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Experimental Investigations on Exhaust Emissions of High Grade Low Heat Rejection Diesel Engine with Pongamia Biodiesel

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ABSTRACT

In the scenario of depletion of fossil fuels, the search for alternative fuels has become pertinent. Vegetable oils are promising substitutes for diesel fuels. Biodiesels derived from vegetable oils present a very promising alternative to diesel fuel since biodiesels have numerous advantages compared to fossil fuels as they are renewable, biodegradable, provide energy security and foreign exchange savings besides addressing environmental concerns and socio-economic issues. Experiments were conducted to determine exhaust emissions of a low heat rejection (LHR) diesel engine with different operating conditions [normal temperature and pre-heated temperature] of pongamia biodiesel with varied injection timing and injector opening pressure. LHR engine consisted of air gap insulated piston, air gap insulated liner and ceramic coated cylinder head. Exhaust emissions of particulate emissions and nitrogen oxide levels were determined at various values of brake mean effective pressure of the engine fuelled with biodiesel. Comparative studies on exhaust emissions were made with diesel working on similar conditions. Particulate emissions decreased while nitrogen oxide levels increased with biodiesel operation on LHR engine over conventional engine. They further improved with increase of injector opening pressure, advanced injection timing and preheating of biodiesel.

KEYWORDS: Vegetable oils, biodiesel, LHR combustion chambers, Fuel performance and exhaust emissions.

1. INTRODUCTION

The rapid depletion of petroleum fuels and their ever increasing costs have lead to an intensive search for alternate fuels. It has been found that the vegetable oils are promising substitute, because of their properties are similar to those of diesel fuel. They are renewable and can be easily produced. Rudolph Diesel, the inventor of the diesel engine that bears his name, experimented with fuels ranging from powdered coal to peanut oil [1].Several researchers experimented the use of vegetable oils as fuel on diesel engine and reported that the performance was poor, citing the problems of high viscosity, low volatility and their polyunsaturated character. [2–7]. The drawbacks associated with crude vegetable oil for use in diesel engine of high viscosity and low volatility were reduced to some extent, if crude vegetable oils are chemical converted into biodiesel. Experiments were conducted with biodiesel in conventional engine. [8–11].They reported that marginal improvement of performance and reduction of particulate emissions and increase of nitrogen oxide levels with biodiesel operation in comparison with diesel operation on conventional engine.

Experiments were conducted on preheated vegetable oils [temperature at which viscosity of the vegetable oils were matched to that of diesel fuel]. [12–15]. They reported that preheated vegetable oils decreased pollution levels of particulate emissions and NO_x emissions.

By controlling the injector opening pressure and the injection rate, the spray cone angle is found to depend on injector opening pressure [16]. Few investigators reported that injector opening pressure has a significance effect on the performance and formation of pollutants inside the direct injection diesel engine combustion.[17–21]. They reported that particulate emissions decreased with increase of injector opening pressure.

The other important engine variable to improve the performance of the engine is injection timing. Performance improved or deteriorated depending on whether injection timing was advanced (injection timing away from TDC) or retarded (injection timing towards TDC). Recommended injection timing was defined by the manufacturer that it is the timing at which maximum thermal efficiency was obtained with minimum pollution levels from the engine. Investigations were carried out on single cylinder water cooled vertical diesel engine with brake power 3.68 kW at a speed of 1500 rpm with varied injection timing from 27–34^o bTDC [21]. They reported reduction of particulate emissions with advanced injection timing.

The drawbacks associated with biodiesel (high viscosity and low volatility) call for hot combustion chamber, provided by low heat rejection (LHR) combustion chamber. The concept of the engine with LHR combustion chamber is reduce heat loss to the coolant with provision of thermal resistance in the path of heat flow to the coolant. Three approaches that are being pursued to decrease heat rejection are (1) Coating with low thermal conductivity materials on crown of the piston, inner

portion of the liner and cylinder head (low grade LHR combustion chamber or LHR-1 engine); (2) air gap insulation where air gap is provided in the piston and other components with low-thermal conductivity materials like superni (an alloy of nickel),cast iron and mild steel(medium grade LHR combustion chamber or LHR-2 engine);and (3).high grade LHR engine or LHR-3 engine contains air gap insulation and ceramic coated components.
Experiments were conducted on engine with high grade LHR combustion chamber with biodiesel. It consisted of an air gap (3 mm) insulation in piston as well as in liner and ceramic coated cylinder head. The engine was fuelled with biodiesel with varied injector opening pressure and injection timing [22–28]. They reported from their investigations, that engine with LHR combustion chamber at an optimum injection timing of 28^o bTDC with biodiesel increased brake thermal efficiency by 10–12%, at full load operation–decreased particulate emissions by 45–50% and increased NO_x levels, by 45–50% when compared with mineral diesel operation on CE at 27^o bTDC.
The present paper attempted to determine the exhaust emissions of the LHR engine. It contained an air gap (3.0 mm) insulated piston, an air gap (3.0 mm) insulated liner and ceramic coated cylinder head with pongamia biodiesel with different operating conditions with varied injection timing and injector opening pressure. Results were compared with CE with biodiesel and also with diesel at similar operating conditions.

