COMBUSTION CHARACTERISTICS OF DIESEL ENGINE OPERATING ON JATROPHA OIL METHYL ESTER

by

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Fuel crisis because of dramatic increase in vehicular population and environmental concerns have renewed interest of scientific community to look for alternative fuels of bio-origin such as vegetable oils. Vegetable oils can be produced from forests, vegetable oil crops, and oil bearing biomass materials. Non-edible vegetable oils such as jatropha oil, linseed oil, mahua oil, rice bran oil, karanji oil, etc., are potentially effective diesel substitute. Vegetable oils have reasonable energy content. Biodiesel can be used in its pure form or can be blended with diesel to form different blends. It can be used in diesel engines with very little or no engine modifications. This is because it has combustion characteristics similar to petroleum diesel. The current paper reports a study carried out to investigate the combustion, performance and emission characteristics of jatropha oil methyl ester and its blend B20 (80% petroleum diesel and 20% jatropha oil methyl ester) and diesel fuel on a single-cylinder, four-stroke, direct injections, water cooled diesel engine. This study gives the comparative measures of brake thermal efficiency, brake specific energy consumption, smoke opacity, HC, NO_x, ignition delay, cylinder peak pressure, and peak heat release rates. The engine performance in terms of higher thermal efficiency and lower emissions of blend B20 fuel operation was observed and compared with jatropha oil methyl ester and petroleum diesel fuel for injection timing of 20° bTDC, 23° bTDC and 26° bTDC at injection opening pressure of 220 bar.

Key words: *diesel engine, biodiesel, performance, emissions, combustion, jatropha oil methyl ester*

Introduction

Diesel engine will be the major power source for automobiles in the twenty-first century. To reduce emissions and solve the energy crisis, designing diesel engines with low emission and less energy consumption has always be an objective for researchers across the globe.

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However, with the development of new technologies, today's diesel engines have better emission characteristics and the less energy consumption compared with its predecessor. But, there is still lot to do on diesel engines aimed to achieve our goal of clean and effective diesel engine. Accordingly, research on a clean burning fuel instead of conventional fuel is advisable, which could not only decrease exhaust gas to a great extent, but, also provide more options of energy sources. The alkyl monoester of fatty acids as bio-diesel which was obtained from renewable oil and fats materials by transesterification reaction is a good alternative. Biodiesel can be obtained from raw vegetable oil by transesterification with methanol or ethanol after chemical reactions [1]. Vegetable oils present a very promising alternative to diesel oil since they are renewable and have similar properties as of diesel. Many researchers have studied the use of vegetable oils in diesel engines. Vegetable oils offer almost the same power output with slightly lower thermal efficiency when used in diesel engine [2-8]. Reduction of engine emissions is a major research facet in engine development with the increasing concern on environmental protection and the stringent exhaust gas regulation. Vegetable oils are a mixture of organic compounds ranging from simple straight chain compared to complex structure of proteins and fat-soluble vitamins. They are usually triglycerides generally with a number of branched chains of different lengths. Research in the direction of vegetable oils as compression ignition (CI) engine fuel has yielded encouraging results [9-14]. The honge, jatropha and sesame oils are extracted from their seeds. The use of neat vegetable oils poses some operational problems when subjected to prolonged usage in CI engines. These problems are attributed to high viscosity, low volatility and polyunsaturated character of neat vegetable oils [4, 13]. Some of the common problems with vegetable oil run in diesel engines are coking and trumpet formation on the injector, carbon deposit, oil-ring sticking and thickening and gelling of lubricating oil as a result of contamination by the vegetable oils. Different methods such as preheating, blending, ultrasonically assisted methanol transesterification and supercritical methanol transesterification [15, 16] are being used to reduce the viscosity and make them suitable for engine applications.

In the present investigation, biodiesel was prepared from non-edible vegetable oil viz. jatropha oil. *Jatropha curcas* is a large plant and belongs to the family of *Euphorbiaceae* occurring almost throughout India. It has a long productive period of around 50 years, yielding handsome returns annually. Biodiesel derived from Jatropha curcas was tested for properties and its combustion, performance and emission characteristics were studied on a four-stroke, single-cylinder, direct-injection, and water cooled diesel engine to check their feasibility as diesel engine fuels.

Availability and economic value of oils

Jatropha curcas is a large shrub or tree commonly found throughout most of the tropical and subtropical regions of the world. The *Jatropha curcas* plant is a drought-resistant, perennial plant living up to 50 years and has the capability to grow on marginal soils. It requires very little irrigation and grows in all types of soils. The production of jatropha seeds is about 0.8 kg/m^2 per year. The oil content of jatropha seed ranges from 30-40% by weight and the kernel itself ranges from 45 to 60%. A fresh jatropha seeds are a slow drying, odorless and colorless oil, and turns yellow after aging.

The price of crude petroleum and the cost of transporting diesel through long distances to remote markets play a key role in evaluating the economical feasibility of biodiesels. The cost of producing methyl/ethyl esters from edible oils is at present much more expensive than hydrocarbon-based diesel fuel. The cost of biodiesels can be reduced if non-edible oils are used instead of edible oils [17]. The economic feasibility of different vegetable oils including edible and non-edible oils have been reported by earlier researchers [18].

