Performance evaluation of a low heat rejection diesel engine with Jatropha oil based bio-diesel

P.V.K.Murthy[a,*], M.V.S. Murali Krishna[b], A.Sitarama Raju[c] , N. Janarthan[b] and P.Ushasri[d]

[a] Vivekananda Institute of Science and Information Technology, Shadnagar, Mahabubnagar-509216,
[b] Mechanical Engineering Department, Chaitanya Bharathi Institute of Technology, Gandipet, Hyderabad-500 075. E-mail: maddalivs@gmail.com
[c] Mechanical Engineering Department, J.N.T. University, Kukatpally, Hyderabad- 500 085
[d] Mechanical Engineering Department, College of Engineering, Osmania University, Hyderabad- 500 007
***Corresponding author:** Krishnamurthy_venkata@yahoo.co.in

ABSTRACT

Investigations were carried out to evaluate the performance of a low heat rejection (LHR) diesel engine consisting of air gap insulated piston with 3-mm air gap, with superni (an alloy of nickel) crown, air gap insulated liner with superni insert and ceramic coated cylinder head with different operating conditions of jatropha oil based bio-diesel with varied injection timing and injection pressure. Performance parameters are determined at various magnitudes of brake mean effective pressure (BMEP). Exhaust emissions of smoke and oxides of nitrogen (NOx) are recorded at different values of BMEP. Combustion characteristics of the engine are 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 bio-diesel (BD) operation at recommended injection timing and pressure. Performance of both version of the engine is improved with advanced injection timing and higher injection pressure when compared with CE with pure diesel operation. Relatively, peak brake thermal efficiency (BTE) increased by 12%, brake specific energy consumption (BSEC) at peak load decreased by 1%, exhaust gas temperature (EGT) at peak load increased by 35 o C , coolant load (CL) at peak load decreased by 10%, volumetric efficiency (VE) at peak load decreased by 10%, smoke levels at peak load decreased by 6% and NOx levels at peak load increased by 47% with LHR engine with biodiesel at recommended injection timing of 27^{o}bTDC, when compared with pure diesel operation on CE at 27^{o}bTDC .

Keywords: *Bio-diesel, LHR engine, fuel performance, exhaust emissions, combustion characteristics.*

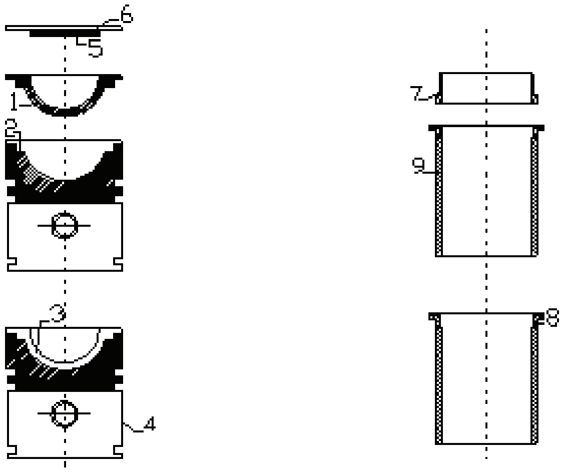
1. Introduction

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. It has been found that the vegetable oils are promising substitute, because of their properties are similar to that of diesel fuel and it is a renewable and can be easily produced. Rudolph Diesel, [1] the inventor of the diesel engine that bears his name, experimented with fuels ranging from powdered coal to peanut oil. Several researchers [2-8] 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. And also the U.S. Department of Energy [9] has stated that, "Raw or refined vegetable oil, or recycled greases that have not been processed into biodiesel, are not biodiesel and should be avoided. The use of raw, unprocessed vegetable oils or animal fats in diesel engines – regardless of blend level – can have significant adverse effects and should not be used as fuel in diesel engines. Raw or refined vegetable oil, or recycled greases have significantly different and widely varying properties that are not acceptable for use in modern diesel engines". For example, the higher viscosity and chemical composition of unprocessed oils and fats have been shown to cause problems in a number of areas: (i) piston ring sticking; (ii) injector and combustion chamber deposits; (iii) fuel system deposits; (iv) reduced power; (v) reduced fuel economy and (vi) increased exhaust emissions. Use of unprocessed oils or fats as neat fuels or blending stock will lead to excessive fuel condensation and corresponding dilution of the engine's lubricating oil that may result in sludge formation. Any or all of these conditions may result in reduced engine life, increased maintenance costs, or catastrophic engine failure. The significantly higher viscosity of raw vegetable oils (27-54 mm^2/s) compared to petroleum diesel fuel (2.6 mm2/s) alters fuel injector spray patterns and spray duration, adds stress on fuel injection systems, and results in incomplete combustion and high dilution of the engine lubricating oil. 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 [10] 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 [11-19] with bio-diesel on CE and reported performance was compatible with pure diesel operation on CE. The drawbacks of the biodiesel call for hot combustion chamber provided by low heat rejection (LHR) diesel engine. The concept of LHR engine is reduce heat loss to the coolant, by providing thermal resistance in the path of heat flow to the coolant thereby gains thermal efficiency. Several methods adopted for achieving LHR to the coolant are i) using ceramic coatings on piston, liner and cylinder head ii) creating air gap in the piston and other components with low-thermal conductivity materials like superni, cast iron and mild steel etc. Studies were made [20-22] on ceramic coated engines with pure diesel operation, and reported that brake specific fuel consumption (BSFC) was increased by 5% and decreased smoke emissions. Experiments were conducted [23-25] with biodiesel with ceramic coated engines and reported BSFC decreased and NOx increased. Creating an air gap in the piston involved the complications of joining two different metals. Investigations were carried out [26] on air gap insulated piston with pure diesel operation, the bolted design employed could not provide complete sealing of air in the air gap. It was made [27] a successful attempt 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. Experiments were conducted [10] on LHR engine which consisted of air gap insulated piston with superni crown and air gap insulated liner with superni insert with advanced injection timings and increased injection pressure with different alternate fuels like alcohols and non-edible vegetable oil and reported improved performance with LHR engine. Experiments were conducted [28] on 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 improved performance and pollution levels.

