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Experimental Investigation on Full Load Combustion Behavior of an Unmodified CI Engine Using Diesel and Biodiesel

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Abstract: *In-cylinder combustion behavior of diesel engines has been enigmatic on many accounts for researchers, but they are significant harbinger of various performance, durability and emission characteristics. With the advent of biodiesel as an alternative fuel for diesel engines, the interest of comparative combustion study of neat diesel (D100) and biodiesel (B100) operation is gaining attention. In this context, a comprehensive experimental trial was envisaged on an unmodified diesel engine to determine and compare several combustion characteristics when operating on neat diesel and neat biodiesel separately. Biodiesel was prepared from locally available "Calophyllum Inophyllum" vegetable oil using transesterification process. The results at full load indicated that in-cylinder pressure data for 51 consecutive cycles showed the cycle to cycle variability among 51 consecutive cycle to be 2.68 bar at 4 crank angle degrees after TDC for D100 operation and 7.47 bars at 3 crank angle degrees after TDC (Top dead center) for B100 operation. Average peak pressure was 68.82 bar at 13.5 crank angle degrees after TDC for D100 and 67.12 bar at 15 crank angle degrees after TDC for biodiesel. 45% Heat release occurred in the premixed phase and 55% in the diffusion phase for D100 operation, whereas for B100 41% heat release was in the premixed phase and 59% in the diffusion phase*

Keywords: *Biodiesel, combustion behavior, cylinder pressure, diesel engines, ignition delay etc.*

1. INTRODUCTION

Since the inception of renaissance in modern Europe, the internal combustion driven heat engines enjoy a near monopoly, as "lone decentralized prime mover", specifically useful in transportation, decentralized power generation and other industrial applications. In the modern times, rapid, robust and efficient mode of transportation became vital indicator of socio-economic prowess across civilizations, hence making these engines as a vital ingredient of growth and prosperity for the mankind. However, the flip side of the heat engines i.e. reliance on petroleum fuels and their baleful environmental consequences like global warming, climate change and fouling of breathable air etc. are starkly visible now, than ever before in the entire course of human history. Furthermore, the sinister shadow of gigantic petroleum fuel consumption is not restricted to environment alone, rather further spilled over wider geo-political and economic ramifications. No single factor affects worldwide stock-markets and financial health more than the volatility of international crude oil prices. Similarly, the conflict between the oil producing and consuming countries along with the underlying politics have long threatened global peace and security. In the light of the above, it is evident that further reliance on petroleum derived fuels to energize our day to day heat engines is not sustainable and there is an urgent need to reduce the burden on petroleum fuels to some other new, renewable and alternative mode of fuels. In this context, a large number of alternatives are investigated worldwide as potential substitutes of petroleum fuels. Amongst these alternatives, fuels derived from biological origin such as vegetable oil and animal fat, otherwise known as biofuels, are very promising as they are widely available and carbon neutral to a great extent. Besides, these fuels have specific utility in diesel engines, known to consume highest share of total petroleum. Biofuels derived from vegetable oil and animal fat possess higher viscosity and density with low heating values. These properties create operational and durability problems in diesel engines such as poor spray and atomization, injector coking, deposit formation etc. Despite these odds, many researchers favor usage of vegetable oils for diesel engine applications meant for rural electrification and decentralized power generation in places blessed with vegetable oil availability. Some of the recent interesting work on vegetable oils application on heat engines was carried out by Mishra and Murthy[1], No[2], Balat[3], Xue[4], Ramadhas[5], Singh et.al[6], Mishra et.al[7] etc. which are reviewed extensively. Apart from neat application, vegetable oils are envisaged to be used along with some additives or diluting agents such as alcohols. In this context the recent works by Pali et.al [8], Mishra et.al [9], Vibhaanshu et.al [10] are noteworthy. However both the neat vegetable form and use of additives are not very popular for large scale diesel engine application. Arguably the fatty acid methyl ester of vegetable oils, otherwise known as biodiesel are the most acceptable application of renewable alternative fuels both in the arena of research and practice. A large number of documented work is available regarding biodiesel application in both unmodified and modified diesel engines.

Only a fraction of these research papers are associated with in-cylinder combustion studies of unmodified diesel engines fuelled with biodiesel, which is of primary significance in Indian context to realize the mitigation targets of carbon emission and energy self-reliance. A review of recent research literatures dealing with the aforementioned topic is discussed below.

Qi et.al[11] investigated the effects of methanol as an additive to biodiesel–diesel blends. The results indicated that the combustion starts later for BDM5 and BDM10 than for BD50 at low engine load, but is almost identical at high engine load. At low engine load and 1500 rpm, BDM5 and BDM10 show the similar peak cylinder pressure and peak of pressure rise rate to BD50, and higher peak of heat release rate than that of BD50. The power and torque output of BDM5 and BDM10 are slightly lower than those of BD50. BDM5 and BDM10 show dramatic reduction of smoke emissions. NO_x and HC emissions are almost similar to those of BD50 at speed characteristics of full engine load. Kannanet.al [12] investigated the use of Ferric-Chlorideas afuel borne catalyst (FBC)for waste cooking palm oil based biodiesel. The engine trial results revealed that the FBC added biodiesel showed lower NO_x emissions and slightly higher CO_2 emissions compared to diesel. CO, HC and smoke emissions of FBC added biodiesel decreased by 52.6%, 26.6% and 6.9% respectively compared to biodiesel without FBC. Higher cylinder pressure, HRR and shorter ignition delay were observed with FBC added biodiesel.Ganapathyet.al [13] studied the effect of injection timing along with engine operating parameters in Jatropha biodiesel engine.