2. MATERIALS AND METHODS

2.1 Preparation of biodiesel

Pongamia seeds have approximately 27% (w/w) oil content. Oil is obtained by crushing the seeds of plant. The chemical conversion of esterification reduced viscosity four fold. Crude Pongamia contains up to 70 % (wt.) free fatty acids. The methyl ester was produced by chemically reacting crude pongamia oil with methanol in the presence of a catalyst (KOH). A two–stage process was used for the esterification of the crude pongamia oil [10]. The first stage (acid-catalyzed) of the process is to reduce the free fatty acids (FFA) content in pongamia oil by esterification with methanol (99% pure) and acid catalyst (sulfuric acid-98% pure) in one hour time of reaction at 55°C. Molar ratio of pongamia oil to methanol was 9:1 and 0.75% catalyst (w/w). In the second stage (alkali-catalyzed), the triglyceride portion of the pongamia oil reacts with methanol and base catalyst (sodium hydroxide–99% pure), in one hour time of reaction at 65°C, to form methyl ester (biodiesel) and glycerol. To remove un–reacted methoxide present in raw methyl ester, it is purified by the process of water washing with air–bubbling. The properties of the Test Fuels used in the experiment were presented in Table–1.

Table.1 Properties of test fuels

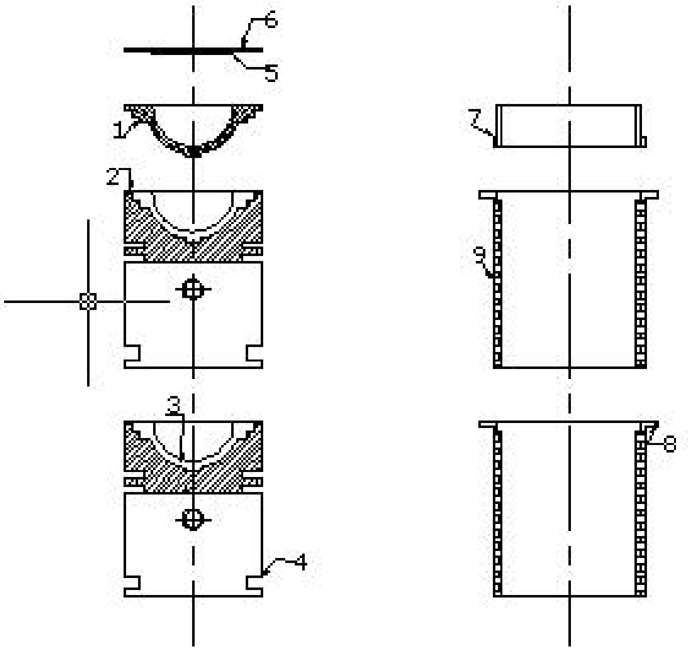
Property	Units	Diesel (DF)	Biodiesel(BD)	ASTM Standard
Carbon Chain	--	C_8–C_{28}	C_{16}–C_{24}	---
Cetane Number	-	51	60	ASTM D 613
Specific Gravity at 15°C	-	0.8275	0.88	ASTM D 4809
Bulk Modulus at 15°C	MPa	1408.3	1750	ASTM D 6793
Kinematic Viscosity @ 40°C	cSt	2.5	5.01	ASTM D 445
Air Fuel Ratio (Stoichiometric)	--	14.86	13.8	--
Flash Point (Pensky Marten's Closed Cup)	°C	120	165	ASTM D93
Cold Filter Plugging Point	°C	Winter 6° C Summer 18°C	180	ASTM D 6371
Pour Point	°C	Winter 3°C Summer 15°C	-2	ASTM D 97
Sulfur	(mg/kg, max)	50	---	ASTM D5453
Low Calorific Value	MJ/kg	42.0	42.2	ASTM D 7314

Oxygen Content	%	0.3	12	--

2.2 Engine with LHR combustion chamber

Fig.1 shows assembly details of insulated piston, insulated liner and ceramic coated cylinder head.

Engine with LHR combustion chamber contained a two–part piston ; the top crown made of superni was screwed to aluminium body of the piston, providing an air gap (3.0 mm) in between the crown and the body of the piston by placing a superni gasket in between the body and crown of the piston. A superni insert was screwed to the top portion of the liner in such a manner that an air gap of 3.2 mm was maintained between the insert and the liner body. At 500 ^{o}C the thermal conductivity of superni and air are 20.92 and 0.057 W/m–K. Partially stabilized zirconium (PSZ) of thickness 500 microns was coated by means of plasma coating technique. The combination of low thermal conductivity materials of air, superni and PSZ provide sufficient insulation for heat flow to the coolant, thus resulting in LHR combustion chamber

