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PERFORMANCE, EXHAUST EMISSIONS AND COMBUSTION CHARACTERISTICS OF COTTON SEED OIL BASED BIODIESEL IN CERAMIC COATED DIESEL ENGINE

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ABSTRACT

Experiments were conducted to evaluate the performance of a LHR diesel engine with ceramic coated cylinder head [ceramic coating of thickness 500 microns was done on inside portion of cylinder head] with different operating conditions [normal temperature and pre-heated temperature] of cotton seed oil in biodiesel form with varied injector opening pressure and injection timing and compared the performance with pure diesel operation on conventional engine. Performance parameters (brake thermal efficiency, brake specific energy consumption, exhaust gas temperature, volumetric efficiency, coolant load, sound levels) and exhaust emissions (smoke levels and oxides of nitrogen) were determined at various values of brake mean effective pressure of the engine, while combustion characteristics were measured at peak load operation of the engine with biodiesel operation. Conventional engine showed compatible performance, while LHR engine showed improved performance with biodiesel at recommended injection timing and pressure. The performance of both versions of the engine improved with advanced injection timing and at higher injector opening pressure with biodiesel operation when compared with conventional engine with pure diesel operation. The optimum injection timing was 33^{o}bTDC for conventional engine while it was 30^{o}bTDC for LHR engine with biodiesel operation.

KEYWORDS: Alternate Fuels, Biodiesel, LHR Engine, Fuel Performance, Exhaust Emissions, Combustion Characteristics

INTRODUCTION

In the context of fast depletion of fossil fuels, increase of pollution levels with fossil fuels and increase of fuel prices in International Market causing economic burden on developing countries, the search for alternate fuels has become pertinent. Vegetable oils are important substitute for diesel fuel as they have properties compatible to diesel fuel. They are renewable in nature.

Several researchers [1-5] 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. Not only that, the common problems of crude vegetable oils in diesel engines are formation of carbon deposits, oil ring sticking, thickening and gelling of lubricating oil as a result of contamination by the vegetable oils. The presence of the fatty acid components greatly affects the viscosity of the oil. The increase in viscosity and crystal formation of fatty acids below cloud point hinders the operation of the injector. Increase saturated hydro carbon content increases the cloud point of the oil. The limitation of unsaturated fatty acids is necessary due to the fact heating higher unsaturated fatty acids results in polymerization of glycerides. This can leads to formation of deposits or to deterioration of lubricating oil. The different fatty acids present in the vegetable oil are palmic, steric, lingoceric, oleic, linoleic and fatty acids. These fatty acids increase smoke emissions and also lead to incomplete combustion due to improper air-fuel mixing. These problems can be solved, if neat vegetable oils are chemically modified to bio-diesel.

The process of chemical modification is not only used to reduce viscosity, but to increase the cloud and pour points. The higher viscosity of the oil affects the spray pattern, spray angle, droplet size and droplet distribution. Bio-diesels derived from vegetable oils present a very promising alternative to diesel fuel since biodiesels have numerous advantages compared to fossil fuels as they are renewable, biodegradable, provide energy security and foreign exchange savings besides addressing environmental concerns and socio-economic issues. Experiments were carried out [6-10] with bio-diesel on CE and reported performance was compatible with pure diesel operation on CE. The drawbacks associated with biodiesel call for hot combustion chamber provided by low heat rejection (LHR) diesel engine.

The concept of LHR engine is to provide thermal insulation in the path of heat flow to the coolant and increase thermal efficiency of the engine. Hence LHR engines are classified as per the degree of insulation. Low grade LHR engines consist of ceramic coating on engine components such as top surface of the piston, cylinder head and cylinder liner. Medium grade LHR engines are air gap insulated engines, where air gap is created in the piston and other components with low-thermal conductivity materials like superni (an alloy of nickel whose thermal conductivity is one sixteenth of that of aluminium alloy), cast iron and mild steel etc. High grade LHR engines are the combination of low grade LHR engines and medium grade LHR engines. Ceramic coatings with pure diesel operation provided adequate insulation and improved brake specific fuel consumption (BSFC) which was reported by various researchers. However previous studies [11-13] revealed that the thermal efficiency variation of LHR engine not only depended on the heat recovery system, but also depended on the engine configuration, operating condition and physical properties of the insulation material. Investigations were carried [14-18] out on ceramic coated diesel engine with biodiesel and it was reported that biodiesel operation on LHR engine increased thermal efficiency of the engine marginally and decreased smoke emissions. However, it increased NOx levels.

Since interest is beginning to build up in the area of bio-diesel, the present paper attempted to evaluate the performance of LHR engine, which contained ceramic coated cylinder head with varied injector opening pressure and injection timing with different operating conditions of crude cotton seed oil with varied engine parameters of injector opening pressure and injection timing and compared with conventional engine at recommended injection timing and injector opening pressure.

METHODOLOGY

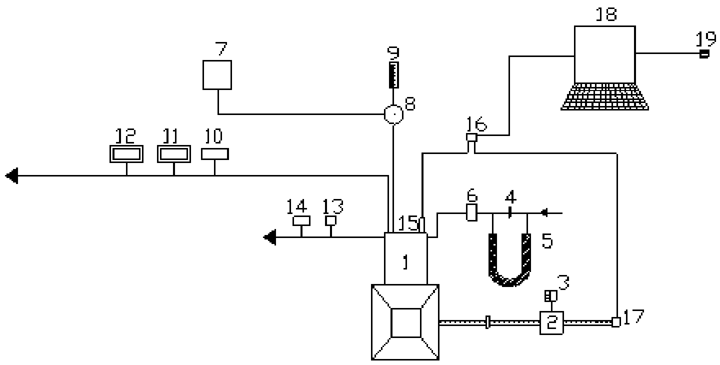
The process of converting the vegetable oil into methyl esters was carried out by heating the vegetable oil with the methanol in the presence of the catalyst (Sodium hydroxide). In the present case, vegetable oil (cotton seed oil) was stirred with methanol at around 60-70^{o}C with 0.5% of NaOH based on weight of the oil, for about 3 hours. At the end of the reaction, excess methanol is removed by distillation and glycerol, which separates out was removed. The methyl esters were treated with dilute acid to neutralize the alkali and then washed to get free of acid, dried and distilled to get pure vegetable oil esters. The properties of the vegetable oil ester and the diesel used in this work are presented in Table-1.

