COMBUSTION AND EMISSION CHARACTERISTICS OF DI COMPRESSION IGNITION ENGINE OPERATED ON JATROPHA OIL METHYL ESTER WITH DIFFERENT INJECTION PARAMETERS

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ABSTRACT

The current paper reports the engine performance, combustion and emissions from a direct injection compression ignition engine operated with different injector opening pressure (IOP) and injection timing (IT) with jatropha oil methyl ester (JOME) (B100), B20 (20% biodiesel and 80% petroleum diesel fuel which are generally called of B20 fuel) and diesel as test fuels. The engine was run on three different IOP viz. 180, 220 and 240bar along with normal IOP 200bar and two IT viz. 20deg. bTDC and 26deg. bTDC along with normal IT 23deg. bTDC. For all IOP and IT tried, the performance parameters such as brake thermal efficiency (BTE), brake specific energy consumption (BSEC), combustion parameters such as peak cylinder pressure, peak heat release rate and ignition delay and emissions such as UBHC, smoke opacity and NOx are reported here. From the experimental investigations it is observed that IOP 220bar and IT 26deg. bTDC showed better performance for all the test fuels. On the other hand, the performance, combustion and emission characteristics of B20 blend fueled direct injection compression ignition engine performed better for entire load range of operation. At higher loads with IOP 220bar and IT 26deg. bTDC emissions such as smoke opacity and UBHC were observed to be lower compared to other IOPs and ITs. But, NOx emission at retard IT 20deg. bTDC was very low compared to other two ITs. BTE of blend B20 fueled compression ignition engine has increased by 1.01% when operated with IOP 220bar at IT 23deg. bTDC and 1.34% with IT 26deg. bTDC at IOP 200bar. On other hand blend B20 fueled direct injection compression ignition engine showed better performance with reasonable higher brake thermal efficiency and lower BSEC, better combustion and emission when compared to biodiesel (B100) and diesel fuel.

Keywards: Jatropha oil methyl ester, Transesterification, Compression Ignition engine, Injection opening pressure, Injection timing.

NOMENCLATURE

bTDC-Before Top Dead Centre
BTE-Brake Thermal Efficiency
BSEC-Brake Specific Energy Consumption
UBHC-Unburned Hydrocarbon
NOx -Oxides of Nitrogen
CO- Carbon monoxide
JOME- Jatropha oil Methyl Ester
IOP-Injector Opening Pressure
IT-Injection Timing
B20-20% biodiesel and 80% Petroleum diesel
B100- Pure biodiesel (JOME)

1. INTRODUCTION

Internal combustion engines particularly of the compression ignition type play a major role in transportation, industrial power generation and in the agricultural sector as well. There is need to search and find ways of using alternative fuels, which are preferably renewable and also contribute low levels of gaseous and particulate emissions from internal combustion engines. In the case of agricultural applications, fuels that can be produced in rural areas in a decentralized manner, near the consumption points will be favoured. The permissible emission levels can also be different in rural areas as compared to urban areas on account of the large differences in the number density of engines.

Ramadhas et al. (2004), Parmani (2003) and Senthil Kumar et al. (2003) reported the vegetable oils are easily available in rural areas, are renewable, have a reasonably high cetane number to be used in compression ignition engines with simple modifications and can be easily blended with diesel in the neat and esterifed (biodiesel) forms. Jatropha oil, Karanji oil, Coconut oil, Sunflower oil, Rapeseed oil and Neem oil are some of the vegetable oils that have been tried as fuels in internal combustion engines earlier. It was also found that the heat release rate is very similar to diesel with vegetable oils. The CO and HC emissions are higher and NOx emissions are lower than that of diesel for vegetable oils with higher smoke levels. Samamga (1983) studied the biodiesel derived from several feed stock have also been investigated extensively. Brake thermal efficiency with the biodiesel is comparable with diesel whereas in case of straight vegetable oil it is less than diesel. Further, with esters HC emissions are lower compared to the raw vegetable oil. Hammetien et al. (1991) investigated the performance and emission characteristics of vegetable oil fuelled engines, several methods like conversion to biodiesel, addition of oxygenates, dual fuelling with a gaseous fuel, use of cetane number improving additives and preheating to lower the viscosity have been tried. Addition of oxygenates and dual fuelling lead to high brake thermal efficiency and also reduction in HC and CO emissions in some cases. Srinivasa Rao et al. (1991), Masjuki et al. (1991) reported the gradual depletion of world petroleum reserves and the impact of environmental pollution due to engine exhaust emissions, there is an urgent need for suitable alternative fuels for use in diesel engines. In view of this, vegetable oil is a promising alternative because it has several advantages. Therefore, in recent years systematic efforts have been made by several research workers to use vegetable oils as fuel in engines. Obviously, the use of non-edible vegetable oils compared to edible oil is very significant because of the tremendous demand for edible oils as food and they are far too expensive to be used as fuel at present.

Saholl et al. (1993) have reported the major problem associated with direct use of vegetable oils is their viscosity. One possible method to overcome the problem of high viscosity is transestrification of oils to produce esters (commonly known as biodiesel) of respective oils. Marshall W. et al. (1995) reported the esters of fatty acids derived from transestrification of vegetable oils have properties closer to petroleum diesel fuel. These fuels tend to burn cleaner with its performance comparable to conventional diesel fuel and combustion similar to diesel fuel. Biodiesel is a non-polluting fuel made from organic oils of vegetable origin. Chemically it is known as free fatty acid methyl ester.

Kato et al. (1989) studied the effect of fuel injection pressures play a vital role in engine exhaust emissions. Higher injection pressures create faster combustion rates which result in higher gas temperatures as compared to the conventional low pressure system. When switching from a low pressure to a high pressure injection system, particulate emission reductions of up to 80% were observed with no change in hydrocarbon emissions and only slightly higher NOx emissions. Foidl et al. (1996) performed the investigation based on his experimental findings; the esters of vegetable oils can be used directly in existing engines without modifications. Biodiesel is virtually non-toxic and biodegradable, potentially providing additional environmental benefits and accepted by EPA (Environmental Protection Agency) as alternative fuel for diesel engine. Biodiesel can provide a substantial reduction in green house gases. Palaniswamy et al. (2005) reported that use of biodiesel in conventional diesel engine results in substantial reduction of unburned HC, CO and particulate matters with lower NOx emissions.

2. EXPERMENTAL WORK

2.1 Transestrification of Jatropha Oil

Widely used and accepted process to reduce the viscosity of triglycerides in vegetable oil is transesterification**.** Anjana Srivastava et al. (2000) reported that 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. About 3-4grams of catalyst (NaOH) was dissolved in 100ml of methanol to prepare alkoxide, which is required to activate the alcohol. Around 15-20minutes 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 60^0C-65^0C was carried out for one hour with water or air cooled condenser. The resulting mixture was then taken out and poured into the separating funnel to separate glycerol from the mixture to get the methyl ester of jatropha oil. Water washing was done in order to remove alcohol and impurities from the biodiesel.

2.2 Experimental setup

The performance and emissions tests were conducted on a single cylinder, four stroke, direct injection and water cooled with eddy current dynamometer compression ignition engine test rig as shown in figure A. The specifications of test rig are depicted in table 1. Engine was directly coupled to an eddy current dynamometer. The engine and dynamometer were interfaced to a control panel, which is connected to a computer for data acquisition. Test parameters such as fuel flow rate, temperatures, air flow rate, load etc. was recorded with data acquisition and used for calculating the engine performance characteristics. The calorific value and the density of a particular fuel was fed to the acquisition software as input variables for necessary calculation. The exhaust gas was made to pass through the probe of exhaust gas analyzer for the measurement of HC, opacity and NOx. Later exhaust was passed through the probe of smoke meter of Hartidge type for the measurement of smoke opacity.

2.3 Experimental Procedure

The whole set of experiments was conducted at the engine speed of 1500rpm and compression ratio 17.5. Firstly, the experiments were conducted at the designed injection timing of 23deg. bTDC and 200bar injector opening pressure for no-load, 20% load, 40% load, 60% load, 80% load and full load for diesel fuel, B100 and its blend B20.

