INFLUENCE OF PREHEATING ON PERFORMANCE PARAMETERS AND COMBUSTION CHARACTERISTICS OF HIGH GRADE SEMI ADIABATIC DIESEL ENGINE WITH COTTON SEED BIODIESEL

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ABSTRACT

Vegetable oils are important substitutes for diesel fuel, as their properties are comparable to diesel fuel and also they are renewable in nature. However, drawbacks associated with vegetable oils of high viscosity and low volatility need to be converted into biodiesel. These 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 volatilitywhich cause combustionproblems 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 lowcalorific value fuel. Investigations were carried out to evaluate the performance and combustion characteristics with low heat rejection combustion chamber with 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. Combustion characteristics were determined by means of Piezo electric pressure transducer, TDC (top dead center) and special pressure- crank angle software package at full load operation. Comparative studies were made for engine with LHR combustion chamber and CE at manufacturer’s recommended injection timing (27^{o}bTDC) with biodiesel operation.Engine with LHR combustion chamber with biodiesel showed improved performance at 27^{o}bTDCover CE.

Key words: Vegetable oil, Biodiesel; LHR combustion chamber;Fuel performance

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 thoseof 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 [1–5].

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[6–10].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 [11–13]. 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.

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 anair 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 [14–20]. 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_xlevels, 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.

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]

Preparation of biodiesel

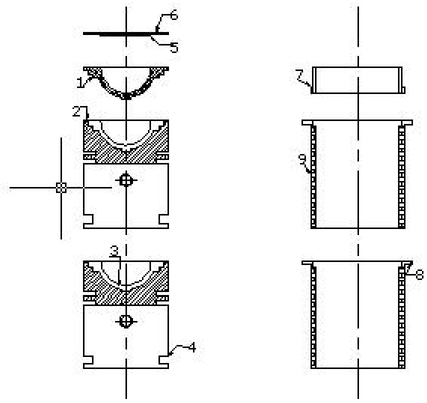
The chemical conversion of esterification reduced viscosity four fold. Crude cotton seed oil contains up to 70 % (wt.) free fatty acids. The methyl ester was produced by chemically reacting crude cotton seed oil with methanol in the presence of a catalyst (KOH). A two–stage process was used for the esterification 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°C	-	0.8275	0.8673	ASTM D 4809
Bulk Modulus at 15°C	MPa	1408.3	1564	ASTM D 6793
Kinematic Viscosity @ 40°C	cSt	2.5	5.44	ASTM D 445
Air Fuel Ratio (Stoichiometric)	--	14.86	13.8	--
Flash Point (Pensky Marten's Closed Cup)	°C	120	144	ASTM D93
Cold Filter Plugging Point	°C	Winter 6° C Summer 18°C	3° C	ASTM D 6371
Pour Point	°C	Winter 3°C Summer 15°C	0°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	--

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 (3.2 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.



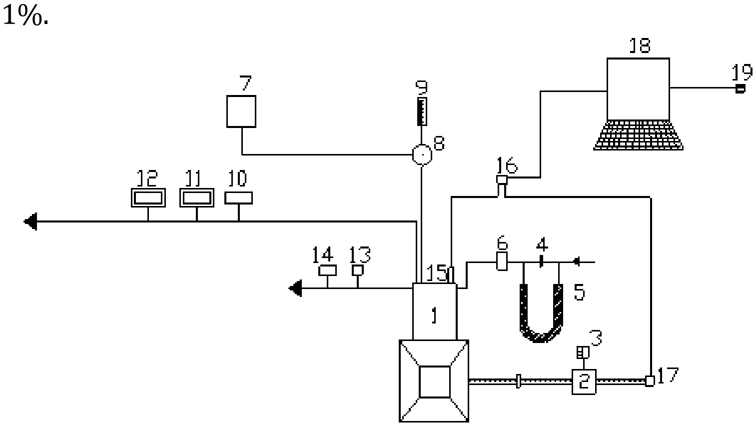
1.Pistoncrown with threads, 2. Superni gasket, 3.Air gap in piston, 4. Body of piston, 5. Ceramic coating on inside portion of cylinder head, 6. Cylinder head, 7.Superni insert with threads, 8.Air gap in liner, 9.Liner

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

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.