It was observed that advance in injection timing from factory settings caused reduction in BSFC, CO, HC and smoke levels and increase in BTE, P_{max}, HRR_{max} and NO_x emission with Jatropha biodiesel operation. The best injection timing for Jatropha biodiesel operation with minimum BSFC, CO, HC and smoke and with maximum BTE, P_{max}, HRR_{max} was found to be 340 CAD.Yasin et.al. [14] conducted an experiment to evaluate the performance and emissions of a small proportion of methanol (5% by volume) in a B20 blend and mineral diesel separately. The first condition was an increase in engine speed from 1500 rpm to 3500 rpm at partial engine load and the second condition involved maintaining a constant speed of 2500 rpm at three different engine loads (0.05 MPa, 0.4 MPa and 0.7 MPa). Vedharajet.al. [15] studied the use of additive 1,4 Dioxane along with Kapok biodiesel and diesel. The additive improved the cold flow, ignition and other thermo-physical properties of the biodiesel. The engine trial results indicated 5.7% improvement for B25-10mL over B25 at full load. The emissions of HC, CO, NOx and smoke were reduced by 25.3%, 22.5%, 15.2% and 24.6% for additive containing biodiesel than neat form. Gogoi and Baruah[16] conducted an experimental investigation on a small direct injection diesel engine with Koroch biodiesel. The BTE and BSFC of the engine operating with biodiesel was inferior to diesel, but start of combustion and ignition delay of the biodiesel fuels were better. Basha and Anand [17] ferreted the effect of carbon nanotubes (CNT) with Jatropha Methyl Ester (JME). The engine trial result suggested improved BTE for CNT added JME. Furthermore, the combined effect of micro explosion and secondary atomization phenomena associated with the CNT blended JME emulsion fuels led to reduction of NO_x and smoke in the tail pipe. Colaçoet.al [18] studied the transient heat conduction in a piston of a diesel engine, subjected to a periodic boundary condition on the surface in contact with the combustion gases.

The heat transfer coefficient at the top surface was modeled taking into account the temperature and pressure inside the combustion chamber. Such instantaneous pressure was measured using a special probe for an engine operating with several blends of diesel and biodiesel, and the temperature was obtained through First Law analysis. The time wise variations of the temperature of several points in the piston were examined for different piston materials and various load conditions.Qi et.al [19] conducted an experimental investigation to evaluate the effects of using diethyl ether and ethanol as additives to biodiesel diesel blends. The results indicated that, compared with B30, there was slightly lower BSFC for BE-1. Drastic reduction in smoke was observed with BE-1 and BE-2 at higher engine loads.Agarwal and Dhar [20] investigated Karanja oil blends with mineral diesel in unheated condition for CI engine application. Detailed combustion study indicated that the combustion duration increased significantly even with smaller concentration of Karanja oil in the fuel blend. HC, CO and smoke emissions were found to decrease by 20-50% (v/v) of Karanja oil content in the fuel blends.

In the light of the review of existing literature, it is evident that a good number of research has already been carried out on biodiesel application (with and without additives) in diesel engines. However the comparative in-cylinder combustion studies of un-modified diesel engines running on neat diesel and neat biodiesel has not been ferreted much. Furthermore biodiesel derived from non-edible vegetable oils is another unexplored area as far as combustion studies is concerned. Therefore an experimental investigation is conceived by the researchers to investigate a full spectrum in-cylinder combustion behavior of an unmodified agricultural diesel engine fuelled with neat diesel and neat biodiesel derived from a high free fatty acid containing non-edible vegetable oil known as Calophyllum Inophyllum.

2. MATERIALS AND METHODOLOGY

2.1 PREPARATION OF TEST FUEL

Calophyllum vegetable oil was converted to biodiesel using standard transesterification process. Due to high free fatty acid content of the oil, an esterification cum transesterification process was carried out. Various process parameters like reaction temperature, reaction time, catalyst concentration etc. were maintained as suggested by Ong et.al.[21]. It was observed that for the esterification stage, FFA of 1.9% was achieved for 1.5% catalyst (PTSA) concentration, 65°C reaction temperature and 60 minutes reaction time. Similarly, in the transesterification stage, 0.88% concentration of catalyst (KOH), 63°C reaction temperature and 90 minutes of reaction time produced yield of 96.48% at constant methanol to oil molar ratio of 1:6. Neat biodiesel and neat diesel fuel samples are shown in Fig. 1.

