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RESEARCH ARTICLE

Experimental Investigations on Cotton Seed Biodiesel Fuelled DI Diesel Engine with Low Heat Rejection Combustion Chamber

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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, as 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 with its significant characteristics of higher operating temperature, maximum heat release, and ability to handle low calorific value fuel. Investigations were carried out to evaluate the performance of a engine with low heat rejection combustion chamber consisting 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. Performance parameters and exhaust emissions were determined at full load operation of the engine. Combustion characteristics of the engine were measured at full load operation with water cooled Piezo-electric pressure transducer, TDC (top dead centre) encoder and pressure–crank angle software package. Finite element method was applied to determine isotherms in order to evaluate the performance of lubricating oil with change of configuration of combustion chamber and results were correlated with experimental investigations. The optimum injection timing with conventional engine (CE) was 31^o bTDC, while it was 28^o 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 (27^o bTDC) and optimum injection timing with biodiesel operation. Engine with LHR combustion chamber with biodiesel showed improved performance over CE at 27^o bTDC and at optimum injection timing. Temperature of lubricating oil predicted by FEM was within the limits.

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 and hinted that vegetable oil would be the future fuel in diesel engine Soo-Young No (2011). 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 causing the problems of piston ring sticking, injector and combustion chamber deposits, fuel system deposits, reduced power, reduced fuel economy and increased exhaust emissions Soo-Young No (2011), Avinash Kumar Agarwal et.al, (2013).

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 Rakopoulos CD, et al, (2008), McCarthy et.al, (2011), Anirudh Gautam et.al, (2013), Krishna Maddali et.al, (2014). It was 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 Pugazhvadivu M et.al, (2005), Agarwal D et.al, (2007), Hanbey Hazar et.al, (2010). Investigations were carried out on engine with preheated vegetable oils. It was 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, Celikten I, (2003), Avinash Kumar Agarwal et,al (2013). 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. It was 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).Combination of low grade and medium grade LHR combustion chambers results in high grade LHR combustion chamber.

Experiments were conducted on engine with high grade LHR combustion chamber, consisting of air gap (3 mm) insulation in piston as well as in liner and ceramic coated cylinder head with biodiesel with varied injector opening pressure and injection timing Venkateswara Rao N, et.al, (2013), Murali Krishna MVS, et.al, (2014), . It was 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 NOx levels, by 45–50% when compared with mineral diesel operation on CE at 27^o bTDC.

The temperature distribution and heat flow in engine with medium grade LHR combustion chamber consisting of air gap insulated piston and air gap insulated liner by employing finite element technique was studied using ANSYS programme in order to evaluate the performance of the lubricating oil with the change of configuration of the combustion chamber Murali Krishna MVS. (2004). Heat flow analysis showed average reduction of 31% of heat flow through the insulated piston and 16% with insulated liner of the engine with LHR combustion chamber in comparison with piston and liner of the CE. Murali Krishna M.V.S. (2004).

The present paper attempted to evaluate the performance of the engine with high grade LHR combustion chamber, which 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 mineral diesel at similar operating conditions. Finite element method was applied to determine the performance of lubricating oil.

Material and Methods

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 Anirudh Gautam et.al, (2013).

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 Anirudh Gautam et.al, (2013). 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, Krishna Maddali et.al, (2014).

Engine with LHR combustion chamber

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.

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.1. 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%.

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%.

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.

In-cylinder pressure was measured by a water-cooled piezoelectric pressure transducer (AVL Austria: QC34D). A precision shaft encoder (AVL Austria: 365x) with a crank angle (CA) resolution of 0.5 crank angle degrees (CAD) was used for determining the crankshaft position. 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.

Thermal Analysis

Finite element (FE) analysis using ANSYS program was adopted for determining the temperature distribution within the piston, liner and cylinder head of CE and insulated engine. The mesh required for the analysis was generated using ANSYS software employing eight–noded quadrilateral isoparametric elements of 970 elements with 1039 nodes**.** Boundary conditions were taken from Reference Murali Krishna MVS. (2004).

Assumptions for FE Analysis: **The heat transfer inside the air gap is by conduction only. The properties of the various materials used are independent of temperature. Film coefficients are constant and same for all the configurations. Air alone exists in between the liner and the piston in the contact area.**

Experiment for validation of FE Results

In order to determine the temperature of air in air gap of insulated liner, holes were drilled through the cylinder liner at three circumferential locations (120^o apart) to insert the thermocouples in the air gap of the insulated liner of LHR combustion chamber. Insert made of superni was removed and these thermocouples were attached to the liner across the air gap by spot welding. Once welding was completed, superni insert was inserted again with the body of the liner by threading. Thermocouples were connected to the temperature sensors for measuring the temperature. The experimental set up employed calculated the average temperature of air in the air gap of the liner of the engine with LHR combustion chamber.

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 uncertainty analysis (Minimum number of trials must be not less than ten).

Result and Discussion

Table.1 Properties of test fuels Anirudh Gautam et.al, (2013).

Property	Units	Diesel	Biodiesel	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**	**--**

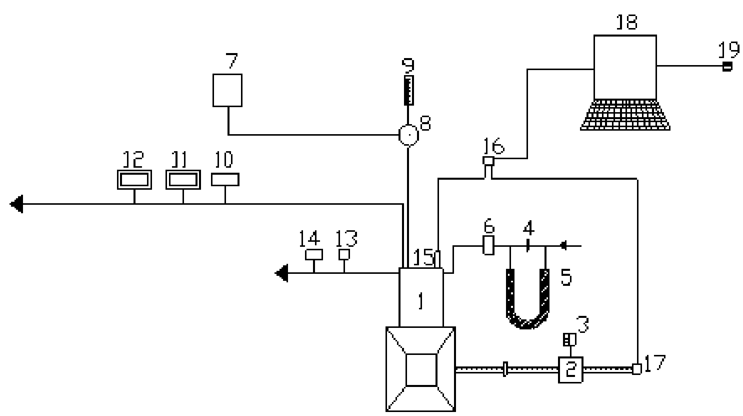

Fig.1 **Schematic diagram of experimental set–up**

1.Engine, 2.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.Smoke opacity meter, 12. NO_x Analyzer, 13.Outlet jacket water temperature indicator, 14. Outlet-jacket water flow meter, 15.Piezo-electric pressure transducer, 16.Console, 17.TDC encoder, 18.Personal Computer and 19. Printer.