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 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 andU–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 thewater 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, Pre-heater 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.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. 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. Piezo electric transducer, (AVL Austria: QC34D). connected to cylinder head, which in turn was connected to console. TDC encoder (AVL Austria: 365x) connected to extended portion of shaft of dynamometer, which in turn was connected to console. Console was connected to compute. From the signals of pressure and crank angle, crank angle diagram was obtained. The accuracy of measurement of pressure is ±1 bar, while it is ± 1^o for crank angle. Combustion parameters such as peak pressure (PP), time of occurrence of peak pressure (TOPP) and maximum rate of pressure rise at the full load operation of the engine were evaluated.

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

RESULTS AND DISCUSSION

Performance parameters

Fig.3 presents bar charts showing the variation of peak brake thermal efficiency with both versions of the engine at recommended injection timing and pressure with biodiesel operation. It showed that CE with biodiesel at 27^o bTDC showed comparable performance. The presence of oxygen in fuel composition might have improved performance with biodiesel operation, when compared with diesel operation on CE at27^{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.

From Fig.3, it is observed that at 27° bTDC, engine with LHR combustion chamber with biodiesel showed the comparable performance 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.

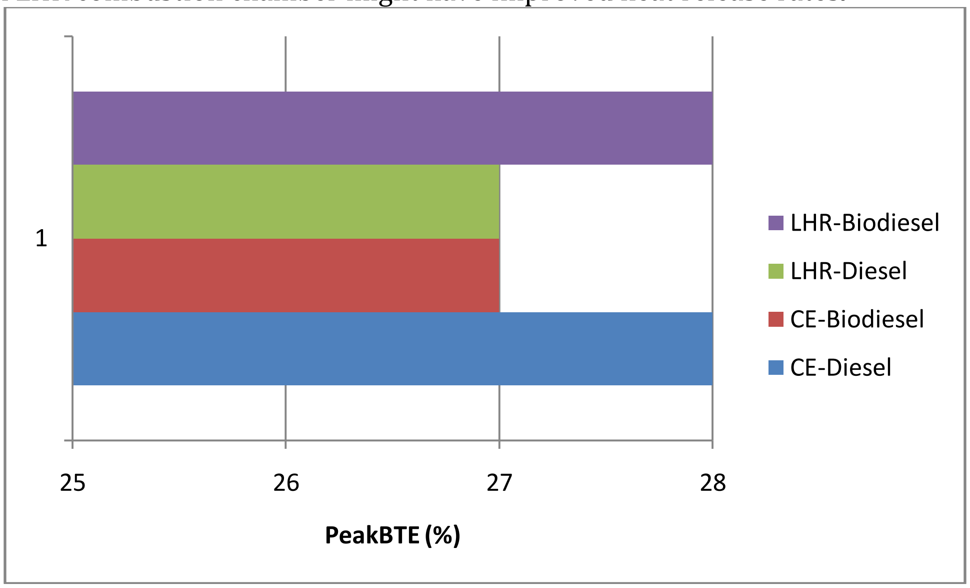


Fig.3 Bar charts showing the variation of peak brake thermal efficiency (BTE) with test fuels with both versions of the engine at recommended injector opening pressure of 190 bar.

Engine with LHR combustion chamber with biodiesel operation increased peak BTE by 14% in comparison with same configuration of the engine with diesel operation. This showed that engine with LHR combustion chamber was more suitable for biodiesel.

Fig.4 presents bar charts showing the variation of brake specific energy consumption (BSEC) at full load with test fuels. BSEC was comparable with biodiesel with CE at 27° bTDC when compared with CE with diesel operation at 27° bTDC. Improved combustion with higher cetane number and presence of oxygen in fuel composition with higher heat release rate with biodiesel may lead to produce comparable BSEC at full load. Engine with LHR combustion chamber with biodiesel decreased BSEC at full load operation by 6% at 27° bTDC when compared diesel operation with engine with LHR combustion chamber at 27° bTDC. This once again confirmed that engine with LHR combustion chamber was more suitable for biodiesel operation than neat diesel operation. Engine with LHR combustion chamber with biodiesel decreased BSEC at full load operation by 3% at 27°bTDC in comparison with CE at 27° bTDC.Improved evaporation rate and higher heat release rate of fuel with LHR combustion chamber might have improved the performance with LHR engine.

