. For the low speed operation, poor atomization and uneven distribution of small portions of fuel across the combustion chamber, along with a low gas temperature, may lead to local oxygen deficiency and incomplete combustion. That could be the answer as to why CO emissions tend to increase for the easy loaded engine. At high revolutions and high radial turbulence intensity in the combustion chamber, the mixing of the fuel rich portions with ambient air should be improved, but on the other hand, the duration of the combustion process expressed in units of time becomes too limited, which results in increasing CO emission. As it is clear from the figure that the curves of CO emissions for all biodiesel blends remain under the curve of pure diesel and decrease as the biodiesel percent increase. Several reasons have been reported to explain the general CO decrease when substituting conventional diesel for biodiesel: (1) The additional oxygen content in the fuel, which enhances a complete combustion of the fuel, thus reducing CO emissions. (2) The increased biodiesel cetane number [30,31]. The higher the cetane number, the lower the probability of fuel- rich zones formation, usually related to CO emissions. (3) As commented in other sections, the advanced injection and combustion when using biodiesel may also justify the CO reduction with this fuel.

3.2.2. NO emission

The emissions of NO against engine speed from various blends are compared in Fig. 7. The biodiesel and biodiesel/diesel blend fuels produced higher NO for all engine speeds as expected. Among these fuels, B50 produced the highest NO for engine speed higher than 1500 while B20 gave the highest NO at engine speeds of 1000 and 1250 rpm. Among engine operations, both engine speeds of 1750 and 2000 rpm produced the highest NO concentration for all types of fuels, while the engine speeds of 1000 and 1250 produced the lowest NO concentration. For all of the blends, the curves for each blend

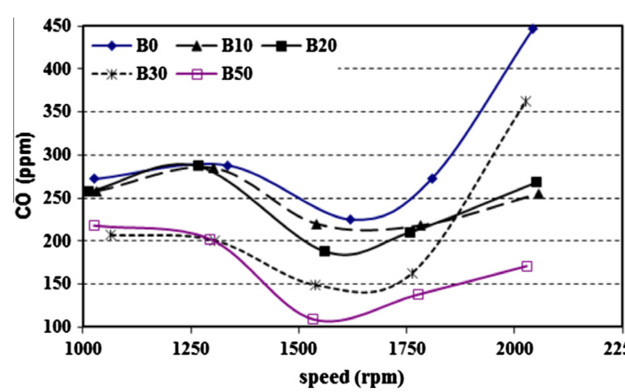

Figure 6 Comparisons of CO emission as a function of speed at full load condition for all blends of Jatropha.

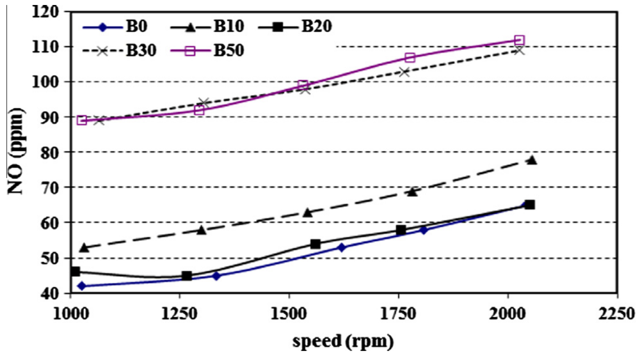

Figure 7 Comparisons of NO emission as a function of speed at full load condition for all blends of Jatropha.

remains over the curve of pure diesel. There were some reasons for this behavior: (1) Regarding the adiabatic flame temperature, some authors state that it is slightly higher for biodiesel because of its oxygenated nature which help for more complete combustion and so higher temperature and NO_x emission. (2) Regarding the reduction in soot formation with biodiesel. Radiation from soot produced in the flame zone is a major source of heat transfer away from the flame, and can lower bulk flame temperatures by 25–125 K, depending on the amount of soot produced at the engine operating conditions. (3) Biodiesel typically contains more double bonded molecules than petroleum derived diesel. These double bonded molecules have a slightly higher adiabatic flame temperature, which leads to the increase in NO_x production for biodiesel.

