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Research Article

Studies on Performance Parameters of Di Diesel Engine with Low Grade LHR Combustion Chamber Fuelled with Cottonseed Biodiesel

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Abstract

Increased environmental awareness and depletion of resources are driving industry to develop alternative fuels that are environmentally more acceptable Investigations were carried out to evaluate the performance of a low grade low heat rejection (LHR) diesel engine with ceramic coated cylinder head with different operating conditions [normal temperature and pre-heated temperature] of cotton seed biodiesel with varied injector opening pressure and injection timing. Performance parameters [brake thermal efficiency, brake specific energy consumption, exhaust gas temperature, coolant load and volumetric efficiency] were evaluated at different values of brake mean effective pressure (BMEP) of the engine. Comparative studies were made with conventional engine (CE) with biodiesel and also with mineral diesel operation working on similar operating conditions. Engine with LHR combustion chamber improved its performance when compared with CE with biodiesel operation. Engine with LHR combustion chamber at optimum injection timing of 30^o bTDC with biodiesel operation increased peak brake thermal efficiency by 7%, decreased coolant load at full load by 15% and reduced volumetric efficiency at full load by 6% in comparison with CE with neat diesel operation at manufacturer's recommended injection timing of 27^o bTDC.

Keywords: *Crude vegetable oil; biodiesel, LHR combustion chamber, fuel performance*

1. Introduction

Fuel crisis because of dramatic increase in vehicular population and environmental concerns have renewed interest of scientific community to look for alternative fuels of bio-origin such as vegetable oils. Vegetable oils can be produced from forests, vegetable oil crops, and oil bearing biomass materials. Non-edible vegetable oils such as linseed oil, mahua oil, rice bran oil, cotton seed oil etc., are potentially effective diesel substitute. Vegetable oils have high-energy content and comparable cetane number to 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 that bears his name, experimented with fuels ranging from powdered coal to peanut oil and hinted that vegetable oil would be the future fuel [Venkateswara Rao. *et al*,2013]. Several researchers experimented the use of vegetable oils as fuel on conventional engines and reported that the performance was poor, citing the problems of high viscosity, low volatility and their polyunsaturated character. [Venkateswara Rao. *et al*,2013;–Avinash Kumar Agarwal and Atul Dhar. *et al*,2013].These problems can be solved to some extent, if neat vegetable oils are chemically modified (esterified) to bio-diesel. Experiments were conducted on conventional diesel engine with biodiesel operation and it was reported that biodiesel increased efficiency marginally and decreased particulate emissions and increased oxides of nitrogen.[McCarthy PM, Rasul MG and Moazzem S. . *et al*, 2011;– Krishna Maddali and Chowdary R. *et al* ,2014].

Experiments were conducted on preheated vegetable oils in order to equalize their viscosity to that of mineral diesel may ease the problems of injection process [Pugazhvadivu, M. and Jayachandran, K. *et al* 2005;Hanbey Hazar and Huseyin Aydin. *et al* 2010]. Investigations were carried out on engine with preheated vegetable oils. It was reported that preheated vegetable oils marginally increased thermal efficiency, decreased particulate matter emissions and NO_x levels, when compared with normal biodiesel.

Increased injector opening pressure may also result in efficient combustion in compression ignition engine [Celikten,I. *et al*,2003; Avinash Kumar Agarwal, Dhananjay Kumar Srivastava, Atul Dhar.*et al*. 2013]. It has a significance effect on performance and formation of pollutants inside the direct injection diesel engine combustion. Experiments were conducted on engine with biodiesel with increased injector opening pressure. It was reported that performance of the engine was improved, particulate emissions were reduced and NO_x levels were increased marginally with an increase of injector opening

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Table.1 Properties Test Fuels