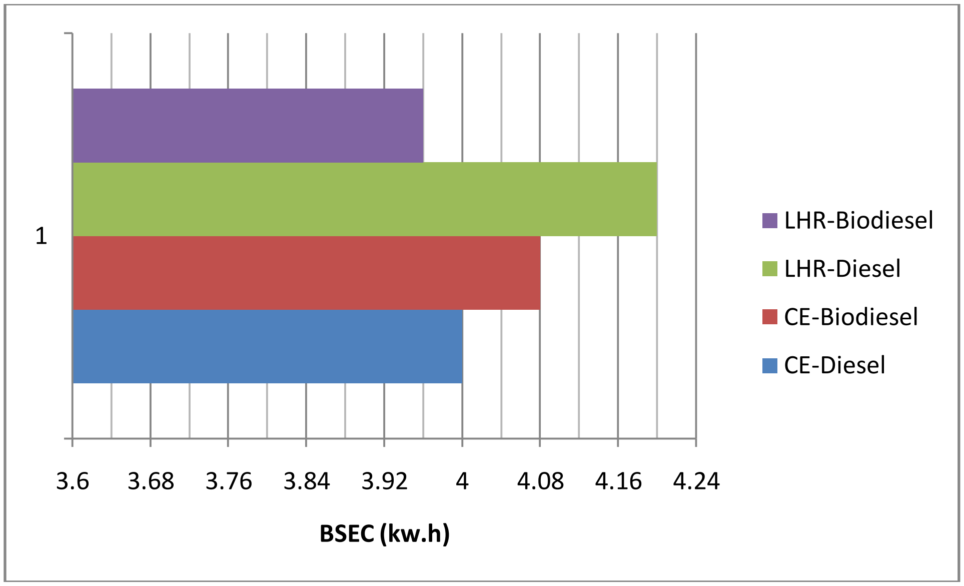


Fig.4.Bar charts showing the variation of brake specific energy consumption (BSEC) at full load operation with test fuels with both versions of the engine at recommended injection timing at an injector opening pressure of 190 bar.

Fig.5 presents bar charts showing variation of exhaust gas temperature (EGT) at full load with test fuels.CE with biodiesel operation increased EGT at full load operation by 6% at 27^{o}bTDC in comparison with CE with neat diesel operation at 27^{o}bTDC. Though calorific value (or heat of combustion) of biodiesel is lower than that of diesel, density of biodiesel is higher, therefore greater amount of heat was released in the combustion chamber leading to produce higher EGT at full load operation with biodiesel operation than neat diesel operation. This was also because of higher duration of combustion of biodiesel causing retarded heat release rate. From Fig.5, it is noticed that engine with LHR combustion chamber with biodiesel operation increased EGT at full load operation by 5% at 27^{o} bTDC when compared with diesel operation on same configuration of the engine at 27^{o} bTDC. High duration of combustion due to high viscosity of biodiesel in comparison with diesel might have increased EGT at full load. Engine with LHR combustion chamber with biodiesel increased EGT at full load operation by 17% at 27^{o}bTDC in comparison with CE at 27^{o}bTDC. This indicated that heat rejection was restricted through the piston, liner and cylinder head, thus maintaining the hot combustion chamber as result of which EGT at full load operation increased with reduction of ignition delay.