Fig. 1. Neat diesel-D100 sample (left) and neat biodiesel-B100 sample (right)

A comparative physico-chemical and fuel characteristics of neat diesel and neat biodiesel was carried out in the lab and the same was compared with the findings of Ong et.al. [21] and mentioned in the table

TABLE 1: Fuel physicochemical properties

Property	Unit	Test method	Limit	D100	B100
Viscosity @40°C	cSt	D445	1.9-4.1	2.95	4.6
Density @15°C	kg/m^3	D6890	858	840.0	850
Flash point	°C	D93	52	70.5	>200
Cloud point	°C	D2500	-	-2	3
Oxidation stability	hours	EN14112	6	26.5	3.96
Cetane number	-	D6890	Min 47	51	-
Acid value	mg KOH/g	D664	0.3	49.7	-
Heating value	MJ/kg	EN14214	35	45.825	41.7
Carbon residue	wt.%	D524	0.35	0.821	-
Ash content	wt.%	D2709	0.05	0.08	-
Copper-strip corrosion	-	EN2160	1	1	-

2.2 METHODOLOGY FOR HEAT RELEASE RATE, MASS FRACTION BURNT AND IGNITION DELAY DETERMINATION

The heat release rate was determined using the ideal gas equation (eq. 1) and first law of thermodynamics (eq. 2) as mentioned below:

$$PV = mRT \qquad (1)$$

$$dE = \delta Q - pdv + \sum_j h_j \, dm_j \qquad (2)$$

The primary heat release equation using the above governing equation is elucidated in equation3. The same derivation is referred form Heywood. [22]

$$\frac{dQ_{chem}}{d\theta} = \frac{\gamma}{\gamma-1} p \frac{dv}{d\theta} + \frac{1}{\gamma-1}\left(v\frac{dp}{d\theta} - \frac{pv}{m}\frac{dm}{d\theta}\right) - \frac{dQ_w}{d\theta} - \sum_j h_j \frac{dm_j}{d\theta} \qquad (3)$$

The wall heat transfer was measured using Woschni correlation [23], mentioned in equation4 and 5.

$$\frac{dQ_w}{d\theta} = \frac{\alpha A}{6N}(T_{cyl} - T_{wall}) \qquad (4)$$

$$\alpha = 130D^{-0.2}T_{cyl}^{-0.53}P_{cyl}^{0.8}\left[C_1 + U_m + C_2\frac{V_{stroke}T_1}{P_1V_1}(P_z - P_0)\right]^{0.8} \qquad (5)$$

Mass fraction burned (MFB) in each individual engine cycle is a normalized quantity with a scale of 0 to 1, describing the process of chemical energy release as a function of crank angle. The determination of MBF is commonly based on burn rate analysis – a procedure developed by Rassweiler and Withrow [22].

$$MFB = \frac{m(i)}{m(total)} = \frac{\sum_0^i \Delta P_c}{\sum_0^N \Delta P_c} \qquad (6)$$

The ignition delay was calculated by Hardenberg and Hasemethod [23] mentioned in equation 7.

$$\theta_{\mathrm{delay}} = (0.36 + 0.22U_m)\exp^{\left[\frac{618,840}{CN+25}\left(\frac{1}{RT}-\frac{1}{17,190}\right)+\left(\frac{21.2}{P-12.4}\right)^{0.63}\right]} \tag{7}$$

2.3 DEVELOPMET OF ENGINE TEST RIG.

A single cylinder, four stroke, vertical, light duty, water cooled, diesel engine of Kirloskar make was chosen for the present engine trials. Such types of engines are generally used for agricultural activities or decentralized power generation purposes in India. The detailed specification of the engine is provided in table 2.

TABLE 2: Test engine specification

Make	Kirloskar
No. of cylinder	1
Strokes	4
Rated Power	3.5 kW@1500rpm
Cylinder diameter	87.5mm
Stroke length	110mm
Connecting rod length	234mm
Compression ratio	17.5:1
Orifice diameter	20mm
Dynamometer arm length	185mm
Inlet Valve Opening	4.5°BTDC
Inlet Valve Closing	35.5°ABDC
Exhaust Valve Opening	35.5°BBDC
Exhaust Valve Closing	4.5°ATDC
Fuel injection timing	23^0BTDC

For the provision of loading, a water cooled eddy current dynamometer of 7.5kW rating was coupled with the engine shaft. A high precision strain gauge type load cell was attached to the dynamometer to accurately transmit the engine loading. Two sets of fuel tanks were provided for the engine set up. One tank was used for diesel and the other tank was meant for biodiesel. The cyclic variation of combustion pressure and the corresponding crank angle was recorded using a "Kubeler" piezoelectric transducer, with a low noise cable, mounted into the engine head. The pressure transducer contained a piezoelectric sensor and charge amplifier. The maximum resolution of the pressure sensor was 0.25°CA. The data were fed to a centralized data acquisition system NI USB-6210, 32-bit. A personal computer with a software package "Enginesoft" was connected to the data acquisition system for online and subsequent offline analysis. The layout of the final experimental test setup is shown in Fig. 2.

