PERFORMANCE, EMISSION AND COMBUSTION ANALYSIS OF C.I ENGINE USING SEA LEMON OIL AS A BIODIESEL WITH DIFFERENT INJECTION TIMING AND INJECTION PRESSURE

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ABSTRACT

The performance, combustion and emissions from a compression ignition engine operated with different injector pressure (IOP) and injection timing (IT) with Sea lemon oil methyl ester(SLOME) and Diethyl ether. The engine was run on three different IP 150,170,190, 230 and 250bar 19deg, 21deg, 23deg, 25deg, 27deg bTDC along with 210 bar The biodiesel was prepared from neat oils and blends prepared with diesel. The different blends were tested for their use as a substitute fuel for diesel in a single cylinder diesel engine at varying loads. The performance, combustion and emission characteristics of B50 blend fueled direct injection compression ignition engine performed better for entire load range of operation. BTE of blend B50 fueled compression ignition engine has increased when operated with IOP 230bar. On other hand blend B50 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 and diesel fuel. A single-cylinder, water-cooled, constant speed direct injection diesel engine was used for experiments. HC, NOx, CO, and smoke of the exhaust gas were measured to estimate the emissions. Various engine performance parameters such as thermal efficiency, and brake specific fuel consumption were calculated from the acquired data The 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. Fuel consumption was increased with increase in blend proportions. It is found that the emission level of CO and HC decreased with increased in proportion in diesel fuel.

Keywords: *Sea lemon oil methyl ester, Injection pressure, Injection timing, Diesel engine, Biodiesel, Diethyl ether*

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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 [15,17,31]. 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 engine[3,7,9].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. Sea lemon 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[2,5,14].The biodiesel derived from several feed stock have also been investigated extensively[22,26,33]. 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 [1,8,12]. 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[28,35,38]. Addition of oxygenates and dual fuelling lead to high brake thermal efficiency and also reduction in HC and CO emissions in some cases [4,6,37]. 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[34]. In view of this, vegetable oil is a promising alternative because it has several advantages[23]. 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 [10,13,25]. 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[11,24]. The esters of fatty acids derived from transestrification of vegetable oils have properties closer to petroleum diesel fuel[29]. 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[12]. 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[30,36]. 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. [6,10]. Performed the investigation based on his experimental findings; the esters of vegetable oils can be used directly in existing engines without modifications[16]. 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 [1,9,11].Reported that use of biodiesel in conventional diesel engine results in substantial reduction of unburned HC, CO and particulate matters with lower NOx emissions[21,27].

2. EXPERMENTAL WORK

2.1 Transestrification of Sealemon Oil

Widely used and accepted process to reduce the viscosity of triglycerides in vegetable oil is transesterification [2,6,19].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[18,20]. 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 sea lemon oil. A continuous stirring of the resulting mixture at temperature between 60°C-65°C 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 sea lemon 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 230bar injector opening pressure for no-load, 20% load, 40% load, 60% load, 80% load and full load for diesel fuel, B100 and its blend B50. 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. 150,170,190,210,230 and 250bar, the performance and emissions parameters were recorded as earlier. Similar experiments were conducted at other injection timing viz. 21deg. bTDC and 25deg. bTDC and performance and emissions parameters were recorded as earlier.

Table 1 Experimental Setup Specifications

Make	Kirloskar AV-1
Type	Single cylinder, water cooled,
Max. power	3.7 kW at 1500 rpm
Displacement	550 CC
Bore x Stroke	80 x 110 mm

Compression ratio	16.5:1
Fuel injection timing	21deg BTDC
Loading device	Eddy current dynamometer

Fig. 1 Experimental Setup

3. RESULTS AND DISCUSSIONS

3.1 Brake Thermal Efficiency

Figure 2, 3,4,5 and 6 illustrates the variation of brake thermal efficiency (BTE) with different IOPs with diesel, B50 and B100 fuel at full load condition and figure 5 illustrates the variation of BTE with BP of diesel,B25, B50, B75 and B100 fuel at injector opening pressure 230bar and injection timing 25deg. bTDC. It is observed that efficiency obtained at full load and part load of blend B50 fuel with injector opening pressure 230bar 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 B50, B75 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 230bar and blend B50 which was selected as optimal injection pressure. Further, increase in the injector opening pressure beyond 230bar to 250bar 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.

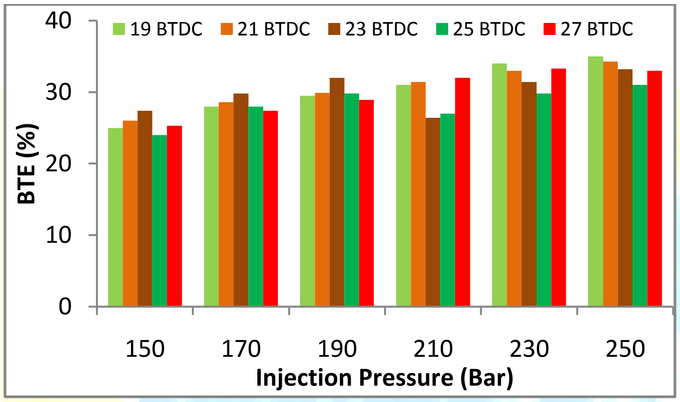

Figure 2 BTE with diesel

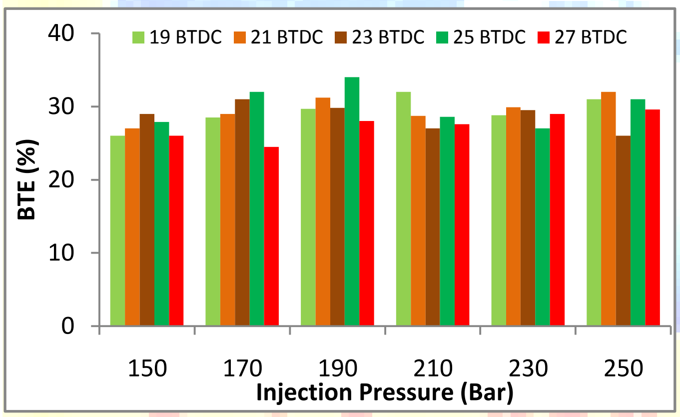

Figure 3 BTE with B50.

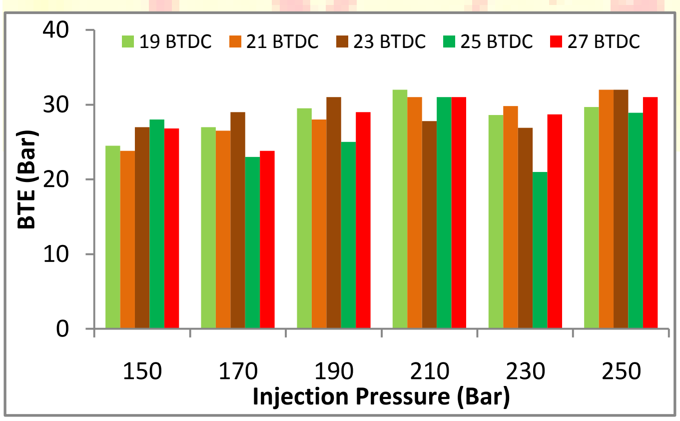

Figure 4 BTE with B75.