Table 1 Experimental Setup Specifications

Engine	Four-stroke, single cylinder, constant speed, water cooled Diesel engine
BHP	7BHP @ 1500 RPM
Bore x Stroke	87.5 x 110 mm
Stroke Volume	661.5 cc
Compression Ratio	17.5:1
Load Measurement	Strain Gage Load cell
Water Flow Measurement	Rota meter
Fuel and Air Measurement	Differential Pressure Unit
Speed Measurement	Rotary Encoder
Interfacing	ADC card

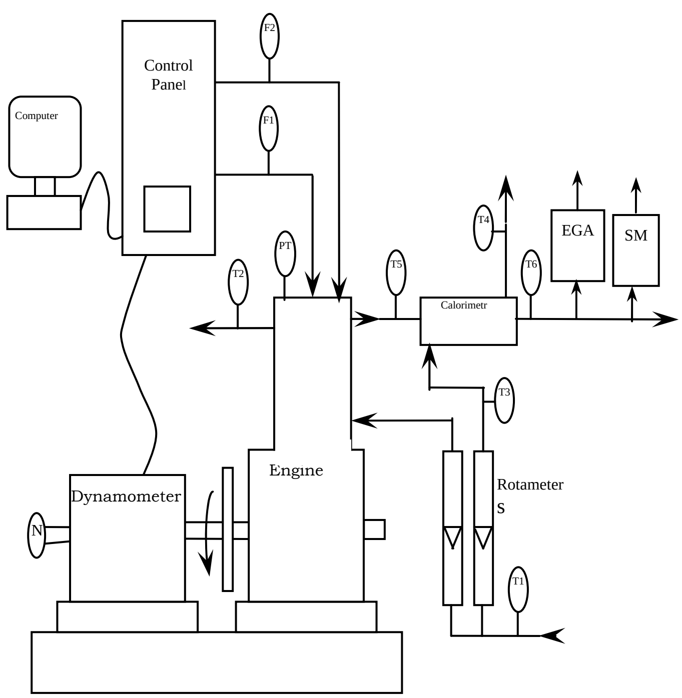

Fig. 1 Experimental Setup

T1, T3 Inlet Water Temperature ^{0}C
T2 Outlet Engine Jacket Water Temperature ^{0}C
T4 Outlet Calorimeter Water Temperature ^{0}C
T5 Exhaust Gas Temperature before Calorimeter ^{0}C
T6 Exhaust Gas Temperature after Calorimeter ^{0}C
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

Data such as exhaust gas temperature, water inlet and outlet temperature, fuel consumption rate, brake power, HC, smoke opacity and NOx was recorded. Similar experiments were conducted at different injector opening pressures viz. 180, 220 and 240bar, the performance and emissions parameters were recorded as earlier. Similar experiments were conducted at other injection timing viz. 20deg. bTDC and 26deg. bTDC and performance and emissions parameters were recorded as earlier.

3. RESULTS AND DISCUSSIONS

3.1 Brake Thermal Efficiency

Figure 2, 3, and 4 illustrates the variation of brake thermal efficiency (BTE) with different IOPs with diesel, B20 and B100 fuel at full load condition and figure 5 illustrates the variation of BTE with BP of diesel, B20 and B100 fuel at injector opening pressure 220bar and injection timing 26deg. bTDC. It is observed that efficiency obtained at full load and part load of blend B20 fuel with injector opening pressure 220bar is higher than the B100 and diesel fuel compared with other injector opening pressures and similar increase in the thermal efficiency was also observed in the remaining loads. It can be observed that the thermal efficiency of all fuels at lower injection pressure is low due to coarse spray formation and poor atomization and mixture formation of biodiesel during injection. However, with higher injection opening pressure due to the fine spray formed during injection and improved atomization, resulted with lower physical delay period yielding in better combustion. This will enhance combustion and in turn improves efficiency. For blend B20 fuel, the brake thermal efficiency is markedly higher than B100 fuel and diesel fuel. The possible reason for the above findings is attributed to the additional lubricity of biodiesel which tend to minimize the frictional losses in the cylinder. The maximum BTE occurred with IOP 220bar and blend B20 which was selected as optimal injection pressure. Further, increase in the injector opening pressure beyond 220bar to 240bar resulted in decrease in the thermal efficiency with all test fuels. This may be due to the fact that, at higher injection opening pressure, the size of fuel droplets decreases drastically. Thus a very fine fuel spray will be injected in to the combustion chamber. Due to reduction in penetration of fuel spray and also reduced momentum of the fuel droplets results in ineffective combustion.

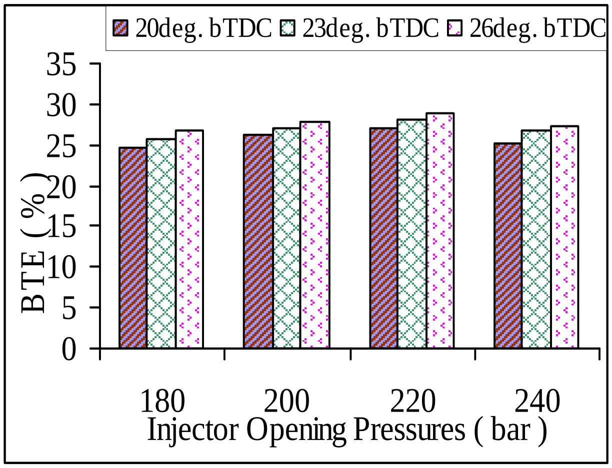

Figure 2 Variation of BTE with different IOPs of diesel fuel at full load conditions.

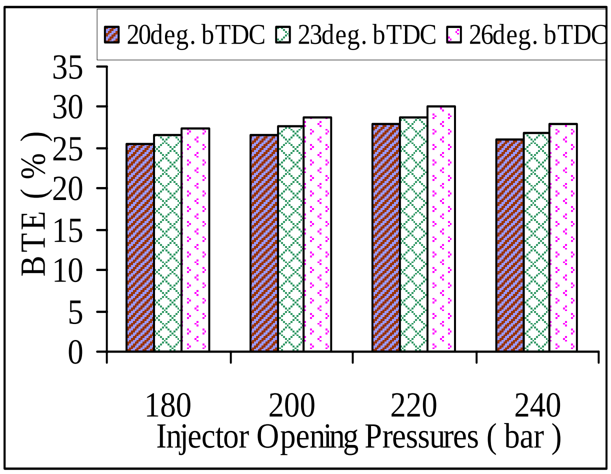

Figure 3 Variation of BTE with different IOPs of B20 fuel at full load conditions.

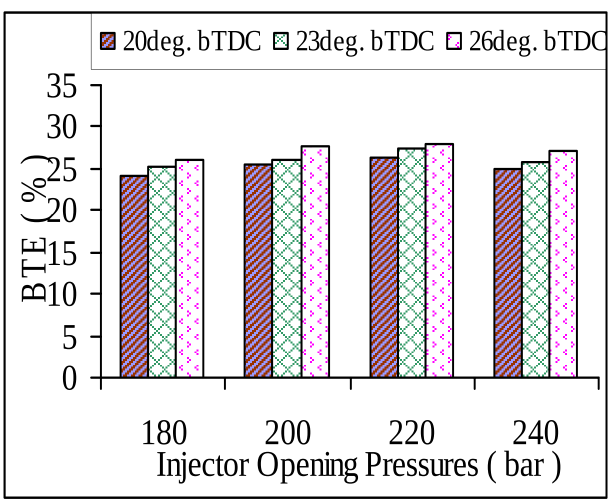

Figure 4 Variation of BTE with different IOPs of B100 at fuel full load conditions.