3.3. Combustion characteristics

In a CI engine, cylinder pressure depends on the burned fuel fraction during the premixed burning phase, i.e., initial stage of combustion. Cylinder pressure characterizes the ability of the fuel to mix well with air and burn. High peak pressure and maximum rate of pressure rise correspond to large amount of fuel burned in premixed combustion stage. The cylinder pressure crank angle history is obtained at different speeds for diesel and Jatropha oil biodiesel blends. Also Peak pressure is recorded at different speeds.

3.3.1. Effect of blends on cylinder pressure

The variations in the cylinder pressure with crank angle for diesel and biodiesel at different engine speeds are shown in Fig. 8. It is clear that the peak cylinder pressure is higher for biodiesel at all engine speeds as it is represented in Fig. 9. At all engine speeds the combustion starts earlier for biodiesel than for diesel. This can be attributed to a short ignition delay and advanced injection timing for biodiesel (because of a higher bulk modulus which will result in higher speed of sound in the fuel injection pipe causing early injection and consequently early combustion which will result in higher cylinder pressure, higher density of biodiesel, and higher cetane number) [35]. The possible reason for the trends in the peak cylinder pressure is because of the longer ignition delay for diesel than for biodiesel, combustion starts later for diesel fuel. As a result, the peak cylinder pressure attains a lower value as it is further away from the TDC in the expansion stroke. However, the cylinder pressure during the late combustion phase for biodiesel and their blends is marginally lower than that of diesel. This

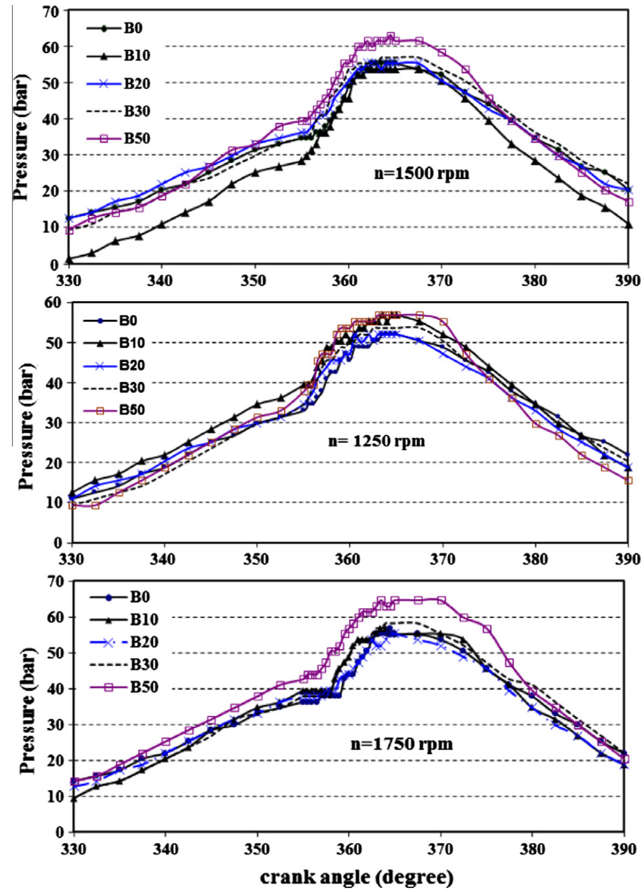

Figure 8 Pressure vs. crank angle for diesel, Jatropha biodiesel blends at full load conditions for various rotational speeds.