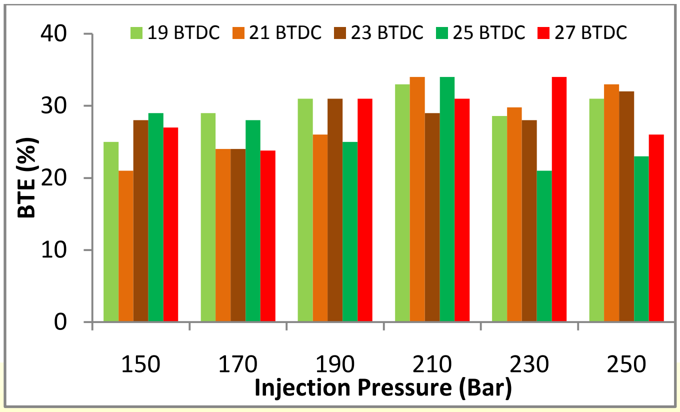

Figure 5 BTE with B100.

It is observed that BTE of retard injection timing i.e. 21deg. bTDC is lower than the other injection timings. But with advance injection timing, the BTE of diesel fuel, blend B25 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 25deg. bTDC and blend B50 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 B50 fuel showed better results than at designed injection timing and retard injection timing. At this injection timing with IOP 230bar, the brake thermal efficiency of blend B50, diesel and B100 was 29.99%, 28.89% and 28.25% respectively for full load operations.

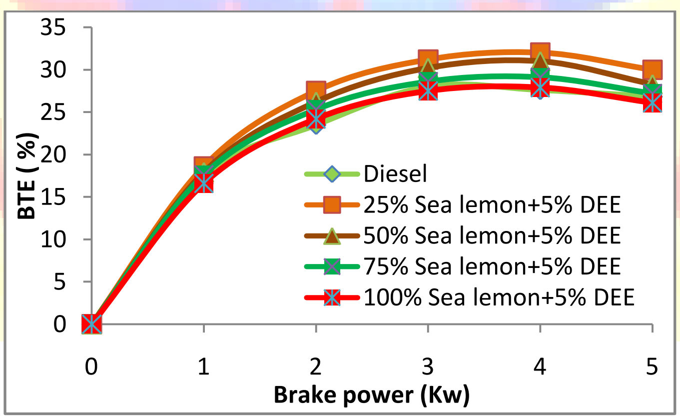

Figure 6 BTE with BP.

3.2 Brake Specific Energy Consumption

Figure 7, 8,9,10 and 11 illustrates the variation of BSEC with different IOPs of diesel,B25,B50,B75 and B100 fuel at full load condition and figure 9 illustrates the variation of BSEC with BP of diesel, B50 and B100 fuel at injector opening pressure 230bar and injection timing 25deg. bTDC. The BSEC for B100 fuels is higher than diesel and blend B50 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 230bar with blend B50 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 230bar, 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, B50 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 B50 fuel than B100 and diesel fuel. This may be due to complete combustion of fuel due to sufficient duration. At advanced injection timing 25deg. bTDC and IOP 230bar, the BSEC of blend B50, diesel and B100 are 11.80, 12.25 and 12.82MJ/kW-hr respectively for full load operations.

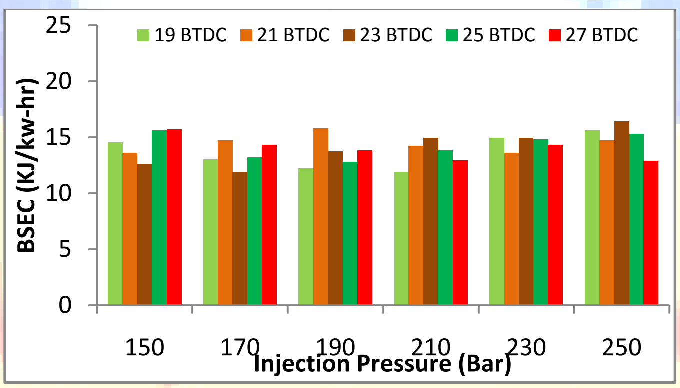

Figure 7 BSEC with of diesel.

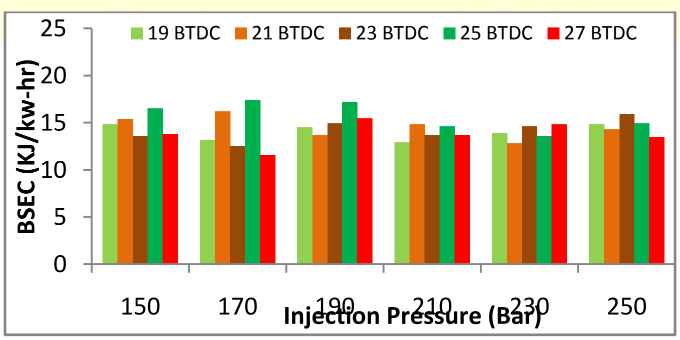

Figure 8 BSEC with B50.

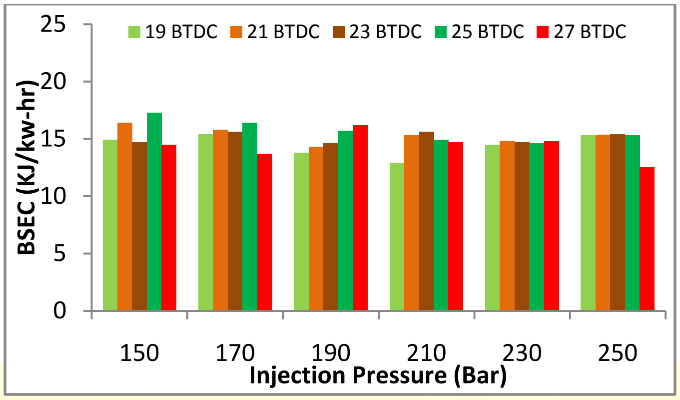

Figure 9 BSEC with B75.

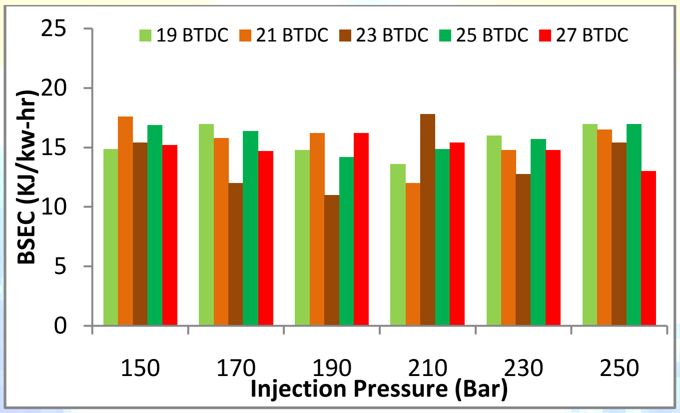

Figure 10 BSEC with B100.

