Available online at www.academicpaper.org
Academic @ Paper
ISSN 2146-9067
International Journal of Automotive Engineering and Technologies
Vol. 1, Issue 2, pp. 16 – 31, 2012

International Journal of Automotive Engineering and Technologies
http://www.academicpaper.org/index.php/IJAET

POTENTIAL OF A MEDIUM GRADE LOW HEAT REJECTION DIESEL ENGINE WITH CRUDE JATROPHA OIL

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Received 4 June 2012; Accepted 09 July 2012

ABSTRACT

Investigations were carried out to evaluate the performance of a medium grade LHR diesel engine consisting of air gap insulated piston with 3-mm air gap, with superni (an alloy of nickel) crown and air gap insulated liner with superni insert with different operating conditions of crude jatropha oil with varied injection timing and injection pressure. Performance parameters of brake thermal efficiency (BTE), exhaust gas temperature (EGT) and volumetric efficiency (VE) were determined at various values of brake mean effective pressure (BMEP). Exhaust emissions of smoke and oxides of nitrogen (NOx) were recorded at different values of BMEP. Combustion characteristics of 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 (TOMRPR) were measured with TDC (top dead centre) encoder, pressure transducer, console and special pressure-crank angle software package. Conventional engine (CE) showed deteriorated performance, while LHR engine showed improved performance with crude jatropha oil (CJO) operation when compared with pure diesel operation at recommended injection timing and pressure. The performance of both version of the engine improved with advanced injection timing and higher injection pressure with test fuels. Peak brake thermal efficiency increased by 4%, volumetric efficiency decreased by 8%, smoke levels decreased by 4% and NOx levels increased by 37% with vegetable oil operation on LHR engine at its optimum injection timing, when compared with pure diesel operation on CE at manufacturer's recommended injection timing.

KEY WORDS: Crude jatropha oil, Diesel, CE, LHR engine, Performance, Exhaust emissions, Combustion characteristics

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1. INTRODUCTION

In the scenario of increase of vehicle population at an alarming rate due to advancement of civilization, use of diesel fuel in not only transport sector but also in agriculture sector leading to fast depletion of diesel fuels and increase of pollution levels with these fuels, the search for alternate fuels on has become pertinent for the engine manufacturers, users and researchers involved in the combustion research. Vegetable oils and alcohols are promising substitutes for diesel fuel as they are renewable in nature. Alcohols have low Cetane number and hence engine modification is necessary [1-2] for use as fuel in diesel engine. On the other hand, vegetable oils have compatible properties in comparison with diesel fuel. The idea of using vegetable oil as fuel has been around from the birth of diesel engine. Rudolph diesel, the inventor of the engine [3] that bears his name, experimented with fuels ranging from powdered coal to peanut oil. Several researchers [4-10] 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. Hence crude vegetable oil was converted [11] into biodiesel by treating crude vegetable 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 or biodiesel. Investigations were carried out [12-18] with biodiesel in CE and reported biodiesel showed compatible performance when compared with pure diesel operation on CE. The drawbacks associated with vegetable oils and biodiesels for use in diesel engines call for LHR engines.

It is well known fact that about 30% of the energy supplied is lost through the coolant and the 30% is wasted through friction and other losses, thus leaving only 30% of energy utilization for useful purposes. In view of the above, the major thrust in engine research during the last one or two decades has been on development of LHR engines. The concept of LHR engine is to reduce heat loss to coolant by providing thermal insulation in the path of heat flow to the coolant. LHR engines are classified depending on degree of insulation such as low grade, medium grade and high grade insulated engines. Several methods adopted for achieving low grade LHR engines are using ceramic coatings on piston, liner and cylinder head, while medium grade LHR engines provide air gap in the piston and other components with low-thermal conductivity materials like superni, cast iron and mild steel etc and high grade LHR engine is the combination of low grade and medium grade engines. Though LHR engines with pure diesel operation provided insulation and they improved brake specific fuel consumption (BSFC), peeling of coating was reported by various researchers [19-21] after certain hours of trials.

Experiments were conducted [22-25] on low grade LHR engines with biodiesel and reported biodiesel improved performance and reduced smoke levels, however, they increased NOx levels. Regarding medium grade LHR engines, creating an air gap in the piston involved the complications of joining two different metals. Though it was observed [26] effective insulation provided by an air gap, the bolted design employed by them could not provide complete sealing of air in the air gap. It was made a successful attempt [27-28] of screwing the crown made of low thermal conductivity material, nimonic (an alloy of nickel) to the body of the piston, by keeping a gasket, made of nimonic, in between these two parts. However, low degree of insulation provided by these researchers [27-28] was not able to burn high viscous fuels of vegetable oils.

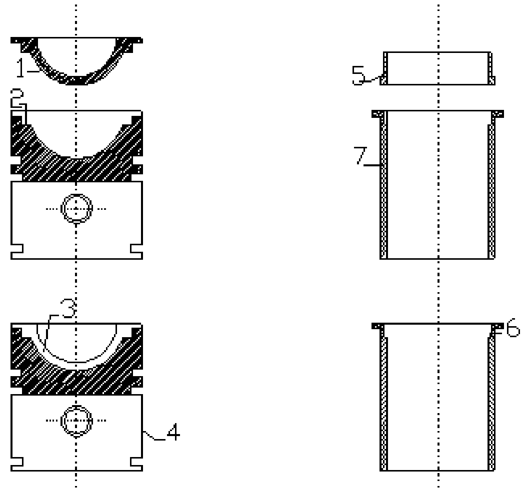
It was studied [29] the performance of a medium grade LHR diesel engine by insulating engine parts employing 2-mm air gap in the piston studded with the body of the piston and the liner with mild steel sleeve fitted with total length of the liner thus attaining a semi-adiabatic condition and reported that the deterioration in the performance of the engine at all loads, when compared to pure diesel operation on CE.