The LHR diesel engine contained ceramic coated cylinder head. Partially stabilized zirconium (PSZ) of thickness 500 microns was coated on inside portion of cylinder head.

Table 1: Properties of Test Fuels

Test Fuel	Kinematic Viscosity at 40°C (Centi-Stroke)	Specific Gravity at 25 °C	Cetane Number	Calorific Value (kJ/kg)
Diesel(DF)	2.25	0.84	55	42000
Esterified Cotton seed oil ECSO, biodiesel	3.0	0.87	50	39500

The experimental setup used for the investigations of LHR diesel engine with biodiesel is shown in Figure 1. The fuel injector had 3-holes of size 0.25-mm. The combustion chamber consisted of a direct injection type with no special arrangement for swirling motion of air. The engine was connected to electric dynamometer for measuring its brake power. Burette method was used for finding fuel consumption of the engine. Air-consumption of the engine was measured by air-box method (with the help of U-tube water manometer, orifice meter). The naturally aspirated engine was provided with water-cooling system in which inlet temperature of water was maintained at 80°C by adjusting the water flow rate. 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 and its effect on the performance of the engine was studied, along with the change of injector opening pressure from 190 bar to 270 bar (in steps of 40 bar) using nozzle testing device. The maximum injector opening pressure was restricted to 270 bar due to practical difficulties involved. Exhaust gas temperature (EGT) was measured with thermocouples made of iron and iron-constantan.



1. Engine, 2. Electical Dynamo Meter, 3. Load Box, 4. Orifice 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 Meter, 12. Netel Chromatograph NOx Analyzer, 13. Outlet Jacket Water Temperature Indicator, 14. Outlet-Jacket Water Flow Meter, 15. Piezo-Electric Pressure Transducer, 16. Console, 17. TDC Encoder, 18. Pentium Personal Computer and 19. Printer.

Figure 1: Experimental Set-up

Exhaust emissions of smoke and NO_x were recorded by AVL smoke meter and Netel Chromatograph NOx analyzer respectively at various values of brake mean effective pressure of the engine. Piezo electric transducer, fitted on the cylinder head to measure pressure in the combustion chamber was connected to a console, which in turn was connected to Pentium personal computer. TDC encoder provided at the extended shaft of the dynamometer was connected to the console to measure the crank angle of the engine. A special P-θ software package evaluated the combustion characteristics such as peak pressure (PP), time of occurrence of peak pressure (TOPP), maximum rate of pressure rise (MRPR) and time of occurrence of maximum rate of pressure rise (TOMRPR) from the signals of pressure and crank angle at the peak load operation of the engine. Pressure-crank angle diagram was obtained on the screen of the personal computer. The photograph of ceramic coated cylinder head was shown in Plate 1. The specifications of the test engine were given in Table-2. The specifications of analyzers were shown in Table-3. The test fuels used in the experimentation were pure diesel and biodiesel. The various configurations of the engine were conventional engine and LHR engine.

Plate 1: Photographic View of the Ceramic Coated Cylinder Head

Table 2: Specifications of the 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
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	5.31 bar
Manufacturer’s recommended injection timing and pressure	27^{o}bTDC × 190 bar
Dynamometer	Electrical dynamometer
Number of holes of injector and size	Three × 0.25 mm
Type of combustion chamber	Direct injection type

Different operating conditions of the vegetable oil were normal temperature and preheated temperature. The different injector opening pressure attempted in this experimentation were 190 bar, 230 bar and 270 bar, while injection timings attempted were 27-34^{o}bTDC.

Table 3: Specifications of Analyzers

Name of the Analyzer	Measuring Range	Precision	Resolution
AVL Smoke meter	0-100 HSU	1 HSU	1 HSU
Netel Chromatograph NOx analyzer	0-2000 ppm	2 ppm	1 ppm
Sound Analyzer	0-150 Decibels	1 decibel	1 decibel

RESULTS AND DISCUSSIONS

Performance Parameters

Curves from Figure 2 indicate that brake thermal efficiency increased with biodiesel operation with conventional engine, up to 80% of the peak load due to increase of fuel efficiency and beyond that load, it decreased due to decrease of combustion efficiency.

Biodiesel operation on CE showed compatible performance when compared with pure diesel operation on CE.

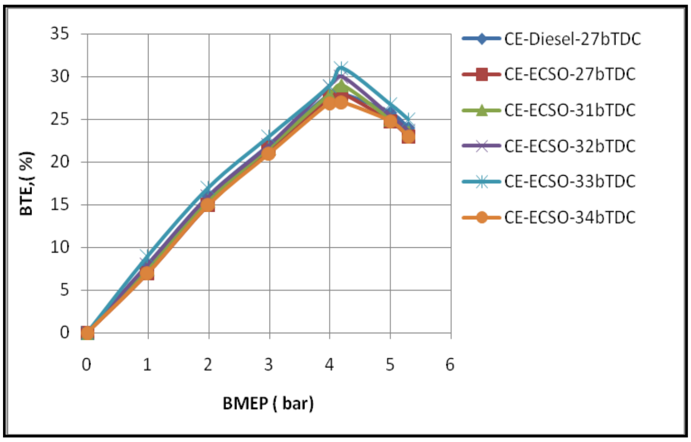


Figure 2: Variation of Brake Thermal Efficiency (BTE) with Brake Mean Effective Pressure (BMEP) in Conventional Engine (CE) at an Injector Opening Pressure of 190 Bar with Biodiesel (ECSO) Operation at Various Injection Timings

This was due to lower calorific value and high viscous nature of biodiesel. Brake thermal efficiency increased at all loads when the injection timing was advanced to 33^{o}bTDC in the conventional engine at the normal temperature of biodiesel. The increase of brake thermal efficiency at optimum injection timing over the recommended injection timing with biodiesel with conventional engine as attributed to its longer ignition delay and combustion duration.