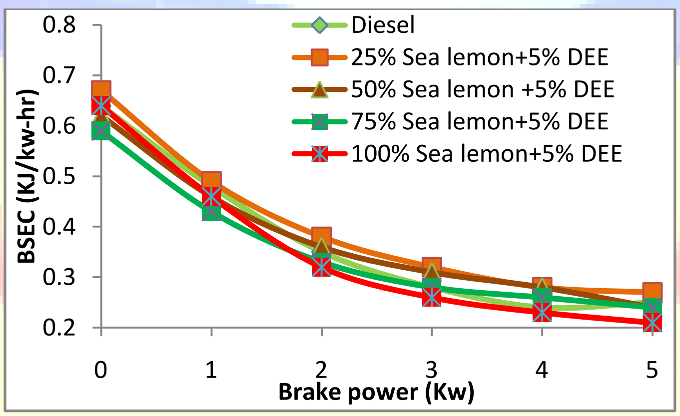

Figure 11 BSEC with BP

3.3 Unburned Hydrocarbon

Figure 12,13,14,15 and 16 illustrates the variation of UBHC with different IOPs of diesel, B25, B50, B75 and B100 fuel at full load condition and figure 13 illustrates the variation of UBHC with BP of diesel, B50 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 170bar and 250bar, the UBHC emissions are exceedingly higher compared to 210bar and 230bar. 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 230bar, the B50 fuel showed significant reduction in UBHC emissions. The Improved performance was observed at IOP 230bar with B50 fuel, though they reasonably high viscosity and lower cetane number.

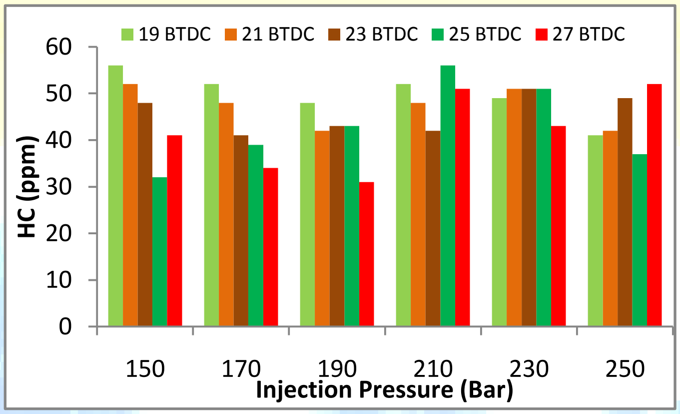

Figure 12 UBHC with diesel.

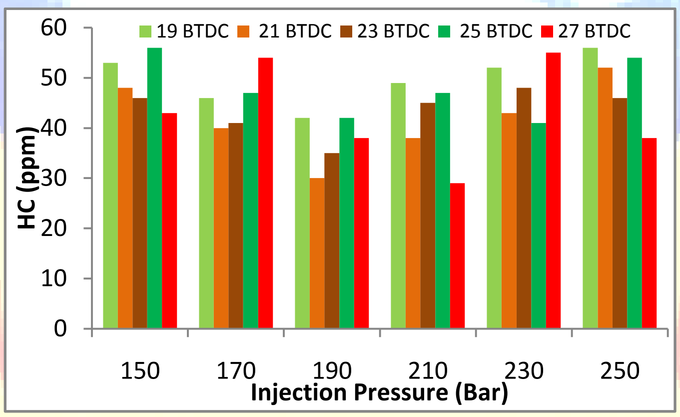

Figure 13 UBHC with B50

At advanced injection timing 25deg. 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 25deg. bTDC and IOP 230bar, UBHC of blend B50, diesel and B100 are 32, 36 and 41ppm respectively for full load operations.

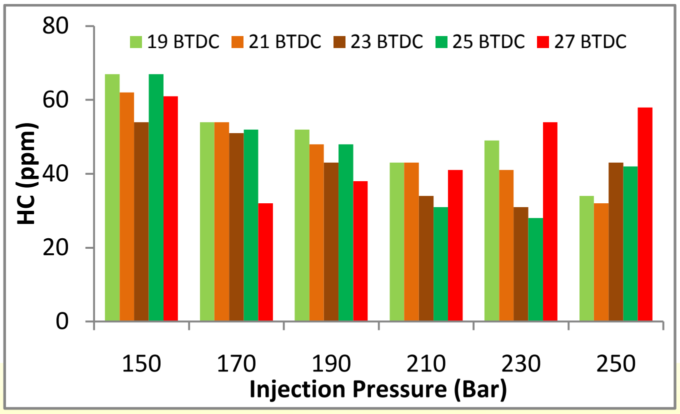

Figure 14 UBHC with B75.

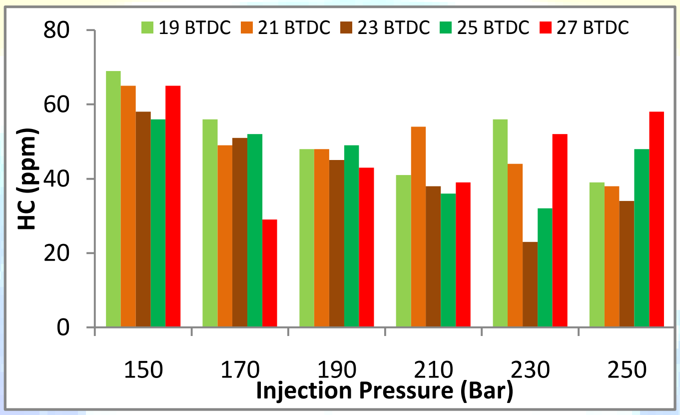

Figure 15 UBHC with B100.

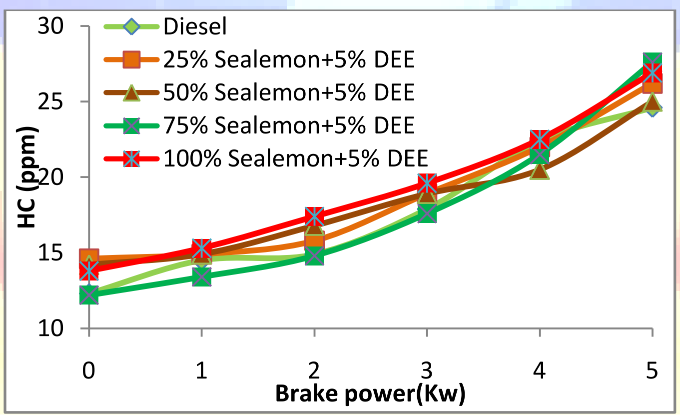

Figure 16 UBHC with BP

3.4 Smoke Opacity

Figure 17,18,19,20 and 21 illustrates the variation of smoke opacity with different IOPs of diesel, B25,B50,B75 and B100 fuel at full load condition and figure 17 illustrates the variation of smoke opacity with BP of diesel, B50 and B100 fuel at injector opening pressure 230bar and injection timing 25deg. bTDC. It is noticeable that smoke opacity is marginally affected by the change in

IOPs. The smoke opacity is marginally lower for blend B25 with IOP 230bar compared to B100 and diesel fuel. This may be due to two main reasons; firstly, the thermal efficiency which is higher for blend B25 fuel represents a better and complete combustion.