It is observed that BTE of retard injection timing i.e. 20deg. bTDC is lower than the other injection timings. But with advance injection timing, the BTE of diesel fuel, blend B20 and B100 was higher than the other two injection timings. This may be due to increase in power produced at advanced injection timing and lower fuel consumption. The maximum brake thermal efficiency occurred at injection timing of 26deg. bTDC and blend B20 which is selected as optimal. This is 3deg. more advanced than that of designed injection timing. It is seen that brake thermal efficiency at advanced injection timing and with blend B20 fuel showed better results than at designed injection timing and retard injection timing. At this injection timing with IOP 220bar, the brake thermal efficiency of blend B20, diesel and B100 was 29.99%, 28.89% and 28.25% respectively for full load operations.

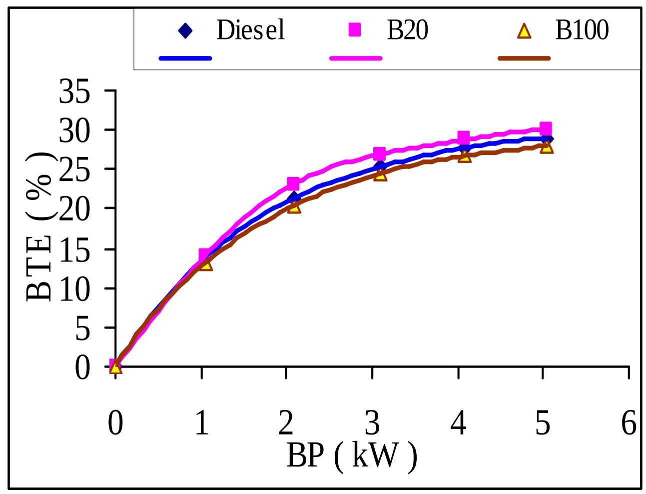

Figure 5 Variation of BTE with BP of diesel, B20 and B100 fuel at IOP 220bar and IT 26deg. bTDC.

3.2 Brake Specific Energy Consumption

Figure 6, 7, and 8 illustrates the variation of BSEC with different IOPs of diesel, B20 and B100 fuel at full load condition and figure 9 illustrates the variation of BSEC with BP of diesel, B20 and B100 fuel at injector opening pressure 220bar and injection timing 26deg. bTDC. The BSEC for B100 fuels is higher than diesel and blend B20 fuel which was observed due to lower calorific values, higher density and lower energy content. Higher the density more will be the discharge of fuel for the same displacement of the plunger of the fuel injection pump. For the injector opening pressure of 220bar with blend B20 fuel, the BSEC of compression ignition engine for the entire load range was lower compared to other injector opening pressures. This may be due to the increased penetration length and spray cone angle and due to more area coverage of spray formed in the combustion chamber and utilizing the air effectively resulting optimum peak pressure, better fuel air mixing and higher spray atomization. However, injector opening pressure 240bar, the performance has suffered significantly because of low penetration of fuel droplets due to low momentum of fuel droplets.

It was observed that at retard injection timing, the BSEC of B100, B20 and diesel fuel was higher than other injection timings under all the load conditions. This may be due to poor and untimely combustion fuels. But at advance injection timing, the BSEC was lower for B20 fuel than B100 and diesel fuel. This may be due to complete combustion of fuel due to sufficient duration. At advanced injection timing 26deg. bTDC and IOP 220bar, the BSEC of blend B20, diesel and B100 are 11.80, 12.25 and 12.82MJ/kW-hr respectively for full load operations.

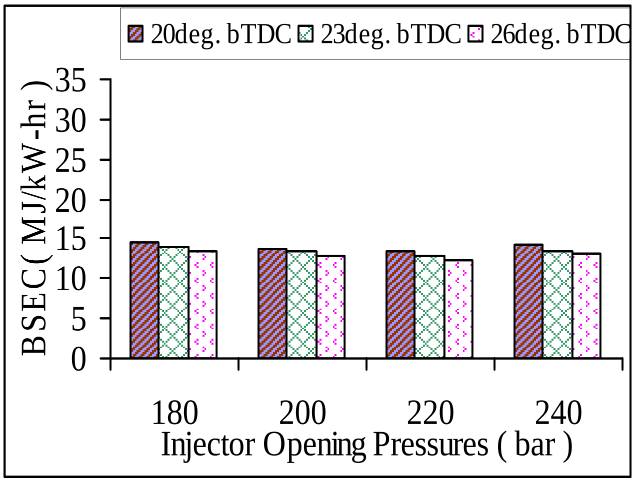

Figure 6 Variation of BSEC with different IOPs of diesel fuel at full load conditions.

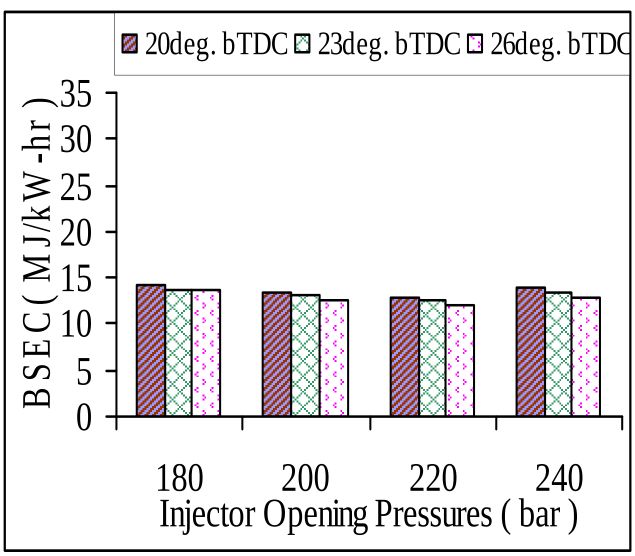

Figure 7 Variation of BSEC with different IOPs of B20 fuel at full load conditions.

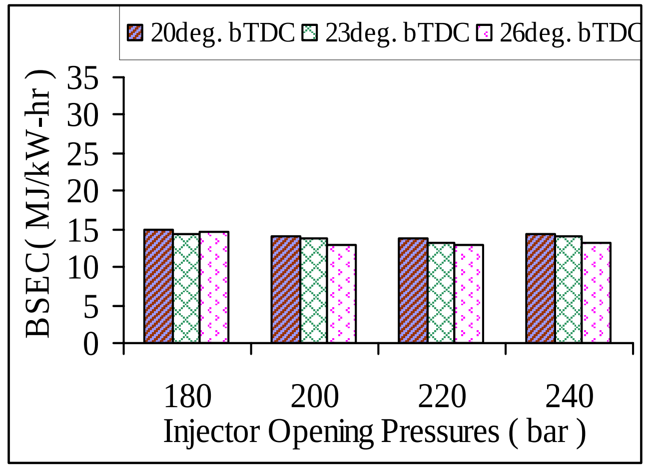

Figure 8 Variation of BSEC with different IOPs of B100 fuel at full load conditions.

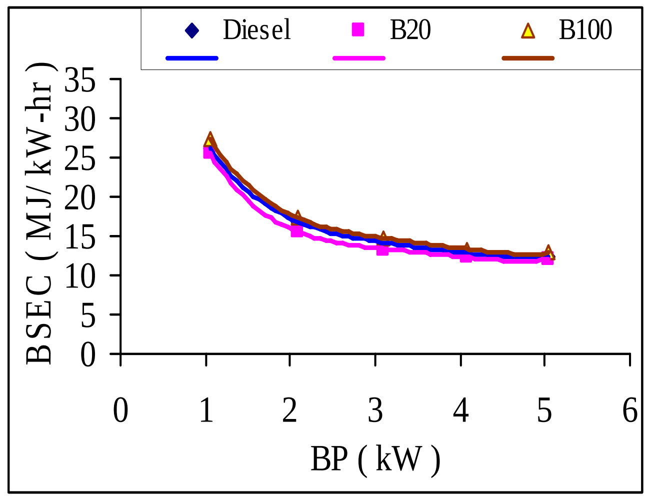

Figure 9 Variation of BSEC with BP of diesel, B20 and B100 fuel at IOP 220bar and IT 26deg. bTDC.