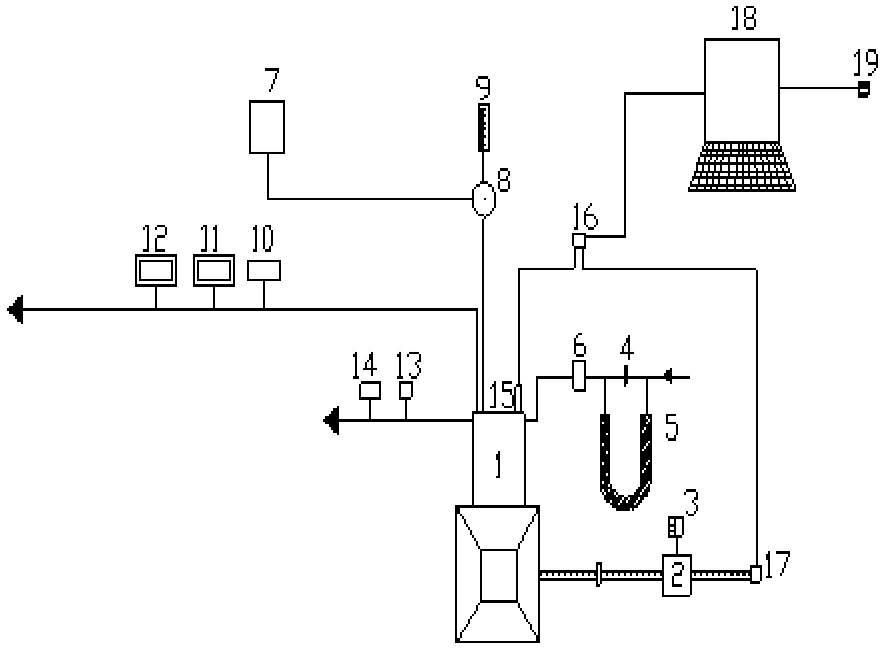


1.Supe 1. Piston crown with threads, 2. Superni gasket, 3. Air gap in piston, 4. Body of piston, 5. Ceramic coating on inside portion of cylinder head, 6. Cylinder head, 7.Superni insert with threads, 8.Air gap in liner, 9.Liner

Figure.1 Assembly details of air gap insulated piston, air gap insulated liner and ceramic coated cylinder head

2.3 Experimental set–up

The schematic diagram of the experimental setup used for the investigations on the engine with LHR combustion chamber with pongamia biodiesel is shown in Fig.2. Specifications of Test engine are given in Table2.The engine was coupled with an electric dynamometer (Kirloskar), which was loaded by a loading rheostat. The fuel rate was measured by Burette. The accuracy of brake thermal efficiency obtained is ±2%. Provision was made for preheating of biodiesel to the required levels (90^{o}C) so that its viscosity was equalized to that of diesel fuel at room temperature. Air-consumption of the engine was obtained with an aid of air box, orifice flow meter and U–tube water manometer assembly. The naturally aspirated engine was provided with water–cooling system in which outlet temperature of water was maintained at 80^{o}C by adjusting the water flow rate. The water flow rate was measured by means of analogue water flow meter, with accuracy of measurement of ±1%.



1.Four Stroke Kirloskar Diesel Engine, 2.Kirloskar Electical Dynamometer, 3.Load Box, 4.Orifice flow meter, 5.U-tube water manometer, 6.Air box, 7.Fuel tank, 8, Preheater 9.Burette, 10. Exhaust gas temperature indicator, 11.AVL Smoke opacity meter,12. Netel Chromatograph NO_xAnalyzer, 13.Outlet jacket water temperature indicator, 14. Outlet-jacket water flow meter,15.AVL Austria Piezo-electric pressure transducer, 16.Console, 17.AVL Austria TDC encoder, 18.Personal Computer and 19. Printer.

Fig.2 Schematic diagram of experimental set–up

Engine oil was provided with a pressure feed system. No temperature control was incorporated, for measuring the lube oil temperature. Copper shims of suitable size were provided in between the pump body and the engine frame, to vary the injection timing. Injector opening pressure was changed from 190 bar to 270 bar using nozzle testing device. The maximum injector opening pressure was restricted to 270 bar due to practical difficulties involved. Coolant water jacket inlet temperature, outlet water jacket temperature and exhaust gas temperature were measured by employing iron and iron-constantan thermocouples connected to analogue temperature indicators. The accuracies of analogue temperature indicators are ±1%.

Table.2 Specifications of Test Engine

Description	Specification
Engine make and model	Kirloskar (India) AV1
Maximum power output at a speed of 1500 rpm	3.68 kW
Number of cylinders ×cylinder position× stroke	One × Vertical position × four-stroke
Bore × stroke	80 mm × 110 mm
Engine Displacement	553 cc
Method of cooling	Water cooled
Rated speed (constant)	1500 rpm
Fuel injection system	In-line and direct injection
Compression ratio	16:1
BMEP @ 1500 rpm at full load	5.31 bar
Manufacturer’s recommended injection timing and injector opening pressure	27^{o}bTDC × 190 bar

Number of holes of injector and size	Three × 0.25 mm
Type of combustion chamber	Direct injection type

Exhaust emissions of particulate matter and nitrogen oxides (NO_x) were recorded by smoke opacity meter (AVL India, 437) and NO_x Analyzer (Netel India;4000 VM) at full load operation of the engine. Table 3 shows the measurement principle, accuracy and repeatability of raw exhaust gas emission analyzers/ measuring equipment for particulate emissions and NO_x levels. Analyzers were allowed to adjust their zero point before each measurement. To ensure that accuracy of measured values was high, the gas analyzers were calibrated before each measurement using reference gases.