Test Fuel	Viscosity at 25^{o}C (Centi-Stroke)	Specific gravity at 25^{o}C	Cetane number	Calorific value (kJ/kg)
Diesel	2.5	0.82	51	42000
Biodiesel (BD)	5.4	0.87	56	39900
ASTM Standard	ASTM D 445	ASTM D 4809	ASTM D 613	ASTM D 7314

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
Engine Displacement	553 cc
Method of cooling	Water cooled
Rated speed (constant)	1500 rpm
Fuel injection system	In-line and direct injection
Compression ratio	16:1
BMEP @ 1500 rpm at full load	5.31 bar
Manufacturer's recommended injection timing and injector opening pressure	27^{o}bTDC × 190 bar
Dynamometer	Electrical dynamometer
Number of holes of injector and size	Three × 0.25 mm
Type of combustion chamber	Direct injection type

pressure. The drawbacks (high viscosity and low volatility) of biodiesel call for LHR engine which provide hot combustion chamber for burning these fuels which got high duration of combustion.

The concept of engine with LHR combustion chamber is to minimize heat loss to the coolant by providing thermal insulation in the path of the coolant thereby increases the thermal efficiency of the engine. Several methods adopted for achieving LHR to the coolant are i) using ceramic coatings on piston, liner and cylinder head (low grade LHR combustion chamber) ii) creating air gap in the piston and other components with low-thermal conductivity materials like superni (an alloy of nickel), cast iron and mild steel etc. (medium grade LHR combustion chamber) and iii) combination of low grade and medium grade LHR combustion chamber resulted in high grade LHR combustion chamber. Investigations were carried out on engine with low grade LHR combustion chamber with neat diesel operation and it was reported that ceramic coatings provided adequate insulation and improved brake specific fuel consumption (BSFC). [Ekrem, B., Tahsin, E. and Muhammet, C.*et al* ,2006; Ciniviz, M., Hasimoglu, C., Sahin, F. and Salman, M.S. *et al* 2008].

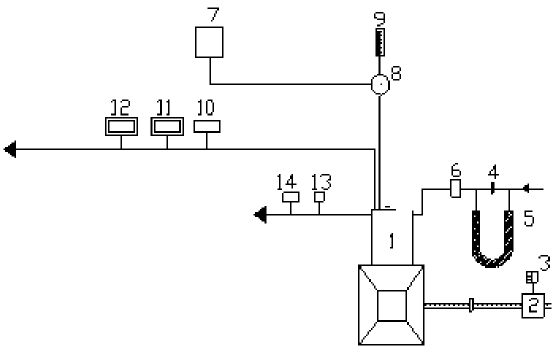
Studies were made on ceramic coated diesel engines with biodiesel and reported that performance was comparable, particulate emissions decreased while NO_x levels increased in comparison with neat diesel operation on CE. [Venkateswara Rao, N., Murali Krishna, M.V.S. and Murthy, P.V.K. *et al* 2013; Chowdary, R.P., Murali Krishna, M.V.S., Kishen Kumar Reddy, T. *et al* 2014] However, comparative studies were not made with mineral diesel operation working on similar conditions.

The present paper attempted to evaluate the performance parameters of engine with LHR combustion chamber which contained ceramic coated cylinder head fuelled with different operating conditions of cotton seed biodiesel with varied injector opening pressure and injection timing and compared with CE with biodiesel operation and also with mineral diesel operation working on similar working conditions.

2. Materials and Methods

*2.1 Preparation of biodiesel:*The chemical conversion of esterification reduced viscosity four fold. Cotton seed oil contains up to 70 % (wt.) free fatty acids. The methyl ester was produced by chemically reacting crude cotton seed oil with methanol in the presence of a catalyst (KOH). A two–stage process was used for the esterification of the crude cotton seed oil [Anirudh Gautam and Avinash Kumar Agarwal. *et al*,2013]. The first stage (acid-catalyzed) of the process is to reduce the free fatty acids (FFA) content in cotton seed oil by esterification with methanol (99% pure) and acid catalyst (sulfuric acid-98% pure) in one hour time of reaction at 55°C. Molar ratio of cotton seed oil to methanol was 9:1 and 0.5% catalyst (w/w). In the second stage (alkali-catalyzed), the triglyceride portion of the cotton seed oil reacts with methanol and base catalyst (sodium hydroxide–99% pure), in one hour time of reaction at 65°C, to form methyl ester (biodiesel) and glycerol. To remove un–reacted methoxide present in raw methyl ester, it is purified by the process of water washing with air–bubbling. The properties of the Test Fuels used in the experiment were presented in Table-1.