3.3 Unburned Hydrocarbon

Figure 10, 11 and 12 illustrates the variation of UBHC with different IOPs of diesel, B20 and B100 fuel at full load condition and figure 13 illustrates the variation of UBHC with BP of diesel, B20 and B100 fuel at optimum injector opening pressure and injection timing. Unburnt hydrocarbons are results of incomplete combustion of fuel. UBHC emissions generally found to be very less in diesel engine compared to petrol engine. With the injector opening pressure 180bar and 240bar, the UBHC emissions are exceedingly higher compared to 200bar and 220bar. This may be attributed to the incomplete and improper mixture formation of the fuel at lower injection and higher injection pressure respectively. Also at very high IOPs considerable portion of the combustion occurs in the diffusion phase. However, with IOP 220bar, the B20 fuel showed significant reduction in UBHC emissions. The Improved performance was observed at IOP 220bar with B20 fuel, though they reasonably high viscosity and lower cetane number.

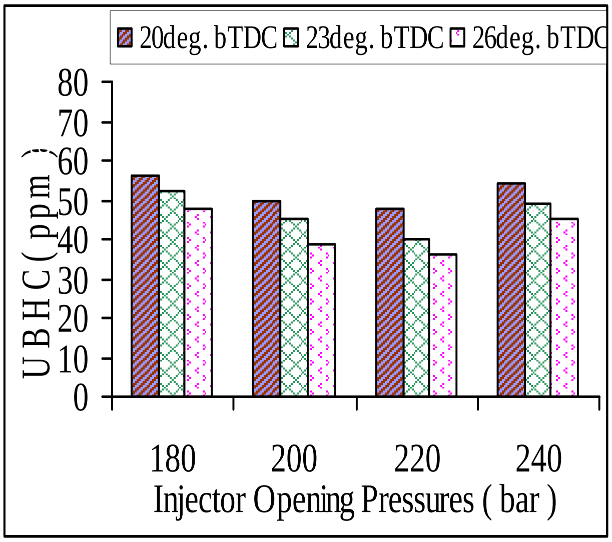

Figure 10 Variation of UBHC with different IOPs of diesel fuel at full load conditions.

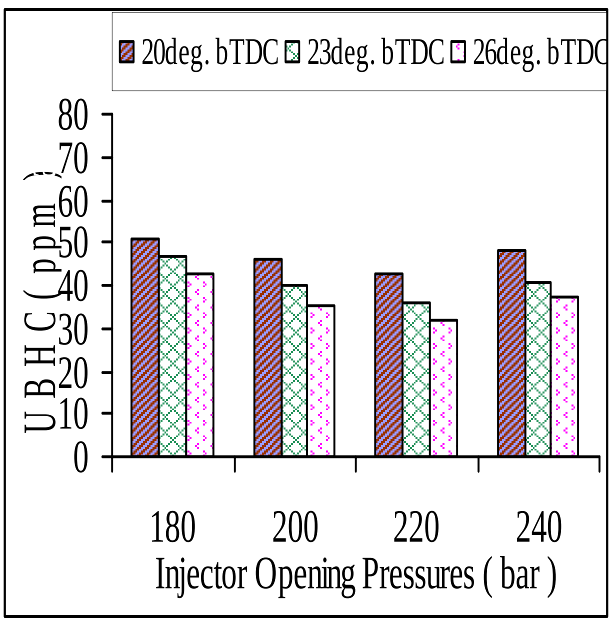

Figure 11 Variation of UBHC with different IOPs of B20 fuel at full load conditions.

At advanced injection timing 26deg. bTDC, the UBHC emission was lower compared to other injection timings tried for all the engine output power. The increase in UBHC at other injection timings may be due to early start of the combustion process yielding extra time for complete combustion. At advanced injection timing 26deg. bTDC and IOP 220bar, UBHC of blend B20, diesel and B100 are 32, 36 and 41ppm respectively for full load operations.

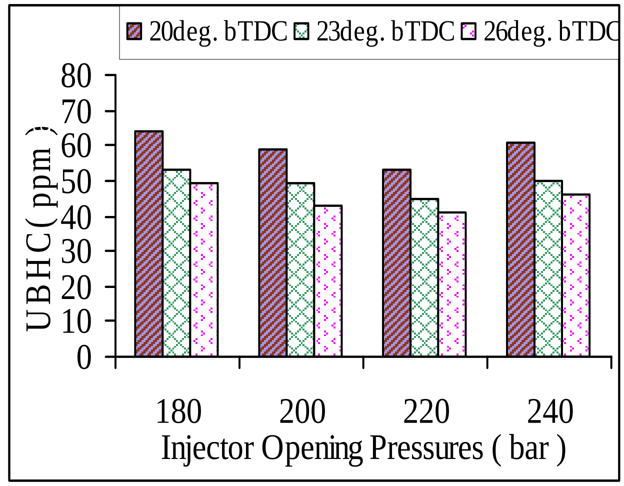

Figure 12 Variation of UBHC with different IOPs of B100 fuel at full load conditions.

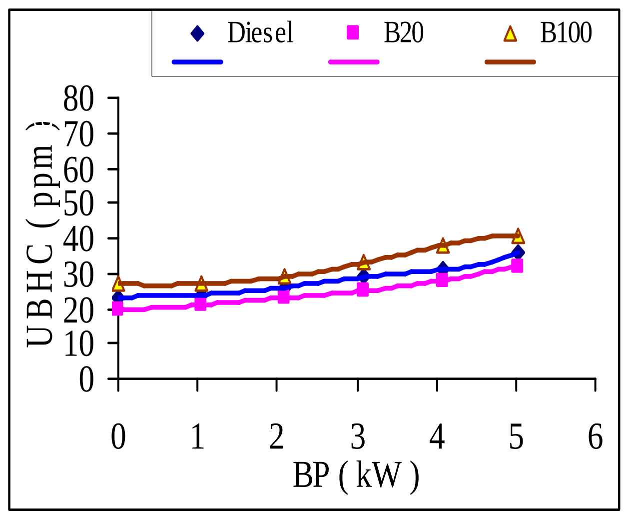

Figure 13 Variation of UBHC with BP of diesel, B20 and B100 fuel at IOP 220bar and IT 26deg. bTDC.

3.4 Smoke Opacity

Figure 14, 15 and 16 illustrates the variation of smoke opacity with different IOPs of diesel, B20 and B100 fuel at full load condition and figure 17 illustrates the variation of smoke opacity with BP of diesel, B20 and B100 fuel at injector opening pressure 220bar and injection timing 26deg. bTDC. It is noticeable that smoke opacity is marginally affected by the change in IOPs. The smoke opacity is marginally lower for blend B20 with IOP 220bar compared to B100 and diesel fuel. This may be due to two main reasons; firstly, the thermal efficiency which is higher for blend B20 fuel represents a better and complete combustion.

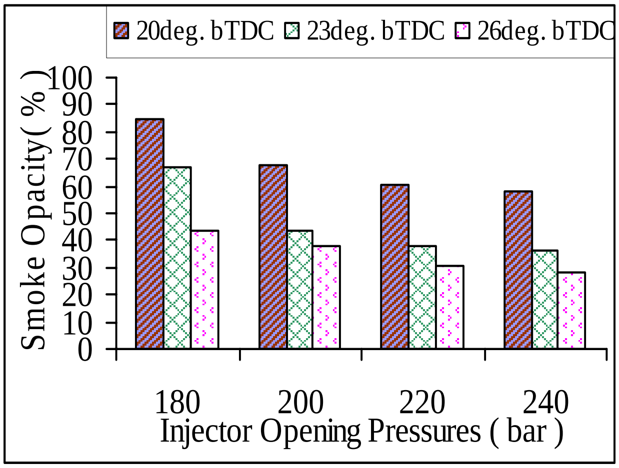

Figure 14 Variation of smoke opacity with different IOPs of diesel fuel at full load conditions.