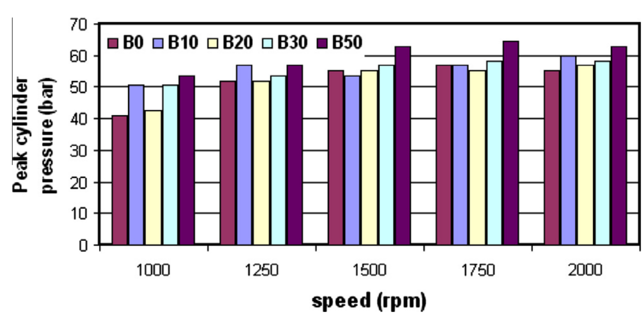

Figure 9 Variation of peak cylinder pressure with engine speed for Jatropha biodiesel blends at full load conditions.

is because the constituents with higher oxygen content of biodiesel blends are adequate to ensure complete combustion of the fuel during the main combustion phase but for diesel fuel, it continues to burn in the late combustion phase because of delay in combustion. The early maximum pressure characteristics warrant careful attention to ensure that, while running with biodiesel and their blends, the peak pressure takes place definitely after TDC for safe and efficient operation. Otherwise, a peak pressure occurring very close to TDC or before that causes severe engine knock, and thus affects engine durability. It is concluded from this discussion that as biodiesel percent increases, the peak cylinder pressure increases.

3.3.2. Effect of blends on ignition delay

The increase in fuel viscosity, particularly for petroleum derived fuels, resulted in poor atomization, slower mixing, increased penetration and reduced flame cone angle. These resulted in longer ignition delay. But biodiesel is not derived from crude petroleum, and the opposite trend is seen in the case of biodiesel and their blends. This study ignition delay (τ, ms) calculated as mentioned in chapter 3 and tabulated in Appendix (D). Fig. 10 compares the delay between neat diesel and biodiesel blends at various speeds. As shown in the figure, as speed increases, the delay period decreases for all blends. This attributed to the decrease in residual gas temperature and wall temperature, which result in lower charge temperature at injection time and lengthening the ignition delay [35]. The ignition delays are consistently shortest for the blend B50. In spite of the slightly higher viscosity and lower volatility of biodiesel, the ignition delay seems to be lower for biodiesel than for diesel. The reason may be that a complex and rapid prefflame chemical reaction takes place at high temperatures. As a result of the high cylinder temperature existing fuel injection, biodiesel may undergo thermal cracking and lighter compounds are produced, which might have ignited earlier to result in a shorter ignition delay [36].

Biodiesel usually includes a small percentage of diglycerides with higher boiling points than diesel. However, the chemical reactions during the injection of biodiesel at high temperature resulted in the break-down of the high-molecular weight esters. These complex chemical reactions led to the formation of gases of low-molecular weight. Rapid gasification of this lighter oil in the fringe of the spray spreads out the jet, and thus volatile combustion compounds ignited earlier and reduced the delay period. Biodiesel derived from Jatropha oil with higher molecular weight is likely to react identically.

3.3.3. Ignition delay correlations

In the present work, the Arrhenus equation modified by EL-Bahnasy and El-Kotb [16] has been considered, with the help of Heywood [18] to calculate the delay period of Jatropha biodiesel blends with diesel fuels as a function of cylinder pressure, cylinder temperature, and equivalence ratio as follows:

$$\tau_{\mathrm{id}} = A \cdot P^{-n} \cdot \Phi_1^{-m}\ \mathrm{EXP}(E_a/R^- \cdot T) \qquad (2)$$

From the ignition delay results using least square fitting technique, one can get the coefficients A, n, m and the general empirical relation of each blend. Some equations were obtained

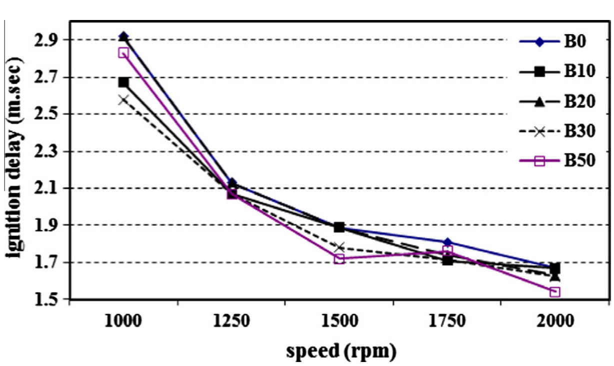

Figure 10 Effect of speed on the ignition delay for diesel and biodiesel blends.