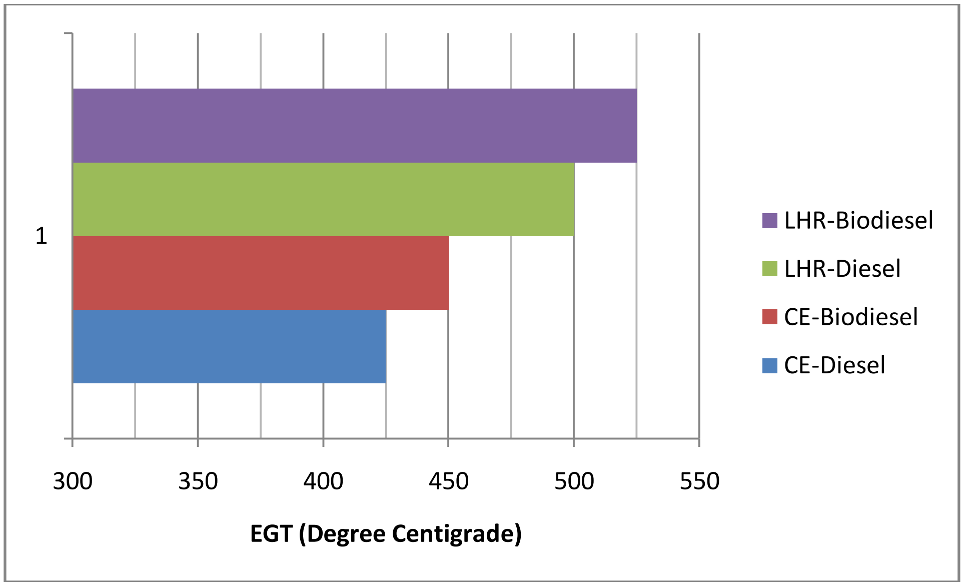


Fig.5 Bar charts showing the variation of exhaust gas temperature (EGT) at full load operation with test fuels with both versions of the engine at recommended injection timings at an injector opening pressure of 190 bar.

Table.3 shows performance parameters of peak BTE, BSEC at full load and EGT at full load,From Table 3, it is noticed that preheating of the biodiesel improved the performance in both versions of the combustion chamber when compared with the biodiesel at normal temperature. Preheating reduced the viscosity of the biodiesel, causing efficient combustion thus improving BTE

Table.3
Comparative data on Peak Brake Thermal Efficiency, Brake Specific Energy Consumption and Exhaust Gas Temperature at full load

Injection timing/ Combustion Chamber Version	Test fuel	Peak Brake Thermal Efficiency (%)		Brake Specific Energy consumption at full load operation (kW.h)		Exhaust Gas Temperature (^{o}C) at full load operation	
		Fuel Operating Condition		Fuel Operating Condition		Fuel Operating Condition	
		NT	PT	NT	PT	NT	PT
27(CE)	DF	28	--	4.0	--	425	---
	BD	27	28	4.08	4.04	450	500
27(LHR)	DF	27	--	4.2	--	500	--
	BD	28	29	3.96	3.92	525	500

From Table.3, it is noticed that EGT at full load operation increased marginally with preheated biodiesel with CE, which indicates the increase of diffused combustion due to high rate of evaporation and improved mixing between methyl ester and air. Therefore, as the fuel temperature increased, the ignition delay decreased and the main combustion phase (that is, diffusion controlled combustion) increased, which in turn raised the temperature of exhaust gases. However, EGT at full load decreased marginally with engine with LHR combustion with preheated biodiesel due to improved combustion.

Fig.6 presents bar charts showing the variation of coolant load with test fuels. CE with biodiesel increased coolant load by 3% at 27°bTDC when compared with neat diesel operation on CE at 27° bTDC as observed from Fig.6.