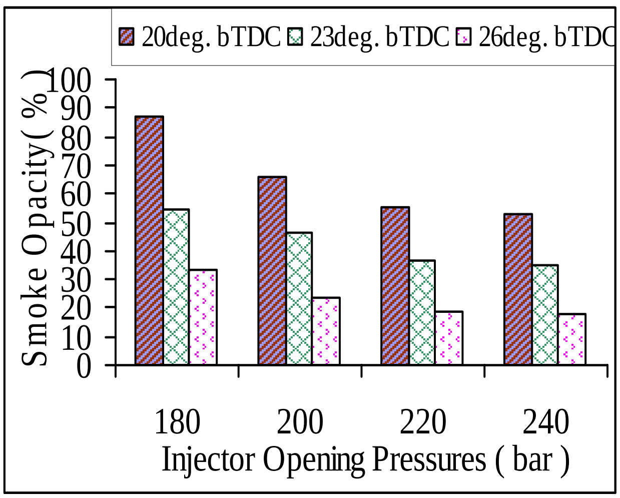

Figure 15 Variation of smoke opacity with different IOPs of B20 fuel at full load conditions.

Thus improving smoke opacity values and secondly, the molecules of biodiesel contain some amount of oxygen that takes part in combustion and this may be a possible reason for more complete combustion. The oxygen molecule present in biodiesel molecular structure may be readily available for oxidation of injected fuel and also indicates that smoke levels steadily fall with increase in the IOP due to improved mixture formation as a result of better atomized spray.

The smoke opacity of diesel, B20 and B100 fuel at retard injection timing is higher when compared with the other injection timings. This is due to retard injection timing and may be incomplete combustion and poor atomization and this leads to higher smoke emission. At advanced injection timing with B20 fuel, smoke opacity was lower compared to diesel fuel and B100 due to lower fuel consumption compared to other fuels. At advanced injection timing 26deg. bTDC and IOP 220bar, smoke opacity of blend B20, diesel and B100 18.4, 30.7 and 43.8% respectively for full load operation.

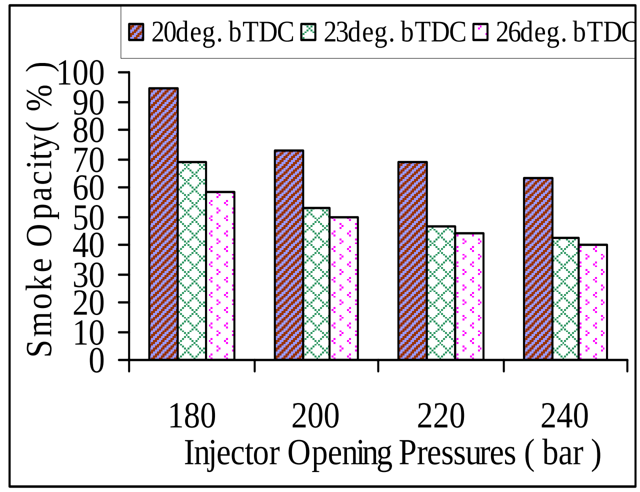

Figure 16 Variation of smoke opacity with different IOPs of B100 fuel at full load conditions.

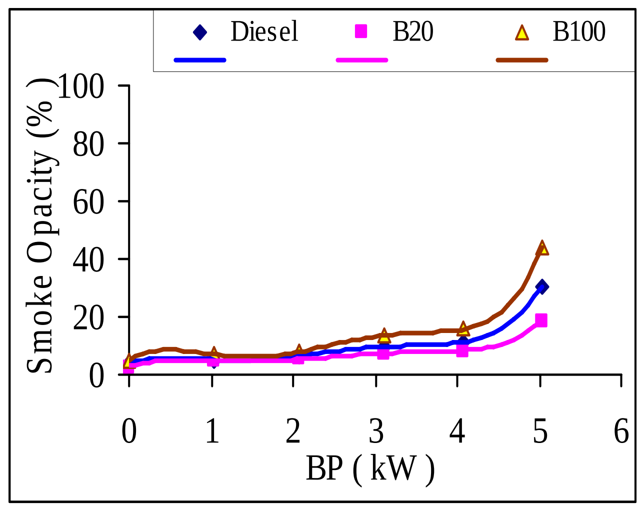

Figure 17 Variation of smoke opacity with BP of diesel, B20 and B100 fuel at IOP 220bar and IT 26deg. bTDC.

3.5 Oxides of Nitrogen

Figure 18, 19 and 20 illustrates the variation of NOx with different IOPs of diesel, B20 and B100 fuel at full load condition and figure 21 illustrates the variation of NOx with BP of diesel, B20 and B100 fuel at injector opening pressure 220bar and injection timing 26deg. bTDC. The nitrogen oxides results from the oxidation of atmospheric nitrogen at high temperature inside the combustion chamber of an engine rather than, resulting from a contaminant present in the fuel. NOx formation is a strongly temperature dependent phenomenon and hence, NOx increases with increase in load for all fuels. It also observed that the NOx is increased with increase in injection pressures and the NOx emission in the case of B100 fuel is slightly higher than the diesel fuel. This may be due to the higher temperatures in the combustion chamber, because of combustion of the fuel at the later part of the expansion stroke.

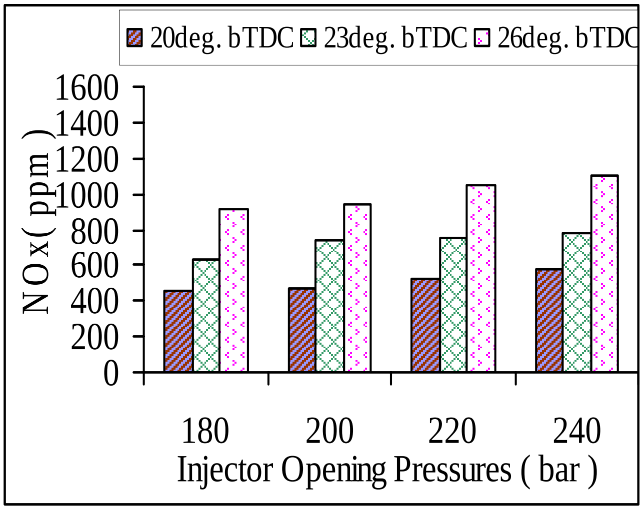

Figure 18 Variation of NOx with different IOPs of diesel fuel at full load conditions.

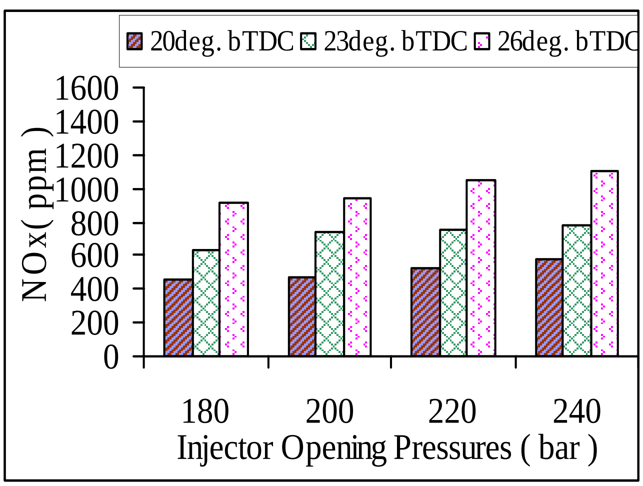

Figure 19 Variation of NOx with different IOPs of B20 fuel at full load conditions.

At advanced injection timing 26deg. bTDC with B100 and B20 fuels, there is higher NOx emission compared to diesel fuel as expected due to increased cylinder gas temperatures. Higher level of NOx emission were recorded due to rise in cylinder peak pressure caused by increased amount of premixed mass burning at advance injection timing. But at retard injection timing, the NOx emission is very low compared to advance, because of lower combustion temperature and pressure. At advanced injection timing 26deg. bTDC and IOP 220brar, the NOx of blend B20, diesel and B100 are 1112, 1055 and 1212ppm respectively.

