Experimental investigations on exhaust emissions of high grade semi adiabatic diesel engine fuelled with cotton seed biodiesel

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Abstract– Biodiesels derived from vegetable oils present a very promising alternative for diesel fuel, since they have numerous advantages compared to fossil fuels. They are renewable, biodegradable, provide energy security and foreign exchange savings besides addressing environmental concerns and socio–economic issues. However drawbacks associated with biodiesel of high viscosity and low volatility which cause combustion problems in CI engines, call for engine with hot combustion chamber. They have significant characteristics of higher operating temperature, maximum heat release, and ability to handle low calorific value fuel. Investigations were carried out to determine exhaust emissions with low heat rejection combustion chamber with crude cotton seed biodiesel. It consisted of an air gap insulated piston, an air gap insulated liner and ceramic coated cylinder head with different operating conditions of cotton seed biodiesel with varied injection timing and injector opening pressure. Exhaust emissions were determined at full load operation of the engine. The optimum injection timing with conventional engine (CE) was 31o bTDC (before top dead centre), while it was 28o bTDC for engine with LHR combustion chamber with biodiesel. Comparative studies were made for engine with LHR combustion chamber and CE at manufacturer's recommended injection timing (27o bTDC) and optimum injection timing with biodiesel operation. Engine with LHR combustion chamber with biodiesel showed reduction of particulate emissions at 27o bTDC and at optimum injection timing over CE.

Index Terms: Biodiesel; LHR combustion chamber; Fuel performance; Exhaust emissions;

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1.Introduction

Fossil fuels are limited resources; hence, search for renewable fuels is becoming more and more prominent for ensuring energy security and environmental protection. It has been found that the vegetable oils are promising substitute for diesel fuel, because of their properties are comparable to those of diesel fuel. They are renewable and can be easily produced. When Rudolph Diesel, first invented the diesel engine, about a century ago, he demonstrated the principle by employing peanut oil. He hinted that vegetable oil would be the future fuel in diesel engine [1]. Several researchers experimented the use of vegetable oils as fuel on conventional engines (CE) and reported that the performance was poor, citing the problems of high viscosity, low volatility and their polyunsaturated character. It caused the problems of piston ring sticking, injector and combustion chamber deposits, fuel system deposits, reduced power, reduced fuel economy and increased exhaust emissions [1] [2] [3] [4] [5].

The problems of crude vegetable oils can be solved to some extent, if these oils are chemically modified (esterified) to biodiesel. Studies were made with biodiesel on CE [6] [7] [8] [9] [10]. They reported from their investigations that biodiesel operation showed comparable thermal efficiency, decreased particulate emissions and increased nitrogen oxide (NO_x) levels, when compared with mineral diesel operation.

Experiments were conducted on preheated vegetable oils in order to equalize their viscosity to that of mineral diesel may ease the problems of injection process [11] [12] [13]. Investigations were carried out on engine with preheated vegetable oils. They reported that preheated vegetable oils marginally increased thermal efficiency, decreased particulate matter emissions and NO_x levels, when compared with normal biodiesel.

Increased injector opening pressure may also result in efficient combustion in compression ignition engine [14] [15]. It has a significance effect on performance and formation of pollutants inside the direct injection diesel engine combustion. Experiments were conducted on engine with biodiesel with increased injector opening pressure. They reported that performance of the engine was improved, particulate emissions were reduced and NO_x levels were increased marginally with an increase of injector opening pressure.

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); (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);and (3).high grade LHR 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 [16] [17] [18] [19] [20] [21] [22]. 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 performance of the engine with high grade LHR combustion chamber. It contained an air gap (3.2 mm) insulated piston, an air gap (3.2 mm) insulated liner and ceramic coated cylinder head with cotton seed 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. Material and method

Cottonseeds have approximately 18% (w/w) oil content. India's cottonseed production is estimated to be around 35% of its cotton output (approximately 4.5millionmetric tons). Approximately 0.30 million metric ton cottonseed oil is produced in India and it is an attractive biodiesel feedstock [5]

2.1 Preparation of biodiesel

The chemical conversion of esterification reduced viscosity four fold. Crude cotton seed oil contains up to 70 % (wt.) free fatty acids. The methyl ester was produced by chemically reacting crude cotton seed oil with methanol in the presence of a catalyst (KOH). A two–stage process was used for the esterification of the crude cotton seed oil [5]. The first stage (acid-catalyzed) of the process is to reduce the free fatty acids (FFA) content in cotton seed 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 cotton seed oil to methanol was 9:1 and 0.75% catalyst (w/w). In the second stage (alkali-catalyzed), the triglyceride portion of the cotton seed 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. [5].