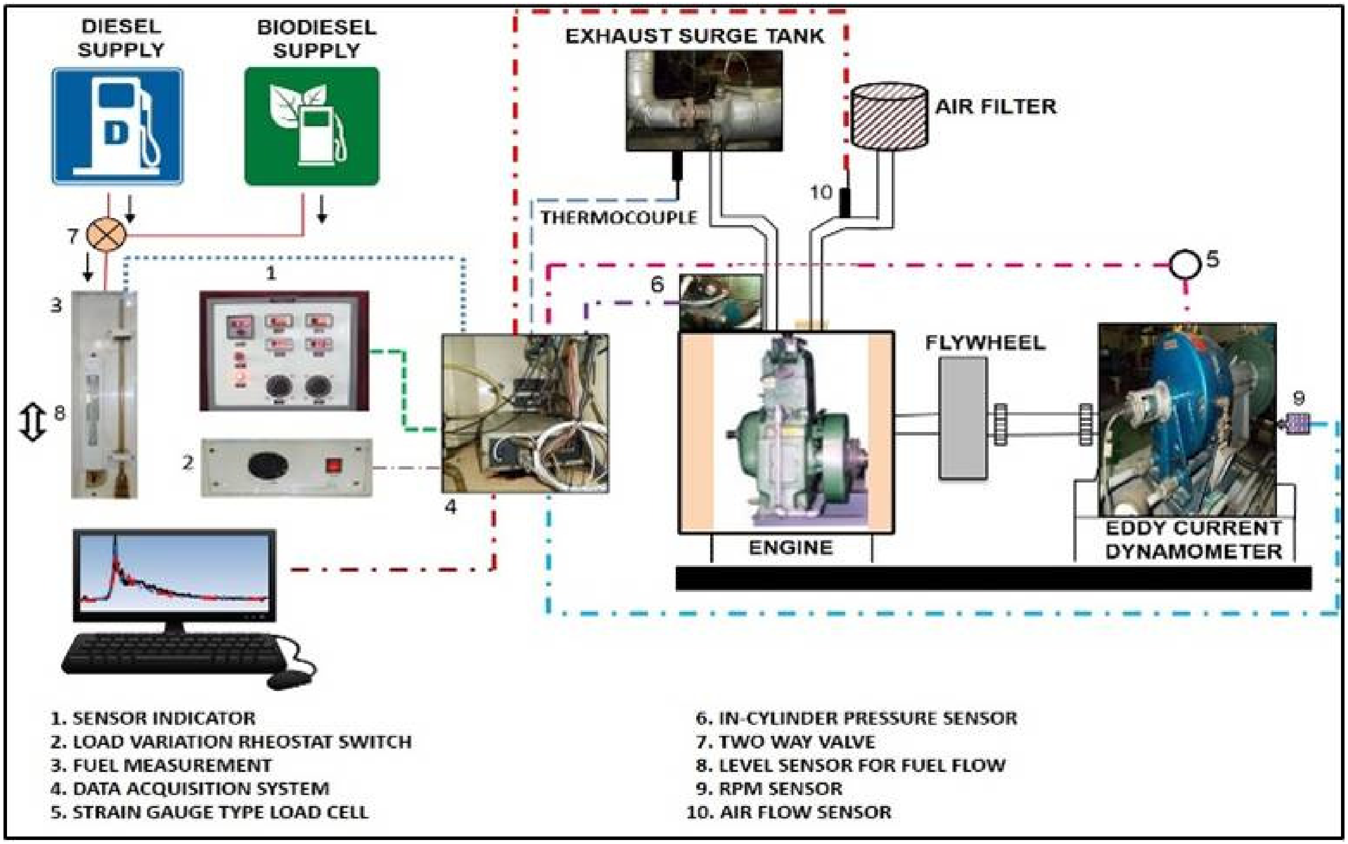

Fig. 2. Engine test rig layout

3. RESULTS AND DISCUSSION

3.1 CYCLIC VARIABILITY STUDY

As discussed earlier, the trial was conducted on a constant speed agriculture diesel engine that usually operates at full load. The pressure-crank angle history at full load for 51 consecutive cycles was plotted in Fig.3 for neat diesel operation and in Fig.4 for neat Calophyllum biodiesel operation. Homogeneity of these cycles indicate a smoother engine operation, whereas large variations is a harbinger of rough engine operation, power loss and poor load response. For a better statistical outlook, the comparative standard deviation at each crank angle for 51 consecutive cycles between diesel and biodiesel operating conditions are plotted in Fig.5.