Table.3 Specifications of the Smoke Opacimeter (AVL, India, 437). And NO_x Analyzer (Netel India ;4000 VM))

Pollutant	Measuring Principle	Range	Least Count	Repeatability
Particulate Emissions	Light extinction	1–100%	1% of Full Scale (FS)	0.1% for 30 minutes
NO_x	Chemiluminiscence	1–5000 ppm	0.5 % F.S	≤0.5% F.S

2.4 Test conditions

Test fuels used in the experiment were neat diesel and biodiesel. Various configurations of the engine were conventional engine and engine with LHR combustion chamber. Different operating conditions of the biodiesel were normal temperature and preheated temperature. Different injector opening pressures attempted in this experiment were 190and 270 bar. Various injection timings attempted in the investigations were manufacturer's recommended injection timing (27^o bTDC) and optimum injection timing.. Each test was repeated twelve times to ensure the reproducibility of data according to uncertainity analysis(Minimum number of trials must be not less than ten).

3. RESULTS AND DISCUSSION

3.1 Performance parameters

The optimum injection timing with CE was 31^{o}bTDC, while it was 28^{o}bTDC for engine with LHR combustion chamber with diesel operation [29]. Fig.3 shows variation of brake thermal efficiency with brake mean effective pressure (BMEP) in conventional engine with biodiesel at various injection timings. BTE increased up to 80% of the full load and beyond that load, it decreased with biodiesel operation at various injection timings. Increase of fuel conversion of efficiency up to 80% of full load and decease of mechanical efficiency and volumetric efficiency beyond 80% of the full load and were the responsible factors for variation of BTE with respect to BMEP. Curves in Fig.3 indicate that CE with biodiesel at 27^{o}bTDC showed comparable performance at all loads.

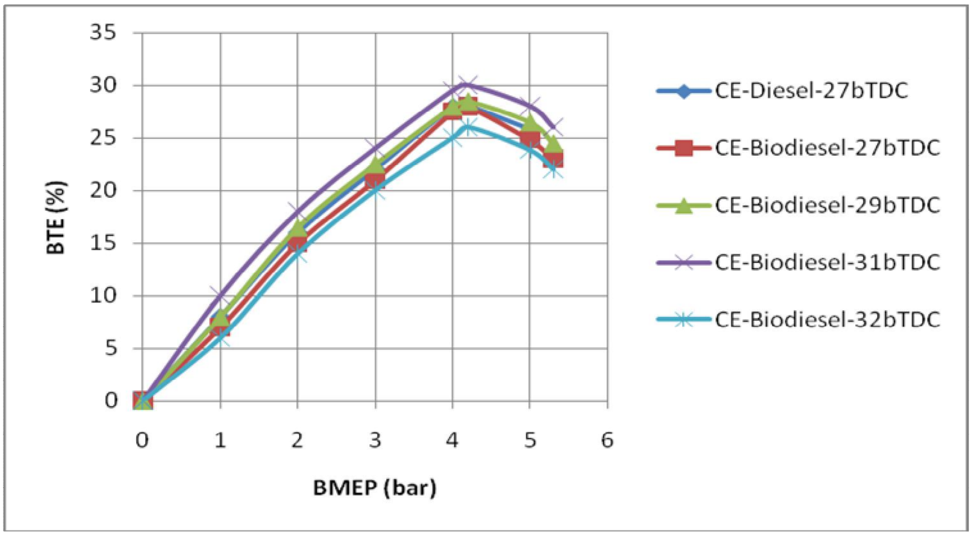


Figure. 3 Variation of brake thermal efficiency (BTE) with brake mean effective pressure (BMEP) in conventional engine (CE) with biodiesel at various injection timings at an injector opening pressure of 190 bar.

The presence of oxygen in fuel composition might have improved performance with biodiesel operation, when compared with mineral diesel operation on CE at 27^o bTDC. CE with biodiesel operation at 27^{o}bTDC decreased peak BTE by 3%, when compared with diesel operation on CE. Low calorific value and high viscosity of biodiesel might have showed comparable performance with biodiesel operation in comparison with neat diesel. CE with biodiesel operation increased BTE at all loads with advanced injection timing, when compared with CE with diesel operation at 27^o bTDC. Initiation of combustion at early period and increase of contact period of fuel with air improved performance with biodiesel when compared with diesel operation at 27^o bTDC. CE with biodiesel operation increased peak BTE by 3% at an optimum injection timing of 31^o bTDC, when compared with diesel operation at 27^o bTDC.
Fig.4 shows variation of brake thermal efficiency with brake mean effective pressure (BMEP) in engine with LHR combustion chamber with biodiesel at various injection timings. This curve followed similar trends with Fig.3. From Fig.4, it is observed that at 27^o bTDC, engine with LHR combustion chamber with biodiesel showed the improved performance at all loads when compared with diesel operation on CE.