Transestrification of jatropha oil

Widely used and accepted process to reduce the viscosity of triglycerides in vegetable oil is transesterification. In the transesterification of vegetable oils, a triglyceride reacts with an alcohol in the presence of a strong acid or base, producing a mixture of fatty acid alkyl esters and glycerol [2]. About 3-4 grams of catalyst (NaOH) was dissolved in 100 ml of methanol to prepare alkoxide, which is required to activate the alcohol. Around 15-20 minutes vigorous stirring was done in a closed container until the alkali was dissolved completely. The alcohol-catalyst mixture was then transferred to the reactor containing moisture free jatropha oil. A continuous stirring of the resulting mixture at temperature between 60-65 °C was carried out for one hour with water or air cooled condenser. The resulting mixture was taken out and poured into the separating funnel and the glycerol was separated from the mixture to get the methyl ester of jatropha oil. Water washing was done later in order to remove moisture and impurities from the biodiesel.

Test fuel

In this paper jatropha oil methyl ester (JOME) was used as bio-diesel and three types of fuel were used. A commercial in use petroleum diesel fuel, called B0 without any oxygenated additives was used as reference fuel and the base fuel for the preparation of biodiesel fuel blend. The second fuel called B20 (*i. e.* 80% petroleum diesel and 20% JOME) by volume. The third test fuel called B100 is pure biodiesel (JOME). The fuel characteristics of the biodiesel, B20 and diesel fuel were compared shown in tab. 1.

Table 1. Fuel characteristics of diesel, B20 biodiesel, and B100 biodiesel

Fuel	Diesel	JOME	B20
Density at 40 °C [kgm^{-3}]	840	880	848
Kinematic viscosity [cSt]	4.59	5.65	4.802
Flash point [°C]	54	170	155.5
Lower calorific value [$kJkg^{-1}$]	43958	38450	42856

Engine and test equipment

The performance test was conducted on a single cylinder, four-stroke, naturally aspirated, open chamber, direct injection, water-cooled, 5.2 kW output computerized diesel engine test-rig. The Kirloskar, India, make engine was directly coupled to an Eddy current dynamometer that permitted engine motoring either fully or partially. The schematic diagram of the experimental setup is as shown in fig. 1. The engine characteristics and other instrument details are

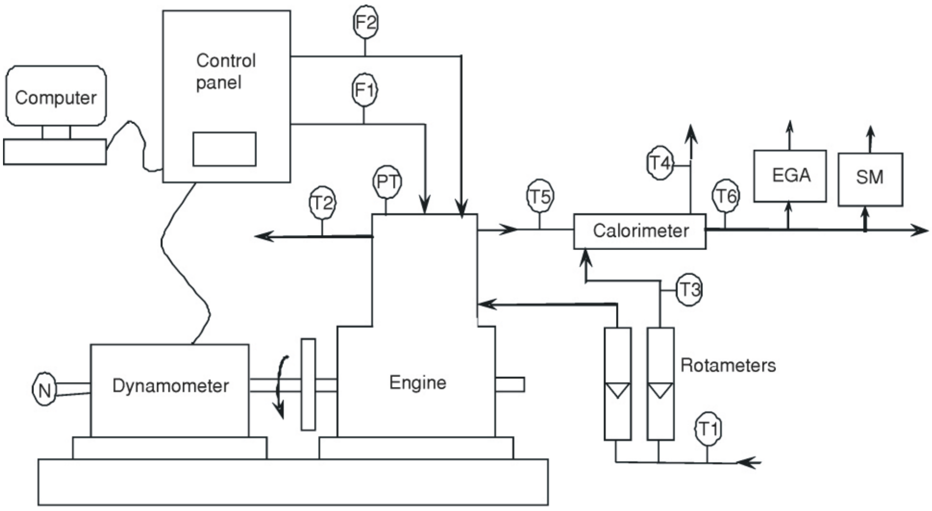

Figure 1. Schematic diagram of the experimental set-up
T1, T3 – Inlet water temperature, T2 – outlet engine jacket water temperature, T4 – outlet calorimeter water temperature, T5 – exhaust gas temperature before calorimeter, T6 – exhaust gas temperature after calorimeter, F1 – fuel flow DP (differential pressure) unit, F2 – air intake DP unit, PT – pressure transducer, N – RPM decoder, EGA – exhaust gas analyzer (5 gas), SM – smoke meter