2.2 Experimental Set-up:Partially stabilized zirconium (PSZ) of thickness 500 microns was coated on inside portion of cylinder head. Experimental setup used for study of exhaust emissions on low grade LHR diesel engine with linseed biodiesel in Fig.1 The specification of the experimental engine is shown in Table.2 The engine was connected to an electric dynamometer (Kirloskar make) for measuring its brake power. Dynamometer was loaded by loading rheostat. The combustion chamber consisted of a direct injection type with no special arrangement for swirling motion of air.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 outlet temperature of water is maintained at 80^{o}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. The naturally aspirated engine was provided with water-cooling system in which outlet temperature of water is maintained at 80^{o}C by adjusting the water flow rate. Injector opening pressure was changed from 190 bar to 270 bar using nozzle testing device. The maximum injector opening pressure was restricted to 270 bar due to practical difficulties involved.



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. 10.Exhaust gas temperature indicator,11. AVL Smoke meter, 12. NOx Analyzer, 13. Outlet water jacket temperature indictor, 14. Water flow meter,

Fig.1 Experimental Set-up

Injection timing was changed by inserting copper shims between pump body and engine frame. Exhaust gas temperature (EGT) and coolant water outlet temperatures were measured with thermocouples made of iron and iron-Constantan attached to the temperature indicator

2.3 Operating Conditions: Different configurations of the combustion chamber used in the experiment were conventional engine and engine with LHR combustion chamber. The various operating conditions of the vegetable oil used in the experiment were normal temperature (NT) and preheated temperature (PT–It is the temperature at which viscosity of the vegetable oil is matched to that of diesel fuel, 80^{o}C). The injection pressures were varied from 190 bar to 270 bar. Various test fuels used in the experiment were biodiesel and diesel. The engine was started and allowed to have a warm up for about 15 minutes. Each test was repeated twelve times to ensure the reproducibility of data according to error analysis (Minimum number of trials must be not less than ten). The results were tabulated and a comparative study of performance parameters, were determined for various loads, injector opening pressures, injection timings at different operating conditions of the fuel.

2.4 Definitions of Parameters

$$m_f = \frac{10 \times \rho \times 3600}{t \times 1000} \tag{1}$$

$$\text{BP} = \frac{V \times I}{\eta_d \times 1000} \tag{2}$$

$$\text{BTE} = \frac{BP \times 3600}{m_f \times CV} \tag{3}$$

$$\text{BSEC} = \frac{1}{BTE} \tag{4}$$

$$BP = \frac{BMEP \times 10^5 \times L \times A \times n \times k}{60000} \tag{5}$$

$$CL = m_w \times (T_O - T_i) \tag{6}$$

$$m_a = C_d \times a \times \sqrt{2 \times 10 \times g \times h \times \rho_a \times 3600} \tag{7}$$