Experiments were conducted [30] on high grade LHR engine, with an air gap insulated piston, air gap insulated liner and ceramic coated cylinder head. The piston with nimonic crown with 2 mm air gap was fitted with the body of the piston by stud design. Mild steel sleeve was provided with 2 mm air gap and it was fitted with the 50 mm length of the liner. The performance was deteriorated with this engine with pure diesel operation, at recommended injection timing. Hence the injection timing was retarded to achieve better performance and pollution levels. Experiments were conducted [31] on high grade LHR engine which contained air gap insulated piston with superni crown with threaded design, air gap insulated liner with superni insert with threaded design and ceramic coated cylinder head with jatropha oil based biodiesel and reported that performance was deteriorated with bio-diesel in CE and improved with LHR engine.

The present paper attempted to evaluate the performance of medium grade LHR engine, which contained air gap piston with superni crown and air gap insulated liner with superni insert with different operating conditions of crude jatropha oil (CJO) with varied injection pressure and injection timing and compared with CE with pure diesel operation at recommended injection timing and injection pressure.

2. MATERIALS AND METHODS

Figure 1. LHR diesel engine contained a two-part piston; the top crown made of low thermal conductivity material, superni-90 screwed to aluminum body of the piston, providing a 3-mm air gap in between the crown and the body of the piston.



1.Superni crown with threads
2.Superni gasket
3.Air gap
4.Body of the piston
5. Superni insert with threads
6. Air gap
7. Liner

Fig 1. Assembly details of air gap piston liner and air gap insulated liner

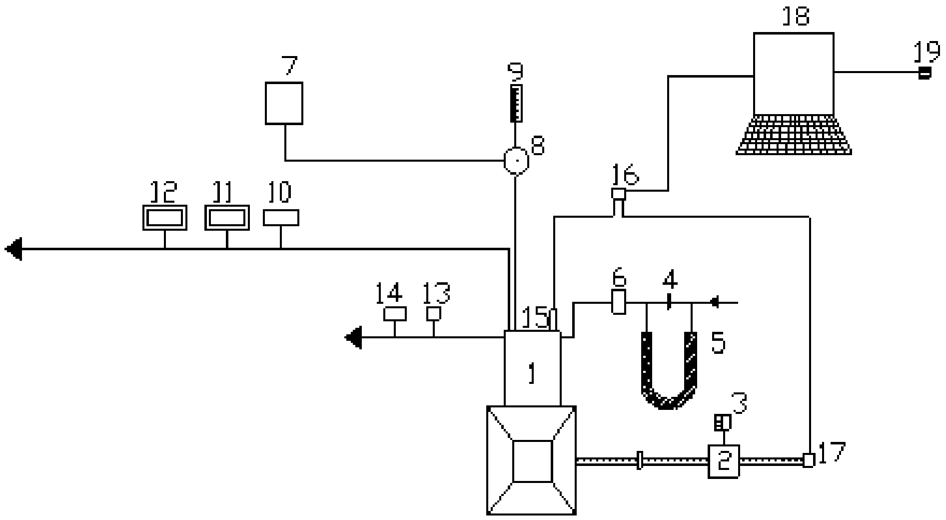
The optimum thickness of air gap in the air gap piston is found to be 3-mm [28], for improved performance of the engine with diesel as fuel. A superni-90 insert was screwed to the top portion of the liner in such a manner that an air gap of 3-mm was maintained between the insert and the liner body. At 500^{o}C the thermal conductivity of superni-90 and air are 20.92 and 0.057 W/m-K respectively. The properties of vegetable oil along with diesel fuel are given in Table 1.

Experimental setup used for the investigations of LHR diesel engine with crude jatropha oil (CJO) operation is shown in Figure 2. CE had an aluminum alloy piston with a bore of 80 mm and a stroke of 110mm. The rated output of the engine is 3.68 kW at a speed of 1500 rpm.

The compression ratio was 16:1 and manufacturer's recommended injection timing and injection pressures were 27^{o}bTDC and 190 bar respectively. 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.

Table 1. Properties of Test Fuels

Test Fuel	Viscosity at 25°C (centi-Poise)	Density at 25 °C	Cetane number	Calorific value (kJ/kg)
Diesel	12.5	0.84	55	42000
Jatropha oil (crude)	125	0.90	45	36000



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.

Fig 2. Experimental Set-up

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. The naturally aspirated engine was provided with water-cooling system in which inlet temperature of water was maintained at 60°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 injection pressures from 190 bar to 270 bar (in steps of 40 bar) using nozzle testing device. The maximum injection 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. The exhaust emissions of smoke and NO_x are recorded by AVL smoke meter and Netel Chromatograph NOx analyzer respectively at different values of BMEP 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.

3. RESULTS AND DISCUSSION

3.1 Performance Parameters

Curves from Figure 3 indicate that BTE increased up to 80% of the peak load operation due to increase of fuel conversion efficiency and beyond that load it decreased due to increase of friction power. CE with vegetable oil showed the deterioration in the performance for entire load range when compared with the pure diesel operation on CE at recommended injection timing.

Although carbon accumulations on the nozzle tip might play a partial role for the general trends observed, the difference of viscosity between the diesel and vegetable oil provided a possible explanation for the deterioration in the performance of the engine with vegetable oil operation. The result of lower jet exit Reynolds numbers with vegetable oil adversely affected the atomization. The amount of air entrained by the fuel spray is reduced, since the fuel spray plume angle is reduced, resulting in slower fuel- air mixing. In addition, less air entrainment by the fuel spay suggested that the fuel spray penetration might increase and resulted in more fuel reaching the combustion chamber walls. Furthermore droplet mean diameters (expressed as Sauter mean) are larger for vegetable oil leading to reduce the rate of heat release as compared with diesel fuel. This also, contributed the higher ignition (chemical) delay of the vegetable oil due to lower Cetane number.