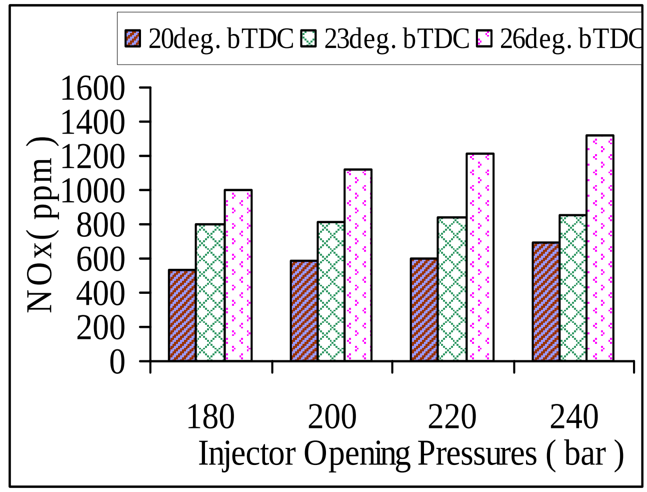

Figure 20 Variation of NOx with different IOPs of B100 fuel at full load conditions.

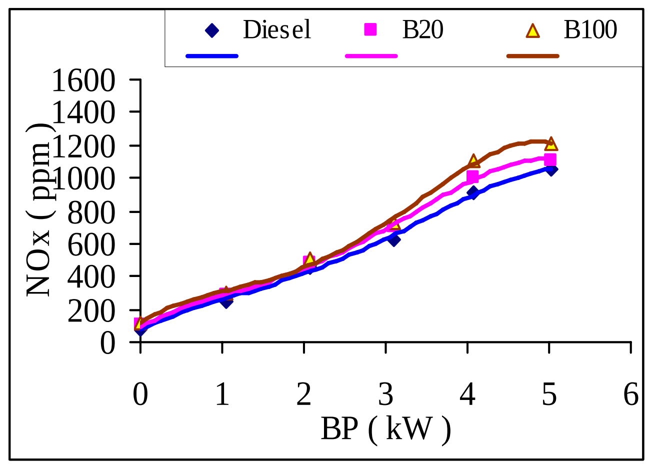

Figure 21 Variation of NOx with BP of diesel, B20 and B100 fuel at IOP 220bar and IT 26deg. bTDC.

3.6 Peak Cylinder Pressure

Figure 22, 23 and 24 illustrates the variation of peak cylinder pressure with different IOPs of diesel, B20 and B100 fuel at full load condition and figure 25 illustrates the variation of peak cylinder pressure with BP of diesel, B20 and B100 fuel at injector opening pressure 220bar and injection timing 26deg. bTDC. The highest peak cylinder pressure was recorded with IOP 220bar which is due to effective and efficient combustion taking place. The peak pressure mainly depends on the combustion rate in the initial stages of combustion which is influenced by the amount of fuel taking part in uncontrolled heat release phase. Also it was observed that the peak cylinder pressure of B100 and its blend was slightly higher compared to that of diesel fuel. This is due to the lower ignition delay for B100 and its blend B20. The oxygen content of biodiesel and its blend which results in better combustion may also contribute in higher peak cylinder pressure compared to diesel.

The peak cylinder pressure of retard injection timing is lower compared to the other two injection timings and also peak cylinder pressure for biodiesel and its blend was higher as compared to diesel. The peak cylinder pressure increases with increase in injection advance, which provides sufficient time for mixture formation and complete combustion. At advanced injection timing 26deg. bTDC and IOP 220bar, peak cylinder pressure of blend B20, diesel and B100 are 70.92, 70.25 and 72.04bar respectively for full load operation.

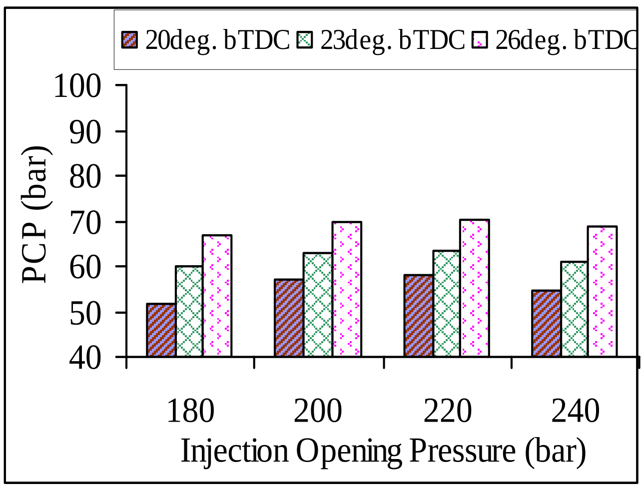

Figure 22 Variation of peak cylinder pressure with different IOPs of diesel fuel at full load conditions.

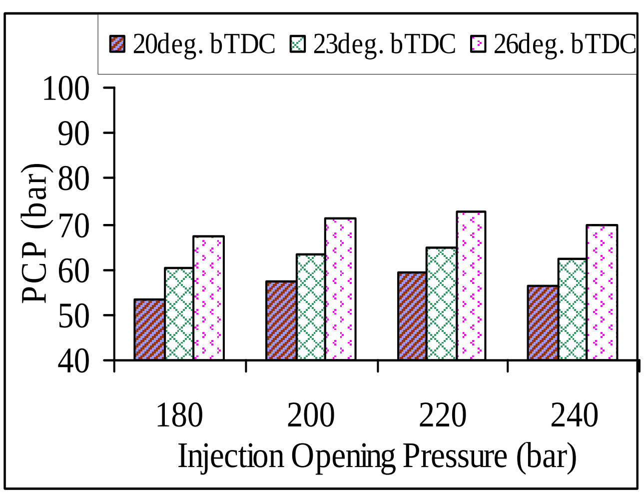

Figure 23 Variation of peak cylinder pressure with different IOPs of B20 fuel at full load conditions.

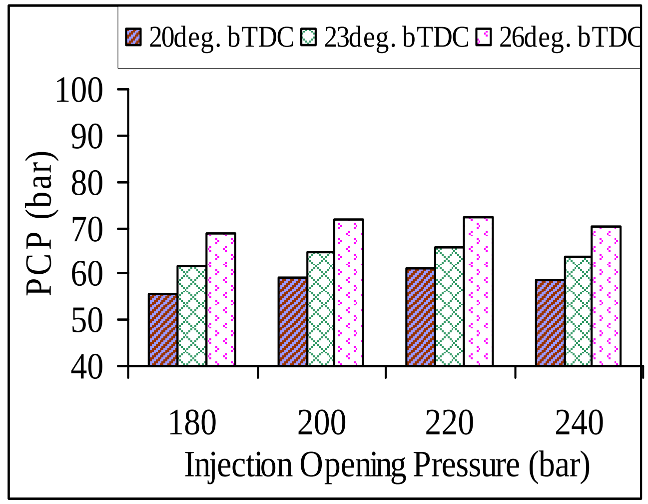

Figure 24 Variation of peak cylinder pressure with different IOPs of B20 fuel at full load conditions.

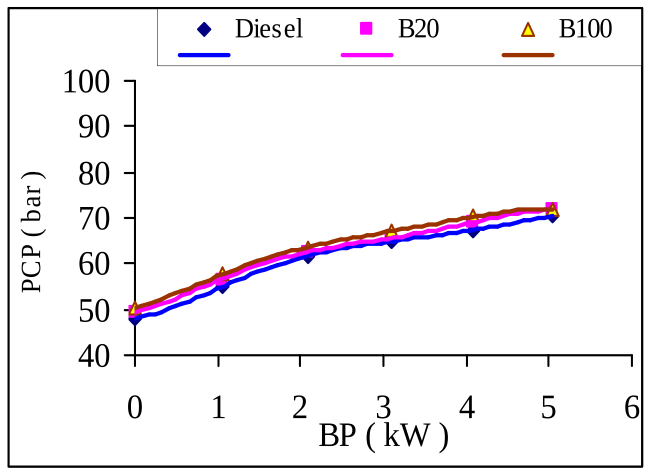

Figure 25 Variation of peak cylinder pressure with BP of diesel, B20 and B100 fuel at IOP 220bar and IT 26deg. bTDC.