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^obTDC × 190 bar
Number of holes of injector and size	Three × 0.25 mm
Type of combustion chamber	Direct injection type

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

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 mineral diesel operation Venkateswara Rao N, (2013). Curves in Fig.2 indicate that CE with biodiesel at 27^o bTDC showed comparable performance at all loads due to improved combustion with the presence of oxygen, when compared with mineral diesel operation on CE at 27^o bTDC.

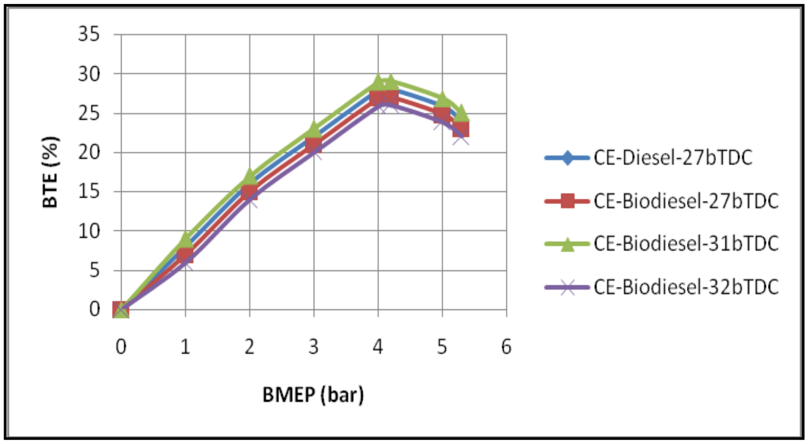

Fig. 2 Variation of brake thermal efficiency (BTE) with brake mean effective pressure (BMEP) in conventional engine (CE) with biodiesel at various injection timings at an injector opening pressure of 190 bar.

CE with biodiesel operation at 27^o bTDC decreased peak BTE by 3%, when compared with diesel operation on CE. This was due to low calorific value and high viscosity of biodiesel. CE with biodiesel operation increased BTE at all loads with advanced injection timing, when compared with CE with biodiesel operation at 27^o bTDC. This was due to initiation of combustion at early period and increase of resident time of fuel with air leading to increase of peak pressures. 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.

From Fig.3, 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 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.

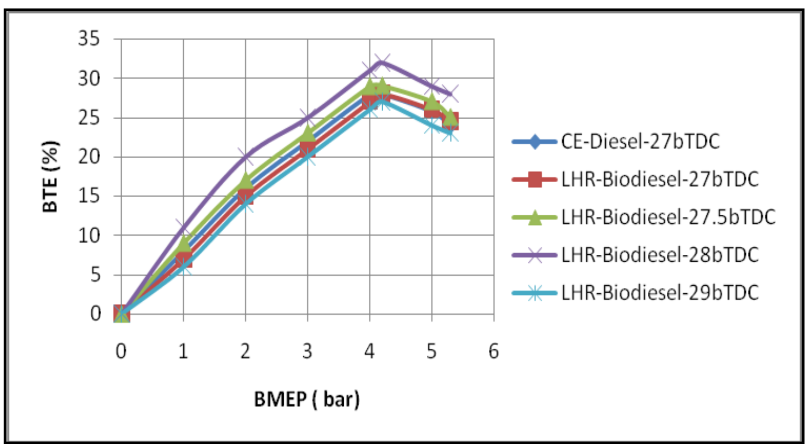

Fig.3 Variation of brake thermal efficiency (BTE) with brake mean effective pressure (BMEP) in engine with LHR combustion chamber with biodiesel at various injection timings at an injector opening pressure of 190 bar.

From Fig.4, it is observed that engine with LHR combustion chamber with biodiesel operation increased peak BTE by 3% at 27^o bTDC and 10% at 31^o bTDC in comparison with CE with biodiesel operation at 27^o bTDC and at 31^o bTDC. This was due to improved evaporation of biodiesel in hot environment provided by the engine with LHR combustion chamber, leading to produce higher peak BTE. Engine with LHR combustion chamber with biodiesel operation showed higher peak BTE than diesel operation on same configuration of the engine. This showed that engine with LHR combustion chamber was more suitable for biodiesel.

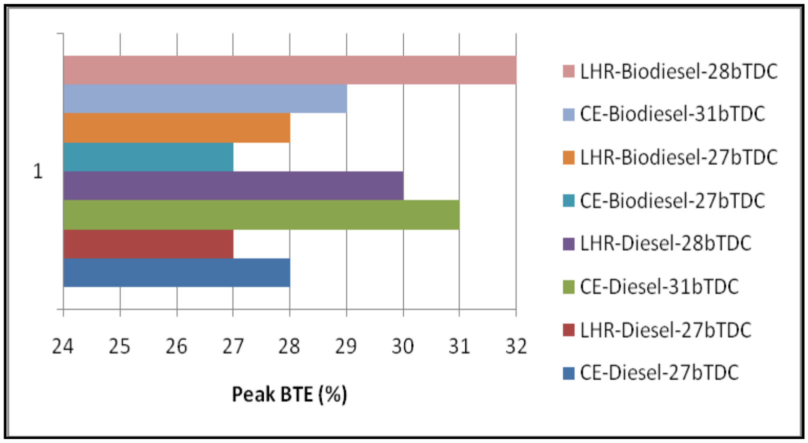

Fig.4 Bar charts showing the variation of peak brake thermal efficiency (BTE) with test fuels with conventional engine (CE) and engine with LHR combustion chamber at recommended and optimized injection timings at an injector opening pressure of 190 bar.