Table 2. The specifications of the engine

Parameters	Specification
Type	TV1 (Kirlosker make)
Software used	Engine soft
Governor type	Mechanical centrifugal type
Number of cylinders	Single cylinder
Number of strokes	Four stroke
Fuel	H. S. diesel
Rated power	5.2 kW (7 HP) at 1500 rpm
Cylinder diameter (bore)	87.5 mm
Stroke length	110 mm
Compression ratio	17.5:1
Air measurement manometer	
Make	13 MX 201
Type	U-type
Range	100-0-100 mm
Eddy current dynamometer	
Model	AG-10
Type	Eddy current
Maximum	7.5 kW at 1500-3000 rpm

mentioned in tab. 2. The fuel is supplied to the test engine by an external tank of about 5 liter capacity, which could easily be drained with the help of three way stop valve for fuel change; a glass burette of 100 cm^3 was also attached in parallel to this tank and was used for fuel flow rate measurement. For each fuel change, the fuel line was purged out with test oil and engine was made to run under load for at least 30 minutes to stabilize on new fuel conditions. It was operated on diesel fuel, blend B20 fuel and JOME at injector opening pressure of 220 bar with three different injection timings (*i. e*. the start of injection) of 20°, 23°, and 26° bTDC. However based on the previous research analysis the injector opening pressures was optimized from 200 bar (designed injection pressure) to 220 bar for JOME and its blend B20. Test-rig was provided with necessary equipment and instruments for combustion pressure and crank-angle measurements. Provision was made for interfacing airflow, fuel flow, temperatures, and load measurement with computer. This set-up facilitates the study of engine performance for brake power (BP), indicated power, frictional power, brake mean effective pressure, indicated mean effective pressure, brake thermal efficiency (BTE), indicated thermal efficiency, mechanical efficiency, volumetric efficiency, specific fuel consumption, A/F ratio, and heat balance. The heat carried away by the exhaust gases can be measured with the help of water calorimeter to prepare the heat balance sheet. During the test, the engine exhaust was measured for the emissions like NO_x, CO, CO_2, and O_2. A calibrated MRU (Germany) Delta 5-Gas analyzer was used for the emission measurement. It consists of flexible probe with stainless steel nose. Once the calibration protocol for 150 s is completed, the probe is then introduced to the sample stream for emission measurement. MRU Delta 5 gas analyzer incorporates a microprocessor technology, which provides instantaneous emission readings with a good accuracy.

A PCB Piziotronics Inc, piezoelectric pressure transducer were installed in engine cylinder. The sensor was flush mounted and it measured the pressure trace in the cylinder with 10 crank angle resolution. The pressure crank-angle data was acquired on a window XP operating system personal computer containing an Appex Innovation Pvt. Ltd data acquisition board. The pressure crank angle data for JOME, its blend B20 and diesel fuel was recorded for all the engine loads tried. The ignition delays were determined from the digitized pressure signals, with the minimum resolution of 1° crank angle. The start of ignition was defined as the crank angle at which the rate of change of the cylinder pressure exceeded a standard pressure derivative curve for the compression process. This agreed well with the point of the rapid increase in the burning rate curve, since it is the deviation of the pressure from the compression process which is in reality used to calculate the combustion rate. In the present work an approach to determine the single zone heat release rate using the experimentally obtained pressure-crank angle data. A simple Microsoft Excel spread sheet program was developed using standard heat release rate equations, correlations, and constants. Heat transfer coefficient h [$Wm^{-2}K^{-1}$] was calculated by using the Nusselt-Reynold's relation by Woschni.

Results and discussions

Performance characteristics

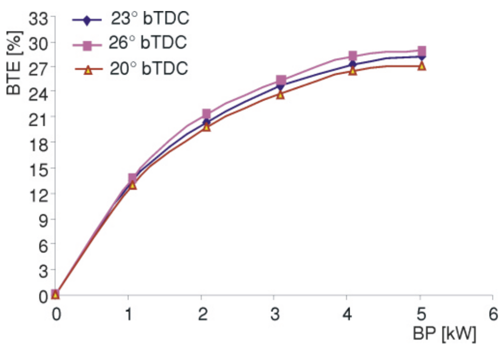

Figure 2. Variation of BTE with BP of diesel fuel for different injection timings

The variation of BTE with BP for different injection timings of diesel, B20 and B100 fuel at different load conditions are shown in figs. 2, 3, and 4, respectively. It is observed that BTE of retard injection timing (IT) is lower than the other IT. But, in advance IT, the BTE of diesel fuel, B20 fuel, and biodiesel was higher than the other two IT. This may be due increase in power produced at advanced IT and lower fuel consumption. The BTE of diesel fuel for injection timing of 20°, 23°, and 26° bTDC are 27.57, 28.26, and 28.55%, respectively, for full load conditions. The BTE of blend B20 fuel for in-