3.7 Peak Heat Release Rate

Figure 26, 27 and 28 illustrates the variation of peak heat release rate with different IOPs of diesel, B20 and B100 fuel at full load condition and figure 29 illustrates the variation of peak heat release rate with BP of diesel, B20 and B100 fuel at injector opening pressure 220bar and injection timing 26deg. bTDC. The peak heat release rate and the peak combustion temperature is correspondingly better for fuel with blend B20 compared to B100 and diesel fuel at injector opening pressure 220bar compared to other injector opening pressures. Hence, looking at all the combustion, performance and emission characteristics, the blend B20 gives overall better performance compared to B100 and diesel fuel. The peak heat release rate is improved when the injector opening pressure is enhanced due to better fuel atomization. This was seen in the case of performance and emissions parameters also. Reduced smoke levels (even lower than base diesel operation) and increased thermal efficiency, but with higher NOx levels were observed when the IOP is increased to 220bar.

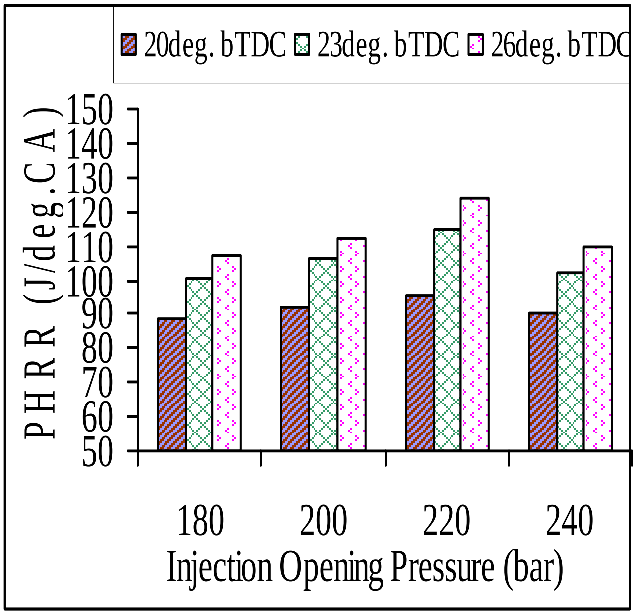

Figure 26 Variation of peak heat release rate with different IOPs of diesel fuel at full load conditions.

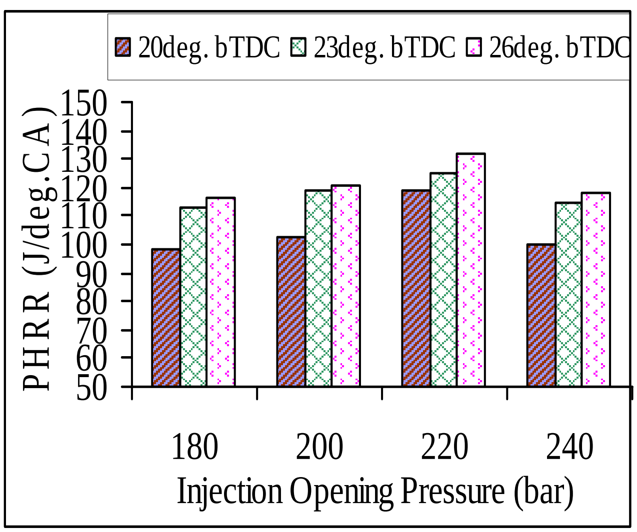

Figure 27 Variation of peak heat release rate with different IOPs of B20 fuel at full load conditions.

The peak heat release rate with B100 is always lower than blend B20 and diesel fuel. At advanced injection timing 26deg. bTDC and IOP 220bar, peak heat release rate of blend B20, diesel and B100 are 131.88, 124.24 and 112.72J/deg. CA respectively for full load operation.

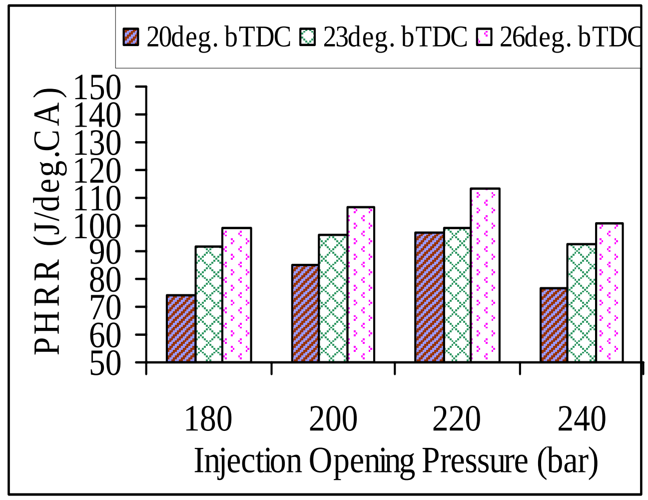

Figure 28 Variation of peak heat release rate with different IOPs of B100 fuel at full load conditions.

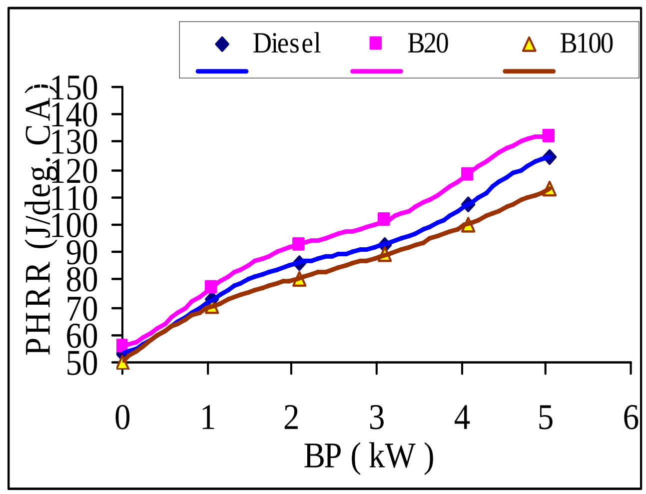

Figure 29 Variation of peak heat release rate with BP of diesel, B20 and B100 fuel at IOP 220bar and IT 26deg. bTDC.

3.8 Ignition Delay

Figure 30, 31 and 32 illustrates the variation of ignition delay with different IOPs of diesel, B20 and B100 fuel at full load condition and figure 33 illustrates the variation of ignition delay with BP of diesel, B20 and B100 fuel at injector opening pressure 220bar and injection timing 26deg. bTDC. It was observed that ignition delay of B100 and its blend B20 fuel are significantly lower than that of diesel fuel and the ignition delay decreases with the increase in brake power. This is due to fact that olieic and linileic fatty acid methyl esters presenting the biodiesel split into smaller compounds when it enters the combustion chamber resulting in higher spray angles and hence earlier ignition. This indicates the biodiesel and its blends have higher cetane number compared to diesel fuel. It is noticed that for all test fuels, the reduction in ignition delay increase in brake power output.

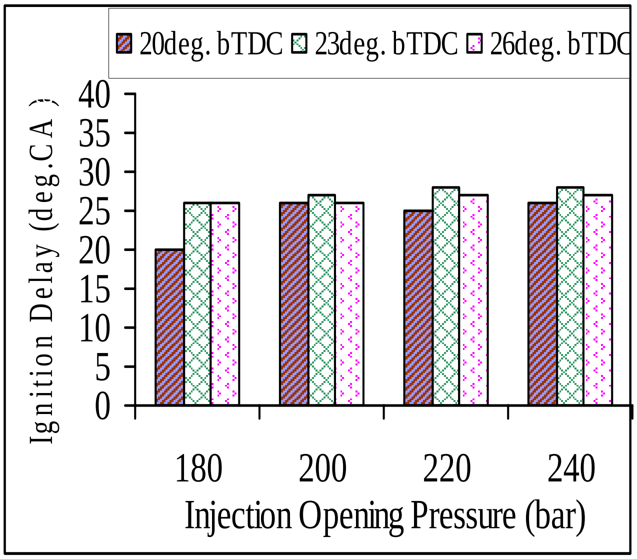

Figure 30 Variation of ignition delay with different IOPs of diesel fuel at full load conditions.