From Fig.5, it is shown that brake specific energy consumption (BSEC) at full load decreased with advanced injection timing with test fuels. This was because of increase of resident time of fuel with air thus improving atomization and thus combustion. BSEC was comparable with biodiesel with CE at 27^o bTDC and 31^o bTDC, when compared with CE with diesel operation at 27^o bTDC and at 31^o bTDC. This was due to improved combustion with higher cetane number and presence of oxygen in fuel composition with higher heat release rate with biodiesel leading to produce comparable BSEC at full load. Engine with LHR combustion chamber with biodiesel decreased BSEC at full load operation by 6% at 27^o bTDC and 3% at 28^o bTDC, when compared mineral diesel operation with engine with LHR combustion chamber at 27^o bTDC and at 28^o bTDC. This once again confirmed that engine with LHR combustion chamber was more suitable for biodiesel operation than mineral diesel operation. Engine with LHR combustion chamber with biodiesel decreased BSEC at full load operation by 3% at 27^o bTDC and 2% at 28^o bTDC, in comparison with CE at 27^o bTDC and at 31^o bTDC with biodiesel. This was due to improved evaporation rate and higher release rate of fuel with LHR combustion chamber.

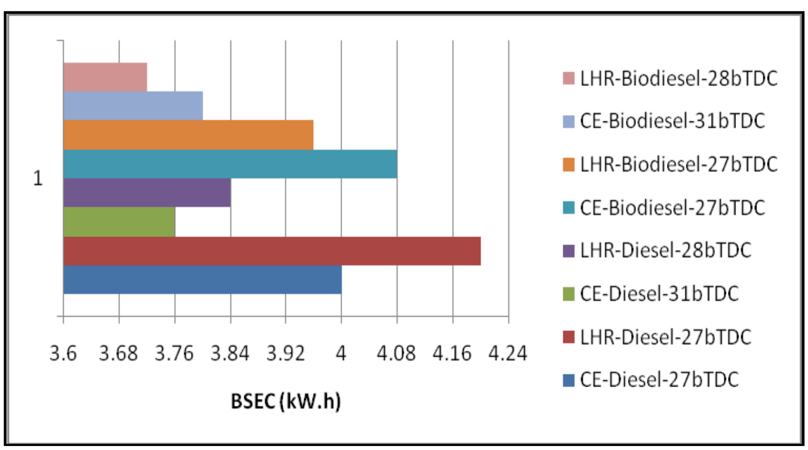

Fig.5 Bar charts showing the variation of brake specific energy consumption (BSEC) at full load operation with test fuels with conventional engine (CE) and engine with LHR combustion chamber at recommended and optimized injection timings at an injector opening pressure of 190 bar.

From Fig.6, it is noticed that, exhaust gas temperature (EGT) at full load operation decreased with advanced injection timing with test fuels. This was because, when the injection timing was advanced, the work transfer from the piston to the gases in the cylinder at the end of the compression stroke was too large, leading to reduce EGT. CE with biodiesel operation increased EGT at full load operation by 12% at 27^o bTDC and 7% at 31^o bTDC in comparison with CE with mineral diesel operation at 27 ^{o} bTDC and at 31^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. Similar findings were obtained by other researchers Krishna Maddali, (2014). From Fig.6, it is noticed that engine with LHR combustion chamber with biodiesel operation decreased EGT at full load operation by 10% at 27^o bTDC and showed a comparable value at 28^o bTDC, when compared mineral diesel operation with engine with LHR combustion chamber at 27^o bTDC and at 28^o bTDC. This was due to reduction of heat rejection and more conversion of exhaust gas enthalpy into useful work with biodiesel operation, in comparison with mineral diesel operation. Engine with LHR combustion chamber with biodiesel increased EGT at full load operation by 16% at 27^o bTDC and 19% at 28^o bTDC, in comparison with CE at 27^o bTDC and at 31^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. Similar observations were reported by previous researchers Venkateswara Rao N et.al, (2013), Venkateswara Rao N et, al, (2013), Murali Krishna MVS et.al, (2014).

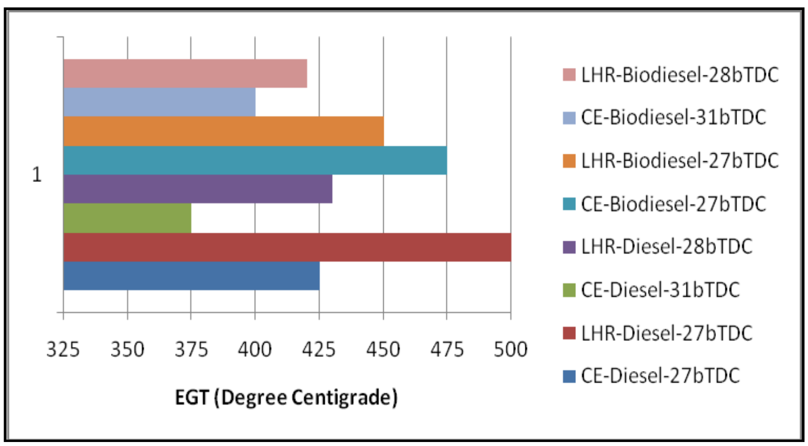

Fig.6 Bar charts showing the variation of Exhaust Gas Temperature (EGT) at full load operation with test fuels with conventional engine (CE) and engine with LHR combustion chamber at recommended and optimized injection timings at an injector opening pressure of 190 bar.