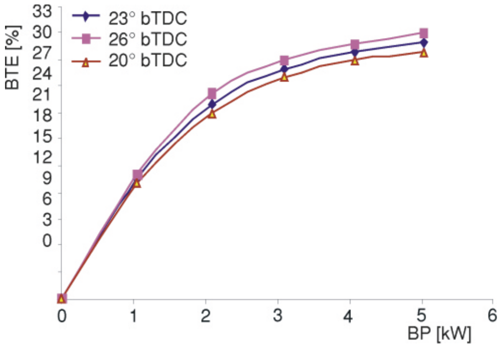

Figure 3. Variation of BTE with BP of B20 fuel for different IT

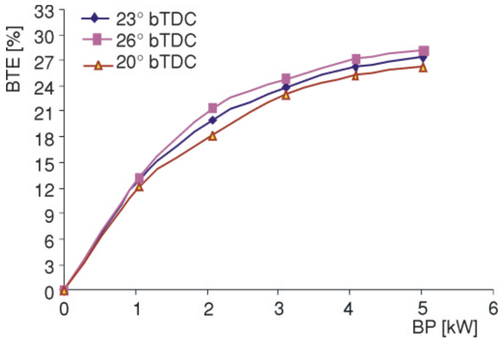

Figure 4. Variation of BTE with BP of B100 fuel for different IT

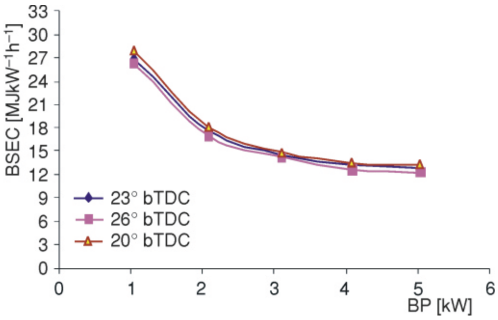

Figure 5. Variation of BSEC with BP of diesel fuel for different IT

jection timing of 20°, 23°, and 26° bTDC are 27.33, 29.61, and 30.10%, respectively, for full load conditions. The BTE of B100 fuel for injection timing of 20°, 23°, and 26° bTDC are 25.34, 28.02, and 28.13%, respectively, for full load conditions. The maximum BTE occurred at injection timing of 26° bTDC and blend B20 performed better. This is 3° more advanced than that of designed IT for diesel fuel operation. It is seen that BTE at advanced IT is better than at deigned IT. At this IT, the BTE of blend B20, diesel, and biodiesel is 30.10, 28.55 and 28.13%, respectively, for full load operation.

The variation of brake specific energy consumption (BSEC) with BP for different injection timings of diesel, B20 and B100 fuel at different load conditions are shown in figs. 5, 6, and 7, respectively. BSEC is higher in case of biodiesel compared to blend B20, and diesel fuel. This may be due to lower calorific value of biodiesel. But at advance injection timing, the blend B20 fuel BSEC is lower than B100 and diesel fuel. This may be due to complete combustion of fuel. Also observed that retarded injection timing, the BSEC increases compared to other injec-

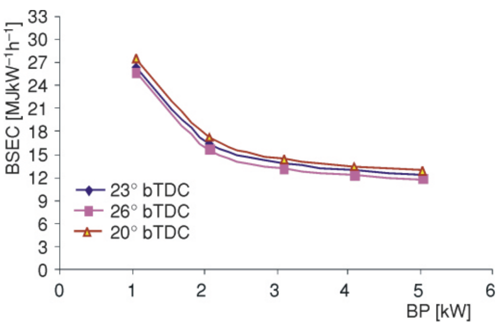

Figure 6. Variation of BSEC with BP of B20 fuel for different IT

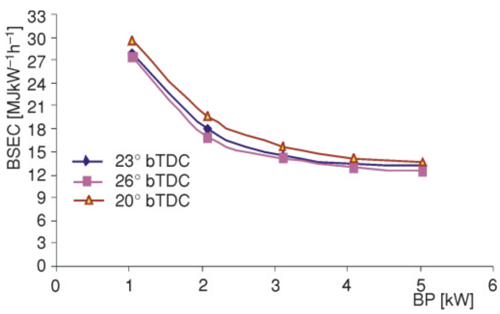

Figure 7. Variation of BSEC with *BP* of B100 fuel for different injection timings

tion timings. The BSEC of diesel fuel for injection timing of 20°, 23°, and 26° bTDC are 13.05, 12.74, and 12.60 MJ/kWh, respectively. The BSEC of blend B20 fuel for 20°, 23°, and 26° bTDC are 13.17, 12.15, and 11.96 MJ/kWh, respectively. The BSEC of B100 fuel for injection timing of 20°, 23°, and 26° bTDC are 14.20, 12.85, and 12.79 MJ/kWh, respectively. At advanced injection timing of 26° bTDC, the BSEC of blend B20, diesel, and biodiesel are 11.96, 12.60, and 12.79 MJ/kWh, respectively, for full load operation.

Emission characteristics

The variation of unburned hydrocarbons (UBHC) with BP for different injection timings of diesel, B20 and B100 fuel at different load conditions are shown in figs. 8, 9, and 10, respectively. It is evident from the figures that UBHC of full load operation is lower at advanced IT compared to other IT. Similar trends are observed for part load operation. The increase in UBHC at other IT may be due to higher fuel consumption. The UBHC of diesel fuel for injection timing of 20°, 23° and 26° bTDC are 47, 42, and 40 ppm, respectively. The UBHC of B20 fuel for injection timing of 20°, 23°, and 26° bTDC are 38, 32, and 27 ppm, respectively. The UBHC of B100 fuel for injection timing of 20°, 23°, and 26° bTDC are 43, 39, and 35 ppm, respectively. At advanced IT of 26° bTDC, UBHC of blend B20, diesel and biodiesel are 27, 37, and 33 ppm, respectively, for full load operation.