According to the qualitative image of the combustion under the crude vegetable oil operation with CE, the lower BTE is attributed to the relatively retarded and lower heat release rates. BTE increased with the advancing of the injection timing in CE with the vegetable oil at all loads, when compared with CE at the recommended injection timing and pressure. This is due to initiation of combustion at earlier period and efficient combustion with increase of air entrainment in fuel spray giving higher BTE. BTE increased at all loads when the injection timing is advanced to 32^{o}bTDC in the CE at the normal temperature of vegetable oil. The increase of BTE at optimum injection timing over the recommended injection timing with vegetable oil with CE could be attributed to its longer ignition delay and combustion duration. BTE increased at all loads when the injection timing is advanced to 32^{o}bTDC in CE, at the preheated temperature (PT) of CJO also.

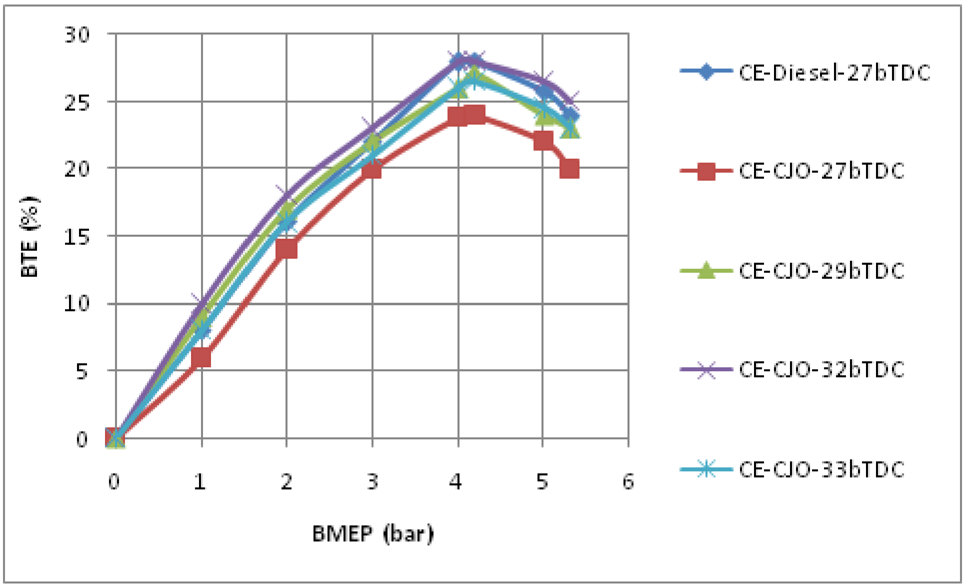


Fig. 3. Variation of brake thermal efficiency (BTE) with brake mean effective pressure (BMEP) in CE with CJO operation at an injection pressure of 190 bar.

From Figure 4, it is observed that LHR version of the engine at recommended injection timing showed the improved performance for the entire load range compared with CE with pure diesel operation.

High cylinder temperatures helped in better evaporation and faster combustion of the fuel injected into the combustion chamber. Reduction of ignition delay of the vegetable oil 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°bTDC with LHR engine with different operating conditions of CJO operation. Since the hot combustion chamber of LHR engine reduced ignition delay and combustion duration and hence the optimum injection timing was obtained earlier with LHR engine when compared with CE with the vegetable oil operation.

Injection pressure is varied from 190 bars to 270 bars to improve the spray characteristics and atomization of the vegetable oils and injection timing is advanced from 27 to 34°bTDC for CE and LHR engine. From Table 2, it is noticed that improvement in BTE at higher injection pressure was due to improved fuel spray characteristics. Peak BTE was higher in LHR engine when compared to CE with different operating conditions of the vegetable oil. The performance improved further in CE with the preheated (It was the temperature, at which viscosity of the vegetable oil was matched to that of diesel fuel, 125°C) vegetable oil compared with normal vegetable oil. It was due to improved spray characteristics of the oil, which reduced the impingement of the fuel spray on combustion chamber walls, causing efficient combustion thus improving BTE. However, the optimum injection timing was not varied even at higher injection pressure with LHR engine, unlike the CE. Hence it is concluded that the optimum injection timing was 32°bTDC at 190 bar, 31°bTDC at 230 bar and 30°bTDC at 270 bar for CE.

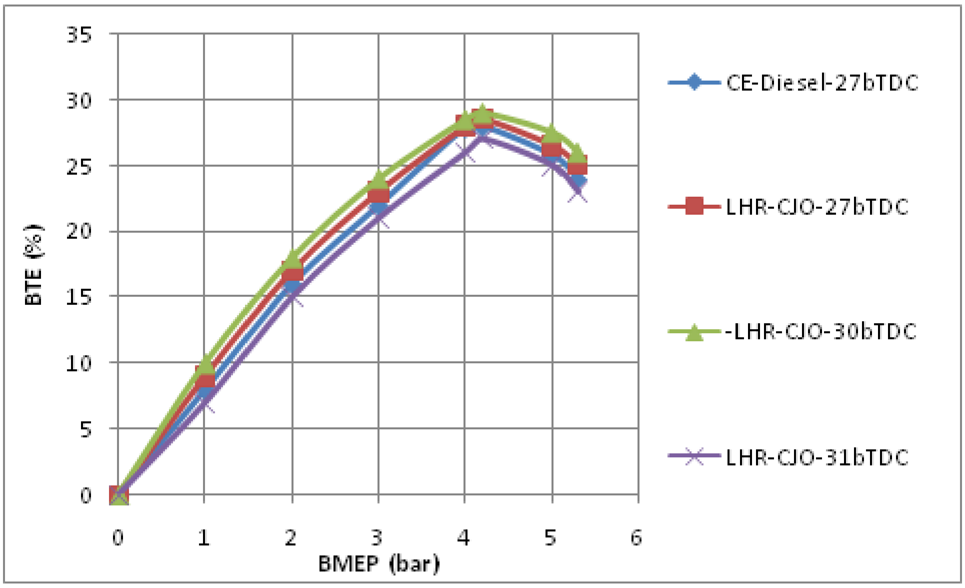


Fig. 4. Variation of brake thermal efficiency (BTE) with brake mean effective pressure (BMEP) in LHR engine with CJO operation at an injection pressure of 190 bar.