IT/ Combustion Chamber Version	Test fuel	Peak Brake Thermal Efficiency (%)				Brake Specific Energy consumption at full load operation (kW.h)				Exhaust Gas Temperature (°C) at full load operation			
		Injector opening pressure (bar)				Injector opening pressure (bar)				Injector opening pressure (bar)			
		190		270		190		270		190		270	
		NT	PT	NT	PT	NT	PT	NT	PT	NT	PT	NT	PT
27(CE)	DF	28	--	30	--	4.0	--	3.84	--	425	---	395	---
	BD	27	28	29	30	4.08	4.04	4.0	3.96	475	500	425	450
27(LHR)	DF	27	--	29	--	4.2	--	4.12	--	500	--	450	---
	BD	28	29	30	31	3.96	3.92	3.88	3.84	450	425	400	375
28(LHR)	DF	30	--	31.5	--	3.84	--	3.76	--	430	---	390	--
	BD	32	32.5	33	33.5	3.72	3.68	3.64	3.6	420	400	380	350
31(CE)	DF	31	--	31	--	3.76	--	3.84	--	375	---	325	--
	BD	29	29.5	30.5	31	3.80	3.76	3.72	3.68	400	425	350	375

CE with biodiesel increased coolant load by 3% at 27° bTDC and 10% at 31° bTDC when compared with mineral diesel operation on CE at 27° bTDC and 31° bTDC as observed from Fig.7.

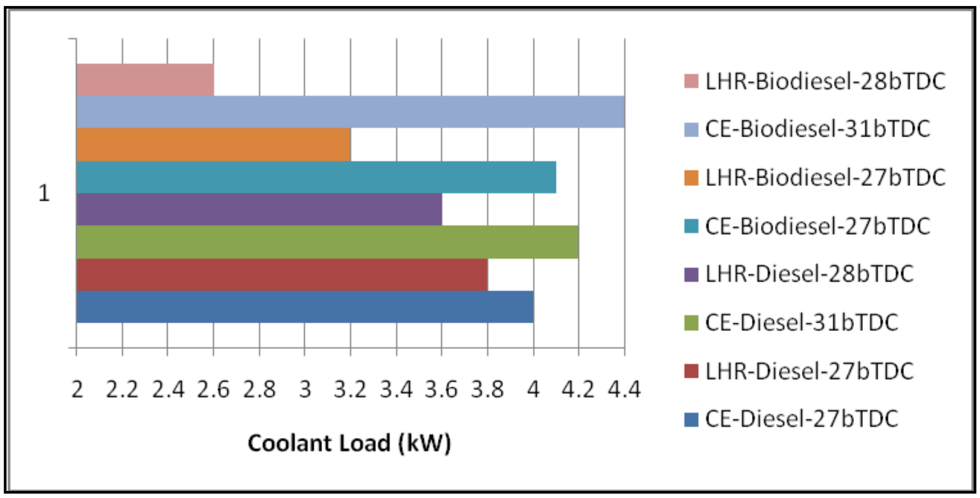

Fig.7 Bar charts showing the variation of coolant load at full load operation with test fuels with conventional engine (CE) and engine with LHR combustion chamber at recommended and optimized injection timings at an injector opening pressure of 190 bar.

This was due to increase of un–burnt fuel concentration at the combustion chamber walls and increase of gas temperatures with biodiesel of higher cetane number than mineral diesel. Similar trends were reported in previous studies Venkateswara Rao N et.al, (2013), Venkateswara Rao N et.al, (2013), Murali Krishna MVS et.al, (2014). Coolant load at full load operation increased in CE, while decreasing the same in engine with LHR combustion chamber with advanced injection timing with biodiesel. In case of CE, un–burnt fuel concentration reduced with effective utilization of energy, released from the combustion, coolant load with test fuels increased marginally at full load operation, with an increase of gas temperatures, when the injection timing was advanced to the optimum value.. The reduction of coolant load in engine with LHR combustion chamber was due to 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 due to recovery from coolant load at their optimum injection timings with test fuels. Engine with LHR combustion chamber with biodiesel operation decreased coolant load operation by 16% at 27^o bTDC and 28% at 28^o bTDC, when compared mineral diesel operation with engine with LHR combustion chamber at 27^o bTDC and at 28^o bTDC. This was due to more conversion of heat into useful work with biodiesel than mineral diesel. Fig.7 indicates that engine with LHR combustion chamber with biodiesel decreased coolant load at full load operation by 7% at 27^o bTDC and 41% at 28^o bTDC, in comparison with CE at 27^o bTDC and at 31^o bTDC. This was due to provision of thermal insulation and improved combustion with engine with LHR combustion chamber. Similar observations were reported by previous researchers Venkateswara Rao N et.al, (2013), Venkateswara Rao N et.al, (2013), Murali Krishna MVS et.al, (2014).

Fig.8 indicates that CE with biodiesel operation decreased volumetric efficiency at full load by 2% at 27^o bTDC and comparable at 31^o bTDC, when compared with mineral diesel operation on CE at 27^o bTDC and 31^o bTDC. This was due to increase of combustion chamber wall temperatures with biodiesel operation on CE.

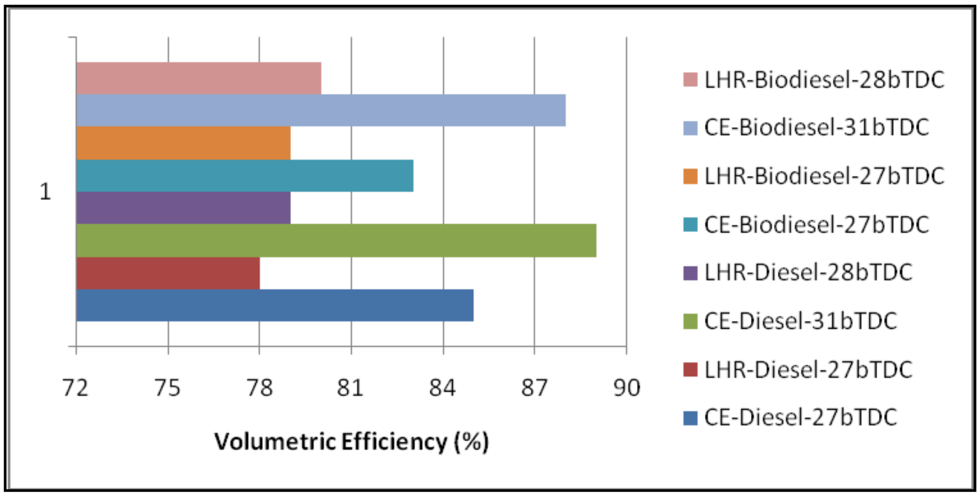

Fig.8 Bar charts showing the variation of volumetric efficiency at full load operation with test fuels with conventional engine (CE) and engine with LHR combustion chamber at recommended and optimized injection timings at an injector opening pressure of 190 bar.