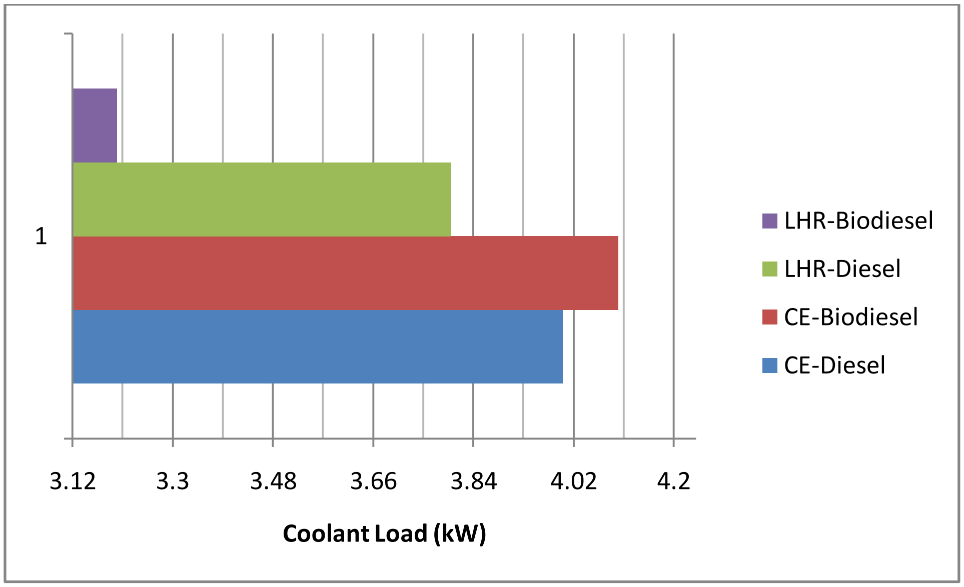


Fig.6 Bar charts showing the variation of coolant load at full load operation with test fuels with both versions of the engine at recommended injection timing at an injector opening pressure of 190 bar.

Increase of un–burnt fuel concentration at the combustion chamber walls may lead to increase of gas temperatures with biodiesel produced higher coolant load than diesel operation. The reduction of coolant load in engine with LHR combustion chamber might be due to the reduction of gas temperatures with improved combustion. Hence, the improvement in the performance of CE was due to heat addition at higher temperatures and rejection at lower temperatures, while the improvement in the efficiency of the engine with LHR combustion chamber was because of recovery from coolant load with test fuels. Engine with LHR combustion chamber with biodiesel operation decreased coolant load operation by 16% at 27° bTDC when compared diesel operation with same configuration of the engine at 27° bTDC. More conversion of heat into useful work with biodiesel than diesel might have reduced coolant load with biodiesel. Fig.6indicates that engine with LHR combustion chamber with biodiesel decreased coolant load at full load operation by 7% at 27°bTDC in comparison with CE at 27°bTDC .

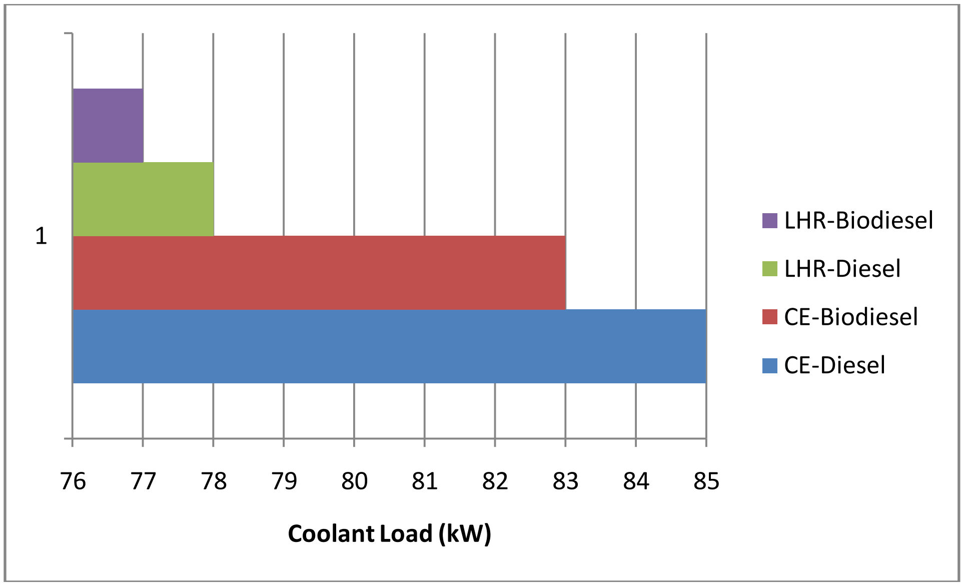


Fig.7 Bar charts showing the variation of volumetric efficiency at full load operation with test fuels with both versions of the engine at recommended injection timing and at an injector opening pressure of 190 bar.

Provision of thermal insulation and improved combustion with engine with LHR combustion chamber might have reduced coolant load with LHR engine in comparison with CE with biodiesel operation.