The optimum injection timing for LHR engine is 30° bTDC irrespective of injection pressure. Improvement in the peak BTE is observed with the increase of injection pressure and with advancing of the injection timing with the vegetable oil in both versions of the engine. Peak BTE is higher in LHR engine when compared with CE with different operating conditions of the vegetable oils. Preheating of the vegetable oil improved the performance in both versions of the engine compared with the vegetable oil at normal temperature. Preheating reduced the viscosity of the vegetable oils, which reduced the impingement of the fuel spray on combustion chamber walls, causing efficient combustion thus improving BTE.

Table 2. Data of Peak BTE

Injection Timing (^{o} bTDC)	Test Fuel	Peak BTE (%)											
		Conventional Engine						LHR Engine					
		Injection Pressure (Bar)						Injection 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	--
	CJO	24	25	25	26	26	27	28.5	29	29	29.5	29.5	30
30	DF	29	---	30	--	30.5	--	29.5	--	30.5	--	31	--
	CJO	26	26.5	26.5	27	28	28.5	29	29.5	29.5	30	30	30.5
31	DF	29.5	--	30	--	31	--	30	--	31	--	31	--
	CJO	27	27.2	28	28.5	27.5	28	--	--	--	--	--	--
32	DF	30		30.5		30.5							
	CJO	28	28.5	27.5	28	27	27.5	--	--	--	---	---	--
33	DF	31		31		30	---	--	--	--	--	--	-

DF-Diesel Fuel, CJO- Crude Jatropha Oil, NT- Normal or Room Temperature , PT- Preheat Temperature

From Table 3, it is noticed that brake specific energy consumption (BSEC) at peak load decreased with the increase of injection pressure and with the advancing of the injection timing at different operating conditions of the vegetable oil in both versions of the engine. This was due to effective energy utilization of the vegetable oil particularly in LHR engine.

Table 3. Data of BSEC at peak load operation

Injection Timing (^{O} bTDC)	Test Fuel	BSEC (kW/ kW)											
		Conventional Engine						LHR Engine					
		Injection Pressure (Bar)						Injection Pressure (Bar)					
		190		230		270		190		230		270	
		NT	PT	NT	PT	NT	PT	NT	PT	NT	PT	NT	PT
27	DF	4.00	--	3.92	--	3.84	--	4.16	---	4.08	--	4.00	--
	CJO	4.90	4.70	4.70	4.65	4.65	4.60	3.96	3.92	3.92	3.88	3.88	3.84
30	DF	3.92	---	3.88	--	3.84	--	4.08	--	4.00	--	3.90	--
	CJO	4.70	4.65	4.65	4.60	3.92	3.88	3.93	3.89	3.89	3.85	3.85	3.81
31	DF	3.84	--	3.80	--	3.77	--	3.86		3.85		3.84	
	CJO	4.45	4.40	3.92	3.88	3.96	3.92	--	--	--	--	--	--
32	DF	3.82	---	3.78	--	3.79	--	--	--	--	--	--	--
	CJO	3.98	3.94	3.94	3.90	3.90	3.86	-	---	--	--	-	----
33	DF	3.77	--	3.77	--	3.84	---	--	----	----	----	---	---

DF-Diesel Fuel, CJO- Crude Jatropha Oil, NT- Normal or Room Temperature , PT- Preheat Temperature

Figure 5 indicates that CE with vegetable oil operation at the recommended injection timing recorded higher EGT at all loads when compared with CE with pure diesel operation. Lower and retarded heat release rates associated with high specific energy consumption caused increase in EGT in CE. Ignition delay in the CE with different operating conditions of vegetable oil increased the duration of the burning phase. LHR engine recorded lower value of EGT when compared with CE with vegetable oil 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 expand in the cylinder giving higher work output and lower heat rejection. This showed that the performance improved with LHR engine over CE with vegetable oil operation.

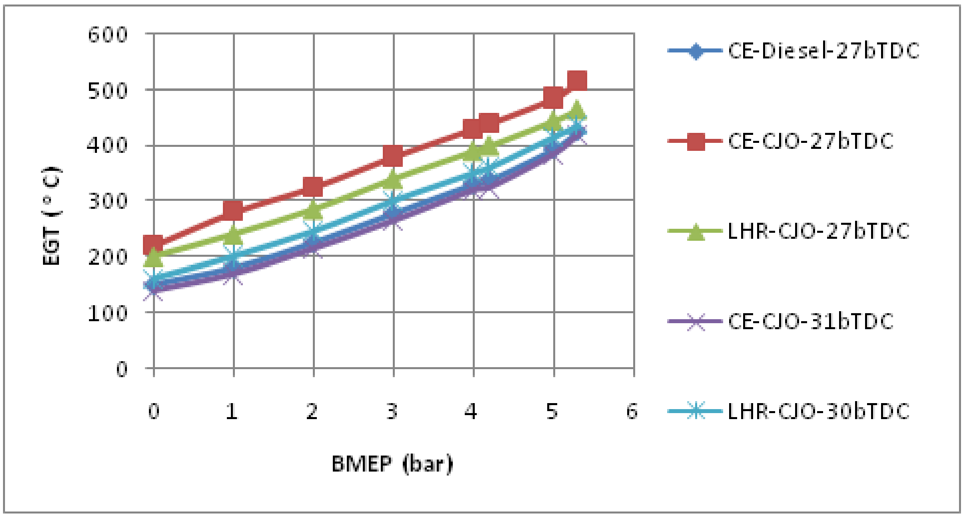


Fig. 5. Variation of exhaust gas temperature (EGT) with BMEP in both versions of the engine at recommended and optimized injection timings with CJO operation at an injection pressure of 190 bar.