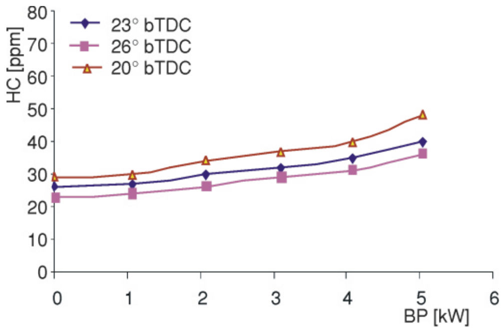

Figure 8. Variation of HC with BP of diesel fuel for different IT

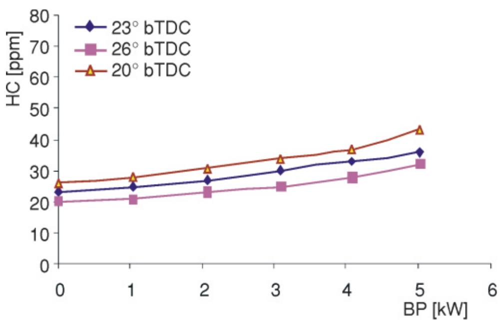

Figure 9. Variation of HC with BP of B20 fuel for different IT

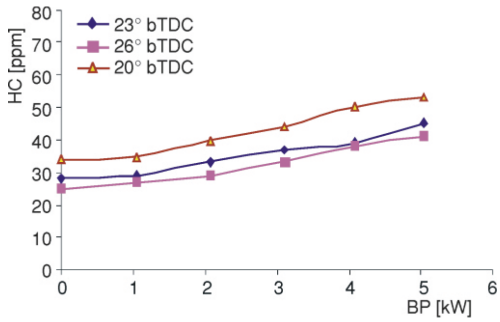

Figure 10. Variation of HC with BP of B100 fuel for different IT

The variation of smoke opacity with brake power for different IT of diesel, B20, and B100 fuel at different load conditions are shown in figs. 11, 12, and 13, respectively. The smoke opacity for diesel, B20, and B100 fuel of IT of 20° is higher in comparison with other IT. It may be due to retarded IT and incomplete combustion due to poor atomization leading to higher smoke emission. The smoke opacity of diesel fuel for injection timing of 20°, 23°, and 26° bTDC are 64.3, 38.3, and 25.2%, respectively. The smoke opacity of B20 fuel for IT of 20°, 23°, and 26°

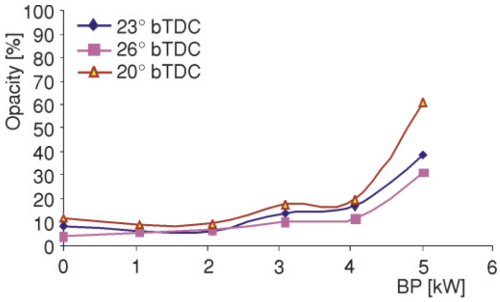

Figure 11. Variation of opacity with BP of diesel fuel for different IT

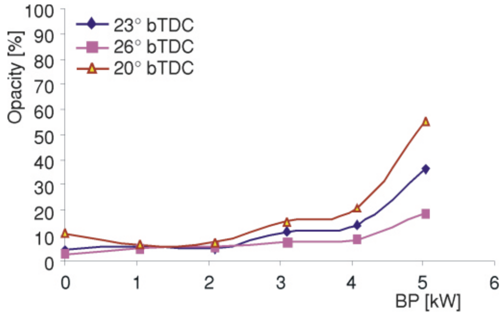

Figure 12. Variation of opacity with BP of B20 fuel for different IT

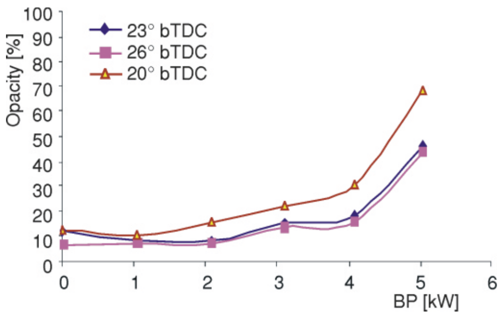

Figure 13. Variation of opacity with BPof B100 fuel for different IT

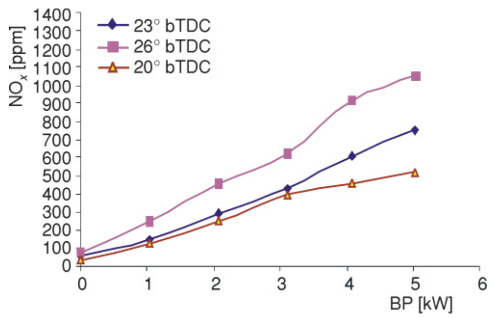

Figure 14. Variation of NO$_x$ with BP of diesel fuel for different IT

bTDC are 55.6, 36.2, and 18.2%, respectively. The opacity of B100 fuel for injection timing of 20°, 23°, and 26° bTDC are 36.9, 24.5, and 18.8%, respectively. At advanced injection timing of 26° bTDC, smoke opacity of blend B20, diesel and biodiesel are 18.2, 25.2, and 18.8 %, respectively, for full load operation.