Fig.7 shows bar charts showing variation of volumetric efficiency at full load with test fuels.

It indicates that CE with biodiesel operation decreased volumetric efficiency at full load by 2% at 27^{o}bTDC , when compared with diesel operation on CE at 27^{o} bTDC. Increase of EGT might have reduced volumetric efficiency at full load, as volumetric efficiency depends on combustion wall temperature, which in turn depends on EGT. From Fig.7, it is noticed that volumetric efficiency at full load operation on engine with LHR combustion chamber at 27^{o} bTDC with biodiesel was marginally lower than diesel operation on same configuration of the engine at 27^{o} bTDC. Increase of EGT was responsible factor for it. Fig.7indicates that engine with LHR combustion chamber with biodiesel decreased volumetric efficiency at full load operation by 7% at 27^{o}bTDC in comparison with CE at 27^{o}bTDC. The reduction of volumetric efficiency with engine with LHR combustion chamber was because of increase of temperatures of insulated components of LHR combustion chamber, which heat the incoming charge to high temperatures and consequently the mass of air inducted in each cycle was lower. Similar observations were noticed by earlier researchers [19–20]. Table.4 shows coolant load and volumetric efficiency at full load. Coolant load at full load operation decreased with preheating of biodiesel, as noticed from Table.4. Improved spray characteristics might have reduced gas temperatures and hence coolant load. Volumetric efficiency at full load operation marginally reduced in CE, while increasing it in engine with LHR combustion chamber with preheated biodiesel as observed from Table.4.This was because of increase of EGT in CE, while decreasing of the same in engine with LHR combustion chamber. Similar trends were noticed by earlier researchers. [21–22]

Table.4 Data of coolant load and volumetric efficiency at full load

IT/ Combustion Chamber Version	Test fuel	Coolant Load (kW)		Volumetric Efficiency (%) at full load operation (%)	
		Fuel Operating Condition		Fuel Operating Condition	
		NT	PT	NT	PT
27(CE)	DF	4.0	-	85	
	BD	4.1	3.9	83	82
27(LHR)	DF	3.8	-	78	
	BD	3.2	3.2	77	78

Combustion Characteristics

Fig 8.presents bar chart showing the variation of peak pressure at full load with both versions of the engine.From Fig, it is noticed that CE with biodiesel increased peak pressure (PP) at full load operation by 4% at 27°bTDC and 5% at 31° bTDC when compared with diesel operation on CE at27°bTDC and at 31° bTDC. Even though biodiesel has lower heat of combustion, it advanced its peak pressure position because of its higher bulk modulus and cetane number. This shift is mainly due to advancement of injection due to higher density and earlier combustion due to shorter ignition delay caused by higher cetane number of biodiesel.Fig.8 indicates that at 27°bTDC, engine with LHR combustion chamber with biodiesel increased peak pressure at full load operation by 35% in comparison with CE. Improved heat release rate with engine with LHR combustion chamber might have increased peak pressure with LHR engine with biodiesel.