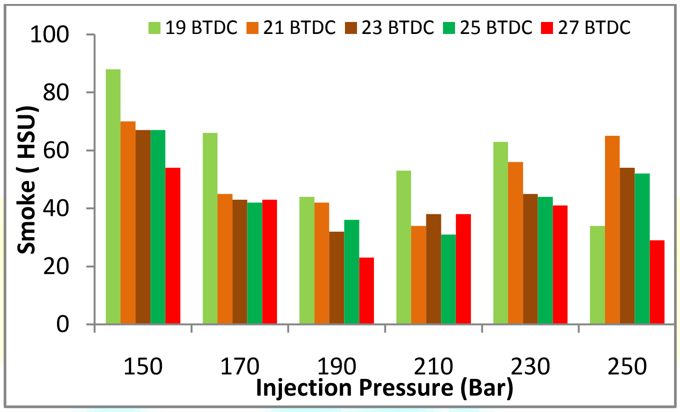

Figure 17 Smoke opacity with diesel.

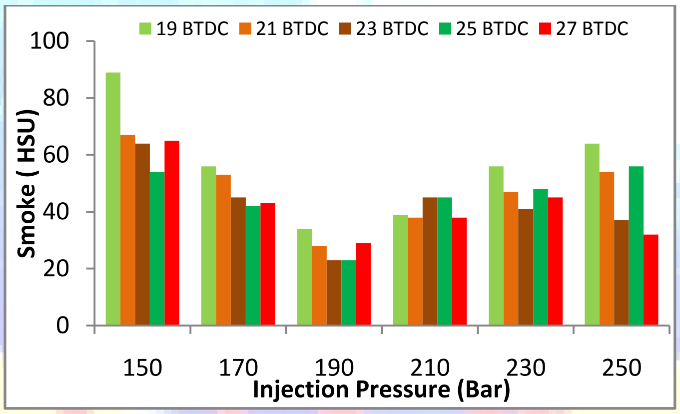

Figure 18 Smoke opacity with B50.

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, B25 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 B50 fuel, smoke opacity was lower compared to diesel fuel and B100 due to lower fuel consumption compared to other fuels. At advanced injection timing25deg. bTDC and IOP 230bar, smoke opacity of blend B50, diesel and B100 18.4, 30.7 and 43.8% respectively for full load operation.

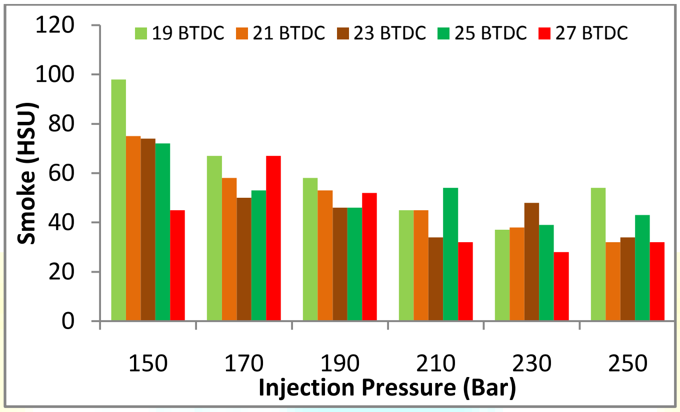

Figure 19 Smoke opacity with B75.

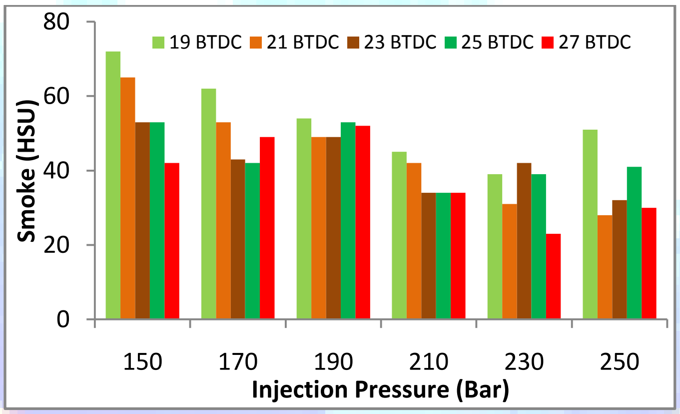

Figure 20 Smoke opacity with B100.

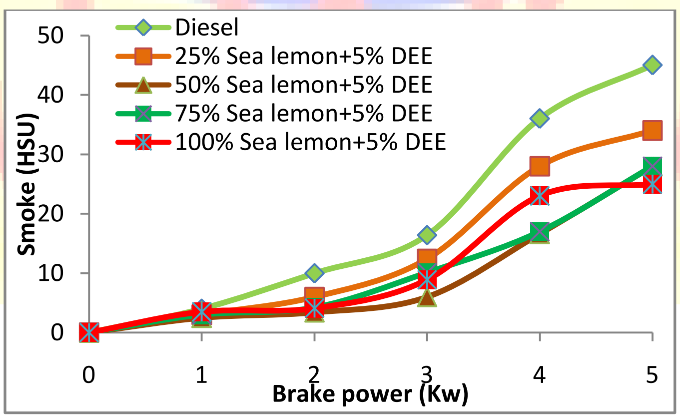

Figure 21 Smoke opacity with BP

3.5 Oxides of Nitrogen

Figure 22,23,24,25 and 26 illustrates the variation of NOx with different IOPs of diesel, B25,B50,B75 and B100 fuel at full load condition and figure 22 illustrates the variation of NOx with BP of diesel, B50 and B100 fuel at injector opening pressure 230bar and injection timing 25deg. 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.

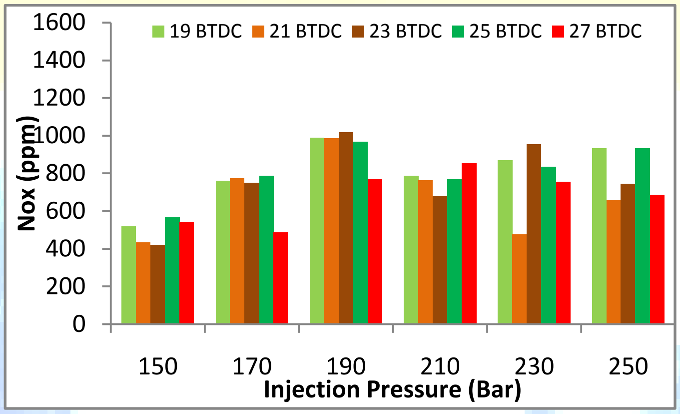

Figure 22 NOx with diesel.