The present paper attempted to evaluate the performance of LHR engine, which contained air gap piston, air gap liner and ceramic coated cylinder head with jatropha oil based bio-diesel with varying engine parameters of change of injection pressure and injection timing and compared with CE with pure diesel at recommended injection timing and injection pressure.

2. Materials and methods

LHR diesel engine contained a two-part piston (Fig. 1); the top crown made of low thermal conductivity material, superni-90 was screwed to aluminum body of the piston, providing a 3mm-air gap in between the crown and the body of the piston. The optimum thickness of air gap in the air gap piston was found to be 3-mm [27], for improved performance of the engine with superni inserts with diesel as fuel.



1. Crown with threads
2. Gasket
3. Air gap
4. Body
5. Ceramic coating
6. Cylinder head

7 Insert with threads
8. Air gap
9. Liner

Insulated piston *Insulated liner* *Ceramic coated cylinder head*

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

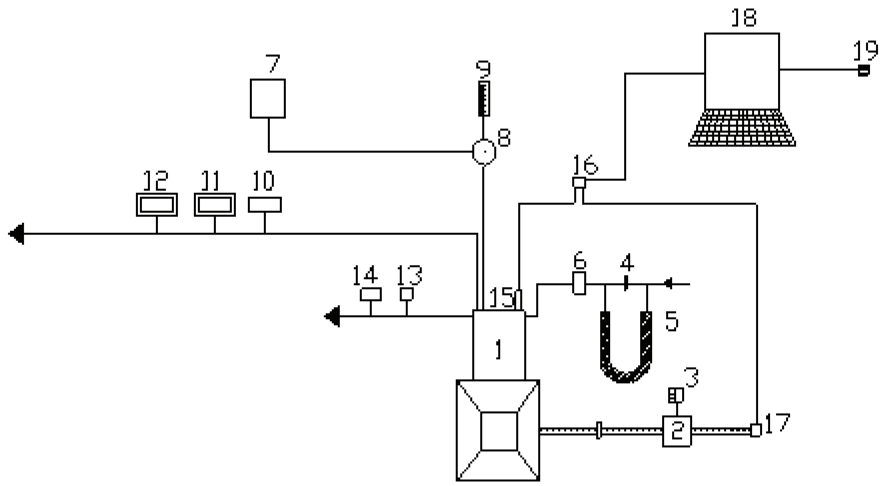
A superni-90 insert was screwed to the top portion of the liner in such a manner that an air gap of 3mm was maintained between the insert and the liner body. At 500^{o}C the thermal conductivity of superni-90 and air are 20.92 and 0.057 W/m-K respectively. Partially stabilized zirconium (PSZ) of thickness 500 microns was coated by means of plasma coating technique.

The process of converting the jatropha oil into methyl esters was carried out by heating the oil with the methanol in the presence of the catalyst (Sodium hydroxide). In the present case, crude jatropha 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 was removed by distillation and glycerol, which separated 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 esters were used in present study. The properties of the vegetable oil ester and the diesel used in this work are presented in Table-1.

Table 1 Properties of Test Fuels

Test Fuel	**Viscosity at 25^{o}C (centi-poise)**	**Density at 25^{o}C**	**Cetane number**	**Calorific value (kJ/kg)**
Diesel	12.5	0.84	55	42000
Bio diesel	53	0.87	55	35500

The experimental setup used for the investigations of LHR diesel engine with jatropha oil based biodiesel is shown in Fig. 2. CE has an aluminum alloy piston with a bore of 80 mm and a stroke of 110mm. The rated output of the engine was 3.68 kW at a rate 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 0.25mm. 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. The naturally aspirated engine was provided with water-cooling system in which inlet temperature of water is maintained at 60^{o}C by adjusting the water flow rate. The 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. Pollution levels of smoke and NO_x were recorded by AVL smoke meter and Netel Chromatograph NOx analyzer respectively at various values of BMEP. 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 is obtained on the screen of the personal computer. The accuracy of the instrumentation used in the experimentation is 0.1%