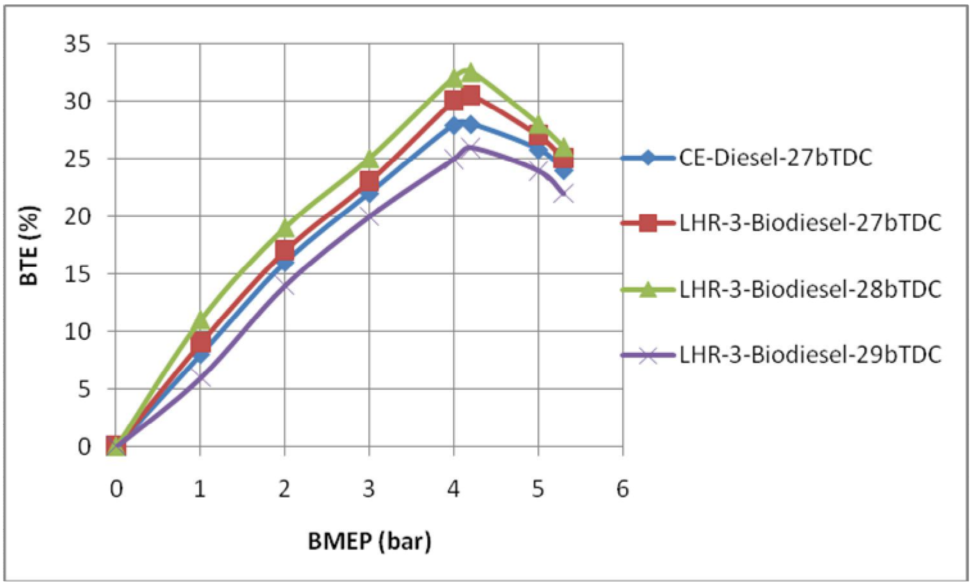


Figure.4. Variation of brake thermal efficiency (BTE) with brake mean effective pressure (BMEP) in engine with LHR combustion chamber with biodiesel at various injection timings at an injector opening pressure of 190 bar.

High cylinder temperatures helped in improved evaporation and faster combustion of the fuel injected into the combustion chamber. Reduction of ignition delay of the biodiesel in the hot environment of the engine with LHR combustion chamber might have improved heat release rates. Engine with LHR combustion chamber with biodiesel operation increased peak BTE by 14% at an optimum injection timing of 28^o bTDC in comparison with mineral diesel operation on CE at 27^o bTDC. Hot combustion chamber of LHR engine reduced ignition delay and combustion duration and hence the optimum injection timing (28^o bTDC) was obtained earlier with engine with LHR combustion chamber when compared with CE (31^o bTDC) with biodiesel operation.

3.2 Exhaust emissions
Particulate emissions and NO_x are the exhaust emissions from diesel engine cause health hazards like inhaling of these pollutants cause severe headache, tuberculosis, lung cancer, nausea, respiratory problems, skin cancer, hemorrhage, etc. [30–32]. In diesel engines, it is rather difficult to lower NO_x and particulate emissions simultaneously due to soot-NO_x tradeoff. High NO_x and particulate emissions are still the main obstacle in the development of next generation conventional diesel engines Therefore, the major challenge for the existing and future diesel engines is meeting the very tough emission targets at affordable cost, while improving fuel economy. It was reported that fuel physical properties such as density and viscosity could have a greater influence on particular emission than chemical properties of the fuel [Avinash Kumar *et al*, 2013]. Fig.5 shows variation of particulate emissions with biodiesel operation on both versions of the engine at recommended injection timing and optimum injection timing.

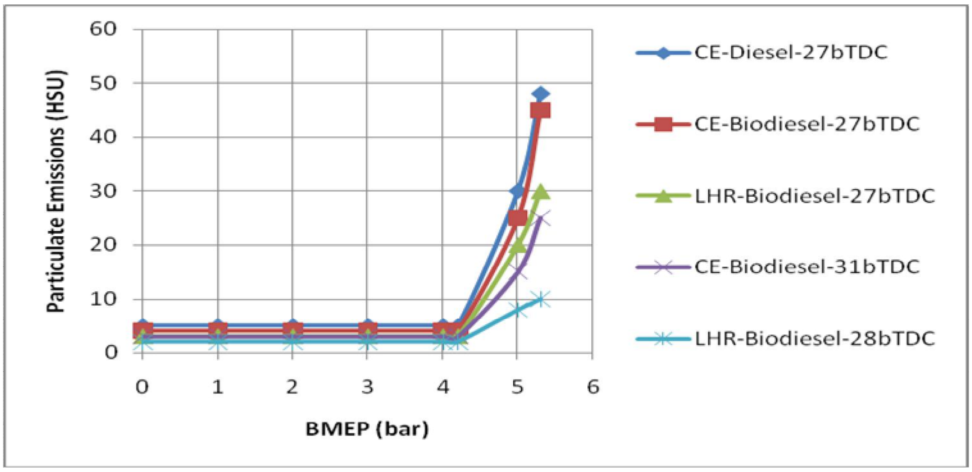


Figure.5 Variation of particulate emissions with biodiesel operation on both versions of the engine at recommended injection timing and optimum injection timing.