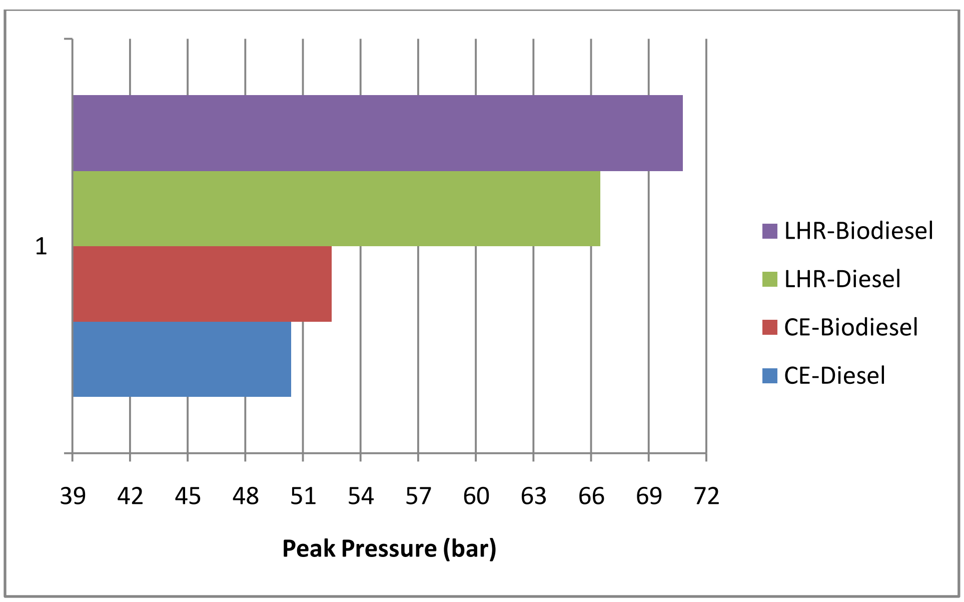


Fig.8. Bar charts showing the variation of peak pressure at full load with both versions of the engine at recommended injection timing with biodiesel operation

Table.5 shows combustion characteristics at full load with biodiesel. From Table.5, it is understood that peak pressure at full load operation reduced with preheating, with both versions of the combustion chamber. When the engine was running on preheated biodiesel, the fuel injection was slightly delayed, due to decrease in bulk modulus of biodiesel with an increase in fuel temperature. The reasons for lower peak pressure with preheated biodiesel was also attributed to earlier combustion caused by short ignition delay (due to faster evaporation of the fuel) at their preheated temperatures.

Table.5
Data of Combustion characteristics at full load

Injection Timing/ Combustion Chamber Version	Test fuel	Peak Pressure (bar)		Maximum Rate of Pressure Rise (bar/degree)		TOPP (degree)	
		Fuel Operating Condition		Fuel Operating Condition		Fuel Operating Condition	
		NT	PT	NT	PT	NT	PT
27(CE)	DF	50.4	---	5.4	----	9	---
	BD	52.5	50.5	4.9	3.9	8	7
27(LHR)	DF	66.5		7.6	---	10	9
	BD	70.8	68.6	6.8	6.1	7	6

Fig.9. presents bar chart showing the variation of maximum rate of pressure rise (MRPR) at full load with both versions of the engine. Fig.9denotes that maximum rate of pressure rise (MRPR) was higher for diesel than biodiesel with both versions of the combustion. High volatile nature of diesel fuel releasing more energy per unit crank angle might have increased MRPR at full load. From Table.5, it is noticed that preheated biodiesel gave lower MRPR when compared with normal biodiesel as in the case of peak pressure. At 27^{o}bTDC Engine with LHR combustion chamber increased MRPR at full load operation by 39% when compared with CE at with biodiesel, which showed that combustion improved with improved air–fuel ratios in hot environment provided by the engine with LHR combustion chamber.