1.Engine, 2.Electical Dynamo meter, 3.Load Box, 4.Orifice meter, 5.U-tube water manometer, 6.Air box, 7.Fuel tank, 8, Three way valve, 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 2 Experimental Set-up

3. Results and discussion

3.1 Performance parameters

Curves from Fig. 3 indicate that CE with bio-diesel 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 bio-diesel provided a possible explanation for the deterioration in the performance of the engine with bio-diesel operation. The result of lower jet exit Reynolds numbers with vegetable oils adversely affected the atomization. The amount of air entrained by the fuel spray is reduced, since the fuel spray plume angle was 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 biodiesel leading to reduce the rate of heat release as compared with diesel fuel. This also, contributed the higher ignition (chemical) delay of the biodiesel due to lower cetane number. According to the qualitative image of the combustion under the biodiesel operation with CE, the lower BTE was attributed to the relatively retarded and lower heat release rates.

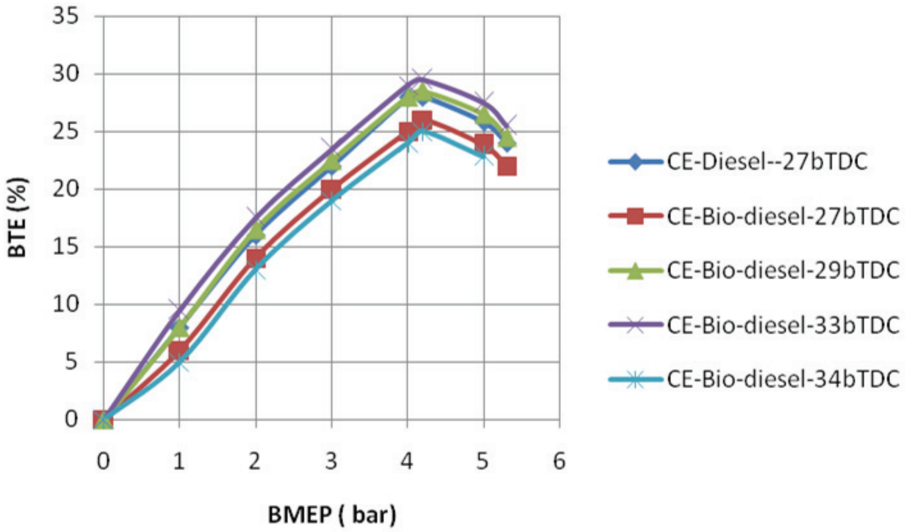


Figure 3 Variation of BTE with BMEP in CE at various injection timings at an injection pressure of 190 bar

BTE increased with the advancing of the injection timing in the CE with the bio-diesel 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 was advanced to 33^{o}bTDC in CE at the normal temperature of bio-diesel. The increase of BTE at optimum injection timing over the recommended injection timing with bio-diesel 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 33^{o}bTDC in CE, at the preheated temperature (125^{o}C) of the bio-diesel. The performance was improved further in CE with the preheated biodiesel for entire load range when compared with normal biodiesel. Preheating of the biodiesel reduced the viscosity, which improved the spray characteristics of the oil.

From Fig. 4, it is observed that LHR version of the engine 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 biodiesel in the hot environment of the LHR engine improved heat release rates and efficient energy utilization. Preheating of biodiesel improved performance further in LHR version of the engine. The optimum injection timing was found to be 31^{o}bTDC with LHR engine with normal bio-diesel operation.

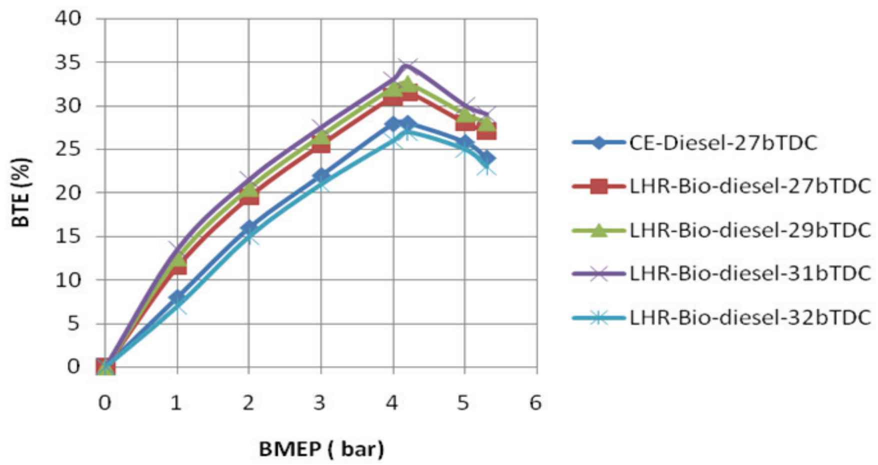


Figure 4 Variation of BTE with BMEP in LHR at various injection timings at an injection pressure of 190 bar

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 biodiesel operation.

Fig. 5 indicates that peak BTE was higher in the LHR engine when compared with CE at all loads with biodiesel operation. This was due to good evaporation of biodiesel in hot environment provided by LHR engine, leading to produce higher BTE. Preheating of the biodiesel improved the performance in both versions of the engine compared with the biodiesel at normal temperature. Preheating reduced the viscosity of the biodiesel, which reduced the impingement of the fuel spray on combustion chamber walls, causing efficient combustion thus improving BTE.