Curves from Figure 3 indicate LHR version of engine with biodiesel operation at recommended injection timing showed improvement in the performance for the entire load range compared with CE with pure diesel. 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 LHR engine improved heat release rates and efficient energy utilization. The optimum injection timing was found to be 30^{o}bTDC with LHR engine with normal biodiesel. Further advancing of the injection timing resulted in decrease in thermal efficiency due to longer ignition delay. Hence it was concluded that the optimized performance of the LHR engine was achieved at an injection timing of 30^{o}bTDC.Since the hot combustion chamber of LHR engine reduced ignition delay and combustion duration and hence the optimum injection timing (30^{o}bTDC) was obtained earlier with LHR engine when compared with CE (33^{o}bTDC) with the biodiesel operation.

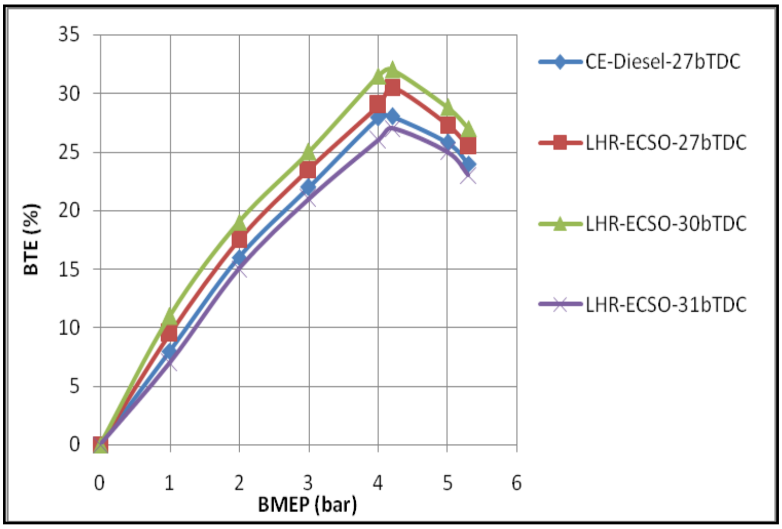


Figure 3: Variation of Brake Thermal Efficiency (BTE) with Brake Mean Effective Pressure (BMEP) in LHR Engine at an Injector Opening Pressure of 190 Bar with Biodiesel (ECSO) Operation at Various Injection Timings

Injector opening pressure was varied from 190 bar to 270 bar to improve the spray characteristics and atomization of the biodiesel and injection timing was advanced from 27 to 34^{o}bTDC for conventional engine and LHR engine. From Table.4, it could be noticed that brake thermal efficiency increased with increase in injector opening pressure in both versions of the engine at different operating conditions of the biodiesel.

The improvement in brake thermal efficiency at higher injector opening pressure was due to improved fuel spray characteristics. However, the optimum injection timing was not varied even at higher injector opening pressure with LHR engine, unlike the conventional engine. Hence it was concluded that the optimum injection timing was 33^{o}bTDC at 190 bar, 32^{o}bTDC at 230 bar and 31^{o}bTDC at 270 bar for CE with biodiesel.

The optimum injection timing for LHR engine was 30^{o}bTDC irrespective of injector opening pressure with biodiesel. Peak brake thermal efficiency was higher in LHR engine when compared with conventional engine with different operating conditions of the biodiesel. Preheating of the biodiesel reduced the viscosity, which improved the spray characteristics of the oil.

Table 4: Data of Peak BTE

Injection Timing (bTDC)	Test Fuel	Peak BTE (%)											
		Conventional Engine (CE)						LHR Engine					
		Injector Opening Pressure (Bar)						Injector Opening Pressure (Bar)					
		190		230		270		190		230		270	
		NT	PT	NT	PT	NT	PT	NT	PT	NT	PT	NT	PT
27	DF	28	--	29	---	30	--	29	--	30	--	30.5	--
	ECSO	28	29	27.5	28	27	27.5	30.5	31	31	31.5	31.5	32
30	ECSO	--	--	--	--	--	--	32	32.5	32.5	33	33	33.5
33	ECSO	31	31.5	31	31.5	31	31.5	--	--	--	--	--	--

From Table.5, brake specific energy consumption (BSEC) (defined as inverse of brake thermal efficiency) at peak load operation decreased with the advanced injection timing and increase of injector opening pressure with both versions of the engine with different operating conditions of biodiesel.

This was due to initiation of combustion at earlier period and efficient combustion with improved air fuel ratios [19] giving lower brake specific energy consumption (BSEC). That shows lower energy substitution and effective energy utilization of biodiesel, which could replace 100% diesel fuel.

Table 5: Data of BSEC at Peak Load Operation

Injection Timing (bTDC)	Test Fuel	Brake Specific Energy Consumption (kW/kW) at Peak Load Operation											
		Conventional Engine (CE)						LHR Engine					
		Injector Opening Pressure (Bar)						Injector Opening Pressure (Bar)					
		190		230		270		190		230		270	
		NT	PT	NT	PT	NT	PT	NT	PT	NT	PT	NT	PT
27	DF	4		3.96		3.92		3.98	--	3.92	--	3.88	--
	ECSO	4.1	3.9	3.9	3.8	3.8	3.7	3.90	3.86	3.86	3.82	3.82	3.78
30	ECSO	3.92	3.88	3.88	3.84	3.84	3.80	3.82	3.78	3.78	3.74	3.74	3.70
33	ECSO	3.86	3.82	3.90	3.86	3.92	3.88	-	--	--	--	--	--

From the Figure 4, it is observed that conventional engine with biodiesel operation at 27^{o}bTDC recorded marginally higher value of exhaust gas temperature at all loads compared with conventional engine with pure diesel operation.