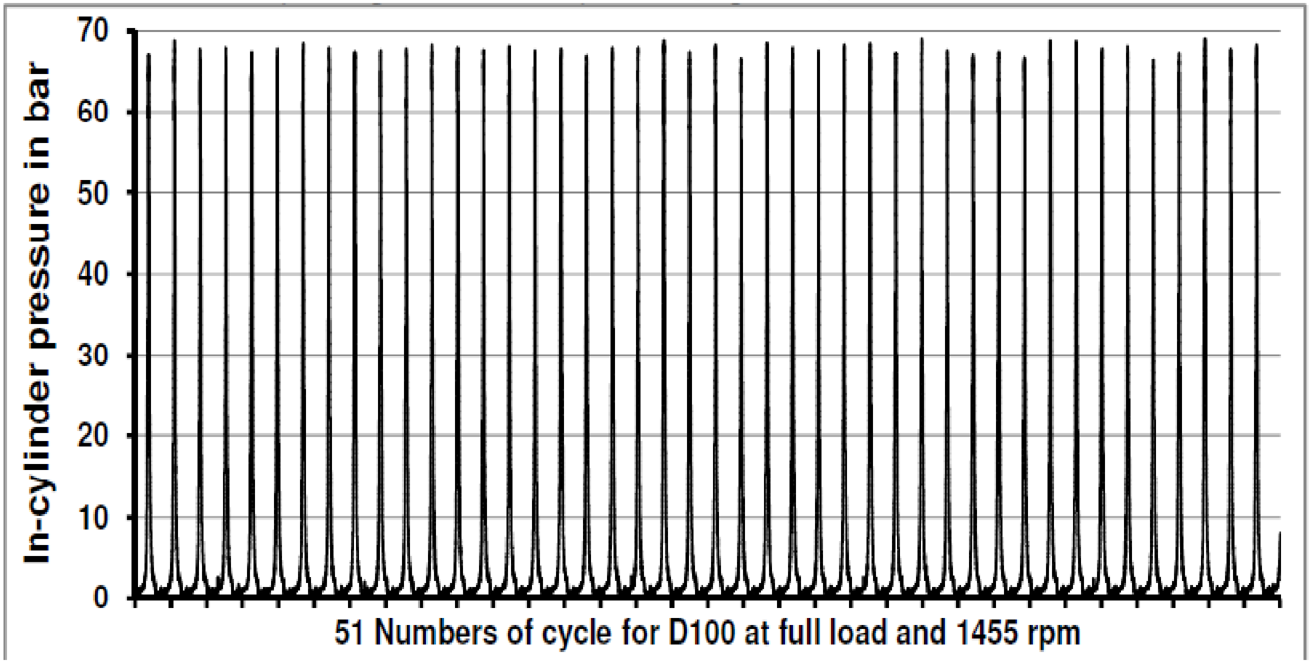

Fig. 3. 51 Consecutive cycles at full load for neat diesel operation

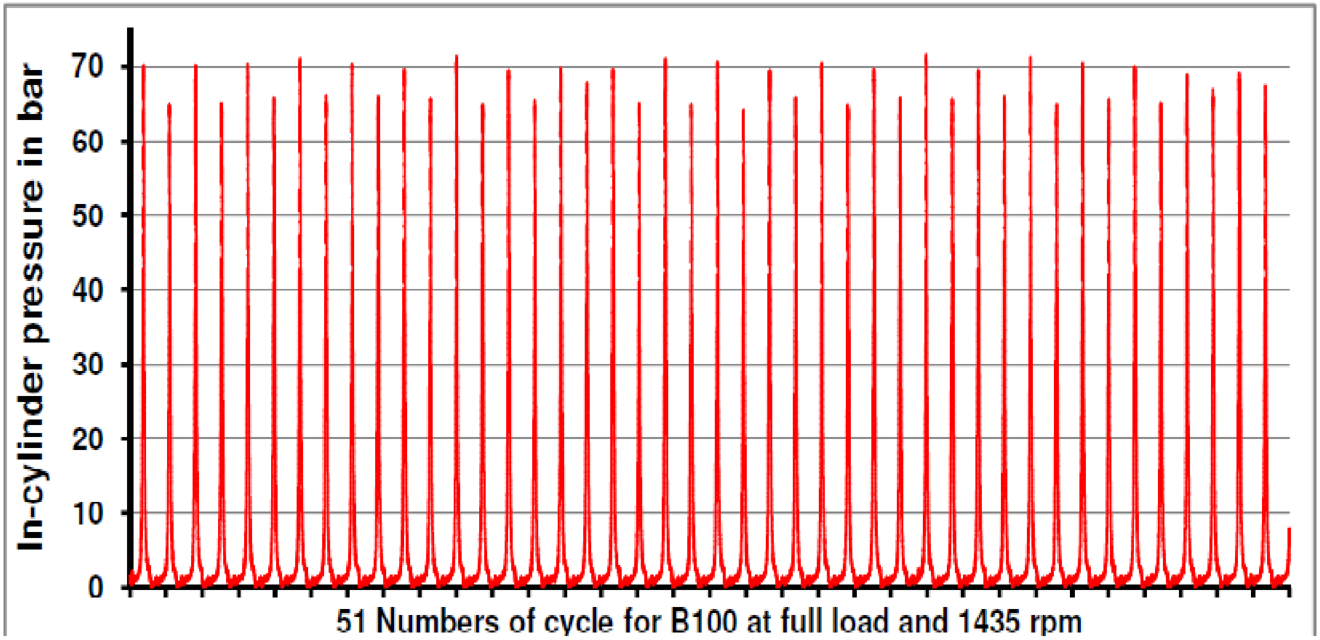

Fig. 4. 51 Consecutive cycles at full load for neat biodiesel operation

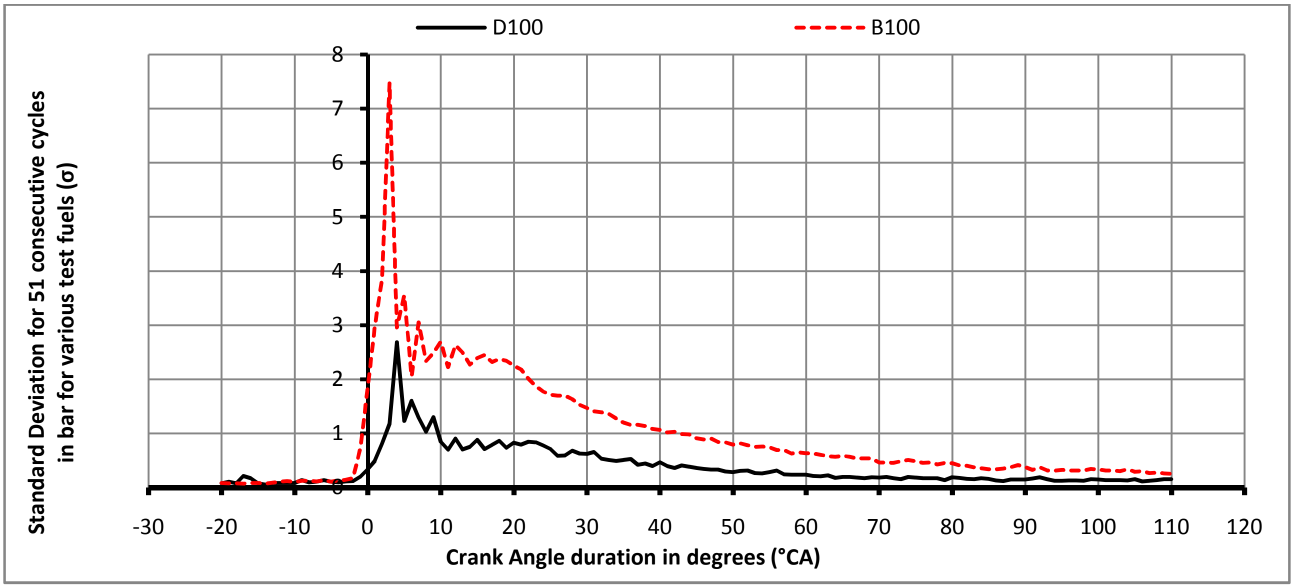

Fig. 5. Standard deviation at each crank angle among 51 cycles for diesel and biodiesel

Fig.3 indicates that the spread across 51 cycles is less than 0.5 bar for the entire duration of engine cycle operation except the period of combustion. A rapid rise in the spread for 51 cycles was observed for all the test fuels in the probable premixed phase of combustion indicating high degree of turbulence. The standard deviation curve (Fig.5) amongst 51 cycles for neat diesel indicated a maximum value of 2.68 bar at 4 crank angle degrees after TDC. The same value for biodiesel operation stood at 7.47 bars at 3 crank angle degrees after TDC. In the probable diffusion phase, there is a reduction in the spread for all the test fuels.It may also be observed that Calophyllum biodiesel operation resulted in a prolific spread of 7.472 bar during the initial phases of combustion, whereas the same for neat diesel operation stood at 2.682 bar. This large spread of biodiesel operating condition indicates poor load response, loss in power and roughness. The same may be attributed towards higher viscosity, density and lower vapor pressure of biodiesel compared to neat diesel. The high viscosity and density impinge spray properties, whereas lower vapor pressure increases the cavitation number which in turn affects the break up and atomization.

3.2 IN-CYLINDER COMBUSTION STUDY

The average of 51 consecutive cycles was chosen for the final pressure crank angle history curve (Fig.6) that was used to predict heat release, mass fraction burnt calculation etc.