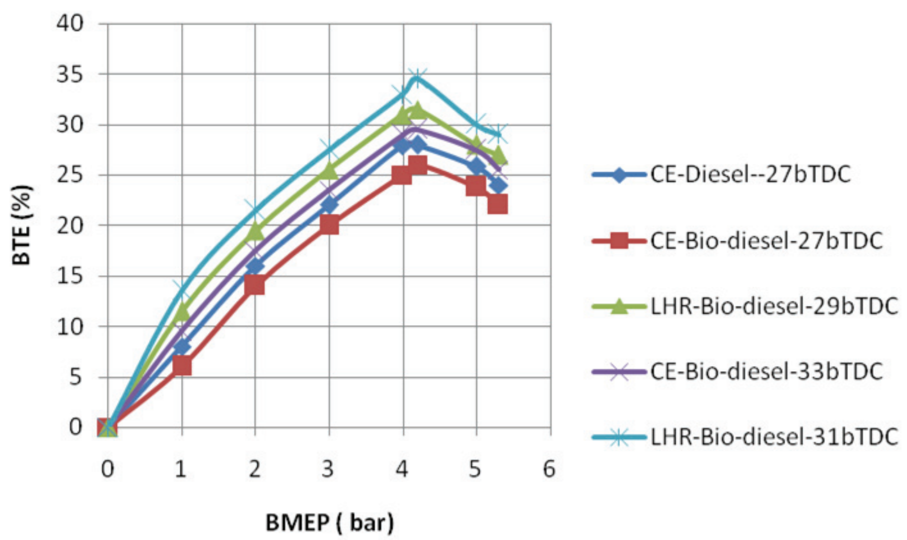


Figure 5 Variation of BTE with BMEP in both versions of the engine at recommend and optimized injection timings at an injection pressure of 190 bar

Injection pressure was varied from 190 bars to 270 bars to improve the spray characteristics and atomization of the biodiesel and injection timing was advanced from 27 to 34^{o}bTDC for CE and LHR engine. The improvement in BTE at higher injection pressure was due to improved fuel spray characteristics. However, the optimum injection timing was not varied even at higher injection pressure with LHR engine, unlike the CE. 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. The optimum injection timing for LHR engine was 31^{o}bTDC irrespective of injection pressure. From the Table 2, it is observed that Improvement in the peak BTE is observed with the increase of injection pressure and with advancing of the injection timing with the biodiesel in both versions of the engine. Peak BTE was higher in the LHR engine when compared with CE with different operating conditions of the biodiesel. Preheating of the biodiesel improved the performance in both versions of the engine compared with the biodiesel at normal temperature. Preheating reduced the viscosity of the biodiesel, 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 (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	28	--	29	---	30	--	29	--	30	--	30.5	--
	BD	26	27	27	28	28	29	31.5	32.5	32.5	33.5	33.5	34.5
30	DF	29	---	30	--	30.5	--	29.5	--	30.5	--	31	--
	BD	27	28	28	29	29	30	32.5	33.5	33.5	34.5	34.5	35.5
31	DF	29.5	--	30	--	31	--	30	--	31	--	31	--
	BD	28	29	29	29.5	29.5	30.5	34.5	35.5	35.5	36	36	36.5
32	DF	30	--	30.5	--	30.5	--						
	BD	29	29	29.5	30.5	27	27.5	31	32	32	32.5	32.5	33
33	DF	31	--	31	--	30	---	--	--	--	--	--	-
	BD	29.5	30.5										

DF- Diesel Fuel; BD- Biodiesel; NT-Normal Temperature; PT- Preheated Temperature ,

From Table 3, it is evident 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. BSEC is defined as energy consumed by the engine in producing unit brake power. With efficient combustion, mass of fuel burned is less with LHR engine leading to produced lower BSEC. With preheated biodiesel, BSEC is less due to improved spraying characteristics.

Table 3 Data of BSEC at peak load operation

Injection Timing (^{O} bTDC)	Test Fuel	BSEC (kW/ kW)											
		Conventional Engine						LHR Engine					
		Injection Pressure (Bars)						Injection Pressure (Bars)					
		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	--
	BD	4.94	4.66	4.66	4.62	4.62	4.58	3.96	3.92	3.92	3.88	3.88	3.84
30	D	3.92	---	3.88	--	3.84	--	4.08	--	4.00	--	3.90	--
	BD	4.66	4.62	4.62	4.58	3.88	3.84	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	
	BD	4.40	4.36	3.88	3.84	3.84	3.80	3.78	3.76	3.76	3.74	3.74	3.72
32	DF	3.82	---	3.78	--	3.79	--	--	--	--	--	--	--
	BD	3.98	3.94	3.84	3.80	3.90	3.86	3.90	3.86	3.86	3.82	3.82	3.78
33	DF	3.77	--	3.77	--	3.84	---	--	----	----	----	---	---
	BD	3.84	3.80	3.88	3.84	3.86	3.82						

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

From Fig. 6, it is noticed that CE with bio-diesel at the recommended injection timing recorded higher exhaust gas temperature (EGT) at all loads compared with CE with pure diesel operation. Lower heat release rates and retarded heat release associated with high specific energy consumption caused increase in EGT in CE. Ignition delay in the CE with different operating conditions of biodiesel increased the duration of the burning phase.