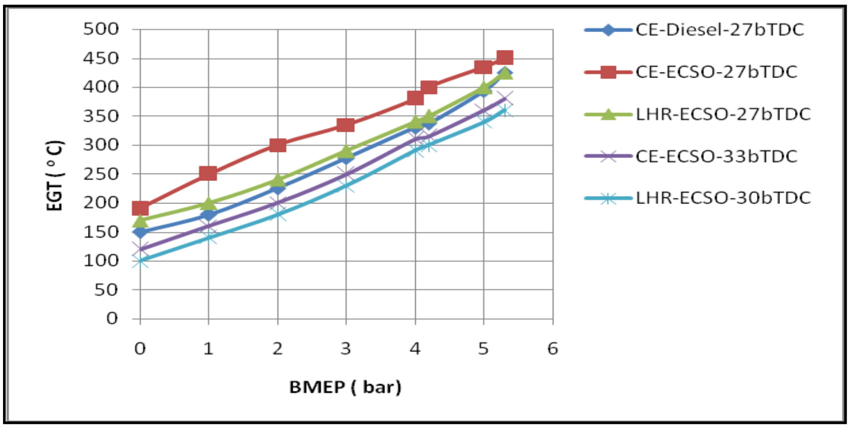


Figure 4: Variation of Exhaust Gas Temperature (EGT) with Brake Mean Effective Pressure (BMEP) in Conventional Engine (CE) and LHR Engine at an Injector Opening Pressure of 190 Bar with Biodiesel (ECSO) Operation at Recommended and Optimized Injection Timings

Lower heat release rates [19] and retarded heat release associated with high specific energy consumption caused increase in value of exhaust gas temperature in conventional engine. Ignition delay in the conventional engine with different operating conditions of biodiesel increased the duration of the burning phase. At recommended injection timing. LHR engine recorded lower value of exhaust gas temperature when compared with conventional engine with biodiesel operation. This was due to reduction of ignition delay in the hot environment with the provision of the insulation in the LHR engine, which caused the gases expanded in the cylinder giving higher work output and lower heat rejection. This showed that the performance improved with LHR engine over conventional engine with biodiesel operation. The value of exhaust gas temperature decreased with advancing of the injection timing with both versions of the engine with biodiesel operation. At the respective optimum injection timings, the value of exhaust gas temperature was lower with LHR engine than that of convention al engine with biodiesel operation. This was due to more conversion of heat into work with LHR engine than conventional engine.

From the Table 6, it is observed that exhaust gas temperature [19] decreased with increase of injector opening pressure and advanced injection timing with both versions of the engine with biodiesel which confirmed that performance increased with increase of injector opening pressure. Preheating of the biodiesel further reduced the value of exhaust gas temperature, compared with normal vegetable oil in both versions of the engine. This was due to improved air fuel ratios [19]. This showed that thermal efficiency increased with preheated condition of the biodiesel when compared with normal condition of the biodiesel leading to less amount of heat rejection and high amount of actual conversion of heat into work.

Table 6: Data of Exhaust Gas Temperature (EGT) at Peak Load Operation

Injection Timing (bTDC)	Test Fuel	Exhaust Gas Temperature (Degree Centigrade) at Peak Load Operation											
		Conventional Engine (CE)						LHR Engine					
		Injector Opening Pressure (Bar)						Injector Opening Pressure (Bar)					
		190		230		270		190		230		270	
		NT	PT	NT	PT	NT	PT	NT	PT	NT	PT	NT	PT
27	DF	425	--	410	---	395	--	460	---	450	--	440	--
	ECSO	450	425	425	400	400	375	425	400	400	375	375	350
30	ECSO	420	400	400	380	380	360	360	340	340	320	320	300
33	ECSO	380	360	400	380	410	390						

It can be observed in Figure 5 that volumetric efficiency (VE) decreased with an increase of brake mean effective pressure in both versions of the engine with biodiesel operation. This was due to increase of gas temperature [19] with the load. At the recommended injection timing, volumetric efficiency decreased at all loads in both versions of the engine with biodiesel operation when compared with conventional engine with pure diesel operation. Volumetric efficiency mainly depends on speed of the engine, valve area, valve lift, timing of the opening or closing of valves and residual gas fraction rather than on load variation. Hence with biodiesel oil operation with conventional engine, volumetric efficiency decreased in comparison with pure diesel operation on conventional engine, as residual gas fraction increased**.** This was due to increase of deposits [19] with biodiesel operation with conventional engine.

The reduction of volumetric efficiency with LHR engine was due increase of temperature of incoming charge in the hot environment created with the provision of insulation, causing reduction in the density and hence the quantity of air with LHR engine. Volumetric efficiency increased marginally in conventional engine and LHR engine at optimized injection timings when compared with recommended injection timing with biodiesel. This was due to decrease of un-burnt fuel fraction in the cylinder leading to increase in volumetric efficiency in conventional engine and reduction of gas temperatures [19] with LHR engine.