The value of EGT decreased at respective optimum injection timings in both versions of the engine with vegetable oil, when compared at recommended injection timing. This confirmed that performance improved at optimum injection timing with both versions of the engine with vegetable oil operation. From Table 4, it is evident that the value of EGT decreased with increase of injection pressure and advanced injection timing with both versions of the engine. This was due to improved spray characteristics and air-fuel ratios with vegetable oil operation. Preheating of the vegetable oils reduced EGT marginally when compared to normal vegetable oils in both versions of the engine. Preheating of the vegetable oil improved the combustion and caused lower exhaust gas temperatures.

Table 4. Data of EGT at peak load operation

Injection timing (° b TDC)	Test Fuel	EGT at the peak load (°C)											
		Conventional Engine						LHR Engine					
		Injection Pressure (Bar)						Injection 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	--	475	---	460	--	445	--
	CJO	515	490	490	480	480	455	465	435	435	405	405	380
30	DF	410	---	400	--	385	---	455	---	450	--	445	--
	CJO	490	470	470	450	450	430	435	405	405	380	380	350
31	DF	400	---	390	--	375	---	450	---	445	---	440	---
	CJO	455	435	435	415	415	395	--	--	--	--	--	--
32	DF	390		380		380		29	--	30	--	30.5	--
	CJO	420	400	430	410	440	430	--	--	--	-	---	-
33	DF	375	---	375	---	400	--	--	--	--	---	--	--

DF-Diesel Fuel, CJO- Crude Jatropha Oil, NT- Normal or Room Temperature, PT- Preheat Temperature

Curves from Figure 6 indicate that that coolant load (CL) increased with BMEP in both versions of the engine with test fuels. However, CL reduced with LHR version of the engine with vegetable oil operation when compared with CE with pure diesel operation.

Heat output was properly utilized and hence efficiency increased and heat loss to coolant decreased with effective thermal insulation with LHR engine. However, CL increased with CE with vegetable oil operation in comparison with pure diesel operation on CE. This was due to concentration of fuel at the walls of combustion chamber. CL decreased with advanced injection timing with both versions of the engine with test fuels. This was due to improved air fuel ratios. From Table 5, it is noticed that CL decreased with advanced injection timing and with increase of injection pressure. This was because of improved combustion and proper utilization of heat energy with reduction of gas temperatures. CL decreased with preheated vegetable oil in comparison with normal vegetable oil in both versions of the engine. This was because of improved spray characteristics.

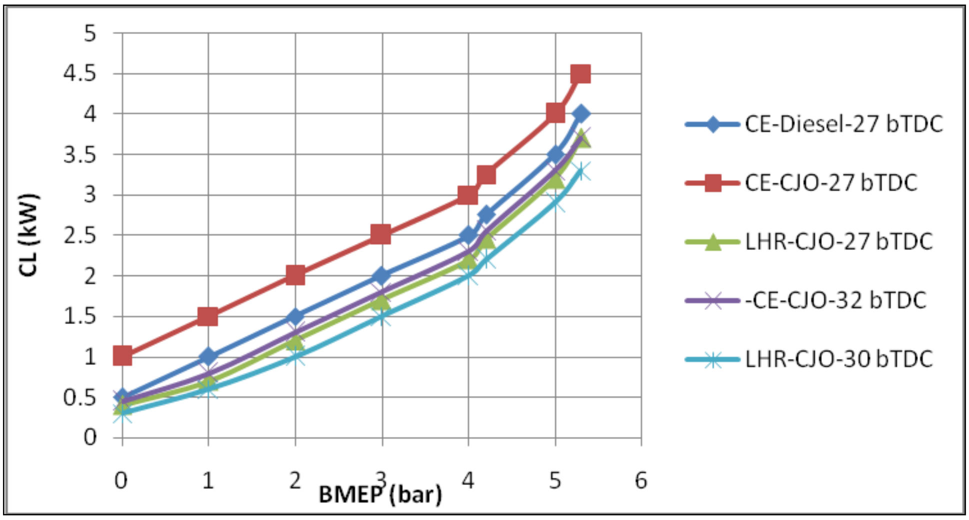


Fig. 6. Variation of coolant load (CL) with BMEP in both versions of the engine at recommended and optimized injection timings with CJO operation at an injection pressure of 190 bar.

Table 5. Data of CL at peak load operation

Injection timing (^{o} bTDC)	Test Fuel	Coolant Load (k W)											
		CE						LHR Engine					
		Injection Pressure (Bar)						Injection Pressure (Bar)					
		190		230		270		190		230		270	
		NT	PT	NT	PT	NT	PT	NT	PT	NT	PT	NT	PT
	DF	4.0	---	3.8	--	3.6	---	4.5	---	4.3	--	4.1	---
27	CJO	4.4	4.2	4.3	4.0	4.2	3.8	3.6	3.5	3.4	3.3	3.2	3.1
29	DF	3.8	--	3.6	---	3.4	--	4.3	--	4.1	--	3.9	--
	CJO	4.2	4.0	4.0	3.8	3.9	3.7	3.4	3.2	3.2	3.0	3.0	2.8
	DF	3.6	--	3.4	--	3.2	---	4.1	--	3.9	---	3.7	--
30	CJO	4.0	3.8	3.8	3.6	3.6	3.4	3.2	3.0	3.0	2.8	2.8	2.6
31	DF	3.4	---	3.2	---	3.0	--						
	CJO	3.8	3.6	3.6	3.4	3.7	3.5						
32	DF	3.2	---	3.0	---	3.2	---						
	CJO	3.6	3.4	3.7	3.5	3.8	3.7						
33	DF	3.0	---	3.2	---	3.4	---						