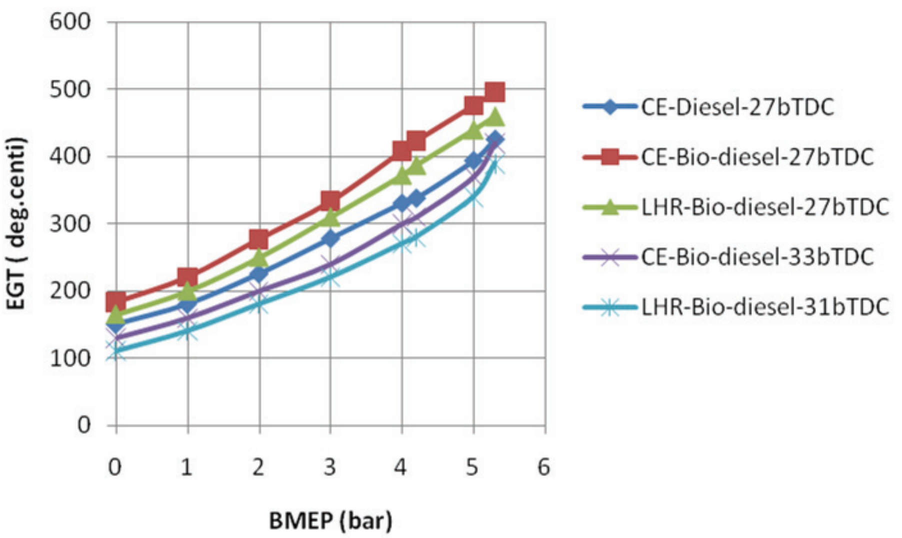


Figure 6 Variation of EGT with BMEP in both versions of the engine at recommend and optimized injection timings at an injection pressure of 190 bar

LHR engine recorded lower value of EGT when compared with CE 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 expand in the cylinder giving higher work output and lower heat rejection. This showed that the performance was improved with LHR engine over the CE with biodiesel operation.

From Table 4, it is noticed that the value of EGT at peak load decreased with advancing of injection timing and with increase of injection pressure in both versions of the engine with biodiesel which confirmed that performance increased with increase of injection pressure. Preheating of the biodiesel further decreased the value of EGT, compared with normal biodiesel in both versions of the engine.

Table 4 Data of EGT at peak load operation

Injection timing (º b TDC)	Test Fuel	EGT at the peak load (ºC)											
		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	425	--	410	---	395	--	460	---	450	--	440	--
	BD	495	475	475	450	450	425	460	430	430	400	400	370
29	DF							440		430		420	
	BD	480	460	460	440	440	420	430	400	410	390	390	370
30	DF	410	---	400	--	385	---	460	---	450	--	440	--
	BD	460	440	440	420	430	410	400	370	370	340	340	310
31	DF	400	---	390	--	375	---	450	---	445	---	440	---
	BD	440	400	430	420	420	400	360	340	340	310	340	310
32	DF	390		380		380							--
	BD	430	375	420	400	440	420	-------	---	---	----	---	-
33	DF	375	---	375	---	400	--	--	--	--	---	--	--
	BD	420	400	440	420	430	410						

Curves from Fig. 7 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 biodiesel 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.

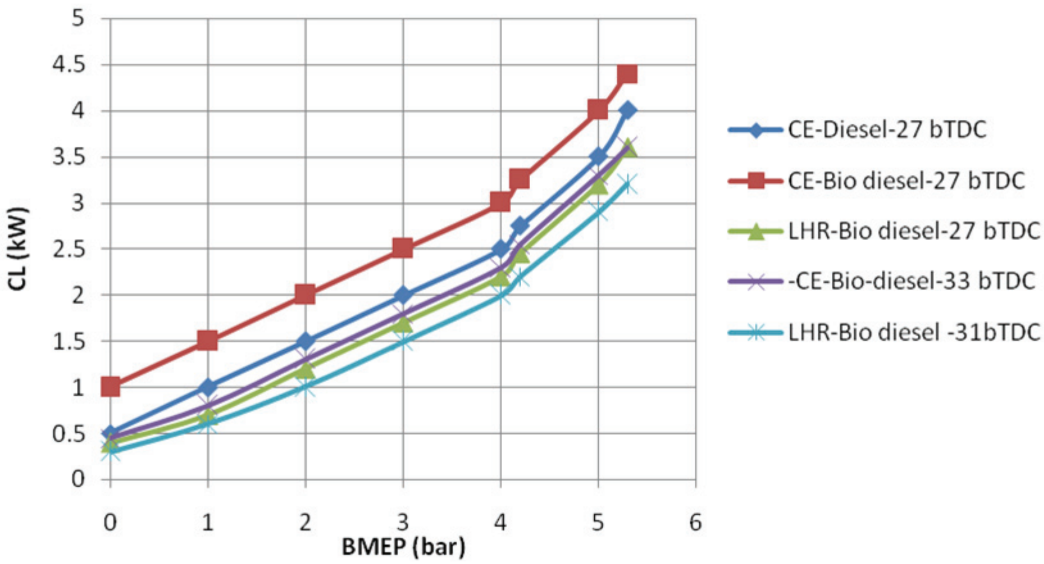


Figure 7 Variation of Coolant load (CL) with BMEP in both versions of the engine at recommend and optimized injection timings at an injection pressure of 190 bar

From Table.5, it is noticed that CL decreased with advanced injection timing and with increase of injection pressure.