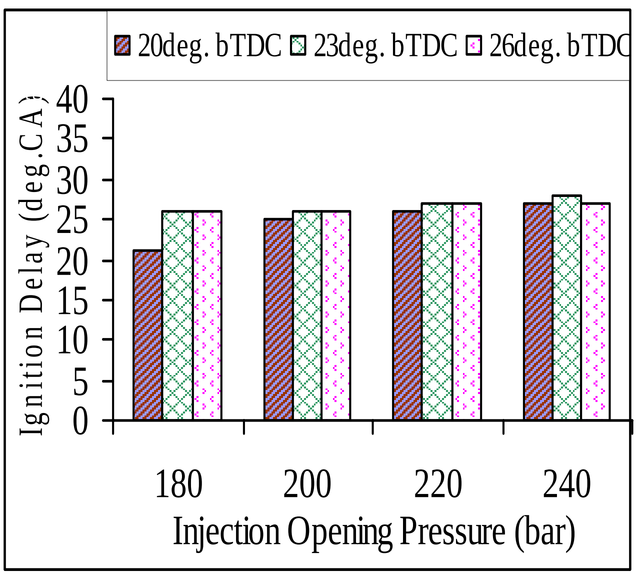

Figure 31 Variation of ignition delay with different IOPs of B20 fuel at full load conditions.

This may be due to higher combustion chamber wall temperature and reduced exhaust gas dilution at higher loads. Ignition delay for all injection timing tried was approximately same. At advanced injection timing 26deg. bTDC and IOP 220bar, ignition delay of blend B20, diesel and B100 are 27, 27 and 26deg. CA respectively for full load operation.

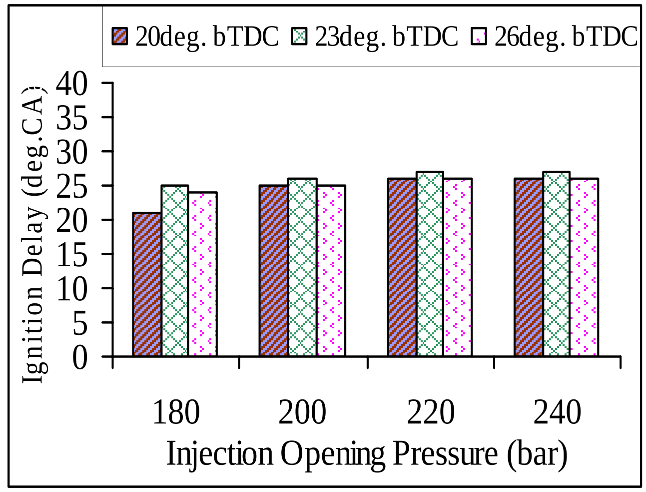

Figure 32 Variation of ignition delay with different IOPs of B100 fuel at full load conditions.

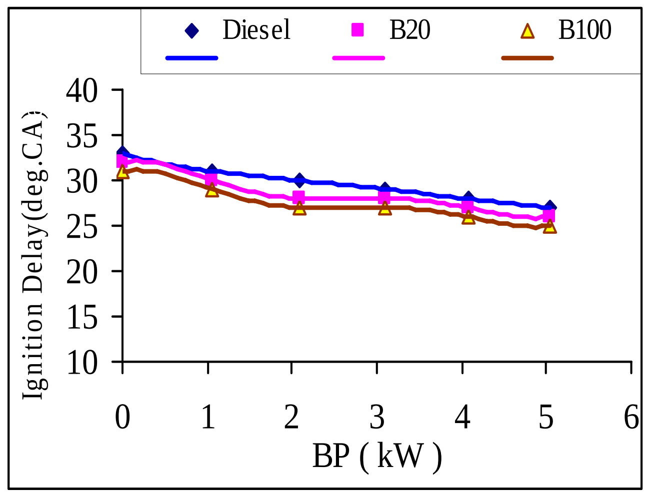

Figure 33 Variation of Ignition delay with BP of diesel, B20 and B100 fuel at IOP 220bar and IT 26deg. bTDC.

4. CONCLUSIONS

The experimental investigation was conducted to explore the performance of B100 and its blend B20 with diesel fuel of varying IOPs and ITs in direct injection compression ignition engine and the results suggest the following conclusions:

Increasing the IOP from 200bar to 220bar and IT from 23deg. bTDC to 26deg. bTDC resulted in a significant improvement in the performance, combustion and emissions of B100 and its blend B20 with diesel due to better spray formation, heat utilization and combustion in premixed part. The following changes were observed at IOP 220bar and 26deg. bTDC full load conditions are:

- The BTE of blend B20 fueled diesel engine has increased by 1.01% at IOP 220bar and 1.34% at IT 26deg. bTDC and the BSEC of decreased by 4.0% when operated at 220bar and 5.5% when operated at IT 26deg. bTDC respectively.

- The emissions such as smoke opacity of blend B20 fuel decreased by 21.0% for IOP 220bar and 49.0% for IT 26deg. bTDC and the UBHC decreased by 9.0% when operated IOP 220bar and 11.0% for IT 26deg. bTDC respectively.

- The NOx emission of blend B20 and B100 fuel increases 4.5% and 5.0% compared to diesel fuel at IOP 220 bar and 5.6% and 6.0% for IT 26deg. bTDC respectively.

- Peak cylinder pressure and corresponding peak heat release rate was higher in both B100 and its blend B20 compared to diesel fuel in both IOP 220bar and IT 26deg. bTDC.

- Ignition delay of B100 and its blend B20 fuel was marginally shorter than that of diesel fuel in both IOP 220bar and IT 26deg. bTDC.

REFERENCES

Anjana Srivastava, Ram Prasad., 2000. Triglycerides -based diesel fuels, Renewable and Sustainable Energy, Vol. 4, pp. 111-133.

Foidl N., Foidl G., Sanchez M., Mittelbach M., Hackel S., 1996. Jatropha Curcas L. as a source for the production of bio-fuel in Nicaragua, Bioresource Technology, Vol. 58, pp. 77-82.

Hammetien N., Korte V., Richter H., 1991. Performance, Emission and Durability of Modern diesel engine running on Rapeseed oil, SAE paper 910848.

Kato T., Tsujimura K., Shintani M., Minami T., 1989. Spray Characteristics and Combustion Improvement of direct injection Diesel Engine with High Pressure fuel Injection, SAE paper 890265.

Masjuki H., Sohif M., 1991, Performance evaluation of palm oil diesel blends on small engine, J. Energy Heat Mass Transfer, Vol. 13, pp. 125-133.

Marshall W., Schumacher L., Howell S., 1995. Engine Exhaust Emissions Evaluation of Cummins L10E When Fueled with a Biodiesel Blend, SAE paper 952363.

Parmanik K., 2003. Properties and uses of Jatropha Curcas oil and diesel fuel blends in Compression Ignition Engine, Renewable Energy, Vol. 28, pp. 239-248.

Palaniswamy E., Manoharan N., 2005. Performance Studies on vegetable oils and their derivatives as alternative fuels for compression ignition engines, Proceedings of the 19th National Conference on Internal Combustion Engines and combustion, India, An over view, pp. 95-100.

Ramadhas A.S., Jayaraj S.M., 2004. Use of Vegetable oils as Internal Combustion Engines fuels-a Review, Renewable Energy, Vol. 29, pp. 727-747.

Senthil Kumar M., Ramesh A., Nagalingam B., 2003. An Experimental Comparison of Methods to use Methanol and Jatropha oil in Compression Ignition Engine, Biomass and Bioenergy, Vol. 25, pp. 309-318.

Samaga B.S., 1983. Vegetable Oils as Alternative fuels for Compression Ignition engine, Eighth National Conference on Internal Combustion engines, Trivandram, India, pp B10 to B-12.

Srinivasa Rao P., Gopalakrishnan V. K., 1991. Vegetable oils and their methyl esters as fuels for diesel engines, J. Energy Heat Mass Transfer, Vol. 29, pp. 293-297.

Scholl K.W., Sorenson S.C., 1993, Combustions of soybean oil methyl ester in a direct injection diesel engine, SAE paper 930934.