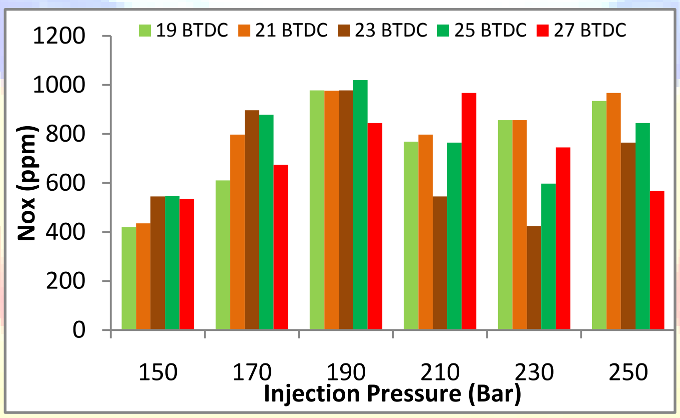

Figure 23 NOx with B50.

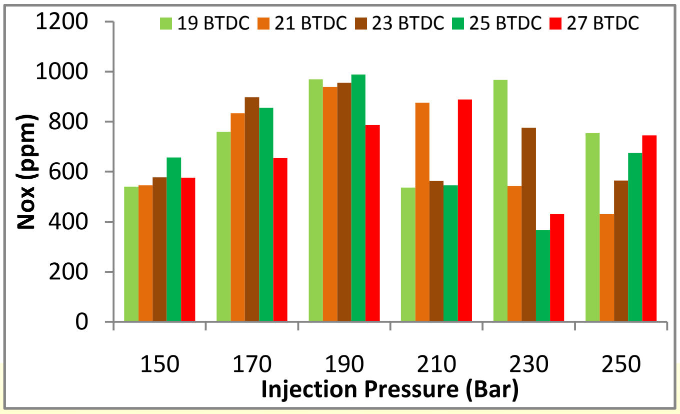

Figure 24 NOx with B75.

At advanced injection timing 25deg. bTDC with B100 and B25, B50 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 25deg. bTDC and IOP 230 bar, the NOx of blend B50, diesel and B100 are 1112, 1055 and 1212ppm respectively.

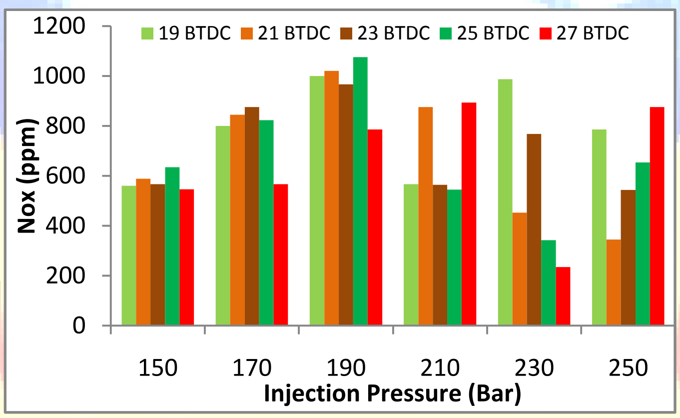

Figure 25 NOx with B100.

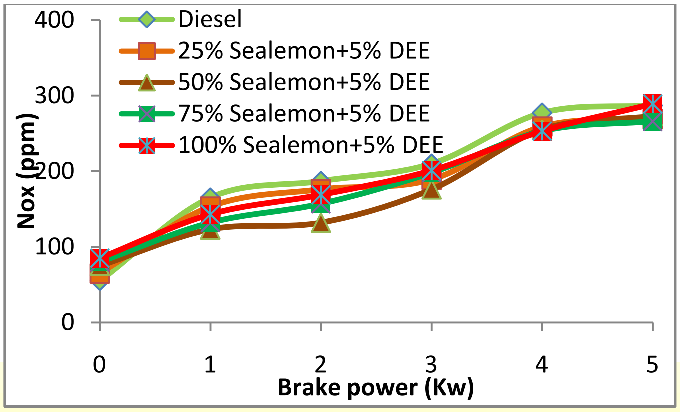

Figure 26 NOx with BP.

3.6 Peak Cylinder Pressure

Figure 27, 28,29,30 and 31 illustrates the variation of peak cylinder pressure with different IOPs of diesel, B50 and B100 fuel at full load condition and figure 27 illustrates the variation of peak cylinder pressure with BP of diesel, B25,B50,B75 and B100 fuel at injector opening pressure 230bar and injection timing 25deg. bTDC. The highest peak cylinder pressure was recorded with IOP 230bar 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 B50. 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 25deg. bTDC and IOP 230bar, peak cylinder pressure of blend B50, diesel and B100 are 70.92, 70.25 and 72.04bar respectively for full load operation.

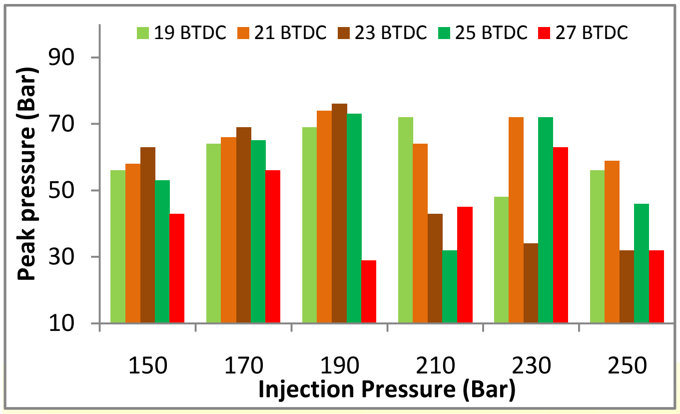

Figure 27 Peak pressure with diesel.

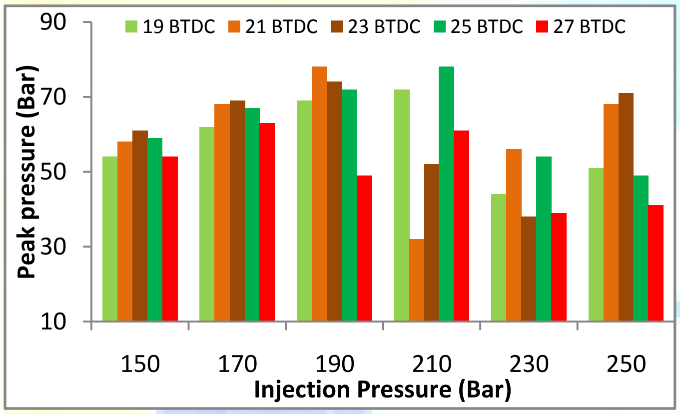

Figure 28 Peak pressure with B50 .