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
27	DF	4.0	---	3.8	--	3.6	---	4.5	---	4.3	--	4.1	---
	BD	4.4	4.0	4.0	3.8	3.8	3.6	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	--
	BD	4.3	4.1	4.1	3.9	3.9	3.7	3.4	3.2	3.2	3.0	3.0	2.8
30	DF	3.6	--	3.4	--	3.2	---	4.1	--	3.9	---	3.7	--
	BD	4.2	4.0	4.0	3.8	3.8	3.6	3.3	3.0	3.0	2.8	2.8	2.6
31	DF	3.4	---	3.2	---	3.0	--						
	BD	4.1	3.9	3.9	3.7	3.6	3.4	3.2	2.8	2.8	2.6	2.4	2.2
32	DF	3.2	---	3.0	---	3.2	---						
	BD	4.0	3.8	3.6	3.4	3.8	3.6						
33	DF	3.0	---	3.2	---	3.4	---						
	BD	3.6	3.4	3.8	3.6	3.7	3.5						

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.

From Fig. 8, it is observed that, the volumetric efficiency (VE) decreased with the increase of BMEP in both versions of the engine. This was due to increase of gas temperature with the load. At the recommended injection timing, VE in the both versions of the engine with bio-diesel 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 bio-diesel operation. This is 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. This was due to better fuel spray characteristics and evaporation at higher injection pressures leading to marginal increase of VE. This is also due to the reduction of residual fraction of the fuel, with the increase of injection pressure. Preheating of the biodiesel 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.

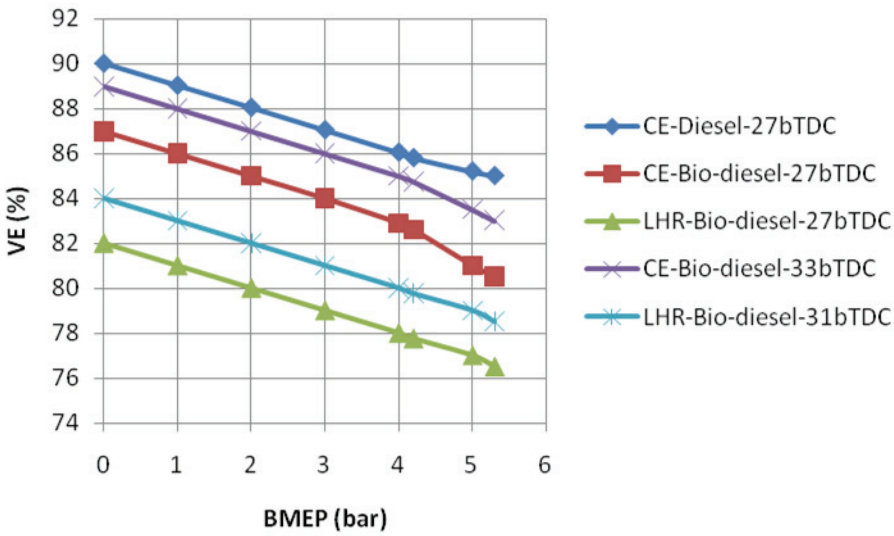


Figure 8 Variation of Volumetric efficiency (VE) with BMEP in both versions of the engine at recommend and optimized injection timings at an injection pressure of 190 bar

Table 6 indicates that VE increased marginally with the advancing of the injection timing and with the increase of injection pressure in both versions of the engine. 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, with the 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

Table 6 Data of volumetric efficiency at peak load operation

Injection timing (^{o} bTDC)	Test Fuel	Volumetric efficiency (%)											
		Conventional Engine						LHR Engine					
		Injection Pressure (Bars)						Injection Pressure (Bars)					
		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	--
	BD	80.5	81.5	81.5	82.5	82.5	83.5	76.5	77.5	77.5	78.5	78.5	79.5
30	DF	86	--	87	--	88	---	80	--	82	--	83	--
	BD	81	82	82	83	83	84	77.5	78.5	78.5	79.5	79.5	80.5
31	DF	87	--	87.5	--	89	--	82	--	83	--	84	--
	BD	82	83	81	82	83	84	78.5	79.5	79.5	80.5	80.5	81.5
32	DF	87.5	--	88	--	87	--	-	--	-	--	--	-
	BD	82.5	83.5	83	84	82	83	--	--	--	--	---	--
33	DF	89	--	89	--	86	--	--	--	--	--	--	--
	BD	83	84	82	83	81	82						