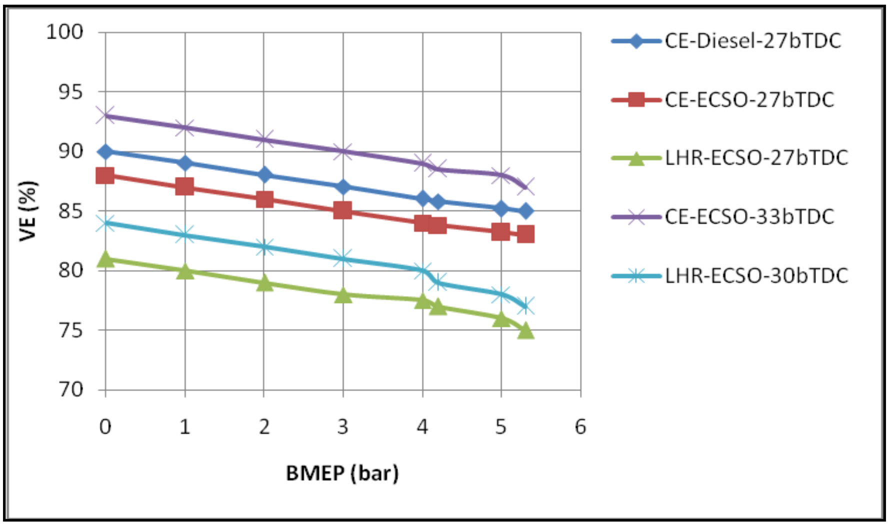


Figure 5: Variation of Volumetric Efficiency (VE) with Brake Mean Effective Pressure (BMEP) in Conventional Engine (CE) and LHR Engine at an Injector Opening Pressure of 190 Bar with Biodiesel Operation (ECSO) at Recommended Injection Timing and Optimized Injection Timing

From Table 7, volumetric efficiency increased with increase of injector opening pressure and with advanced injection timing in both versions of the engine with vegetable oil. This was also due to improved fuel spray characteristics and evaporation at higher injector opening pressure leading to marginal increase of volumetric efficiency. This was also due to the reduction of residual fraction of the fuel with the increase of injector opening pressure. Preheating of the biodiesel marginally improved volumetric efficiency in both versions of the engine, because of reduction of un-burnt fuel concentration with efficient combustion, when compared with the normal temperature of the biodiesel.

Curves from Figure 6 indicate that that coolant load (CL) increased with increase of brake mean effective pressure (BMEP) in both versions of the engine with test fuels. This was due to increase of gas temperatures with increase of fuel consumption. Coolant load was observed to be higher with conventional engine with biodiesel operation when compared with diesel operation on conventional engine.

This was because of increase of un-burnt fuel concentration at the walls of combustion chamber. However, coolant load decreased with LHR version of the engine with biodiesel operation when compared with conventional engine with pure diesel operation. Heat output was properly utilized and hence thermal efficiency increased and heat loss to coolant decreased with effective thermal insulation with LHR engine.

Table 7: Data of Volumetric Efficiency at Peak Load Operation

Injection Timing (° bTDC)	Test Fuel	Volumetric Efficiency (%) at Peak Load Operation											
		Conventional Engine (CE)						LHR Engine					
		Injector Opening Pressure (Bar)						Injector Opening Pressure (Bar)					
		190		230		270		190		230		270	
		NT	PT	NT	PT	NT	PT	NT	PT	NT	PT	NT	PT
27	DF	85	--	86	--	87	--	78		79		80	
	ECSO	83	84	84	85	85	86	75	76	76	77	77	78
30	ECSO	84	85	85	86	86	87	77	78	78	79	79	80
33	ECSO	87	88	86	87	85	86	--	--	--	--	--	--

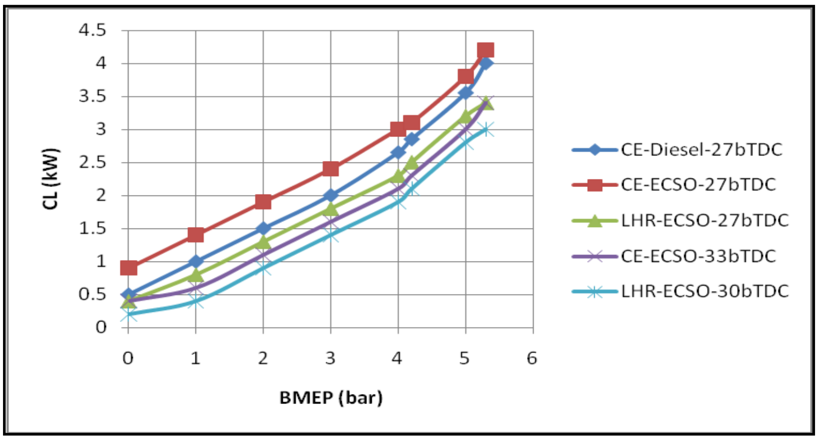


Figure 6: Variation of Coolant Load (CL) with Brake Mean Effective Presure (BMEP) in Conventional Engine (CE) and LHR Engine at an Injector Opening Pressure of 190 Bar with Biodiesel (ECSO) Operation at Recommended Injection Timing and Optimized Injection Timing

Coolant load decreased with advanced injection timing with both versions of the engine with biodiesel operation. This was due to improved air fuel ratios [19] and reduction of gas temperatures. From Table 8, it is noticed that coolant load decreased with advanced injection timing and with increase of injector opening pressure with biodiesel. This was because of improved combustion with increase of air fuel ratios [19] and reduction of gas temperatures [19]. Coolant load decreased with preheated condition of biodiesel in comparison with normal biodiesel in both versions of the engine. This was because of improved spray characteristics.