Figure 7 indicates that volumetric efficiency (VE) decreased with an increase of BMEP in both versions of the engine with test fuels. This is due to increase of gas temperature with the load. At the recommended injection timing, VE in the both versions of the engine with CJO operation decreased at all loads when compared with CE with pure diesel operation. This 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. VE increased marginally in CE and LHR engine at optimized injection timings when compared with recommended injection timings with vegetable oil operation. This was due to decrease of un-burnt fuel fraction in the cylinder leading to increase in VE in CE and reduction of gas temperatures with LHR engine.

VE increased marginally with the advancing of the injection timing and with the increase of injection pressure in both versions of the engine, as it was evident from the Table 6.

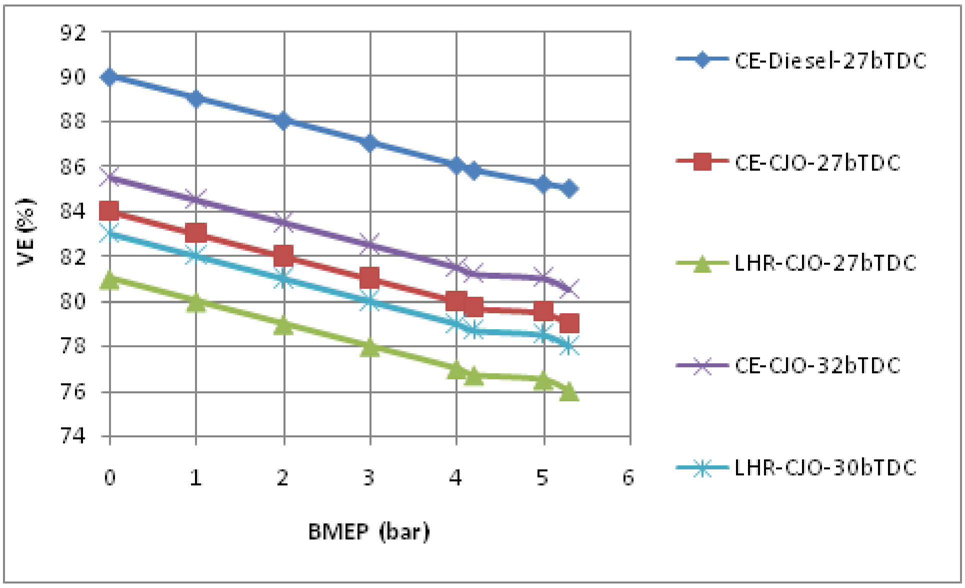


Fig.7. Variation of volumetric efficiency (VE) with BMEP in both versions of the engine at recommended and optimized injection timings with CJO operation at an injection pressure of 190 bar.

Table 6. Data of Volumetric Efficiency at peak load operation

Injection timing (^{o}bTDC)	Test Fuel	Volumetric Efficiency (%)											
		CE						LHR Engine					
		Injection Pressure (Bar)						Injection 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	--	80	--	82	--
	CJO	79	80	80	81	81	82	76	77	77	78	78	79
30	DF	86	--	87	--	88	---	80	--	82	--	83	--
	CJO	79.5	80.5	80.5	81.5	81.5	82.5	78	78.5	79	80	80	81
31	DF	87	--	87.5	--	89	--	82	--	83	--	84	--
	CJO	80	81	81	82	82	83	--	--	--	--	--	--
32	DF	87.5	--	88	--	87	--	-	--	-	--	--	-
	CJO	80.5	81.5	81.5	82.5	82.5	83.5	--	--	--	--	---	--
33	DF	89	--	89	--	86	--	--	--	--	--	--	--

This was due to better fuel spray characteristics and evaporation at higher injection pressures leading to marginal increase of VE. This was also due to the reduction of residual fraction of the fuel and improved combustion with improved air fuel ratios, due to increase of injection pressure. Preheating of the vegetable oil marginally improved VE 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 oil.

3.2 Exhaust Emissions

It was reported [32] reported that fuel physical properties such as density and viscosity could have a greater influence on smoke emission than the fuel chemical properties. From Figure 8, it is noticed that smoke levels were lower at low load and drastically higher at loads higher than 80% of the full load operation, as the availability of oxygen was less.

The magnitude of smoke intensity increased from no load to full load in both versions of the engine. 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, leading to incomplete combustion, producing more soot density. The variation of smoke levels with the BMEP 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. Drastic increase of smoke levels was observed at the peak load operation in CE at different operating conditions of the vegetable oil, compared with pure diesel operation on CE. This was due to the higher magnitude of the ratio of C/H of CJO (0.83) when compared with pure diesel (0.45). The increase of smoke levels was also due to decrease of air-fuel ratios and VE with vegetable oil compared with pure diesel operation. Smoke levels are related to the density of the fuel. Since vegetable oil has higher density compared to diesel fuels, smoke levels are higher with vegetable oil. However, LHR engine marginally reduced 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 vegetable oil compared with the CE. Density influences the fuel injection system. Decreasing the fuel density tends to increase spray dispersion and spray penetration. Preheating of the vegetable oils reduced smoke levels in both versions of the engine, when compared with normal temperature of the vegetable oil. This is due to i) the reduction of density of the vegetable oils, as density is related to smoke levels, ii) the reduction of the diffusion combustion proportion in CE with the preheated vegetable oil, iii) the reduction of the viscosity of the vegetable oil, with which the fuel spray does not impinge on the combustion chamber walls of lower temperatures rather than it directs into the combustion chamber. Smoke levels decreased at optimized injection timings and with increase of injection pressure, in both versions of the engine, with different operating conditions of the vegetable oil as it is noticed from Table 7.