From Fig.5, it is noticed that during the first part, particulate emissions were more or less constant, as there was always excess air present. However, at the higher load range there was an abrupt rise in particulate emissions due to less available oxygen, causing the decrease of air–fuel ratio, leading to incomplete combustion, producing more particulate emissions. From Fig.5, it is noticed that particulate emissions at all loads reduced marginally with CE with biodiesel operation in comparison with diesel operation on CE. Improved combustion with improved cetane number and also with presence of oxygen in composition of fuel might have reduced particulate emissions. Particulate emissions further reduced with engine with LHR-3 combustion chamber, when compared with CE. Improved combustion with improved heat release rate might have further reduced particulate emissions. Particulate emissions at full load reduced with advanced injection timing with both versions of the combustion chamber. Increase of resident time and more contact o fuel with air leading to increase atomization have reduced particulate emissions.

Fig.6 presents bar charts showing variation of particulate emissions at full load with test fuels. From Fig.6, it is noticed that CE with biodiesel operation decreased particulate emissions at full load by 6% at 27^o bTDC and 17% at 31^o bTDC, when compared with neat diesel operation on CE at 27^o bTDC and at 31^o bTDC. Earlier studies have suggested following reasons for relatively lower particulate emissions with biodiesel (a) presence of fuel oxygen, (b) increase in the O/C ratio at the flame lift-off length, [The O/C (w/w) ratio here refers to the total oxygen (air and fuel) (w/w) in the combustible mixture to total carbon in the fuel. For biodiesel, carbon and oxygen content in the fuel was obtained from GC analysis. Oxygen originates from air and fuel (biodiesel) both. For diesel, the standard formula given in the published literature has been used to calculate the O/C ratio [7]. (c) longer flame liftoff length due to higher injection velocity obtained with biodiesel, and (d) superior fuel atomization due to higher injection pressures with biodiesel [7].

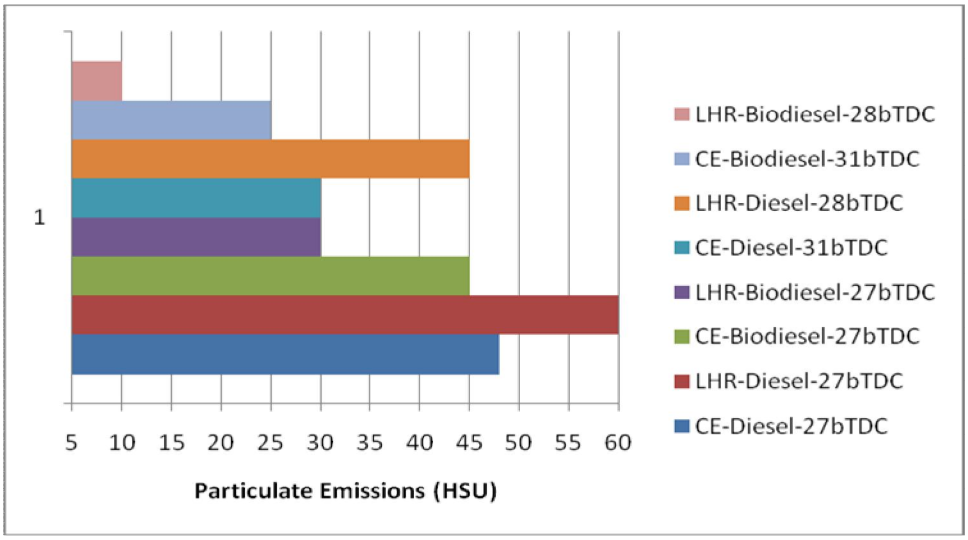


Figure.6. Bar charts showing the variation of particulate emissions at full load operation with test fuels with conventional engine (CE) and engine with LHR combustion chamber at recommended and optimized injection timings at an injector opening pressure of 190 bar.

From Fig.6, it is noticed that particulate emissions decreased with advanced injection timings, in both versions of the combustion chamber, with different operating conditions of the biodiesel. Increase of air entrainment might have caused lower particulate emissions with advanced injection timings. From Fig.6,it is observed that engine with LHR combustion chamber with biodiesel operation decreased particulate emissions at full load by 50% at 27^o bTDC and 65% at 28^o bTDC, when compared diesel operation on engine with LHR combustion chamber at 27^o bTDC and at 28^{o}bTDC. Improved combustion of higher cetane value biodiesel in the hot environment provided by engine with LHR combustion chamber might have reduced particulate emissions with test fuels.

Fig.6 indicates that engine with LHR combustion chamber with biodiesel decreased particulate emissions at full load operation by 33% at 27^o bTDC and 40% at 28^o bTDC, in comparison with CE with biodiesel at 27^o bTDC and at 31^o bTDC. Improved combustion of biodiesel with improved oxygen– fuel ratios might have reduced particulate emissions in the LHR version of the combustion chamber.