for each blend as a function of each parameter (P, T, Φ) as presented in Fig. 11. As it is clear from the figure, the delay period decrease with the increase of both pressure and temperature for all blends. This may be attributed to the fact that with increasing the pressure the mixture molecules become closer and the probability of increasing the active collisions between these molecules is higher, as a result the chemical reactions will be accelerated to complete the combustion with shorter delay period. The experiments are carried out in lean and slightly rich mixture zones, (Φ = 0.6–1.2), so the delay period decreases with the equivalence ratio. It is expected that if the experiments cover the rich mixture zone, the delay period returns to increase with the increase of the equivalence ratio.

From the equations used to draw Fig. 11, we got more general formulas for the ignition delay as a function of all parameters as follows:

For B0 :

$$\tau_{\mathrm{B0}} = 26.06P^{-1.21}\Phi^{-1.36}\ \mathrm{EXP}[1038/T] \qquad (3)$$

For B10:

$$\tau_{\mathrm{B10}} = 79.51P^{-1.45}\Phi^{-0.81}\ \mathrm{EXP}[1028/T] \qquad (4)$$

For B20:

$$\tau_{\mathrm{B20}} = 74.32P^{-1.32}\Phi^{-1.39}\ \mathrm{EXP}[1022/T] \qquad (5)$$

For B30:

$$\tau_{\mathrm{B30}} = 1.55.10^3P^{-2.12}\Phi^{-0.1.01}\mathrm{EXP}[1015.2/T] \qquad (6)$$

For B50:

$$\tau_{\mathrm{B50}} = 3.461.10^3P^{-2.27}\Phi^{-1.06}\ \mathrm{EXP}[1003/T] \qquad (7)$$

other empirical formulas of delay period of biodiesel blends are concluded in the present work from the previous equations as follows:

$$\tau = 124.3 * e^{0.0893*\%\mathrm{B}} * P^{(-0.0234*\%\mathrm{B}-1.1715)} \qquad (8)$$

$$\tau = (0.0039 * \%\mathrm{B} + 1.7031) * \Phi^{(0.0072*\%\mathrm{B}-1.3916)} \qquad (9)$$

$$\tau = (22.104 * \%\mathrm{B} + 1131)/T \qquad (10)$$

where %B is the blend percent.

4. Conclusions

Jatropha curcas as biodiesel renewable alternative fuel source for diesel engine is preferred to be planted over large desert areas with municipal waste water. The wide extension of Jatropha hektars aims to use pure oil extracted from its seeds to produce biodiesel as an alternative non-edible biodiesel oil source. This work proves that the combustion characteristics of this biodiesel is comparable to that of diesel fuel without any modification of the diesel engines. According to the results obtained from the present work, one can draw the following conclusions:

1. Non-edible vegetable (green) oil derived from some plants as Jatroha that can be grown in an un-fertilized lands must be widely cultivated to produce bi-diesel fuel instead of using the edible oil plants such as corn, sun-flower, soybean.