$$\rho_a = \frac{P_a \times 10^5}{750 \times R \times T_a} \tag{8}$$

$$a = \frac{\pi \times d^2}{4}$$

$$\eta_v = \frac{m_a \times 2}{60 \times \rho_a \times N \times V_s} \tag{9}$$

$$V_S = A \times L$$

2.5. Methodology

Mass of the fuel (m_f) consumed in kg/h was calculated by knowing density of fuel (ρ) in gm/cc measured with hydrometer and time taken for 10 cc of fuel measured with stop watch by using the equation–1. Brake power (BP) in kW at different percentages of load was calculated by knowing the voltmeter signal (V) and ammeter signal (I) and efficiency of dynamometer [(η_d) generally assumed as 0.85] by using the equation–2. Brake thermal efficiency (BTE) was determined by knowing mass of fuel consumed, brake power and calorific value of the fuel (CV) by using the equation–3. Brake specific energy consumption (defined as energy consumed by the engine in producing unit brake power, a parameter to compare two fuels with different properties on same configuration of the engine) was determined by using the equation–4, knowing the value of BTE. Brake mean effective pressure (BMEP) of the engine in bar was determined by knowing area of cylinder (A) in square meter, bore of the cylinder (D=0.080 m), stroke of piston (L= 0.110 m), number of power cycles per minute (n), which is equal to N/2, where N is the speed of the engine (1500 rpm) and number of cylinders (k=1) by using the equation–5. Coolant load (CL) in kW was calculated by knowing mass flow rate of coolant, (m_w) measured known quantity of coolant in unit time, specific heat of coolant (4.18 k J/kg–K), inlet temperature of coolant (T_i) and outlet temperature of coolant (T_o) by using the equation–6. Mass flow rate of air (ma) inducted in engine in kg/h was calculated by knowing coefficient of discharge (c_d=0.65), area of orifice meter (a) in square meter (diameter of orifice meter, d=0.020 m), difference of water column in U–tube water manometer (h in cm) and density of air (ρ_a) in kg/m^3 using the equation–7. Density of air was calculated from equation–8, by knowing pressure of air (P_a) in mm of mercury measured by barometer and temperature of ambient air (T_a) in Kelvin. Volumetric efficiency of engine (ηv) was calculated by equation–9, by knowing speed of the engine (N=1500 RPM), mass flow rate of air and stroke volume of cylinder(V_s), in m^3 which is equal to area of cylinder and stroke length of piston.

3. Results and Discussion

3.1 Fuel Performance

The optimum injection timing was 31^o bTDC with CE, while it was 30^o bTDC for engine with low grade LHR combustion chamber with mineral diesel (DF)operation [Venkateswara Rao, N., Murali Krishna, M.V.S. and Murthy, P.V.K. *etal*,2013; Srikanth, D., Murali Krishna, M.V.S., Ushasri, P. and Krishna Murthy, P.V. *etal*,2013]. From Fig.2, it is observed CE with biodiesel at 27^o bTDC showed comparable performance at all loads due to improved combustion with the presence of oxygen, when compared with mineral diesel operation on CE at 27^o bTDC. CE with biodiesel operation at 27^o bTDC decreased peak BTE by 3%, when compared with diesel operation on CE. This was due to low calorific value and high viscosity of biodiesel. CE with biodiesel operation increased BTE at all loads with advanced injection timing, when compared with CE with biodiesel operation at 27^o bTDC. This was due to initiation of combustion at early period and increase of resident time of fuel with air leading to increase of peak pressures. CE with biodiesel operation increased peak BTE by 4% at an optimum injection timing of 31^o bTDC, when compared with diesel operation at 27^o bTDC.

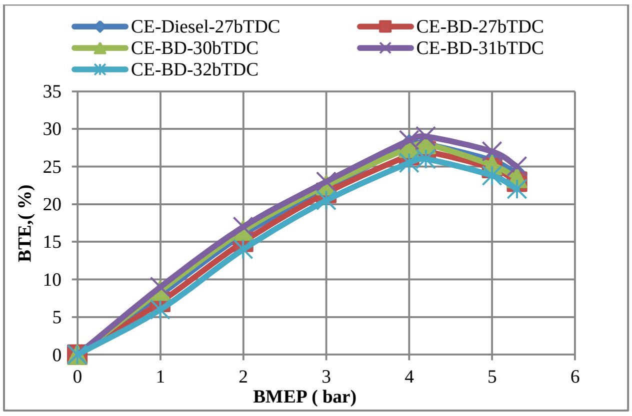


Fig.2 Variation of brake thermal efficiency (BTE) with brake mean effective pressure (BMEP) in conventional engine (CE) and with various injection timings at an injector opening pressure of 190 bar with biodiesel