The temperature and availability of oxygen are the reasons for the formation of NO_x levels. Fig.7 presents variation of nitrogen oxide levels with brake mean effective pressure with biodiesel operation with both versions of the engine at recommended injection timing and optimum injection timing. NO_x concentrations raised steadily with increasing BMEP at constant injection timing. At part load, NO_x concentrations were less in both versions of the engine. Availability of excess oxygen and high temperatures with consumption of fuel increased NO_x levels with both versions of the engine. At remaining loads, NO_x concentrations steadily increased with the load in both versions of the engine. This was because, local NO_x concentrations raised from the residual gas value following the start of combustion, to a peak at the point where the local burned gas equivalence ratio changed from lean to rich. Biodiesel operation increased NOx levels with both versions of the engine, in comparison with neat diesel operation on CE. The increase in NO_x emission might be an inherent characteristic of biodiesel due to the presence of long chain mono-unsaturated fatty acids (MUFA) and of poly–unsaturated fatty acids (PUFA). [33–34]. Presence of oxygen (10%) in the methyl ester, which leads to improvement in oxidation of the nitrogen available during combustion. This will raise the combustion bulk temperature responsible for thermal NO_x formation. The production of higher NO_x with biodiesel fueling is also attributable to an inadvertent advance of fuel injection timing due to its higher bulk modulus (1750 MPa) of compressibility, with the in-line fuel injection system. Similar observations were made by earlier researchers. [7].

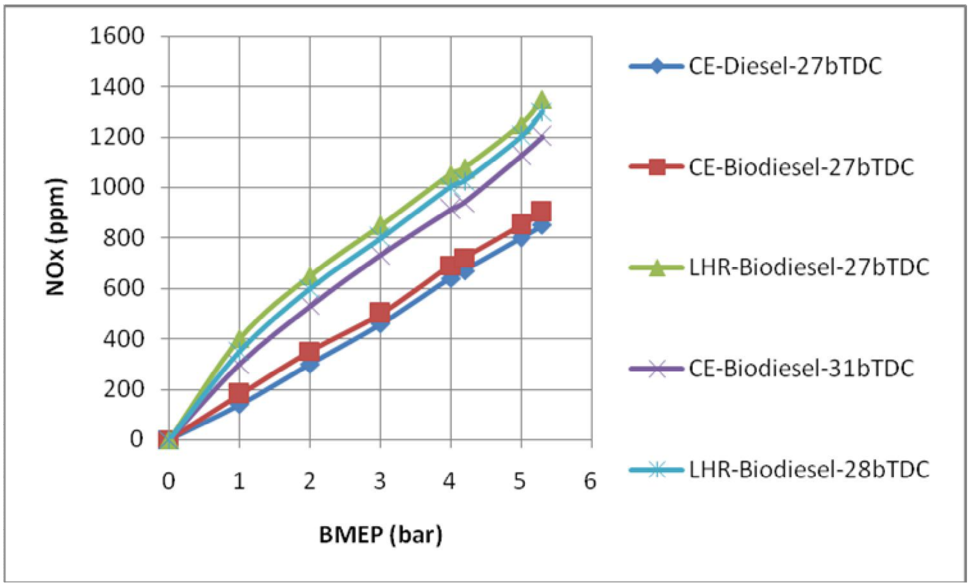


Figure.7. Variation of nitrogen oxide levels with biodiesel operation on both versions of the engine at recommended injection timing and optimum injection timing.

From Fig.7, it was observed that advanced injection timing increased NO_x levels in CE, while decreasing them in engine with LHR combustion chamber with test fuels. Increase of combustion temperatures and resident time lead to produce more NO_x concentration in the exhaust of CE, while reduction of gas temperatures with improved air–fuel ratios decreased NO_x levels in engine with engine with LHR combustion chamber with advanced injection timing.
Fig.8 presents bar charts showing the variation of NOx levels at full load with both versions of the engine with test fuels at recommended injection timing and at optimum injection timing. From Fig.8, it is observed that CE with biodiesel operation increased NO_x levels at full load by 6% at 27^o bTDC and 5% at 31^o bTDC, when compared with diesel operation on CE at 27^o bTDC and at 31^o bTDC

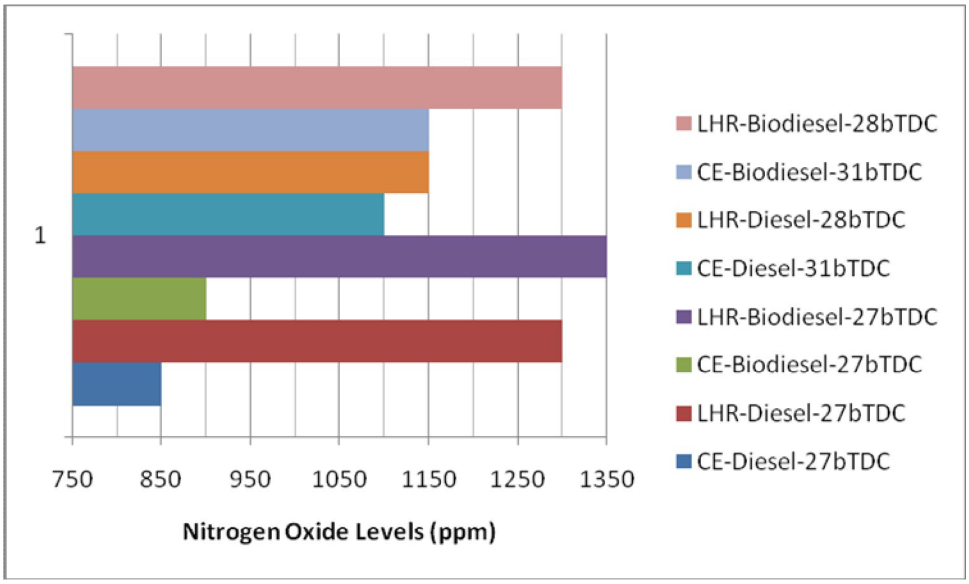


Figure.8. Bar charts showing the variation of nitrogen oxide levels at full load operation with test fuels with conventional engine (CE) and engine with LHR combustion chamber at recommended and optimized injection timings at an injector opening pressure of 190 bar.