Table 8: Data of Coolant Load at Peak Load Operation

Injection Timing (bTDC)	Test Fuel	Coolant Load (kW) at Peak Load Operation											
		Conventional Engine (CE)						LHR Engine					
		Injector Opening Pressure (Bar)						Injector Opening Pressure (Bar)					
		190		230		270		190		230		270	
		NT	PT	NT	PT	NT	PT	NT	PT	NT	PT	NT	PT
27	DF	4.0	---	3.8	--	3.6	---	3.8	---	3.6	--	3.4	---
	ECSO	4.2	4.0	3.8	3.6	3.6	3.4	3.4	3.2	3.2	3.0	3.0	2.8
30	ECSO	4.0	3.8	3.6	3.4	3.4	3.3	3.0	2.8	2.8	2.6	2.6	2.4
33	ECSO	3.4	3.2	3.6	3.4	3.8	3.6	--	--	--	--	--	--

Figure 7 indicates that sound intensity increased(initiation of combustion) up to 80% of the peak load operation fuel and at 80% of the peak load it decreased marginally (combustion efficiency is maximum) and beyond that load it increased again with test fuels at recommended and optimized injection timings. Sound intensities marginally increased in conventional engine at recommended injection timing with biodiesel operation in comparison with conventional engine with pure diesel operation. Higher viscosity, duration of combustion and poor volatility caused moderate combustion of biodiesel leading to generate higher sound levels. LHR engine decreased sound intensity when compared with pure diesel operation on conventional engine. This was because of hot environment in LHR engine improved the combustion of biodiesel. This was also due to decrease of density at higher temperatures leading to produce lower levels of sound with LHR engine. When injection timings were advanced to optimum, sound intensities decreased for both versions of the engine, due to early initiation of combustion and improved air fuel ratios [18].

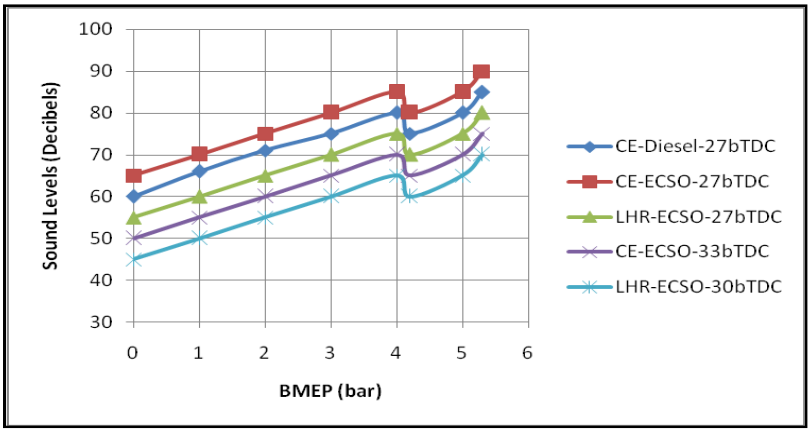


Figure 7: Variation of Sound Levels with Brake Mean Effective Pressure (BMEP) in Conventional Engine (CE) and LHR Engine at an Injector Opening Pressure of 190 Bar with Biodiesel Operation (ECSO) at Recommended Injection Timing and Optimized Injection Timing

Table 9 denotes that the sound intensity decreased with increase of injector opening pressure for both versions of the engine with the biodiesel. This was because of improved combustion with increased air fuel ratios [19]. This was due to improved spray characteristic of the fuel, with which there was no impingement of the fuel on the walls of the combustion chamber leading to produce efficient combustion. Sound intensities were lower at preheated condition of vegetable oil when compared with their normal condition. This was due to improved spray characteristics, decrease of density

Table 9: Data of Sound Intensity at Peak Load Operation

Injection Timing (°bTDC)	Test Fuel	Peak BTE (%)											
		Conventional Engine (CE)						LHR Engine					
		Injector Opening Pressure (Bar)						Injector Opening Pressure (Bar)					
		190		230		270		190		230		270	
		NT	PT	NT	PT	NT	PT	NT	PT	NT	PT	NT	PT
27	DF	85	--	80	--	95	--	90	--	85	--	80	--
	ECSO	90	85	85	80	80	75	80	75	75	70	70	65
30	ECSO	85	80	80	75	75	70	70	65	65	60	60	55
33	ECSO	75	70	80	75	82	78	--	--	--	--	-	--

Exhaust Emissions

Figure 8 indicates that the value of smoke intensity increased from no load to full load in both versions of the engine with test fuels. During the first part, the smoke level was more or less constant, as there was always excess air present. However, in the higher load range there was an abrupt rise in smoke levels due to less available oxygen, causing the decrease of air-fuel ratio [18], leading to incomplete combustion, producing more soot density. The variation of smoke levels with the brake power, typically showed a U-shaped behavior due to the pre-dominance of hydrocarbons in their composition at light load and of carbon at high load. Marginal increase of smoke levels at all loads with conventional engine fuelled with biodiesel was observed when compared with pure diesel operation on CE. This was due to the higher value of ratio of C/H (C= Number of carbon atoms, H= Number of hydrogen atoms in fuel composition) (0.6) when compared with pure diesel (0.45). The increase of smoke levels was also due to decrease of air-fuel ratios [19] and volumetric efficiency. Smoke levels were related to the density of the fuel. Smoke levels were higher with biodiesel due to its high density. However, LHR engine marginally decreased smoke levels due to efficient combustion and less amount of fuel accumulation on the hot combustion chamber walls of the LHR engine at different operating conditions of the biodiesel compared with the conventional engine. Smoke levels decreased at the respective optimum injection timing with both versions of the engine with biodiesel. This was due to initiation of combustion at early period with both versions of the engine.