Curves in Fig.3 indicate that LHR version of the engine at recommended injection timing showed the improved performance at all loads except at full load compared with CE with pure diesel operation. High cylinder temperatures helped in improved 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 combustion chamber improved heat release rates and efficient energy utilization. The optimum injection timing was found to be 30^{o}bTDC with LHR combustion chamber with different operating conditions of biodiesel operation. Since the hot combustion chamber of LHR combustion chamber reduced ignition delay and combustion duration and hence the optimum injection timing was obtained earlier with LHR combustion chamber when compared to conventional engine with the biodiesel operation.

Part load variations were very small and minute for the performance parameters. The effect of varied injection timing on the performance was discussed with the help of bar charts while the effect of injector opening pressure and preheating of biodiesel was discussed with the help of Tables.

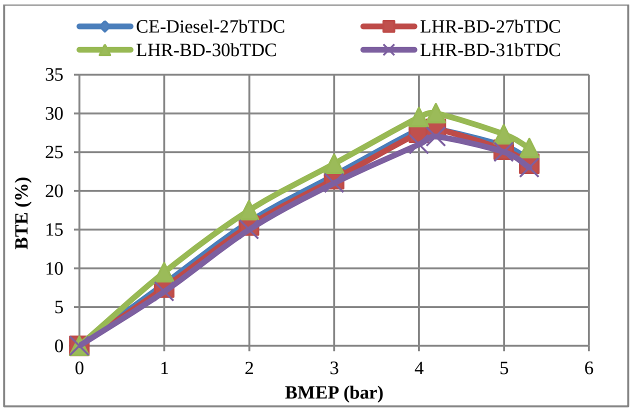


Fig.3 Variation of brake thermal efficiency (BTE) with brake mean effective pressure (BMEP) in LHR combustion chamber at different injection timings with biodiesel (LSOBD) operation.

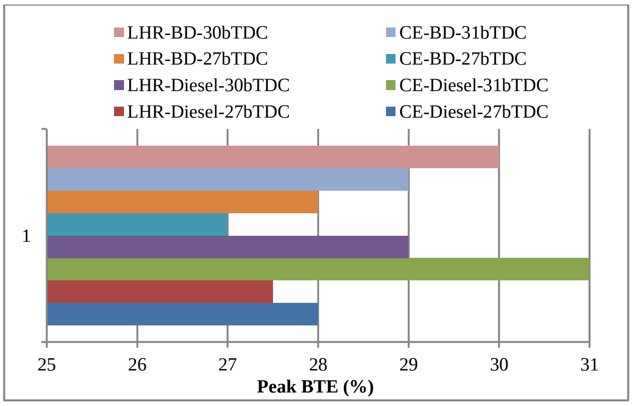


Fig.4 Bar charts showing the variation of peak brake thermal efficiency (BTE) with test fuels at recommended and optimized injection timings at an injector opening pressure of 190 bar in conventional engine and ceramic coated LHR combustion chamber

From Fig.4, it is noticed that CE with biodiesel operation decreased peak BTE by 4% at 27^o bTDC and 7% at 31^o bTDC when compared with neat diesel operation on CE. This was due to high viscosity and low calorific value and volatility of biodiesel. Engine with LHR combustion chamber with biodiesel operation increased peak BTE by 2% at 27^o bTDC and 3% at 30^o bTDC when compared with neat diesel operation on same configuration of the combustion chamber. This was due to improved combustion with higher cetane value of biodiesel in hot environment provided by the LHR combustion chamber

However, engine with LHR combustion chamber with biodiesel increased peak BTE by 4% at 27^o bTDC and 3% at 30^o bTDC in comparison with CE at 27^o bTDC and at 31^o bTDC. This was due to provision of insulation on cylinder head which reduced heat rejection leading to improve the thermal efficiency. This was also because of improved evaporation rate of the biodiesel. High cylinder temperatures helped in improved evaporation and faster combustion of the fuel injected into the combustion chamber. Reduction of ignition delay of biodiesel in the hot environment of the engine with LHR combustion chamber improved heat release rates and efficient energy utilization.