Table.1 Properties of test fuels [5]

Property	Units	Diesel (DF)	Biodiesel(BD)	ASTM Standard
Carbon Chain	--	C_8–C_{28}	C_{16}–C_{24}	---
Cetane Number	-	51	56	ASTM D 613
Specific Gravity at 15^{o}C	-	0.8275	0.8673	ASTM D 4809
Bulk Modulus at 15^{o}C	MPa	1408.3	1564	ASTM D 6793
Kinematic Viscosity @ 40^{o}C	cSt	2.5	5.44	ASTM D 445
Air Fuel Ratio (Stoichiometric)	--	14.86	13.8	--
Flash Point (Pensky Marten's Closed Cup)	^{o}C	120	144	ASTM D93
Cold Filter Plugging Point	^{o}C	Winter 6^o C Summer 18^{o}C	3^o C	ASTM D 6371
Pour Point	^{o}C	Winter 3^{o}C Summer 15^{o}C	0^{o}C	ASTM D 97
Sulfur	(mg/kg, max)	50	42	ASTM D5453
Low Calorific Value	MJ/kg	42.0	39.9	ASTM D 7314
Oxygen Content	%	0.3	11	--

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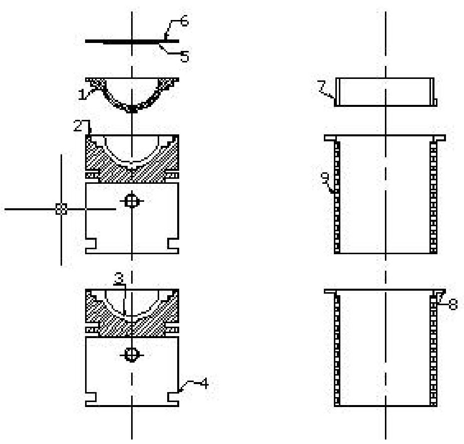
2.3 Engine with LHR combustion chamber

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



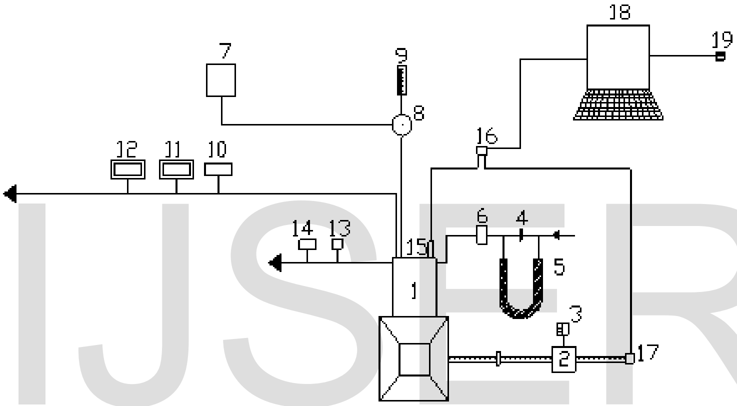
1. Piston crown with threads, 2. Superni gasket, 3. Air gap in piston, 4. Body of piston, 5. Ceramic coating on inside portion of cylinder head, 6. Cylinder head, 7.Superni insert with threads, 8.Air gap in liner, 9.Liner

Fig.1 Assembly details of air gap insulated piston, air gap 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 (3.2 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.

2.4 Experimental set–up

The schematic diagram of the experimental setup used for the investigations on the engine with LHR combustion chamber with cotton seed biodiesel is shown in Fig.2. Specifications of Test engine are given in Table 2. 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_x Analyzer, 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.

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

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%.

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 %	0.1% of Full Scale (FS)	0.1% for 30 minutes
NO_x	Chemiluminiscence	1–5000 ppm	0.5 % F.S	≤0.5% F.S

2.7 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 190 and 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 [23] [24]. 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.