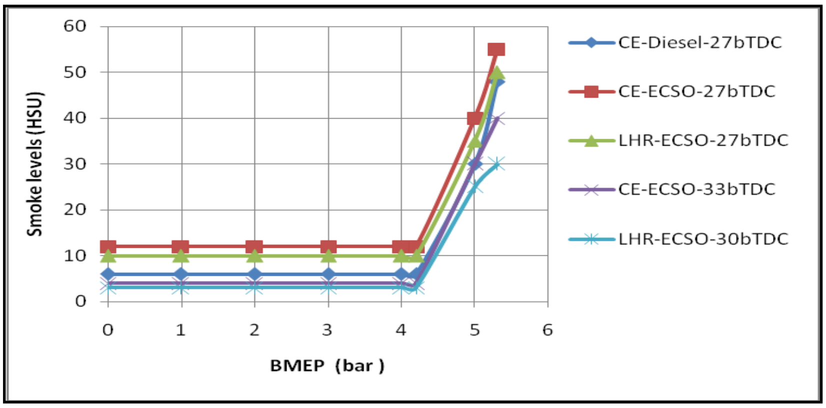


Figure 8: Variation of Smoke Levels with Brake Mean Effective Pressure (BMEP) in Conventional Engine (CE) and LHR Engine at an Injector Opening Pressure of 190 Bar With Biodiesel (ECSO) Operation at Recommended Injection Timing and Optimized Injection Timing

The data from Table 10 shows smoke levels decreased with increase of injection timing and the injector opening pressure in both versions of the engine, with different operating conditions of the biodiesel. This was due to improvement in the fuel spray characteristics with higher injector opening pressure and increase of air entrainment, at the advanced injection timings, causing lower smoke levels. Preheating of the biodiesel decreased smoke levels in both versions of the engine, when compared with normal temperature of the biodiesel. This was due to i) the reduction of density of the biodiesel, as density was directly related to smoke levels, ii) the reduction of the diffusion combustion proportion in conventional engine with the preheated biodiesel, iii) 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 was directed into the combustion chamber**.**

Table 10: Data of Smoke Levels at Peak Load Operation

Injection timing (bTDC)	Test Fuel	Smoke Levels (Hartridge Smoke Unit)											
		CE						LHR Engine					
		Injector Opening Pressure (Bar)						Injector Opening Pressure (Bar)					
		190		230		270		190		230		270	
		NT	PT	NT	PT	NT	PT	NT	PT	NT	PT	NT	PT
27	DF	48	--	38	--	34	--	55	--	50	--	45	--
	ECSO	55	50	50	45	45	40	50	45	45	40	40	35
30	ECSO							30	25	25	20	20	15
33	ECSO	40	35	45	40	43	40						

Availability of oxygen and high temperatures are favorable conditions to form NOx levels. Figure 9 indicates for both versions of the engine, NOx concentrations raised steadily as the fuel/air ratio increased with increasing brake mean effective pressure at constant injection timing. At part load, NOx concentrations were less in both versions of the engine. This was due to the availability of excess oxygen. At remaining loads, NOx concentrations steadily increased with the load in both versions of the engine. This was because, local NOx 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. At peak load, with higher peak pressures, and hence temperatures, and larger regions of close-to-stoichiometric burned gas, NOx levels increased in both versions of the engine. Thus NOx emissions should be roughly proportional to the mass of fuel injected (provided burned gas pressures and temperature do not change greatly).

It is noticed that NOx levels were marginally higher in conventional engine while they were drastically higher in LHR engine at different operating conditions of the biodiesel at the peak load when compared with diesel operation. This was due to lower heat release rate because of high duration of combustion causing lower gas temperatures [19] with the biodiesel operation on conventional engine, which marginally increased NOx levels. Increase of combustion temperatures with the faster combustion and improved heat release rates [19] associated with the availability of oxygen in LHR engine caused drastically higher NOx levels in LHR engine.

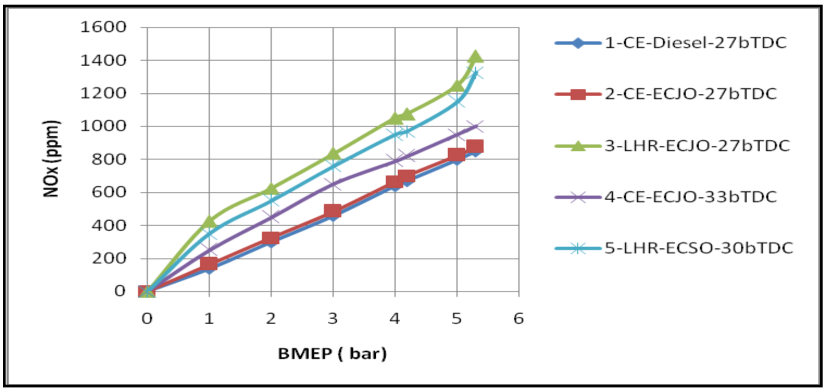


Figure 9: Variation of NOx Levels with Brake Mean Effective Pressure (BMEP) in Conventional Engine (CE) And LHR Engine at an Injector Opening Pressure of 190 Bar with Biodiesel (ECSO) Operation at Recommended Injection Timing and Optimized Injection Timing

The data in Table-11 shows that, NOx levels increased with the advancing of the injection timing in CE with different operating conditions of biodiesel. Residence time and availability of oxygen had increased, when the injection timing was advanced with biodiesel which caused higher NOx levels in conventional engine. However, NOx levels decreased marginally with increase of injection timing with in LHR engine at different operating conditions of biodiesel. This was due to decrease of gas temperatures [19] with the increase of air-fuel ratios [19]. NOx levels decreased with increase of injector opening pressure with different operating conditions of biodiesel. With the increase of injector opening pressure, fuel droplets penetrate and find oxygen counterpart easily.

Turbulence of the fuel spray increased the spread of the droplets which caused decrease of gas temperatures [19] marginally thus leading to decrease in NOx levels. Marginal decrease of NOx levels was observed in LHR engine, due to decrease of combustion temperatures [19] with improved air fuel ratios [19]. The fuel spray properties may be altered due to differences in viscosity and surface tension. The spray properties affected may include droplet size, droplet momentum, degree of mixing, penetration, and evaporation. The change in any of these properties may lead to different relative duration of premixed and diffusive combustion regimes. Since the two burning processes (premixed and diffused) have different emission formation characteristics, the change in spray properties due to preheating of the vegetable oil were lead to reduction in NOx formation. As fuel temperature increased, there was an improvement in the ignition quality, which caused shortening of ignition delay. A short ignition delay period lowered the peak combustion temperature which suppressed NOx formation. Lower levels of NOx was also attributed to retarded injection, improved evaporation, and well mixing of preheated biodiesel due to its low viscosity at preheated temperature of 80°C. Hence lower levels of NOx were observed with preheated biodiesel in comparison with normal biodiesel.