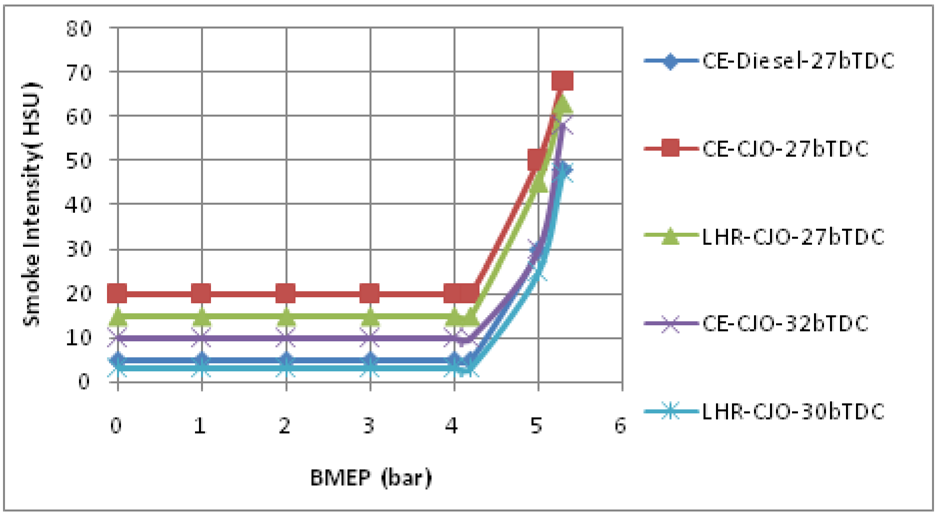


Fig. 8. Variation of smoke levels with BMEP in both versions of the engine at recommended and optimized injection timings with CJO operation at an injection pressure of 190 bar.

Table.7. Data of smoke levels in Hartridge Smoke Units (HSU) at peak load operation

Injection timing (^{o}bTDC)	Test Fuel	Smoke intensity (HSU)											
		Conventional Engine						LHR Engine					
		Injection Pressure (Bar)						Injection 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	--
	CJO	68	63	63	58	58	54	63	58	58	53	53	48
30	DF	36	--	34	--	32	--	45	--	42	--	41	--
	CJO	64	61	61	58	58	55	46	44	44	42	42	40
31	DF	33	---	32	--	30	--	43	--	41	--	40	--
	CJO	61	58	58	55	55	52	--	--	--	--	--	--
32	DF	32	--	31	--	32	--	--	--	--	---	--	--
	CJO	58	55	55	52	52	49	--	--	--	---	--	--
33	DF	30	---	30	--	35	--	-	--	--	--	--	--

This is due to improvement in the fuel spray characteristics at higher injection pressures and increase of air entrainment, at the advanced injection timings, causing lower smoke levels.

Temperature and availability of oxygen are two factors responsible for formation of NOx levels. Figure 9 indicates that NOx levels were lower in CE while they are higher in LHR engine at 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 with the vegetable oil operation on CE, which reduced NOx levels. Increase of combustion temperatures with the faster combustion and improved heat release rates in LHR engine cause higher NOx levels. At respective optimized injection timing, NOx levels increased in CE while they decreased in LHR engine.

This is due to increase of residence time with CE and decrease of combustion temperatures with improvement of air fuel ratios with LHR engine. NOx levels increased with the advancing of the injection timing in CE with different operating conditions of vegetable oil as it is noticed from Table 8. This was due to increase of residence time, when the injection timing was advanced with the vegetable oil operation, which caused higher NOx levels. With the increase of injection pressure, fuel droplets penetrate and find oxygen counterpart easily. Turbulence of the fuel spray increased the spread of the droplets thus leading to decrease NOx levels.

However, decrease of NOx levels was observed in LHR engine, due to decrease of combustion temperatures**,** when the injection timing was advanced and with increase of injection pressure**.** As expected, preheating of the vegetable oil further decreased NOx levels in both versions of the engine when compared with the normal vegetable oil. This was due to improved air fuel ratios with which combustion temperatures decreased leading to decrease NOx emissions.

3.3 Combustion Characteristics

From Table 9, it is observed that peak pressures are lower in CE while they were higher in LHR engine at the recommended injection timing and pressure, when compared with pure diesel operation on CE. This is due to increase of ignition delay, as vegetable oils require large duration of combustion. Mean while the piston started making downward motion thus increasing volume when the combustion takes place in CE. LHR engine increased the mass-burning rate of the fuel in the hot environment leading to produce higher peak pressures. The advantage of using LHR engine for vegetable oil is obvious as it could burn low cetane and high viscous fuels.