Volumetric efficiency at full load operation improved marginally with advanced injection timing with test fuels with both configurations of the combustion chamber. This was due to improved combustion, reduction of EGT and hence combustion wall temperatures with advanced injection timing. From Fig.8, it is noticed that volumetric efficiencies at full load operation on engine with LHR combustion chamber at 27^o bTDC and at 28^o bTDC with biodiesel were marginally higher than mineral diesel operation on same configuration of the engine at 27^o bTDC and 28^o bTDC. This was due to lower EGT with biodiesel operation than mineral diesel. Fig.8 indicates that engine with LHR combustion chamber with biodiesel decreased volumetric efficiency at full load operation by 5% at 27^o bTDC and 9% at 28^o bTDC, in comparison with CE at 27^o bTDC and at 31^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 Venkateswara Rao N et.al, (2013), Venkateswara Rao N et.al, (2013), Murali Krishna MVS et.al, (2014).

Coolant load at full load operation decreased with preheating of biodiesel, as noticed from Table.5. This was because of improved spray characteristics leading to reduce gas temperatures and hence coolant load.

From Table 4, 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. Similar observations were noticed by previous researchers. Pugazhvadivu M, et.al, (2005), Agarwal D, et,al, (2007), Hanbey Hazar et.al, (2010). .Peak BTE improved marginally with an increase of injector opening pressure in both versions of the combustion chamber with test fuels. As injector opening pressure increased, droplet diameter decreased influencing the atomization quality, and more dispersion of fuel particle, resulting in enhanced mixing with air, leads to improved oxygen-fuel mixing rate, as extensively reported in the literature Celikten I, (2003), Avinash Kumar et.al, (2013).

From Table.4, it is observed that BSEC at full load operation decreased marginally with an increase of injector opening pressure. This was because of improved heat release rate with reduction of size of fuel particle. BSEC at full load operation decreased with preheating of biodiesel in both versions of the combustion chamber. This was due to improved spray characteristics with the reduction of viscosity of biodiesel.

From Table.4, 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. EGT at full load reduced marginally with an increase of injector opening pressure in both versions of the combustion chamber as seen from Table.4. This was due to improved combustion with improved oxygen–fuel ratios.

Table.4

Comparative data on Peak Brake Thermal Efficiency, Brake Specific Energy Consumption and Exhaust Gas Temperature

Table.5 Comparative data on Coolant Load & Volumetric Efficiency at full load operation

IT/ Combustion Chamber Version	Test fuel	Coolant Load (kW)				Volumetric Efficiency (%)			
		Injector opening pressure (bar)				Injector opening pressure (bar)			
		190		270		190		270	
		NT	PT	NT	PT	NT	PT	NT	PT
27(CE)	DF	4.0	-	4.4	--	85		87	
	BD	4.1	3.9	4.5	4.3	83	82	85	84
27(LHR)	DF	3.8	-	3.4	--	78		80	
	BD	3.2	3.2	2.8	2.6	79	80	81	82
28(LHR)	DF	3.6	-	3.0	--	79	-	81	-
	BD	2.6	2.4	2.2	2.0	80	81	82	83
31(CE)	DF	4.2	-	4.6	--	89		87	
	BD	4.4	4.2	4.8	4.6	88	87	90	89

From Table.5, it is observed that that coolant load at full load operation marginally increased in CE, while decreasing it in engine with LHR combustion chamber with increased injector opening pressure with test fuels. This was due to the fact that increased injector opening pressure increased nominal fuel spray velocity resulting in improved fuel–air mixing with which gas temperatures increased in CE. The reduction of coolant load in engine with LHR combustion chamber was due to improved fuel spray characteristics and increase of oxygen–fuel ratios causing decrease of gas temperatures and hence the 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.5. This was because of increase of EGT in CE, while decreasing of the same in engine with LHR combustion chamber. Volumetric efficiency at full load operation was marginally increased with an increase of injector opening pressure in both versions of the combustion chamber with test fuels. This was due to improved oxygen–fuel ratios and reduction of EGT. However, these variations were very small.

Exhaust emissions

Particulate emissions and NO_x are the exhaust emissions from diesel engine causes health hazards Khopkar SM. (2011). 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 Heywood JB, (2013). From Fig.9, 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 mineral diesel operation on CE at 27^o bTDC and at 31^o bTDC.

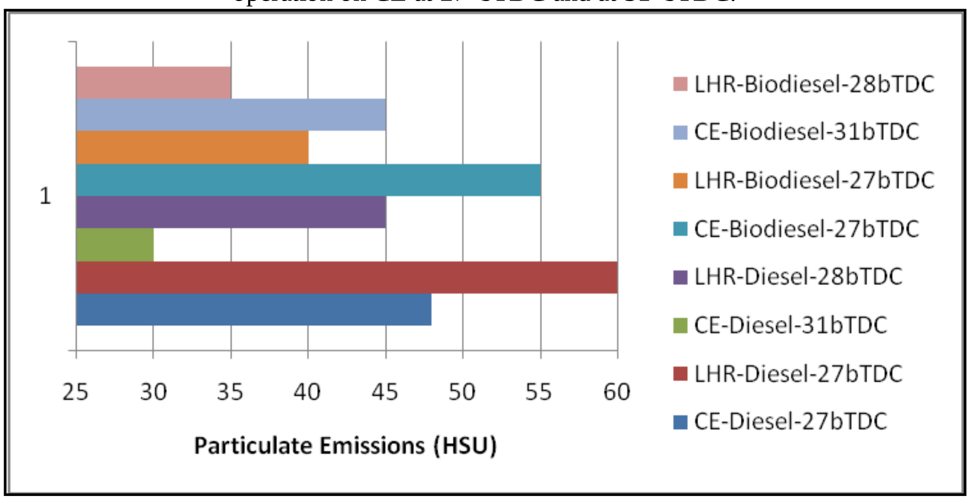

Fig.9. Bar charts showing the variation of particulate emissions at full load operation with test fuels with conventional engine (CE) and engine with LHR combustion chamber at recommended and optimized injection timings at an injector opening pressure of 190 bar.