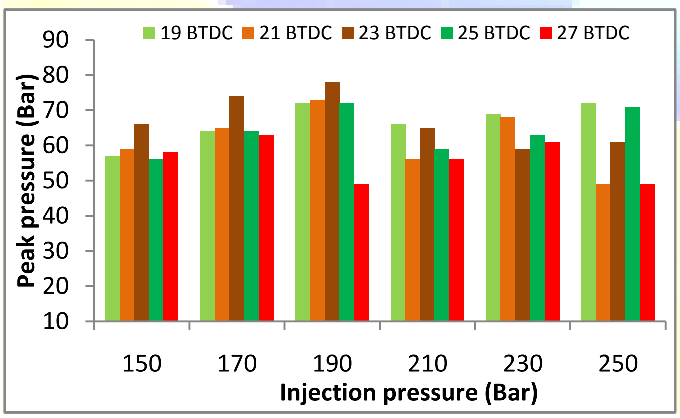

Figure 29 Peak pressure with B75.

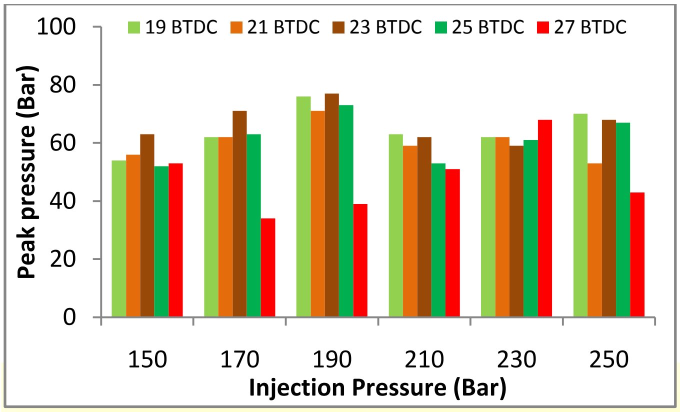

Figure 30 Peak pressure with B100 .

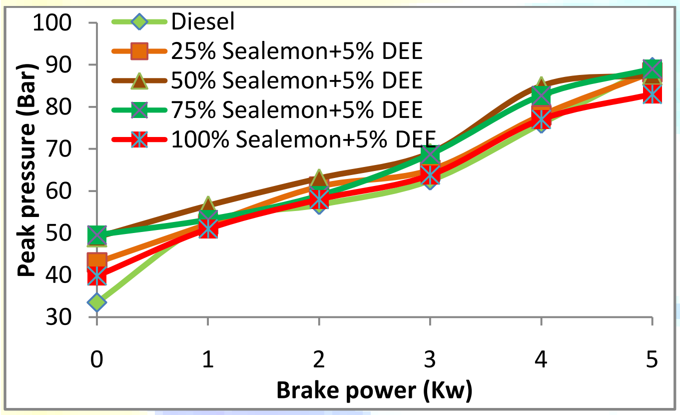

Figure 31 Peak pressure with BP

3.7 Heat Release Rate

Figure 32,33,34,35 and 36 illustrates the variation of heat release rate with different IOPs of diesel, B25,B50,B75 and B100 fuel at full load condition and figure 32 illustrates the variation of peak heat release rate with BP of diesel, B50 and B100 fuel at injector opening pressure 230bar and injection timing 25deg. bTDC. The peak heat release rate and the peak combustion temperature is correspondingly better for fuel with blend B50 compared to B100 and diesel fuel at injector opening pressure 230bar compared to other injector opening pressures. Hence, looking at all the combustion, performance and emission characteristics, the blend B50 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 230bar.

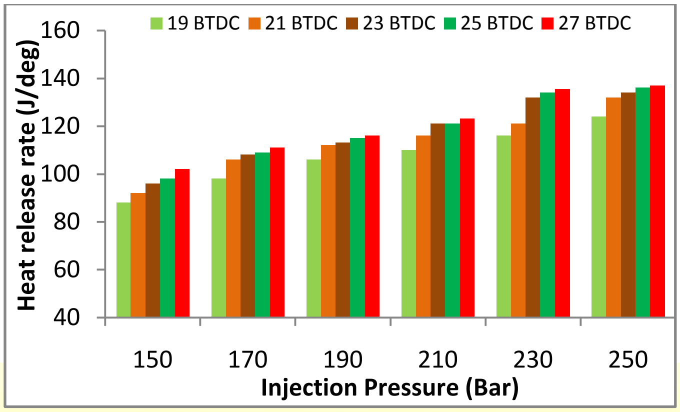

Figure 32 Heat release rate with diesel.

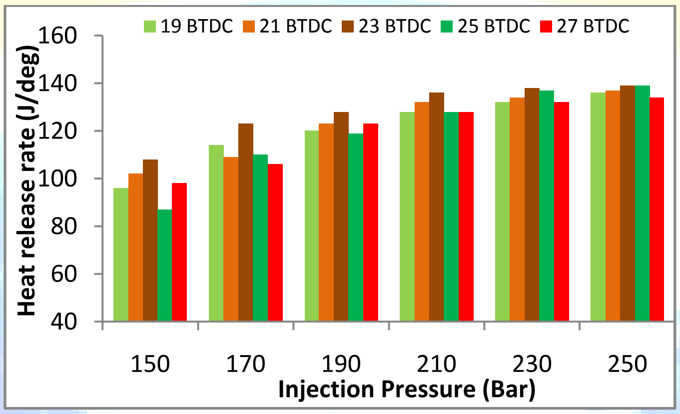

Figure 33 Heat release rate with B50.

The peak heat release rate with B100 is always lower than blend B50 and diesel fuel. At advanced injection timing 25deg. bTDC and IOP 230bar, peak heat release rate of blend B50, diesel and B100 are 131.88, 124.24 and 112.72J/deg. CA respectively for full load operation.

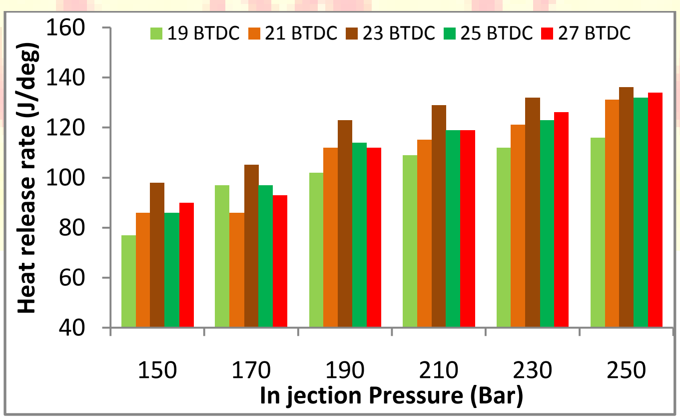

Figure 34 Heat release rate with B75.