3.2 Exhaust emissions

It was reported [29] that fuel physical properties such as density and viscosity could have a greater influence on smoke emission than the fuel chemical properties. From Fig. 9, it is noticed that up to 80% of peak load operation, smoke levels were lower with test fuels and beyond that load it increased drastically with both versions of the engine. A rich fuel–air mixture resulted in higher smoke because of the availability of oxygen was lower. 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 brake power/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. Smoke levels were higher with CE with biodiesel in comparison with pure diesel operation on CE. This was due to the higher magnitude of the ratio of C/H of bio-diesel (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 bio-diesel compared with pure diesel operation. Smoke levels are related to the density of the fuel. Since biodiesel has higher density compared to diesel fuels, smoke levels are higher with biodiesel. 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 CE. Density influences the fuel injection system. Decreasing the fuel density tends to increase spray dispersion and spray penetration. Preheating of the biodiesel reduced smoke levels in both versions of the engine, when compared with normal temperature of the biodiesel. This is due to i) the reduction of density of the biodiesel, as density is directly proportional to smoke levels, ii) the reduction of the diffusion combustion proportion in CE 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 directs into the combustion chamber.

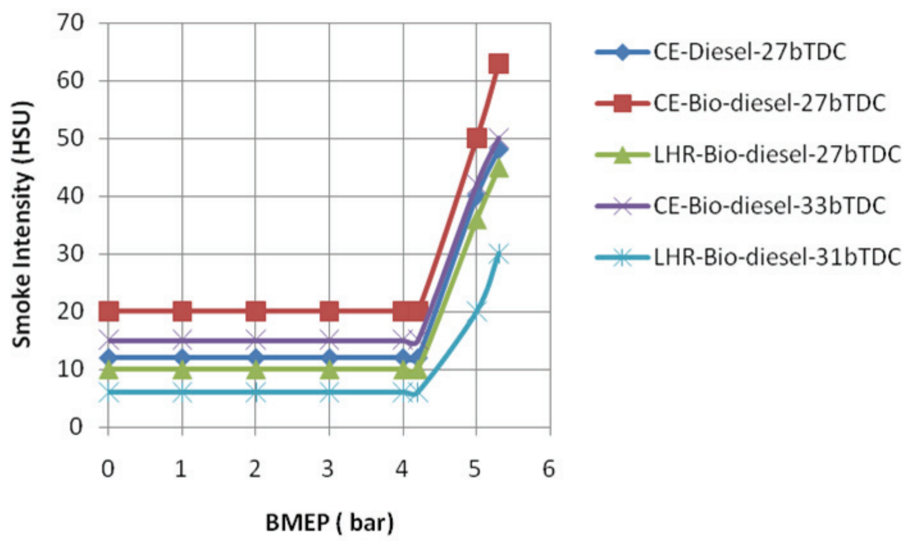


Figure 9 Variation of Smoke intensity (HSU) with BMEP in both versions of the engine at recommend and optimized injection timings at an injection pressure of 190 bar

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 biodiesel as it is noticed from the Table.7.

This was 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.

Table 7 Data of smoke levels at peak load operation

Injection timing (°bTDC)	Test Fuel	Smoke intensity (HSU)											
		Conventional Engine						LHR Engine					
		Injection Pressure (Bars)						Injection Pressure (Bars)					
		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	--
	BD	63	60	61	58	58	54	45	40	40	35	35	30
30	DF	36	--	34	--	32	--	45	--	42	--	41	--
	BD	60	55	55	50	45	55	40	35	35	30	30	25
31	DF	33	---	32	--	30	--	43	--	41	--	40	--
	BD	55	50	50	45	50	45	30	25	25	20	20	18
32	DF	32	--	31	--	32	--	--	--	--	---	--	--
	BD	52	48	50	45	52	49	--	--	--	---	--	--
33	DF	30	---	30	--	35	--	-	--	--	--	--	--
	BD	50	45	55	50	52	48						

The temperature and availability of oxygen were the reasons for the formation of NOx. Curves from Fig. 10 indicate that for both versions of the engine, NOx concentrations raised steadily as the fuel/air ratio increased with increasing BP/BMEP, 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. Though amount of fuel injected decreased proportionally as the overall equivalence ratio was decreased, much of the fuel still burns close to stoichiometric. Thus NOx emissions should be roughly proportional to the mass of fuel injected (provided burned gas pressures and temperature do not change greatly). From Fig. it is noticed that NOx levels were lower in CE while they are higher in LHR engine at different operating conditions of the biodiesel at different loads 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 biodiesel 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. As expected, preheating of the biodiesel decreased NOx levels in both versions of the engine when compared with the normal vegetable oil. This was due to the improvement in air-fuel ratios leading to decrease NOx levels.