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 Heywood JB, (2013). (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 Anirudh Gautam et.al, (2013). From Fig.9, 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. This was due to increase of air entrainment, causing lower particulate emissions. From Fig.9, 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 mineral diesel operation on engine with LHR combustion chamber at 27^o bTDC and at 28^o bTDC. This was due to improved combustion of higher cetane value biodiesel in the hot environment provided by engine with LHR combustion chamber. Fig.9 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. This was due to improved combustion of biodiesel with improved oxygen– fuel ratios in the LHR version of the combustion chamber. The temperature and availability of oxygen are the reasons for the formation of NO_x levels.

From Fig.10, it is observed that CE with biodiesel operation increased NOx levels at full load by 6% at 27^o bTDC and 5% at 31^o bTDC, when compared with mineral diesel operation on CE at 27^o bTDC and at 31^o bTDC. 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). [18]. this was also due to the 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 (1564 MPa) of compressibility, with the in-line fuel injection system. Similar observations were made by earlier researchers. Anirudh Gautam et.al, (2013), Rao PV, (2011). From Fig.10, 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.

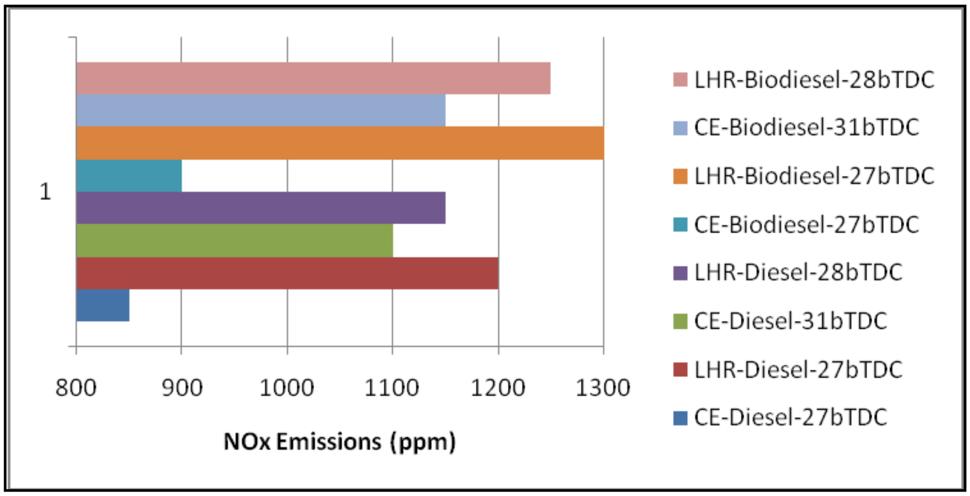

Fig.10. Bar charts showing the variation of nitrogen oxide (NO_x) levels at full load operation with test fuels with conventional engine (CE) and engine with LHR combustion chamber at recommended and optimized injection timings at an injector opening pressure of 190 bar.

From Fig.10, 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 mineral diesel operation on engine with LHR combustion chamber at 27^o bTDC and at 28^o bTDC. This was due to higher cetane value of biodiesel than mineral diesel. 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.

Decreasing the fuel density tends to increase spray dispersion and spray penetration. Particulate emissions at full load were decreased with preheating of biodiesel in both versions of the combustion chamber, as seen in Table.6. This is due to 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.6

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

IT/ Combustion Chamber Version	Test fuel	Particulate emissions (HSU)				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.6, 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. Anirudh Gautam et.al, (2013), Celikten I, (2003), Avinash Kumar Agarwal et.al, (2013). From Table6, it is noticed that NOx 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.6, 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. This was due to enhanced spray characteristics, thus improving fuel air mixture preparation and evaporation process in CE. Improved combustion with improved oxygen–fuel ratios in engine with LHR combustion chamber reduced particulate emissions.

Combustion Characteristics

From Table.7, it is noticed that CE with biodiesel increased peak pressure (PP) at full load operation by 4% at 27^{o} bTDC and 5% at 31^{o} bTDC when compared with mineral diesel operation on CE at 27^{o} bTDC and at 31^{o} bTDC. Even though biodiesel has lower heat of combustion, it advanced its PP 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. When, a high density (or high bulk modulus) fuel is injected, the pressure wave travels faster from pump end to nozzle end, through a high pressure in-line tube Rao PV, (2011). This causes early lift of needle in the nozzle, causing advanced injection. Hence, the combustion takes place very close to TDC and the PP was slightly high due to existence of smaller cylinder volume near TDC. Similar trends were noticed by earlier studies Anirudh Gautam et.al, (2013), Rao PV, (2011). PP at full load operation with biodiesel increased with CE, while marginally decreasing it in engine with LHR combustion chamber with advanced injection timings. This was due to increase of ignition delay of fuel in CE, causing sudden increase of pressures with accumulated fuel with advanced injection timings. Improved combustion with improved air–fuel ratios decreased PP at full load operation on engine with LHR combustion chamber. Table.7 indicates that engine with LHR combustion chamber with biodiesel increased PP at full load operation by 35% at 27^{o} bTDC and 5% at 28^{o} bTDC, in comparison with CE at 27^{o} bTDC and at 31^{o} bTDC. This was because of improved heat release rate with engine with LHR combustion chamber. However, PP at full load operation reduced with preheating, with both versions of the combustion chamber, as observed from Table.7. 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 PP of preheated biodiesel was also attributed to earlier combustion caused by short ignition delay (due to faster evaporation of the fuel) at their preheated temperatures. Similar trends were noticed by earlier researchers Pugazhvadivu M et.al, (2005), Agarwal D et.al, (2007), Hanbey Hazar et.a. (2010). PP at full load operation increased in CE, while marginally reducing it with engine with LHR combustion chamber with an increase of injector opening pressure with test fuels. This may be due to smaller sauter mean diameter shorter breakup length, better dispersion, and better spray and atomization characteristics. This improves combustion rate in the premixed combustion phase in CE. Similar tends were reported by earlier studies Celikten I, (2003), Avinash Kumar Agarwal et.al, (2013). Improved combustion with improved air–fuel ratios marginally reduced PP at full load operation with engine with LHR combustion chamber. Similar trends were reported by earlier researchers Venkateswara Rao N et.al, (2013), Venkateswara Rao N et.al, (2013), Murali Krishna MVS et.al, (2014).