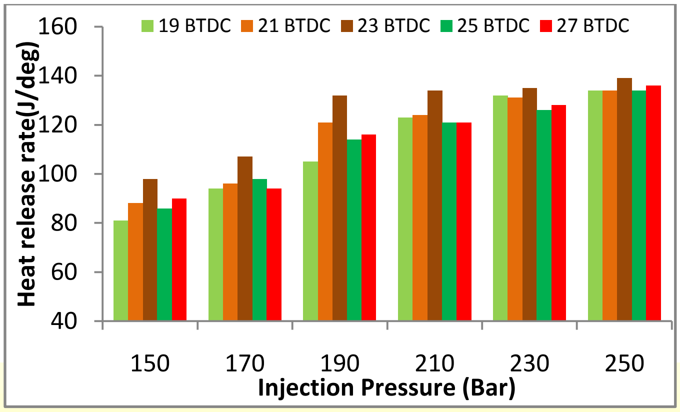

Figure 35 Heat release rate with B100.

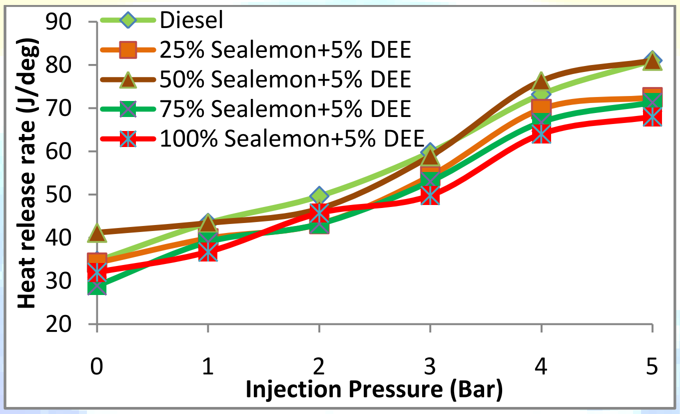

Figure 36 Heat release rate with BP.

3.8 Ignition Delay

Figure 36,37,38,39 and 40 illustrates the variation of ignition delay with different IOPs of diesel, B25, B50, B75 and B100 fuel at full load condition and figure 33 illustrates the variation of ignition delay with BP of diesel, B25, B50 and B100 fuel at injector opening pressure 230bar and injection timing 25deg. bTDC. It was observed that ignition delay of B100 and its blend B50 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.

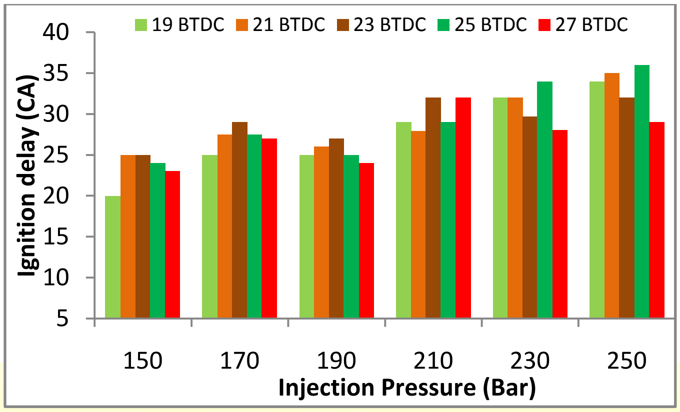

Figure 37 Ignition delay with injection pressure Diesel.

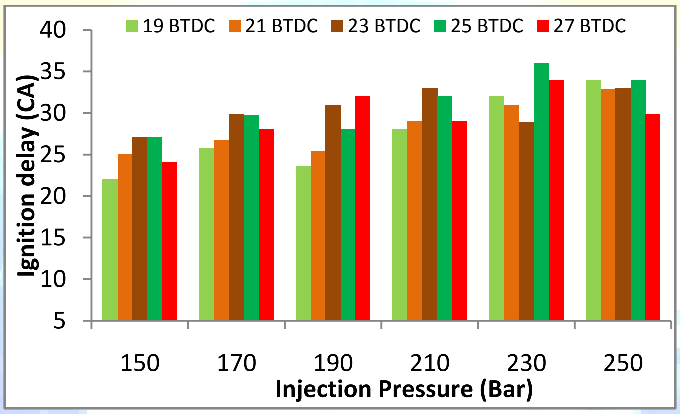

Figure 38 Ignition delay with Injection pressure B50.

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 25deg. bTDC and IOP 230bar, ignition delay of blend B50, diesel and B100 are 21, 23 and 25deg. CA respectively for full load operation

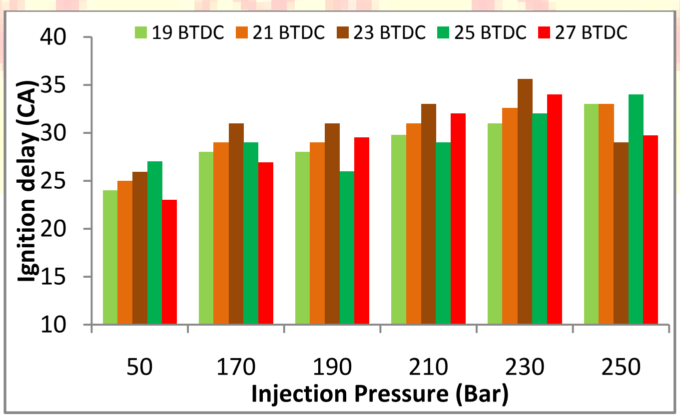

Figure 39 Ignition delay Vs injection pressure B75

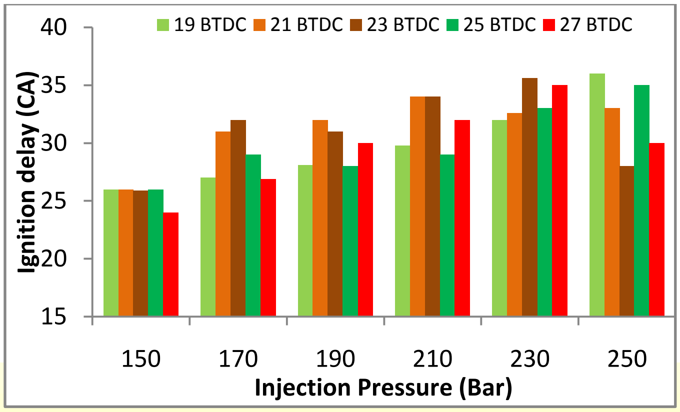

Figure 40 Ignition delay Vs injection pressure B100

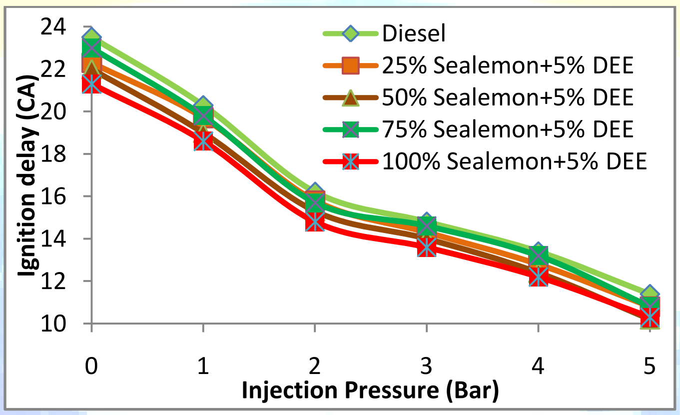

Figure 41 Ignition delay with injection pressure.