The variation of NO$_x$ with BP for different injection timings of diesel, B20, and B100 fuel at different load conditions are shown in figs. 14, 15, and 16, respectively. At advanced IT

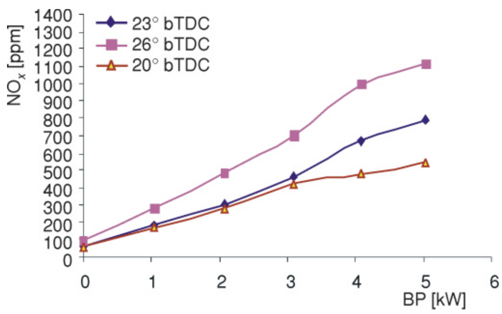

Figure 15. Variation of NO$_x$ with BP of B20 fuel for different IT

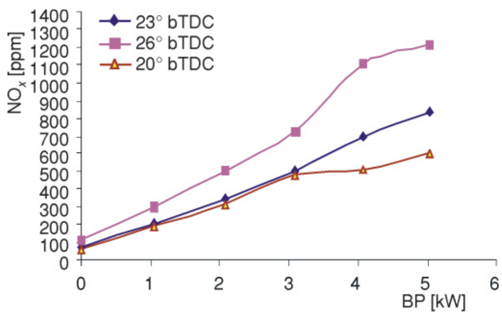

Figure 16. Variation of NO$_x$ with BP of B100 fuel for different IT

with biodiesel and its blend B20 fuel there is higher NO_x emission compared to diesel fuel as expected due to increased cylinder gas temperature. This may be due to reduced premixed mass because of shorter delay period. There is higher level of NO_x emission due to rise in cylinder peak pressure caused by increased amount of premixed mass burning due to advance IT. But at retarded injection timing, the NO_x emission is very low when compared to advanced as well as normal IT. The NO_x of diesel fuel for IT of 20°, 23°, and 26° bTDC are 423, 588, and 970 ppm, respectively. The NO_x of blend B20 fuel for IT of 20°, 23°, and 26° bTDC are 419, 582, and 1020 ppm, respectively. The NO_x of B100 fuel for IT of 20°, 23°, and 26° bTDC are 478, 648, and 1077 ppm, respectively. At retarded IT of 20° bTDC, NO_x of blend B20, diesel, and biodiesel are 419, 423, and 478 ppm, respectively for full load operation.

Combustion characteristics

The variation of peak cylinder pressure with brake power for optimal different injection timings of diesel, B20, and B100 fuel at different load conditions are shown in figs. 17, 18, and 19, respectively. The fig. 20 depicts the variation of cylinder pressure with crank angle at IOP of 220 bar and IT of 26° bTDC of different fuel. The peak cylinder pressure for IT of 20° bTDC is lower compared to the other two injection timings. And also peak cylinder pressure for biodiesel and its blend is higher as compared to diesel. The peak cylinder pressure increases with increase in injection advance, because it provides sufficient time for mixture formation and

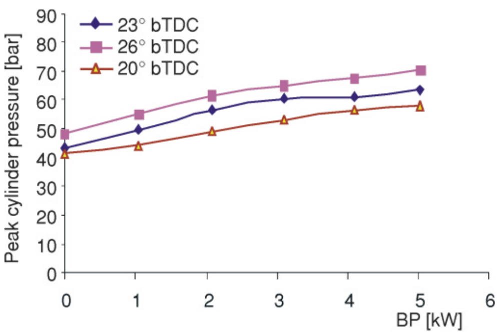

Figure 17. Variation of peak cylinder pressure with BP of diesel fuel for different IT

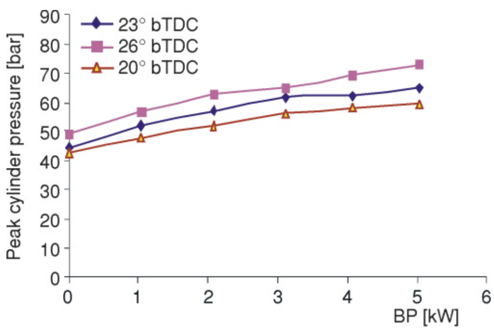

Figure 18. Variation of peak cylinder pressure with BP of B20 fuel for different IT

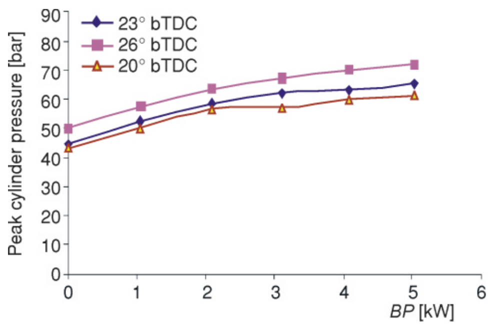

Figure 19. Variation of peak cylinder pressure with BP of B100 fuel for different IT