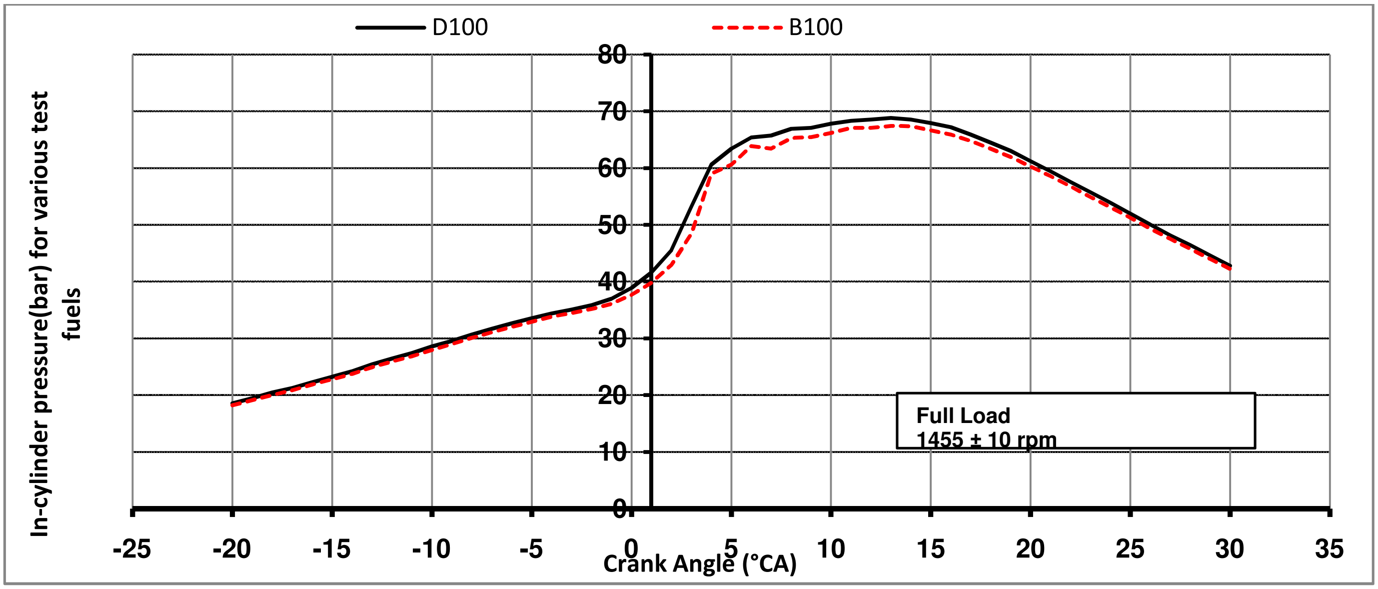

Fig. 6. Pressure crank angle history (Average of 51 cycles) for diesel and biodiesel

Analysis of Fig.6 indicatedan average peak pressure of 68.82 bar at 13.5 crank angle degrees after TDC for D100 operation. The same for B100 was found to be 67.12 bar at 15 crank angle degrees after TDC. On average basis this difference was not significant, but with large cyclic variability, it may be easily inferred that D100 operations resulted in large number of lower peak pressure and lower mean effective pressure cycles referring drop of power and poor load response. Heat release rate curve was plotted using equation (3) and shown in Fig. 7 and mass fraction burnt curve is shown in Fig. 8.