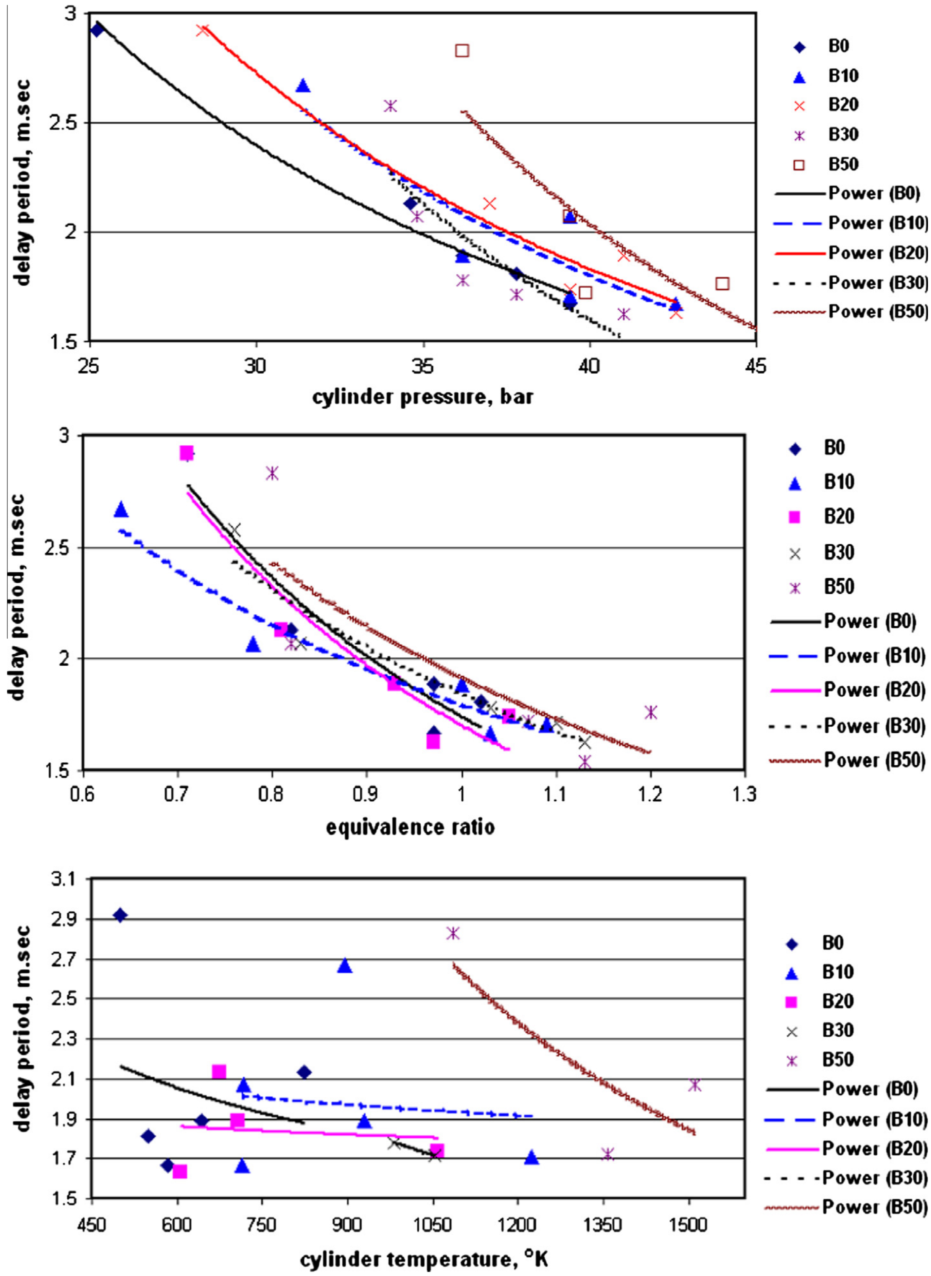

Figure 11 The ignition delay results obtained for each blend as a function of cylinder pressure, equivalence ratio, and cylinder temperature respectively.

2. Peak pressure of B50 is higher at low and high engine speed, while that of B10 and B20 are optimum at economic engine speed (medium speed).
3. Higher percentage of NO_x in case of biodiesel compared with that of diesel is attributed to the higher combustion temperature of oxygenated biodiesel resulted from advanced injection.

References

[3] G. Labeckas, S. Slavinskas, The effect of rape seed oil methyl-ester on direct injection diesel engine performance and exhaust emission, Energy Conversion and Management 47 (2006) 11954–11967.

[4] J.P. Szybist, J. Song, M. Alam, A.L. Boehman, Bio-diesel combustion, emission and emission control, Fuel Processing Technology 88 (2007) 679–691.

[5] O.M. Nwafor, Emission characteristics of diesel engine running on vegetable oil with elevated fuel inlet temperature, Biomass and Bio-energy 27 (2004) 509–511.

[6] Ya-fen. Lin, G. Wu Yo-Ping, C.T. Chang, Combustion characteristics of waste oil produced bio-diesel, diesel fuel blends, Fuel 86 (2007) 1772–1789.