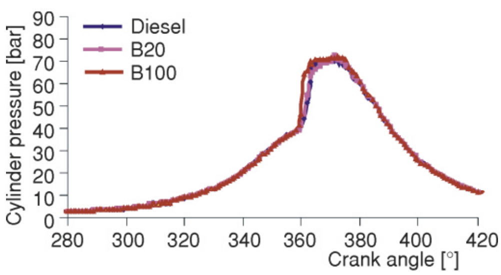

Figure 20. Variation of cylinder pressure with crank angle at IOP of 220 bar and IT of 26° bTDC of different fuel

yields in complete combustion. The peak cylinder pressure of diesel fuel for IT of 20°, 23°, and 26° bTDC are 58.24, 63.44, and 70.25 bar, respectively. The peak cylinder pressure of blend B20 fuel for injection timing of 20°, 23° and 26° bTDC are 59.34, 64.83, and 72.92 bar, respectively. The peak pressure of B100 fuel for IT of 20°, 23°, and 26° bTDC are 61.06, 65.05, and 72.04 bar, respectively. At advanced IT of 260 bTDC, peak cylinder pressure of blend B20, diesel, and biodiesel are 72.92, 70.25, and 72.04 bar, respectively, for full load operation.

The variation of peak heat release rate with BP for different injection timings of diesel, B20, and B100 fuel at different load conditions are shown in fig. 21, 22, and 23, respectively. The fig. 24 depicts the variation of heat release rate with crank angle at IOP of 220 bar and IT of 26° bTDC of different fuel. The peak heat release rate with biodiesel is always lower than blend B20 and diesel fuel. The start of combustion for the same IT was observed to be delayed indicat-

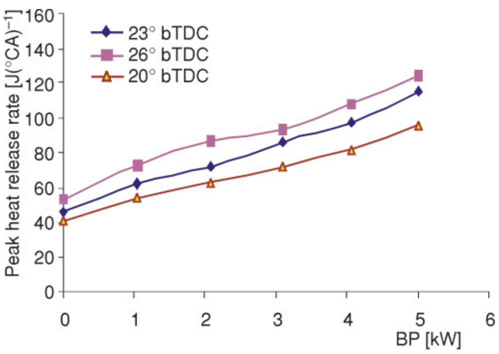

Figure 21. Variation of peak heat release rate with BP of diesel fuel for different IT

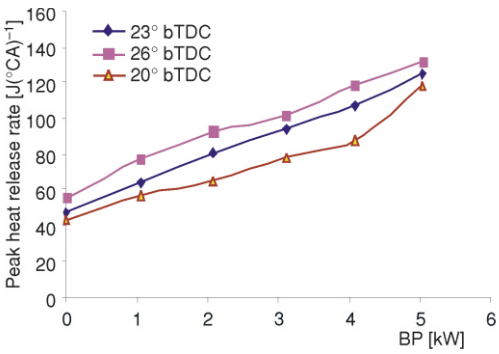

Figure 22. Variation of peak heat release rate with BP of B20 fuel for different IT

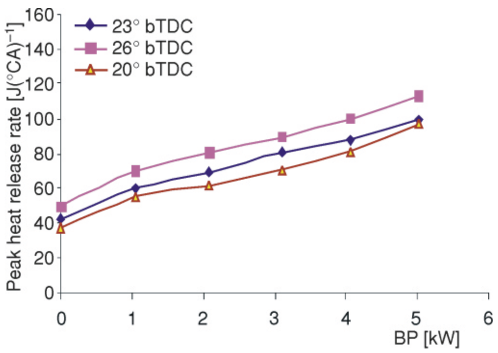

Figure 23. Variation of peak heat release rate with BP of B100 fuel for different IT

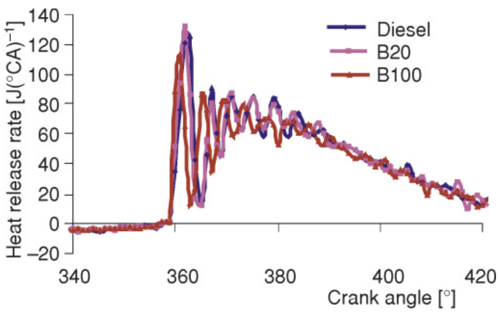

Figure 24. Variation of heat release rate with crank angle at IOP of 220 bar and IT of 260 bTDC of different fuel

ing an increase in the ignition delay with biodiesel. As we advance the IT the ignition delay increases because fuel will be injected earlier in to the combustion chamber. This leads to greater accumulation of the fuel in the premixed part of the combustion. This also results in shorter diffusion combustion and thus smoke formation is lowered. The peak heat release rate of diesel fuel for IT of 20°, 23°, and 26° bTDC are 95.19, 114.97, and 124.24 J/°CA, respectively. The heat release rate of B20 fuel for IT of 20°, 23°, and 26° bTDC are 118.56, 124.68, and 134.88 J/°CA respectively. The heat release rate of B100 fuel for IT of 20°, 23°, and 26° bTDC is 96.90, 98.95, and 112.73 J/°CA, respectively. At advanced IT of 260 bTDC, peak heat release rate of blend B20, diesel, and biodiesel are 131.88, 124.24, and 112.72 J/°CA, respectively, for full load operation.