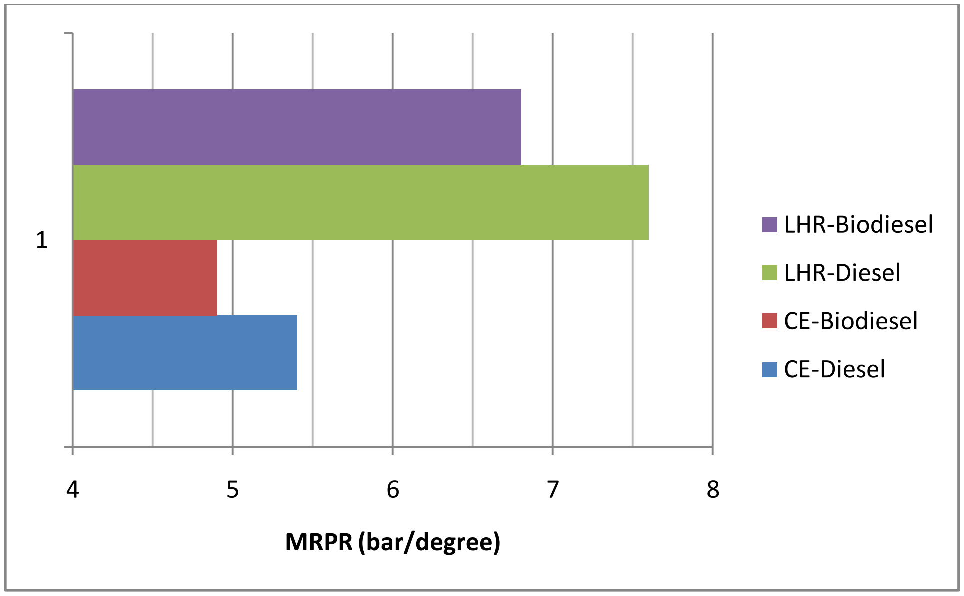


Fig.9. Bar charts showing the variation of maximum rate of pressure rise (MRPR) at full load with both versions of the engine at recommended injection timing with biodiesel operation

Fig.10 presents bar chart showing the variation time of occurrence of peak pressure (TOPP) at full load with both versions of the engine.

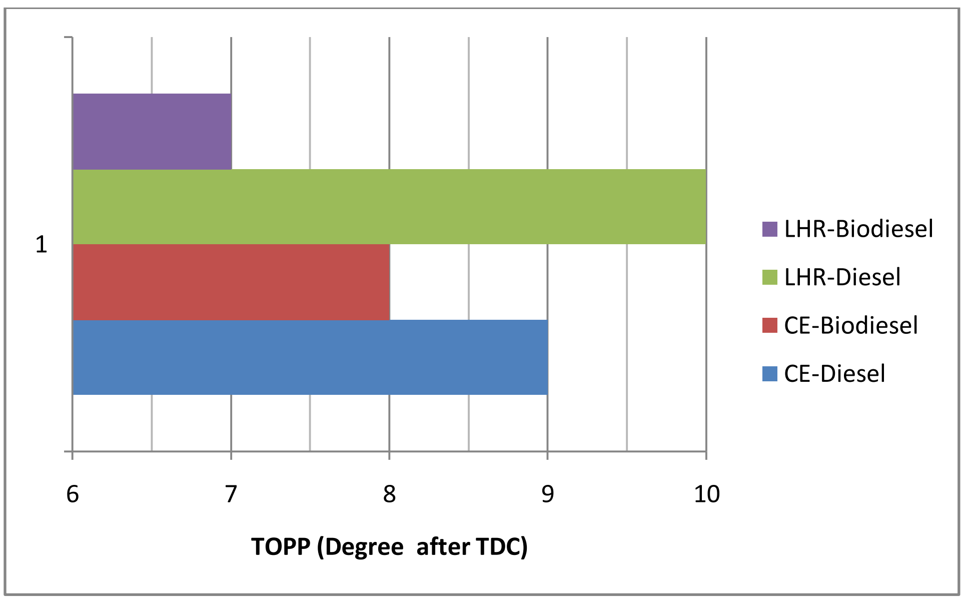


Fig.10. Bar charts showing the variation of time of occurrence of peak pressure (TOPP) at full load with both versions of the engine at recommended injection timing with biodiesel operation

AT recommended injection timing, biodiesel operation marginally decreased time of occurrence of peak pressure (TOPP) at full load operation on CE, while drastically decreasing it with engine with LHR combustion chamber, when compared diesel operation, as noticed from Table.5. Higher bulk modulus of rigidity and cetane number of biodiesel, in comparison with neat diesel operation might have reduced TOPP at full load operation. TOPP at full load decreased with preheated biodiesel with both versions of the combustion chamber due to reduction of viscosity of fuel. This once again confirmed by observing the lower TOPP, the performance of the engine improved with the preheated biodiesel compared with the normal biodiesel.

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 and combustion characteristics over CE at recommended injection timing.
3. The performance and combustion characteristics of the engine improved with preheating of biodiesel with both versions of the combustion chamber with biodiesel.

Novelty

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 performanceand combustion characteristics of the engine.

Highlights

- Fuel injection pressure affects engine performance and combustion characteristics.
- Performanceand combustion characteristics improved with preheating of biodiesel
- Change of combustion chamber design improved the performance of the engine

Future Scope of Work

Performance and combustion characteristics of LHR engine can further be improved by varying injection timing.

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