From Fig.8,it is observed that NOx levels at full load operation on engine with LHR combustion chamber with biodiesel increased by 4% at 27^o bTDC and 13% at 28bTDC, when compared diesel operation on engine with LHR combustion chamber at 27^o bTDC and at 28^o bTDC. Higher cetane value of biodiesel might have improved NO_x levels with biodiesel operation. Engine with LHR combustion chamber with biodiesel increased NO_x levels at full load operation by 50% at 27^o bTDC and 13% at 28 bTDC, in comparison with CE at 27^o bTDC and at 31^o bTDC. Increase of combustion temperatures with the faster combustion and improved heat release rates caused higher NO_x levels in the engine with LHR combustion chamber in comparison with CE with biodiesel operation.
Table.4 shows exhaust emissions at full load with test fuels. Decreasing the fuel density tends to increase spray dispersion and spray penetration. Particulate emissions at full load decreased with preheating of biodiesel in both versions of the combustion chamber, as seen in Table.4.The factors responsible for reduction of particulate emissions with preheated biodiesel might be i) the reduction of density of the biodiesel, as density is directly related to particulate emissions, ii) the reduction of the diffusion combustion proportion with the preheated biodiesel, iii) the reduction of the viscosity of the biodiesel, with which the fuel spray does not impinge on the combustion chamber walls of lower temperatures rather than it is directed into the combustion chamber.

Table.4 Comparative data on Particulate Emissions & NO_x Levels at full load operation

IT/ Combustion Chamber Version	Test fuel	Particulate emissions (Hartridge Smoke Unit)				NO_x levels (ppm)			
		Injector opening pressure (bar)				Injector opening pressure (bar)			
		190		270		190		270	
		NT	PT	NT	PT	NT	PT	NT	PT
27(CE)	Diesel	48	--	34	--	850	---	950	---
	Biodiesel	45	40	35	30	900	850	1000	950
27(LHR)	Diesel	60	---	50	---	1200	---	1100	---
	Biodiesel	40	35	30	35	1250	1200	1150	1100
30(LHR)	Diesel	45	---	35	---	1050	---	950	---
	Biodiesel	20	15	10	09	1200	1150	1100	1050
31(CE)	Diesel	30	---	35	--	1100	--	1200	---
	Biodiesel	25	20	30	25	1150	1100	1250	1200

From Table.4.it is noticed that particulate emissions at full load reduced with an increase of injector opening pressure in both versions of the combustion chamber, with different operating conditions of the biodiesel. Higher fuel injection pressures improved fuel–air mixing followed by faster combustion which directly influences pollutant formation leading to reduce particulate emissions. At higher injector opening pressure, particulate emissions in the exhaust reduced due to relatively superior fuel–air mixing. An increase in fuel injection pressure induces improvement in spray atomization, combustion and particulate emissions. Similar observations were reported by earlier studies. [7,17,33]. From Table.4, it is noticed that NO_x levels reduced with preheating of the biodiesel. The change of the properties of viscosity and surface tension of fuel with preheating may lead to different relative duration of premixed and diffusive combustion regimes, which have different emission formation characteristics. As fuel temperature was increased, there was an improvement in the ignition quality, which will cause shortening of ignition delay. A short ignition delay period lowers the peak combustion temperature which suppresses NO_x formation. From Table.4, it is noted that NO_x levels increased in CE, while decreasing them in engine with LHR combustion chamber with different operating conditions of biodiesel with an increase of injector opening pressure. Enhanced spray characteristics, thus improving fuel air mixture preparation and evaporation process in CE might have increased gas temperatures with CE, which increased NO_x levels. Improved combustion with improved oxygen–fuel ratios in engine with LHR combustion chamber reduced particulate emissions.

4.CONCLUSIONS

1. The optimum injection timing with conventional engine (CE) was 31^o bTDC, while it was 28^o bTDC with low heat rejection (LHR) engine with biodiesel operation.
2. Engine with LHR combustion chamber is efficient for alternative fuel like biodiesel rather than neat diesel.
3. Engine with LHR combustion chamber with biodiesel reduced particulate emissions and increased nitrogen oxide levels at full load operation over CE at recommended injection timing and optimized timing.
4. The exhaust emissions were improved with advanced injection timing, increase of injector opening pressure and with preheating with both versions of the combustion chamber with biodiesel

Future scope of work

Engine with LHR combustion chamber gave higher NO_x levels, which can be controlled by means of the selective catalytic reduction (SCR) technique using lanthanum ion exchanged zeolite (catalyst-A) and urea infused lanthanum ion exchanged zeolite (catalyst-B) with different versions of combustion chamber at full load operation of the engine [Janardhan, *et al.*, 2012].

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