The variation of ignition delay with BP for different IT of diesel, B20, and B100 fuel at different load conditions are shown in figs. 25, 26, and 27, respectively. Since the possibility exists for the decomposition of the methyl ester during the ignition delay period, and the viscosity of the biodiesel blend is somewhat higher than that of typical diesel fuel, one might expect to observe some differences in the combustion behaviour of blend fuel as compared to diesel fuel. From a close analysis of the pressure crank angle diagram, it is seen that ignition delay for all injection timings were found approximately same for all the test fuels. The ignition delay of diesel fuel for IT of 20°, 23°, and 26° bTDC are 25, 28, and 27 °CA, respectively. The ignition delay of B20 fuel for IT of 20°, 23°, and 26° bTDC are 26, 27, and 27 °CA, respectively. The ignition delay of B100 fuel for IT of 20°, 23°, and 26° bTDC are 25, 27, and 26 °CA, respectively. At advanced IT of 26° bTDC, ignition delay of blend B20, diesel, and biodiesel are 27, 27, and 26 °CA, respectively, for full load operation. Though the biodiesel blend has viscosity higher than diesel, the ignition delay is more or less same as diesel. This fact can be attributed to faster burning of OH molecule present in the biodiesel blend.

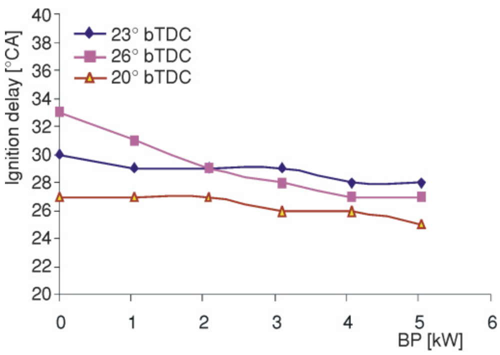

Figure 25. Variation of ignition delay with BP of diesel fuel for different IT

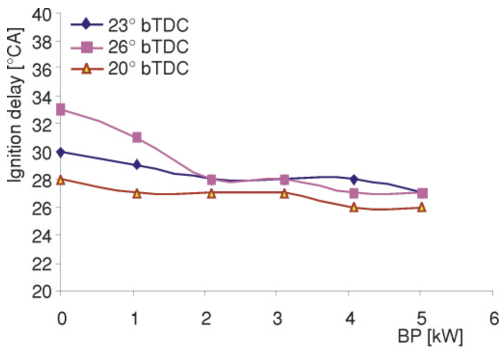

Figure 26. Variation of ignition delay with BP of B20 fuel for different IT

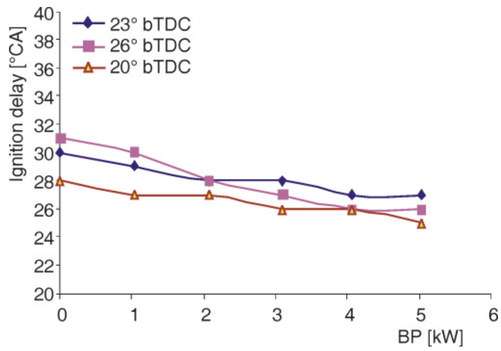

Figure 27. Variation of ignition delay with BP of B100 fuel for different IT

Conclusions

Based on the experimental work on a diesel engine fuelled with jatropha oil methyl ester and its blend B20 fuel, the following conclusions are drawn.

The advance injection timing of 26° bTDC has resulted in a significant improvement in the performance and emissions with JOME and its blend B20 fuel with diesel engine due to better combustion. Advanced injection timing increases the ignition delay and leads to more prominent initial phase of combustion in the premixed part. Advancing the injection timing by 3 °CA results in the following improvement at full load conditions.

- The injection timing of 26° bTDC was found to be optimum injection timing for B20 fuel as better performance and emissions were obtained.
- BTE of blend B20 fueled diesel engine as well as BSEC are found to be better compared to other test fuels.
- The emissions such as smoke opacity, UBHC and NO_x were found to be lower at optimum injection timing with blend B20 fuel.
- Peak cylinder pressure and corresponding peak heat release rate were higher with both biodiesel and its blend B20 compared to diesel fuel.
- Ignition delay of biodiesel and its blend B20 fuel was marginally shorter than that of diesel fuel.

Acronyms

BP – brake power
BSEC – brake specific energy consumption
bTDC – before top dead centre
BTE – brake thermal efficiency
IOP – injector opening pressure
IT – injection timing
JOME – jatropha oil methyl ester
UBHC – unburned hydrocarbon

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