Table 11: Data of NOx Levels at Peak Load Operation

Injection timing (bTDC)	Test Fuel	NOx Levels (ppm) at Peak Load Operation											
		CE						LHR Engine					
		Injector Opening Pressure (Bar)						Injector Opening Pressure (Bar)					
		190		230		270		190		230		270	
		NT	PT	NT	PT	NT	PT	NT	PT	NT	PT	NT	PT
27	DF	850	--	800	--	750	--	1200	--	1150	--	1100	--
	ECSO	875	825	825	775	775	725	1425	1375	1375	1325	1325	1275
30	ECSO	950	900	900	850	850	800	1325	1275	1275	1225	1225	1175
33	ECSO	1000	950	950	900	900	850	--	--	--	--	--	--

Combustion Characteristics

From Table 12, it is observed that peak pressures were compatible in conventional engine while they were higher in LHR engine at the recommended injection timing and pressure with biodiesel operation, when compared with pure diesel operation on conventional engine. Peak pressure was slightly higher with biodiesel than that of diesel fuel in LHR engine, even though the calorific value was lower with biodiesel. When, a high density fuel was injected, the pressure wave traveled faster from pump end to nozzle end, through a high pressure in-line tube. This caused early lift of needle in the nozzle, causing advanced injection. Hence, the combustion took place very close to TDC and the peak pressure slightly high due to existence of smaller cylinder volume near TDC. But in case of conventional engine, combustion was not proper with high viscous fuel like biodiesel and hence peak pressures were always lower than those of diesel fuel.

The advantage of using LHR engine for biodiesel was obvious as it could burn low cetane and high viscous fuels. Preheated biodiesel registered marginally higher value of peak pressure than normal biodiesel. This was due to reduction of ignition delay. Peak pressures increased with the increase of injector opening pressure and with the advancing of the injection timing in both versions of the engine, with biodiesel. Higher injector opening pressure produced smaller fuel particles with low surface to volume ratio, giving rise to higher peak pressure. With the advancing of the injection timing to the optimum value with the conventional engine, more amount of the fuel accumulated in the combustion chamber due to increase of ignition delay as the fuel spray found the air at lower pressure and temperature in the combustion chamber. When the fuel- air mixture burns, it produced more combustion temperatures and pressures due to increase of the mass of the fuel. With LHR engine, peak pressures increases due to effective utilization of the charge with the advancing of the injection timing to the optimum value.

It is observed that, peak pressure was higher and time of occurrence of peak pressure was lower with biodiesel operation even though biodiesel has lower CV than that of diesel as biodiesel has compatible cetane number. The value of time of occurrence of peak pressure decreased with the advancing of the injection timing and with increase of injector opening pressure in both versions of the engine, at different operating conditions of the biodiesel. Time of occurrence of peak pressure was found to be higher with different operating conditions of the biodiesel in conventional engine, when compared with pure diesel operation on CE. Preheating of the biodiesel showed lower time of occurrence of peak pressure (TOPP), compared with biodiesel at normal temperature. This once again confirmed by observing the lower TOPP and higher peak pressure, the performance of the both versions of the engine improved with the preheated biodiesel compared with the normal biodiesel. Maximum rate of pressure rise showed similar trends as those of peak pressure in both versions of the engine at different operating conditions of the biodiesel. This trend of increase of maximum rate of pressure rise indicated improved and faster energy substitution and utilization by biodiesel in LHR engine, which could replace 100% diesel fuel. Hence, these combustion characters were within the limits so that biodiesel can be effectively substituted for diesel fuel.

Table 12: Data of PP, MRPR and TOPP at Peak Load Operation

Injection Timing (bTDC)/Test Fuel	Engine Version	PP (Bar) Injector Opening Pressure (Bar)				MRPR (Bar/deg) Injector Opening Pressure (Bar)				TOPP (Deg) Injector Opening Pressure (Bar)			
		190		270		190		270		190		270	
		NT	PT	NT	PT	NT	PT	NT	PT	NT	PT	NT	PT
27/Diesel	CE	50.4	--	53.5	---	3.1	---	3.4	--	9	-	8	--
	LHR	48.1	--	53.0	--	2.9	--	3.1	--	10	--	9	--
27/ECSO	CE	50.1	51.1	53.5	54.5	3.3	3.4	3.5	3.6	10	9	10	9
	LHR	53.8	54.5	55.5	56.8	3.4	3.5	3.6	3.7	9	8	9	8
30/ECSO	LHR	63.4	64.5	65.3	66.5	3.6	3.7	3.7	3.8	9	8	9	8
33/ECSO	CE	55.8	56.7	56.8	58.8	3.5	3.6	3.6	3.7	10	9	11	10

CONCLUSIONS

Relatively, peak brake thermal efficiency increased by 14%, at peak load operation-brake specific energy consumption decreased by 4%, exhaust gas temperature decreased by 65^{o}C, volumetric efficiency decreased by 9%, coolant load decreased by 25%, sound intensity decreased by 18%, smoke levels decreased by 38%, NOx levels increased by 56% and peak pressure increased by 26% with biodiesel operation on LHR engine at its optimum injection timing, when compared with pure diesel operation on CE at manufacturer’s recommended injection timing. Performance parameters, exhaust emissions and combustion characteristics of biodiesel in both versions of the engine improved with preheating and increase of injector opening pressure.

RESEARCH FINDINGS

LHR engine with ceramic coated cylinder head improved the performance with biodiesel in comparison with conventional engine with pure diesel operation. However, it increased NOx emissions and hence research in reduction of these emissions is a worthy.

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