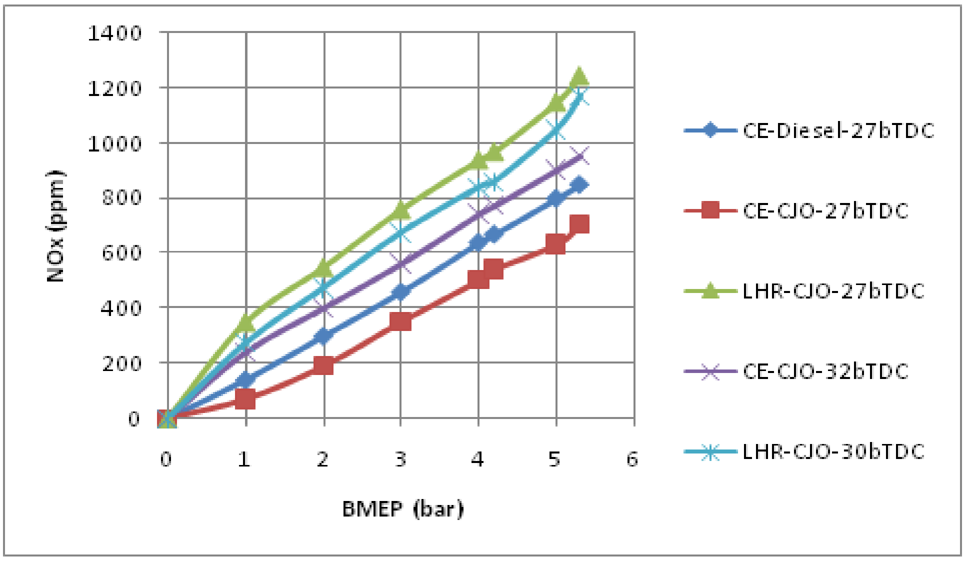


Figure.9. Variation of NOx levels with BMEP in both versions of the engine at recommended and optimized injection timings with CJO operation at an injection pressure of 190 bar.

Table 8. Data of NOx Levels at peak load operation

Injection timing (° bTDC)	Test Fuel	NOx levels (ppm)											
		Conventional Engine						LHR Engine					
		Injection Pressure (Bar)						Injection Pressure (Bar)					
		190		230		270		190		230		270	
		NT	PT	NT	PT	NT	PT	NT	PT	NT	PT	NT	PT
27	DF	850	----	890	----	930	---	1300	--	1280	--	1260	--
	CJO	700	680	680	660	660	640	1245	1230	1230	1210	1180	1115
30	DF	935	---	980	---	1020	--	1225	--	1205	--	1185	--
	CJO	750	720	720	690	690	660	1170	1150	1150	1120	1120	1100
31	DF	1020	---	1070	---	1190	---	1150	--	1130	--	1110	--
	CJO	810	780	780	750	750	720	--	--	--	--	--	-
32	DF	1105	----	1150	---	1235	---	--	--	--	--	--	--
	CJO	950	920	920	890	890	860	--	--	--	--	--	-
33	DF	1190	----	1230	---	1275	---	--	--	--	--	--	-

Peak pressures increased with the increase of injection pressure and with the advancing of the injection timing in both versions of the engine, with the vegetable oil operation. Higher injection pressure produces smaller fuel particles with low surface to volume ratio, giving rise to higher PP. With the advancing of the injection timing to the optimum value with the CE, 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 produces more combustion temperatures and pressures due to increase of the mass of the fuel. With LHR engine, peak pressures increased due to effective utilization of the charge with the advancing of the injection timing to the optimum value. The value of TOPP decreased with the advancing of the injection timing and with increase of injection pressure in both versions of the engine, at different operating conditions of vegetable oils. TOPP was more with different operating conditions of vegetable oils in CE, when compared with pure diesel operation on CE. This is due to higher ignition delay with the vegetable oil when compared with pure diesel fuel. This once again established the fact by observing lower peak pressures and higher TOPP, that CE with vegetable oil operation showed the deterioration in the performance when compared with pure diesel operation on CE. Preheating of the vegetable oil showed lower TOPP, compared with vegetable oil at normal temperature. This once again confirmed by observing the lower TOPP and higher PP, the performance of the both versions of the engine improved with the preheated vegetable oil compared with the normal vegetable oil. This trend of increase of MRPR and decrease of TOMRPR indicated better and faster energy substitution and utilization by vegetable oil, which could replace 100% diesel fuel. However, these combustion characters were within the limits hence the vegetable oil could be effectively substituted for diesel fuel.

Table.9 Data of PP, TOPP, MRPR and TOMRPR at peak load operation

Injection timing (^{o}bTDC)/ Test fuel	Engine version	PP(bar)				MRPR (Bar/deg)				TOPP (Deg)				TOMRPR (Deg)			
		Injection pressure (Bar)				Injection pressure (Bar)				Injection pressure (Bar)				Injection pressure (Bar)			
		190		270		190		270		190		270		190		270	
		NT	PT	NT	PT	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	--	0	0	0	0
	LHR	48.1	--	53.0	--	2.9	--	3.1	--	10	--	9	--	0	0	0	0
27/CJO	CE	46.9	47.7	49.9	50.3	2.4	2.5	2.9	3.0	11	10	11	10	1	1	1	1
	LHR	59.5	60.9	63.6	64.5	3.3	3.4	3.5	3.6	10	9	9	8	1	1	1	1
30/CJO	LHR	61.5	61.9	66.3	67.1	3.4	3.5	3.6	3.7	9	8	8	7	0	0	0	0
32/CJO	CE	51.5	52.8			3.3		3.4		8		8		0		0	

4. CONCLUSIONS

The optimum injection timing was found to be 32^{o}bTDC with CE while it was 30^{o}bTDC for LHR engine with CJO operation. At recommended injection timing, peak brake thermal efficiency increased by 2%, exhaust gas temperature increased by 40^{o}C, volumetric efficiency decreased by 10%, BSEC at peak load operation decreased by 1%, coolant load decreased by 10%, smoke levels increased by 31%, and NOx levels increased by 46% with LHR engine in comparison with CE with pure diesel operation. Also, peak pressure, MRPR increased and TOPP decreased with LHR engine with CJO operation in comparison with pure diesel operation on CE. Preheated vegetable oil improved the performance when compared with normal CJO in both versions of the engine. Performance improved with advanced injection timing and with increase of injection pressure with both versions of the engine at different operating conditions of the vegetable oil.

ACKNOWLEDGMENTS

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

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