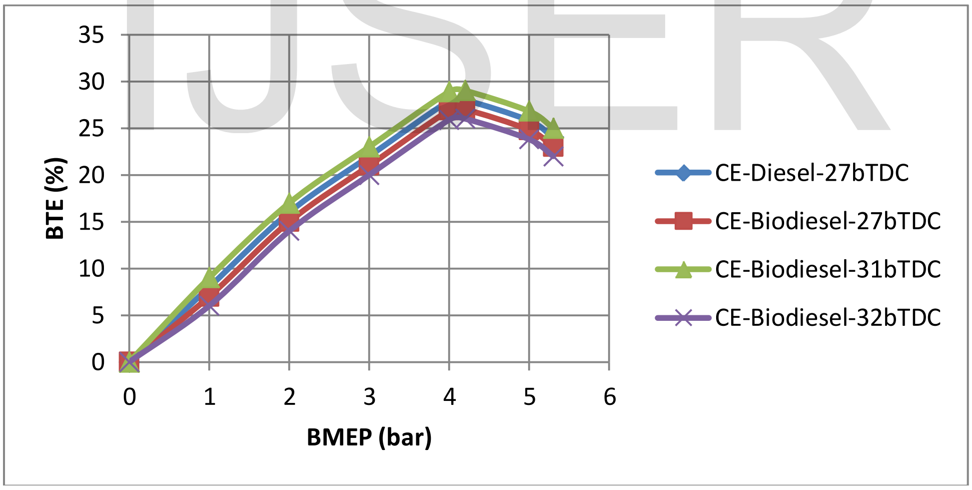


Fig. 3 Variation of brake thermal efficiency (BTE) with brake mean effective pressure (BMEP) in conventional engine (CE) with biodiesel at various injection timings at an injector opening pressure of 190 bar.

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. 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.

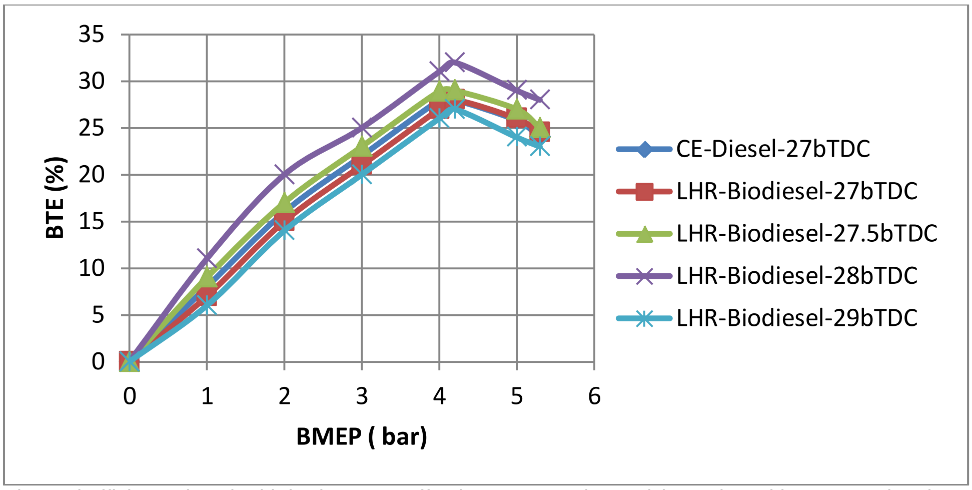


Fig.4. Variation of brake thermal efficiency (BTE) with brake mean effective pressure (BMEP) in engine with LHR combustion chamber with biodiesel at various injection timings at an injector opening pressure of 190 bar.

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. [25] [26] [27]. 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 [17]. Fig.5 shows variation of particulate emissions with biodiesel operation on both versions of the engine at recommended injection timing and optimum injection timing.