3.9 Brake specific fuel consumption

The curve shows that, Bsfc for biodiesel blends is higher at low 50% load. And it decreases with the increase in 100% load, it is also observed from the curve that, specific fuel consumption increase with the increase in biodiesel blend. This is mainly due to the relationship among volumetric fuel injection system, fuel specific gravity, viscosity and heating value. As a result more biodiesel blend is needed to produce the same amount of energy due to its higher density and lower heating value in comparison to conventional diesel fuel. Again as biodiesel blends have different viscosity than diesel fuel, so biodiesel causes poor atomization and mixture formation and thus increases the fuel consumption rate to maintain the power.

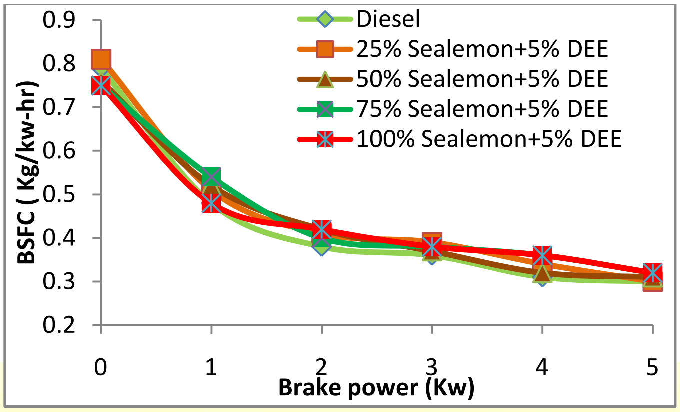

Figure 42 BSFC with Brake power B100.

4.0 Indicated thermal efficiency

The variation of Indicated thermal efficiency with load for sea lemon oil addition DF blends is shown in figure 43.It can be observed that the thermal efficiency is 22.44% at full load for diesel. Because of the increase in indicated power, the Indicated thermal efficiencies of sea lemon oil -DF blends are low particularly at full load. It can be inferred from the graph that indicated thermal efficiency (which is the ratio of indicated power to the heat energy supplied by the fuel during combustion) of B50 mixture shows a lead than diesel and blends of biodiesel. Indicated thermal efficiency depends on both brake power produced by the engine as well as frictional power offered by the engine, as it is the summation of both. Diesel curve leads all the blends of biodiesel as load applied on an engine increases, except B50. Diesel curve also shows a smooth linear curvature. Thus B50 has shown competent and satisfactory performance characteristics results when compared mainly with petroleum-diesel and other blends of biodiesel.

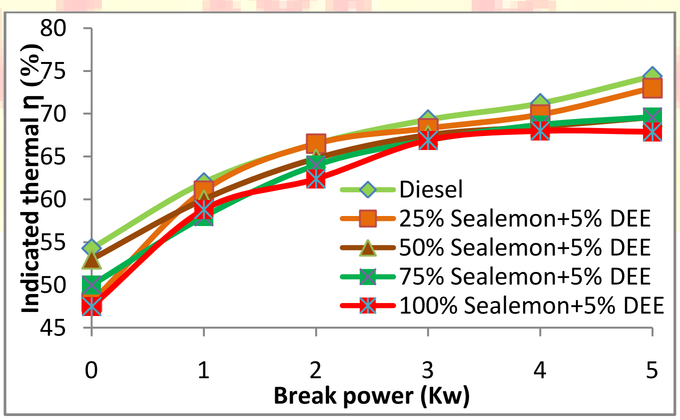

Figure 43 Indicated thermal η with Brake power B100

4.1 Mechanical efficiency

From the figure 44, the variation of mechanical efficiency consumption with brake power can be observed. The values of mechanical efficiency are higher compared to DF with 57.63%. As the engine produces power output the frictional losses are reduced and hence increase in the mechanical efficiency. It can be studied from the graph of η_{mech} vs. B.P, that the mechanical efficiency (the ratio of B.H.P to I.H.P) of biodiesel and petro-diesel follows a smooth linear curvature, and almost crosses near each other for a same brake power. B50 leads all other blends of biodiesel and shows a higher hand to others, even though the difference among them for a given brake power is less appreciable.

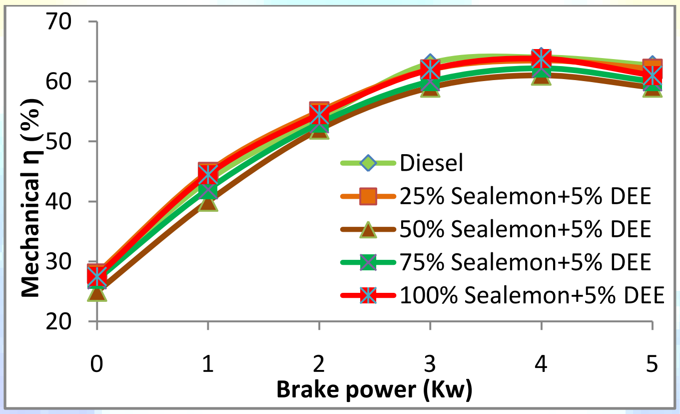

Figure 44 Mechanical η with Brake power B100

4. CONCLUSIONS

- Increasing the IOP from 210bar to 230bar and IT from 23deg. bTDC to 25deg. bTDC resulted in a significant improvement in the performance, combustion and emissions of B100 and its blend B25 with diesel due to better spray formation, heat utilization and combustion in premixed part.
- The BTE of blend B50 fueled diesel engine has increased by 1.01% at IOP 230bar and 1.34% at IT 25deg. bTDC and the BSEC of decreased by 4.0% when operated at 230bar and 5.5% when operated at IT 25deg. bTDC respectively.
- The emissions such as smoke opacity of blend B50 fuel decreased by 21.0% for IOP 230bar and 49.0% for IT 25deg. bTDC and the UBHC decreased by 9.0% when operated IOP 230bar and 11.0% for IT 25deg. bTDC respectively.
- The NOx emission of blend B50 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 25deg. bTDC respectively.

- Peak cylinder pressure and corresponding peak heat release rate was higher in both B100 and its blend B50 compared to diesel fuel in both IOP 230bar and IT 25deg. bTDC.
- Ignition delay of B100 and its blend B50 fuel was marginally shorter than that of diesel fuel in both IOP 230bar and IT 25deg. bTDC.

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