Table.7 Data of combustion characteristics at full load operation

IT/ Combustion Chamber Version	Test fuel	PP (bar)				MRPR (bar/deg)				TOPP (deg)			
		Injector opening pressure (bar)				Injector opening pressure (bar)				Injector opening pressure (bar)			
		190		270		190		270		190		270	
		NT	PT	NT	PT	NT	PT	NT	PT	NT	PT	NT	PT
27(CE)	DF	50.4	--	53.5	---	5.4	--	6.0	--	9		9	
	BD	52.5	50.5	55.5	53.4	4.9	3.9	5.2	4.2	8	7	7	6
27(LHR)	DF	66.5	--	60.6	--	7.6		6.8		10		9	
	BD	70.8	68.6	67.6	65.7	6.8	6.1	6.3	5.9	7	6	6	5
28(LHR)	DF	63.5	-	59.5	--	7.2		6.4		9		8	
	BD	68.5	66.6	63.6	61.6	6.5	5.5	5.7	5.5	6	5	5	4
31(CE)	DF	62.4	--	60.6	--	6.2	--	5.8	--	8		9	
	BD	65.4	63.6	62.6	59.5	5.6	4.4	6.0	4.8	7	6	8	7

Table.7 denotes that maximum rate of pressure rise (MRPR) was higher for mineral diesel than biodiesel with both versions of the combustion. This was due to highly volatile nature of diesel fuel releasing more energy per unit crank angle. Similar trends were reported by earlier researcher Rao PV, (2011). MRPR at full load operation followed similar trends at different operating conditions as in case of PP at full load operation. Preheated biodiesel gave lower MRPR when compared with normal biodiesel as in the case of peak pressure. Engine with LHR combustion chamber increased MRPR at full load operation by 39% at 27° bTDC and 16% at 28° bTDC with engine, when compared with CE at 27° bTDC and at 31° bTDC with biodiesel, which showed that combustion improved with improved air–fuel ratios in hot environment provided by the engine with LHR combustion chamber.

AT recommended injection timing and optimum 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 mineral diesel operation, as noticed from Table.7 This was because of higher bulk modulus of rigidity and cetane number of biodiesel, in comparison with neat diesel operation. TOPP at full load operation decreased (towards TDC) with the advanced injection timing at different operating conditions of biodiesel with both versions of the combustion chamber as noticed from Table.7, which showed that combustion improved with advanced injection timing. TOPP at full load operation decreased marginally with an increase of injector opening pressure with test fuels as seen from same table. This was due to improved spray characteristics and reduction of size of fuel particle. This once again established the fact by observing drastic increase of peak pressure and lower TOPP at full load operation, engine with LHR combustion chamber showed improved performance over CE. 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.

3.4 Finite Element Analysis

It is desirable that the temperatures and distances are non-dimensionalized in order to show the trends instead of presenting data in absolute values. From Fig.11,, it is noticed for insulated piston, there was reduction of surface temperature from 689°C on the axis of the piston to 120°C on its periphery, while in the case of insulated liner, the temperature was reduced from 608°C at inner edge of the liner to 227°C on the its outer periphery, and for cylinder head, temperature was deceased from 771°C at the inner edge to 590°C on the outer periphery. The temperature drop was due cooling of components and resistance offered by them against heat flow. Similar trends were noticed by other researcher in the case of air gap insulated piston and air gap insulated liner. [15].

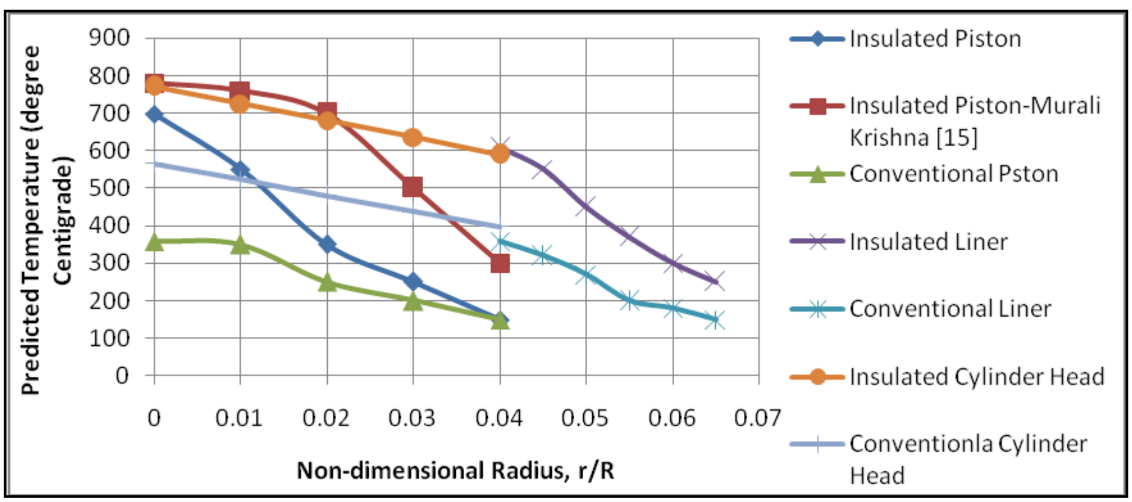

Fig.11. **Variation of the predicted temperature of piston, liner and cylinder head with radius of these components inboth versions of the combustion chamber.**

From fig, 12, it may be noted that a reduction of heat flow from 65% at axis of the piston to about 5% on the periphery was observed by the authors, which gave average heat reduction of 35% while Murali Krishna reported 70% reduction from axis to 5% on periphery which worked out to be average heat reduction of 37.5%. Murali Krishna MVS. (2004).