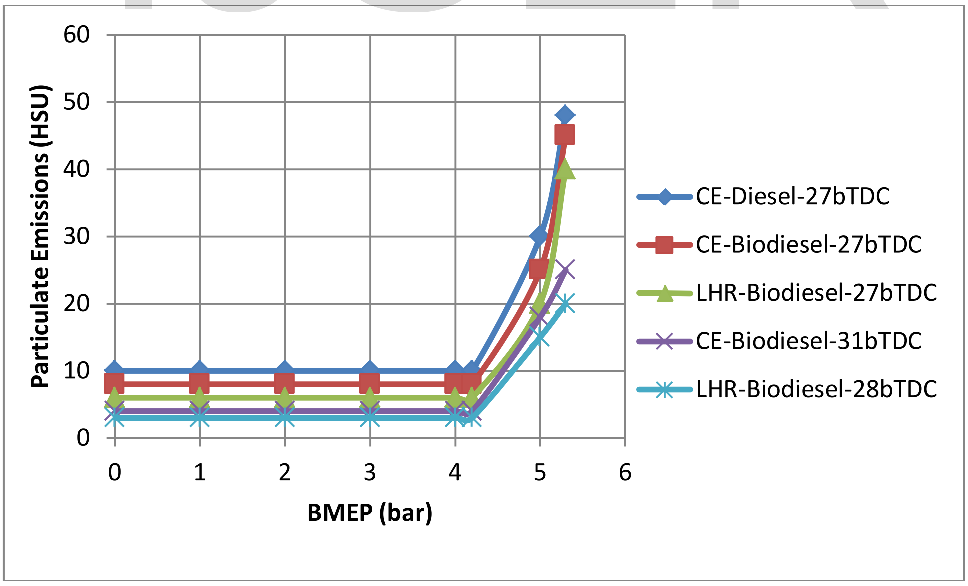


Fig.5. Variation of particulate emissions with brake mean effective pressure (BMEP) with biodiesel with both versions of the engine at recommended injection timing and optimum injection timing.

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.

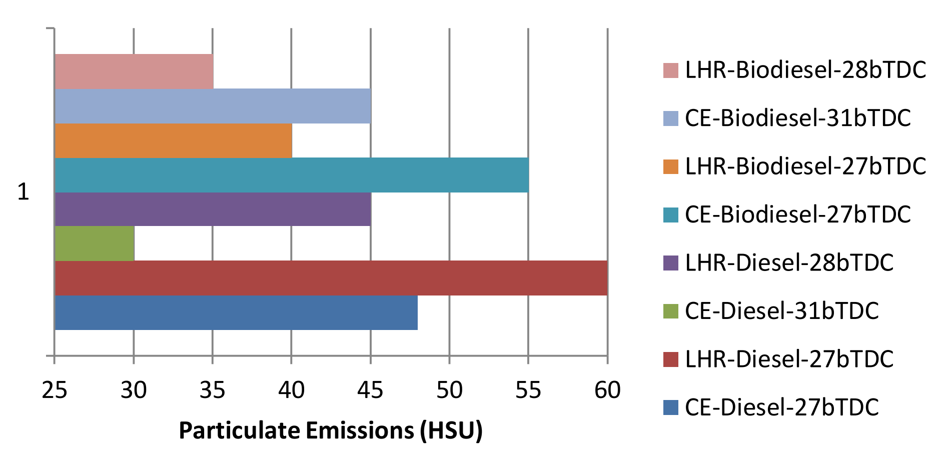


Fig.6. Bar charts showing the variation of particulate emissions at full load operation with test fuels with conventional engine (CE) and engine with LHR combustion chamber at recommended and optimized injection timings at an injector opening pressure of 190 bar.

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 [17]. (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 [5]. 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 33% at 27^{o} bTDC and 55% 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 11% at 27^{o} bTDC and 20% at 28^{o} bTDC, in comparison with CE 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.

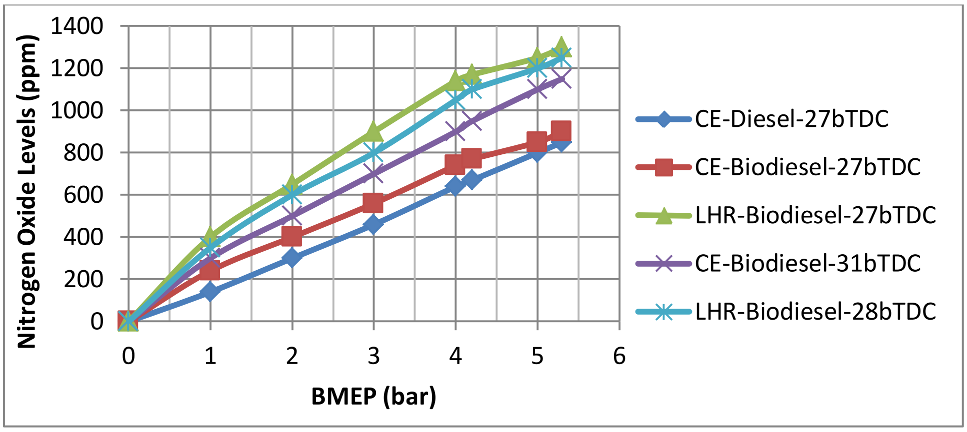


Fig.7. Variation of nitrogen oxide levels with brake mean effective pressure (BMEP) with biodiesel with both versions of the engine at recommended injection timing and optimum injection timing.