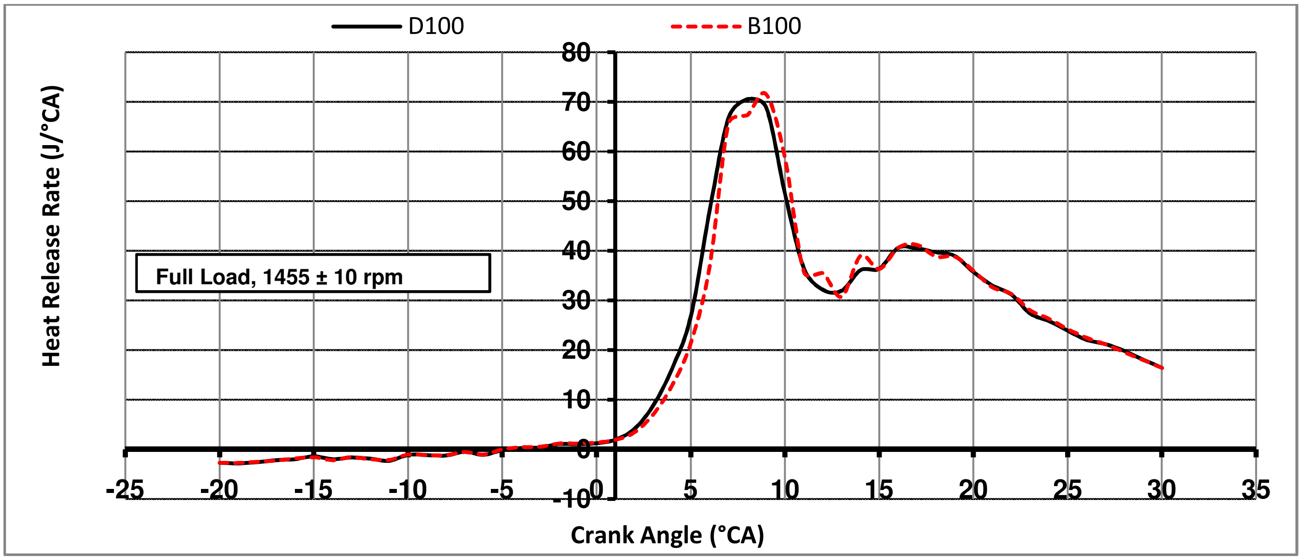

Fig. 7. Heat release rate plot for diesel and biodiesel

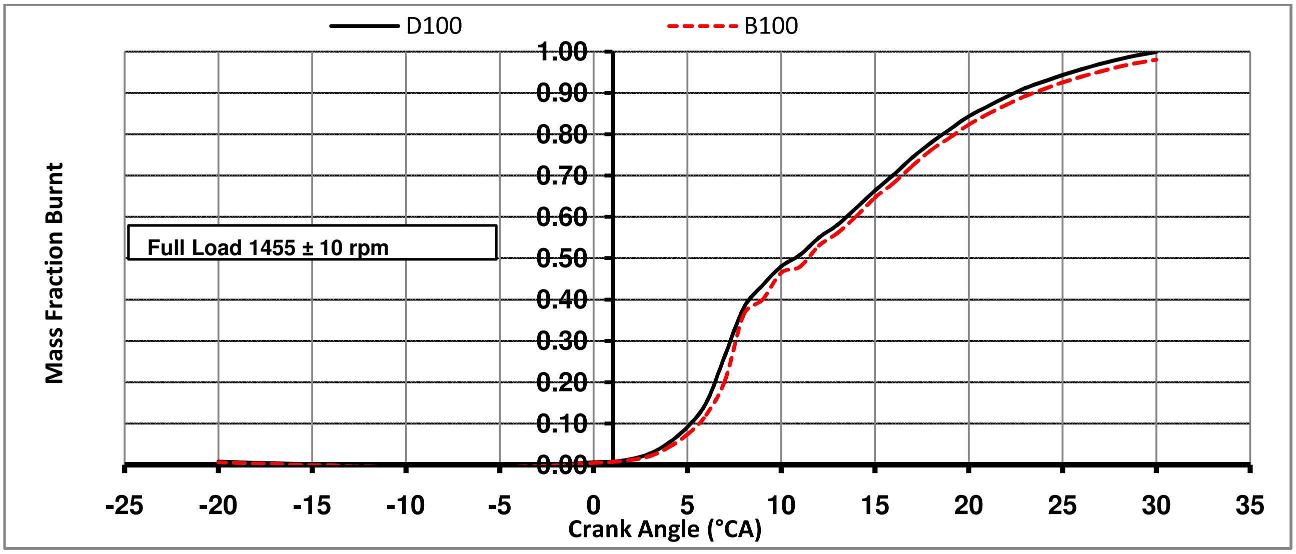

Fig. 8. Mass fraction burnt curve for diesel and biodiesel

Fig.7 showed that peak HRR was 70.5 J/°CA for neat diesel and 71.51 J/°CA for biodiesel. The HRR curve for B100 was slightly shifted rightwards indicating higher heat release in the probable diffusion phase and larger ignition delay. The MFB results indicated 32.1 crank angle degrees of combustion duration with 7.75 crank angle degrees of ignition delay for neat diesel operation and 29.5 crank angle degrees of combustion duration with 8.5 crank angle degrees of ignition delay for neat diesel operation. An overall phase wise heat release estimate suggested 45% heat release in the premixed phase and 55% in the diffusion phase for D100 operation, whereas for B100 41% heat release was in the premixed phase and 59% in the diffusion phase indicating comparatively less pressure rise rate of biodiesel operation. The phase wise heat release is mentioned in Fig. 9

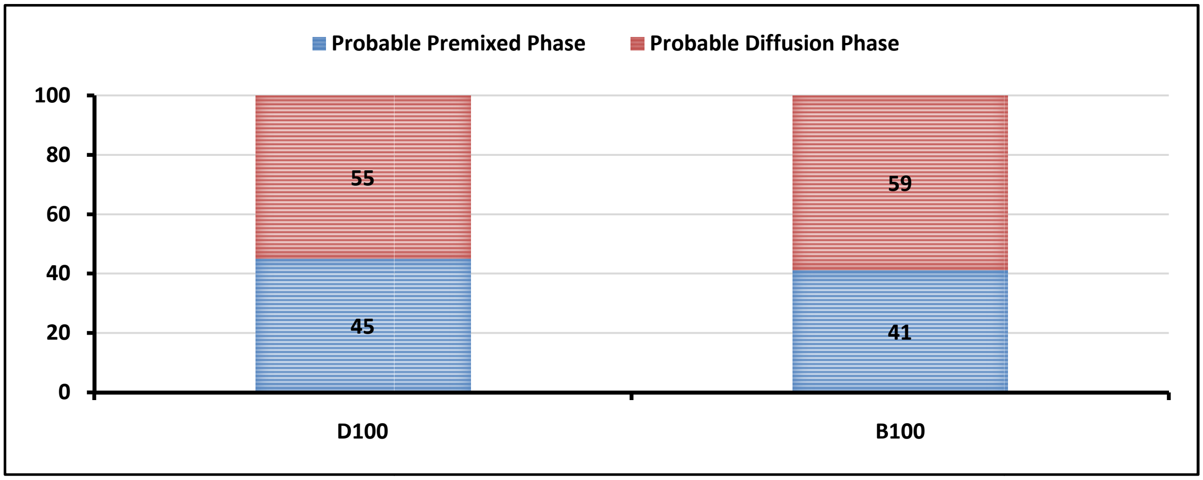

Fig. 9. Phase wise heat release for diesel and biodiesel

4. CONCLUSIONS

The present study was aimed at evaluating and comparing the combustion characteristics of an agriculture diesel engine fuelled with neat diesel and neat biodiesel. The results of the study are mentioned below.

1. At full load the cycle to cycle variability among 51 consecutive cycle showed a maximum spread of 2.68 bar at 4 crank angle degrees after TDC for D100 operation and 7.47 bars at 3 crank angle degrees after TDC for B100 operation.
2. Average peak pressure of 68.82 bar at 13.5 crank angle degrees after TDC was observed for D100 operation. The same for B100 was found to be 67.12 bar at 15 crank angle degrees after TDC.
3. Peak HRR was 70.5 J/°CA for neat diesel and 71.51 J/°CA for biodiesel. HRR curve was found to shift slightly rightwards for B100 operation.
4. Mass fraction burnt and HRR analysis showed 32.25 crank angle degrees of combustion duration with 7.75 crank angle degrees of ignition delay for neat diesel operation and 29.5 crank angle degrees of combustion duration with 8.5 crank angle degrees of ignition delay for neat diesel operation.
5. 45% Heat release occurred in the premixed phase and 55% in the diffusion phase for D100 operation, whereas for B100 41% heat release was in the premixed phase and 59% in the diffusion phase.

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