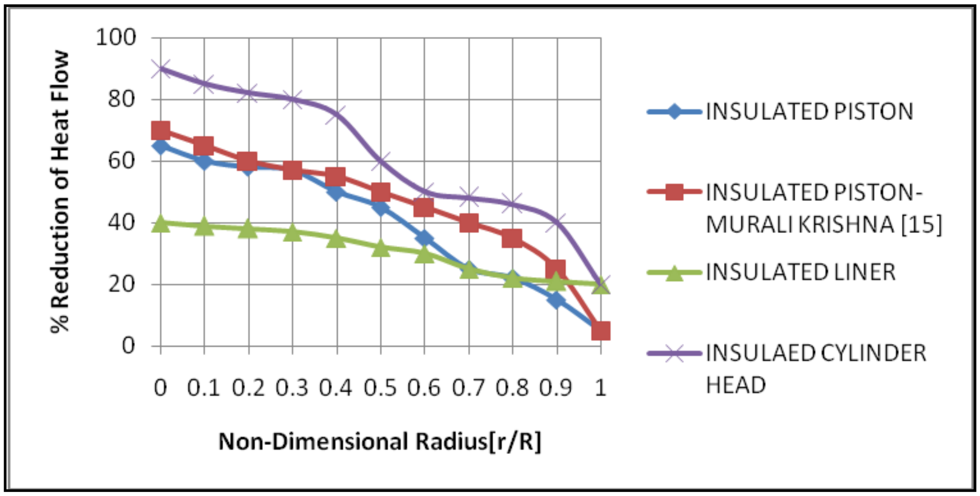

Fig.12 Variation of the percentage reduction in the heat flux with non- dimensional radius for the insulated piston insulated liner and insulated cylinder head over the similar components of conventional engine.

From the same Figure, it is noticed that, that a reduction of heat flux for insulated liner was 40% at axis to 20% on the periphery, which gave average heat reduction of 30% when compared to the conventional liner. It was estimated the reduction in heat flow with ceramic coated cylinder head was 90% at axis and 20% at periphery, which gave average reduction in heat flux was 55%, when compared to the conventional cylinder head.

Validation of FEM Results

Table.8 shows the comparison on the variation of temperature of the air in the air gap of the insulated liner at a radial distance of 41.5 mm obtained theoretically by means of FEM and experiment with provision of thermocouples. The difference in results obtained by authors and other researcher may be due to ceramic coating provided in the cylinder head. Murali Krishna MVS. (2004). However, the analytical analysis predicted higher values of temperatures over the experimental results as expected due to the assumptions made in analysis. The deviation of theoretical result predicted by FEM was 4% from experimental result. Uncertainty, which is a measure of accuracy involved for measuring temperature with temperature sensor, was 3.8% by uncertainty analysis. However, trends were matching well with those of other researcher. Murali Krishna MVS. (2004).

Table.8

Researcher	Temperature of air in air gap of the insulated piston (^{o}C)		Temperature of air in air gap of the insulated liner (^{o}C)	
	Predicted value (^{o}C)	Measured value (^{o}C)	Predicted value (^{o}C)	Measured value (^{o}C)
Authors	312	--	270	260
Murali Krishna [15]	431	--	162	150

Comparison with the other researcher on temperature distribution across insulated piston and insulated liner

From Fig.13, it is observed that the temperature in lubricating oil zone determined by FEM method was within the limits (less than 250^{o}C) and hence the concept of LHR combustion chamber can be safely employed as lubricating oil was not deteriorated at higher temperatures.

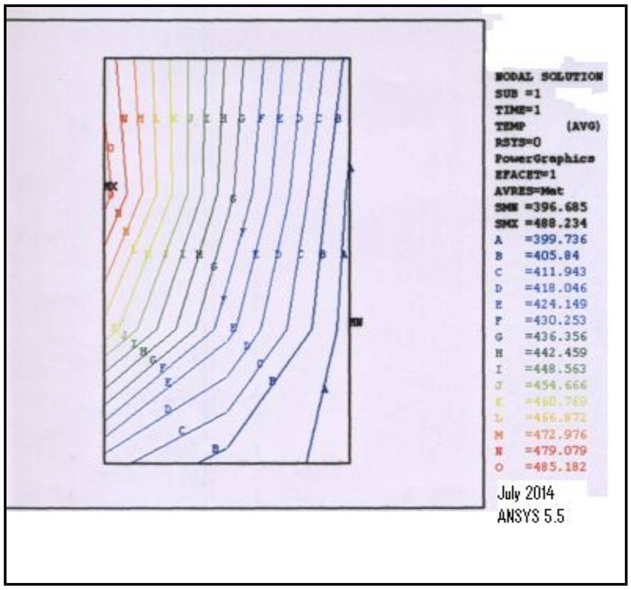

Fig13. Isotherms from the finite element analysis in the vicinity of lubricating oil of the insulated engine

1. 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 performance, exhaust emissions and combustion characteristics were improved with advanced injection timing, increase of injector opening pressure and with preheating with both versions of the combustion chamber with biodiesel.**
   4. **The temperature of the air in the air gap of insulated liner of the engine with LHR combustion chamber predicted by FE analysis matched well with the experimental investigations.**
   5. Heat flow analysis showed average reduction of 40% heat flow through the insulated components in comparison with CE.
   6. Temperature of the lubricating oil determined by FEM was within the limit with insulated engine.

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.

Recommendations

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 [19].

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