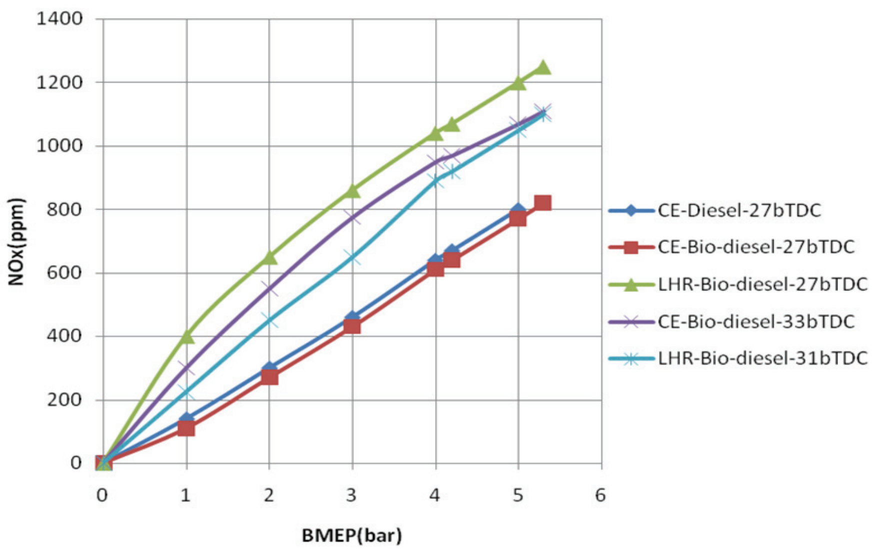


Figure 10 Variation of NOx levels intensity (HSU) with BMEP in both versions of the engine at recommend and optimized injection timings at an injection pressure of 190 bar

NOx levels increased with the advancing of the injection timing and with increase of injection pressure in CE with different operating conditions of biodiesel as noticed in Table.8. Residence time and combustion temperatures had increased, when the injection timing was advanced with the biodiesel 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 in NOx levels.

Table 8 Data of NOx emissions at peak load operation

Injection timing (^{o} b TDC)	Test Fuel	NOx levels (ppm)											
		Conventional Engine						LHR Engine					
		Injection Pressure (Bars)						Injection Pressure (Bars)					
		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	--
	BD	820	770	770	720	720	670	1250	1200	1200	1150	1150	1100
30	DF	935	---	980	---	1020	--	1225	--	1205	--	1185	--
	BD	920	870	870	820	820	770	1150	1100	1100	1050	1050	1000
31	DF	1020	---	1070	---	1190	---	1150	--	1130	--	1110	--
	BD	970	920	920	870	870	820	1100	1050	1050	1000	1000	950
32	DF	1105	----	1150	---	1235	---	--	--	--	--	--	--
	BD	1020	970	970	920	920	870	--	--	--	--	--	-
33	DF	1190	----	1230	---	1275	---	--	--	--	--	--	-
	BD	1110	1060	1060	1010	1010	960						

3.3 Combustion characteristics

From Table 9, it is evident that biodiesel operation, peak pressures were 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 was 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 biodiesel oils is obvious as it could burn low cetane and high viscous fuels. 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 oils operation. Higher injection pressure produced 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 burned, it produced more combustion temperature and pressure 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. The magnitude 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 biodiesel. TOPP was higher with different operating conditions of biodiesel in CE, when compared with pure diesel operation on CE. This was due to higher ignition delay with the biodiesel when compared with pure diesel fuel. This once again established the fact by observing lower peak pressures and higher TOPP, that CE with biodiesel operation showed the deterioration in the performance when compared with pure diesel operation on CE. Preheating of the biodiesel showed lower TOPP, compared with biodiesel at normal temperature. This once again confirmed by observing the lower TOPP and higher PP, the performance of the both versions of the engine is improved with the preheated biodiesel compared with the normal biodiesel. This trend of increase of MRPR and decrease of TOMRPR indicated better and faster energy substitution and utilization by biodiesel, which could replace 100% diesel fuel. However, these combustion characters were within the limits hence the biodiesel could be effectively substituted for diesel fuel.

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

Injection timing (°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	46.1	--	51.1	--	2.7	--	2.9	--	11	--	9	--	0	0	0	0
27/ Bio-diesel	CE	48.6	49.6	51.3	52.4	2.6	2.7	2.9	3.0	11	10	11	10	1	1	1	1
	LHR	62.9	63.8	67.3	67.5	3.6	3.7	3.8	3.9	9	8	9	9	1	1	1	1
31/Bio-diesel	LHR	65.8	66.5	67.8	68.6	3.7	3.9	3.9	4.1	8	8	8	8	0	0	0	0
33/Bio-diesel	CE	51.8		52.7		3.3		3.4		8		8		0		0	

CE-Conventional engine, LHR-Low heat rejection, NT-Normal temperature, PT-Preheated temperature.

4. Conclusions

The optimum injection timing was found to be 33°bTDC for CE, while it was 31°bTDC for LHR engine at an injection pressure of 190 bar. Relatively, Peak BTE increased by 23%, BSEC at load operation decreased by 5%, EGT at full load operation decreased by 65°C, CL at peak load operation decreased by 20%, VE at peak load decreased by 8%, smoke levels at peak load decreased by 37%, NOx levels at peak load increased by 29% and PP increased by 30% with LHR engine at its optimized injection timing when compared with pure diesel operation on CE at its recommended injection timing of 27°bTDC.

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