[9] D.H. Qi, L.M. Geng, H. Chen, Y.ZH. Bian, J. Liu, X.C.H. Ren, Combustion and performance evaluation of a bio-diesel engine fuelled with bio-diesel produced from soybean crude oil, Renewable Energy xxx (2009) 1–8, http://dx.doi.org/10.1016/J.renene.2009.05.004.

[10] M.M.K. Kandasan, M. Thangavel, R. Ganern, Operational characteristics of diesel engine run by ester of sunflower oil and compare with diesel fuel operation, Journal of Sustainable Development 2 (2) (2009).
[13] P.K. Sahoo, L.M. Das, Combustion analysis of Jatropha, Karanja and Polanga based bio-diesel as fuel in a diesel engine, Fuel 88 (2009) 994–999.
[16] S.H.M. El-Bahnasy, M.M. El-Kotb, Shock Tube Study for The Measurement of Ignition Delay Period of 1,2-Epoxy Propane and *n*-Hexane. In: 6th International Conference in Fuel Atomization and Spray Systems, ICLASS-94, Ruene, France, 1994.
[18] J.B. Heywood, Internal Combustion Engine Fundamentals, Mc-Graw Hill Book Co., 1988.
[20] Z. Utlu, M.S. Koçak, The effect of biodiesel fuel obtained from waste frying oil on direct injection diesel engine performance and exhaust emissions, Renew Energy 33 (2008) 1936–1941.
[21] C. Kaplan, R. Arslan, A. Sürmen, Performance characteristics of sunflower methyl esters as biodiesel, Energy Sources Part A 28 (2006) 751–755.
[22] M. C- etinkaya, Y. Ulusay, Y. Tekì n, F. Karaosmanoglu, Engine and winter road test performances of used cooking oil originated biodiesel, Energy Conversion and Management 46 (2005) 1279–1291.
[23] C. Carraretto, A. Macor, A. Mirandola, A. Stoppato, S. Tonon, Biodiesel as alternative fuel: experimental analysis and energetic evaluations, Energy 29 (2004) 2195–2211.
[24] F.N. Silva, A.S. Prata, J.R. Teixeira, Technical feasibility assessment of oleic sunflower methyl ester utilization in diesel bus engines, Energy Conversion and Management 44 (2003) 2857–2878.
[25] Y.C. Lin, W.J. Lee, T.S. Wu, C.T. Wang, Comparison of PAH and regulated harmful matter emissions from biodiesel blends and paraffinic fuel blends on engine accumulated mileage test, Fuel 85 (2006) 2516–2523.
[26] M.J. Haas, K.M. Scott, T.L. Alleman, R.L. McCormick, Engine performance o f biodiesel fuel prepared from soybean soapstock: a high quality renewable fuel produced from a waste feedstock, Energy Fuels 15 (2001) 1207–1212.
[27] S. Murillo, J.L. Mi´ guez, J. Porteiro, E. Granada, J.C. Mora´ n, Performance and exhaust emissions in the use of biodiesel in outboard diesel engines, Fuel 86 (2007) 1765–1771.
[28] Handbook of biodiesel: emissions reductions with biodiesel. <http://www.cytoculture.com/Biodiesel%20Handbook.htmS>, 1999.
[29] Assessment and Standards Division (Office of Transportation and Air Quality of the US Environmental Protection Agency). A comprehensive analysis of biodiesel impacts on exhaust emissions, EPA420-P-02-001, 2002.
[30] K.F. Hansen, M.G. Jensen, Chemical and biological characteristics of exhaust emissions from a D I diesel engine fuelled with rapeseed oil methyl ester (RME), SAE paper, 1997, 971689.
[31] A.C. Pinto, L.L.N. Guarieiro, J.C. Rezende, N.M. Ribeiro, E.A. Torres, E.A. Lopes, et al, Biodiesel: an overview, Journal of the Brazilian Chemical Society 16 (6B) (2005) 1313–1330.
[32] X. Shi, Y. Yu, H. He, S. Shuai, J. Wang, R. Li, Emission characteristics using methyl soyate-ethanol-diesel fuel blends on a diesel engine, Fuel 84 (2005) 1543–1549.
[33] A. Serdari, K. Fragioudakis, C. Teas, F. Zannikos, S. Stournas, E. Lois, Effect of biodiesel addition to diesel fuel on engine performance and emissions, Journal of Propulsion and Power 15 (2) (1999) 224–231.
[34] K. Hamasaki, E. Kinoshita, H. Tajima, K. Takasaki, D. Morita, Combustion characteristics of diesel engines with waste vegetable oil methyl ester, in: The 5th international symposium on diagnostics and modeling of combustion in internal combustion engines (COMODIA 2001), 2001.
[35] A. Senatore, M. Cardone, V. Rocco, M.V. Prati, A comparative analysis of combustion process in DI diesel engine freed with biodiesel and diesel fuel, SAE Paper, 2001, 39–47 20 01-01-0 691.
[36] P.K. Sahoo, S.N. Naik, L.M. Das, Studies on biodiesel production technology from Jatropha curcas and its performance in a CI engine, Journal of Agricultural Engineering Indian Society of Agricultural Engineering (ISAE) 42 (2) (2005) 18–24.