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). [28] [29]. resence 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 (1564 MPa) of compressibility, with the in-line fuel injection system. Similar observations were made by earlier researchers. [5]. 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.

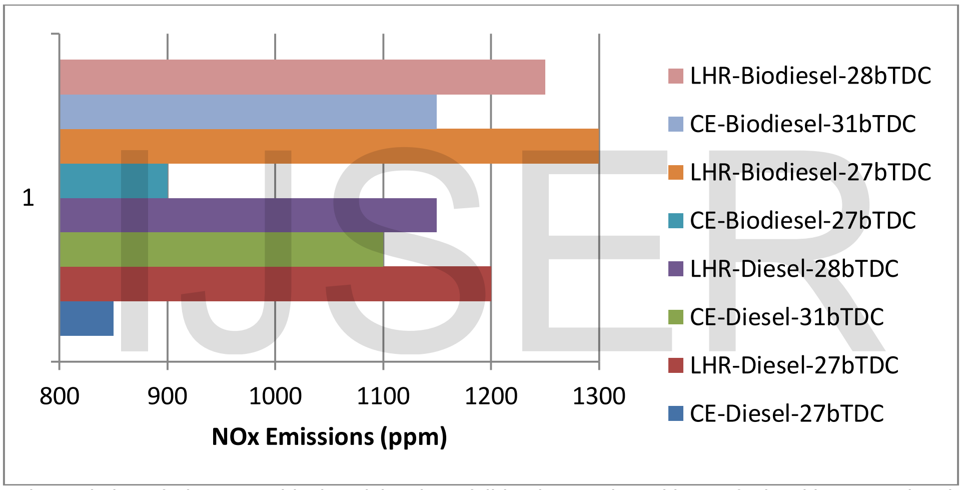


Fig.8. Bar charts showing the variation of nitrogen oxide (NO_x) levels at full load operation with test fuels with conventional engine (CE) and engine with LHR combustion chamber at recommended and optimized injection timings at an injector opening pressure of 190 bar.

From Fig.8. it is observed that NOx levels at full load operation on engine with LHR combustion chamber with biodiesel increased by 8% at 27^o bTDC and 9% at 28^o bTDC, 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 44% at 27^o bTDC and 9% at 28^o 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)	DF	48	--	34	--	850	--	950	--
	BD	45	40	35	30	900	850	1000	950
27(LHR)	DF	60	---	50	---	1200	---	1100	--
	BD	40	35	30	25	1300	1250	1200	1150
28(LHR)	DF	45	---	35	---	1150	--	1050	--
	BD	20	15	10	10	1250	1200	1150	1100
31(CE)	DF	30	---	35	--	1100	--	1200	---
	BD	25	20	35	30	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. [14] [15] [28]. 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.

Summary

1. Engine with LHR combustion chamber is efficient for alternative fuel like biodiesel rather than neat diesel.
2. Engine with LHR combustion chamber with biodiesel improved its performance over CE at recommended injection timing and optimized timing.
3. 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.

4.2 Novelty

Engine parameters (injection timing and injection pressure) fuel operating conditions (normal temperature and preheated temperature) and different configurations of the engine (conventional engine and engine with LHR combustion chamber) were used simultaneously to improve performance, exhaust emissions and combustion characteristics of the engine. Change of injection timing was accomplished by inserting copper shims between pump frame and engine body. The performance of the lubricating oil was determined by FE method with the change of configuration of combustion chamber design from conventional to LHR combustion chamber. FE results were correlated with experimental results.

Highlights

- Fuel injection pressure & timings affect engine performance, exhaust emissions and combustion characteristics.
- Performance, exhaust emissions and combustion characteristic improve with preheating of biodiesel
- Engine with LHR combustion chamber was safe, as the performance of lubricating oil was not deteriorated.

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 [30].

Acknowledgments

Authors thank authorities of Chaitanya Bharathi Institute of Technology, Hyderabad for providing facilities for carrying out this research work. Financial assistance provided by All India Council for Technical Education (AICTE), New Delhi is greatly acknowledged.

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