Further reading

[1] O.M. Nwafor, Combustion characteristics of dual fuel diesel engine using pilot injection ignition, Institute of Engineers (India) Journal-Mc 84 (2003) 22–25.
[2] S.H.M. El-Bahmasy, Effect of pilot injection diesel fuel on dual fuel engine characteristics, faculty of engineering mattaria, Engineering Research Journal 109 (2007), February.
[7] S. Murllo, J.L. Miaquez, J. Porteiro, E. Granda, T.C. Moran, Performance and exhaust emission in the use of bio-diesel in out board diesel engine, Fuel 86 (2007) 1765–1771.
[8] M. Lapuerta, O. Armas, J. Rodrighez, J. Rodriquez-Fernerdez, Effect of bio-diesel fuel on diesel engine emission, Progress in Energy and Combustion Science 34 (2008) 198–223.
[11] D.H. Qi, H. Chen, R.D. Mathews, Y.Z.H. Bian, Combustion and emission characteristics of ethanol-biodiesel-water micro-emulsion used in a direct injection compression ignition engine, Fuel xxx (2009), http://dx.doi.org/10.101/j.fuel.2009.06.029.
[12] S.A. Basha, K.R. Gopal, S. Jebaraj, A review on bio-diesel production, combustion, emission and performance, Renewable and Sustainable Energy Reviews 13 (2009) 1628–1634.
[14] M. El-Kasaby, S.H.M. El-Bahnasy, A. Moussa, W. Abdel Khafar, M. Nemat Allah, Methyl ester bio-fuel characteristics as a substitute renewable I.C.E. fuel: 1-fuel preparation and performance tests of waste vegetable oil as bio-diesel, Alexandria Engineering Journal 47 (6) (2008).
[15] D.N. Assanis, Z.S. Filip, S.B. Fiveland, M. Syrimis, A Predictive Ignition Delay Correlation Steady State and Transient Operation of A Direct Injection Diesel Engine, Private, 1999 (file name: rod 0533.doc).
[17] H.O. Hardenberg, F.W. Hase, An Empirical Formula for Computing the Pressure Rise Delay of a Fuel from its Cetane Number and from the Relevant Parameters of DI Diesel Engine, SAE Paper 790493, SAE Trans vol. 88, 1979.
[19] K. Gelhard, C.M. Andrew, W.R-III. Thomas, Cetane number of branched and straight chain fatty ester determined in an ignition quality tester, Fuel 82 (2003) 971–975.
[37] C.W. Yu, S. Bari, A. Ameen, A comparison of combustion characteristics of waste cooking oil with diesel as fuel in a direct injection diesel engine, Proceedings of the Institution of Mechanical Engineers Part D: Journal of Automobile Engineering 216 (2002) 237–243.