Brake specific fuel consumption, is not used to compare the two different fuels, because their calorific value, density, chemical and physical parameters are different. Performance parameter, brake specific energy consumption (BSEC), is used to compare two different fuels by normalizing brake specific energy consumption, in terms of the amount of energy released with the given amount of fuel.

From Fig.5, it is evident that engine with LHR combustion chamber with mineral diesel decreased BSEC at full load operation by 8% at 27^o bTDC and 6% at 30^o bTDC in comparison with CE at 27^o bTDC and at 31^o bTDC. This was due to reduction of ignition delay with neat diesel operation with LRH engine as hot combustion chamber was provided with engine with LHR combustion chamber.

CE with biodiesel operation showed comparable BSEC at full load at 27^o bTDC, while increasing it by 6% at 31^o bTDC when compared with neat diesel operation on CE. This was due to low calorific value of biodiesel requiring higher energy to produce unit brake power. Engine with LHR combustion chamber with biodiesel operation decreased BSEC at full load by 3% at 27^o bTDC and 1% at 30^o bTDC when compared with neat diesel operation on same configuration of the combustion chamber. This was due to reduction of ignition delay and producing peak pressures at near TDC with biodiesel operation. However, engine with LHR combustion chamber with biodiesel decreased BSEC at full load operation by 2% at 27^o bTDC and 1% at 30^o bTDC in comparison with CE at 27^o bTDC and at 31^o bTDC. BSEC was higher with CE due to due to higher viscosity, lower volatility and reduction in heating value of biodiesel lead to their poor atomization and combustion characteristics. The viscosity effect, in turn atomization was more predominant than the oxygen availability leads to lower volatile characteristics and affects combustion process**.** BSEC was improved with LHR combustion chamber with lower substitution of energy in terms of mass flow rate.

BSEC decreased with advanced injection timing with test fuels. This was due to initiation of combustion and increase of atomization of fuel with more contact of fuel with air. BSEC of biodiesel is almost the same as that of neat diesel fuel as shown in Figure.4. Even though viscosity of biodiesel is slightly higher than that of neat diesel, inherent oxygen of the fuel molecules improves the combustion characteristics. This is an indication of relatively more complete combustion.

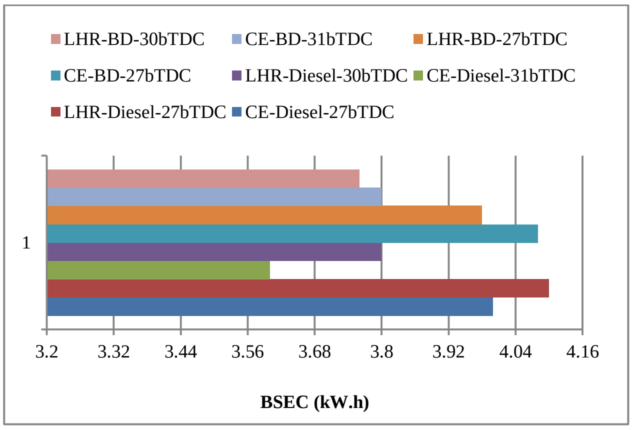


Fig.5 Bar charts showing the variation of brake specific energy consumption (BSEC) at full load operation with test fuels at recommended and optimized injection timings at an injector opening pressure of 190 bar in CE and LHR combustion chamber.

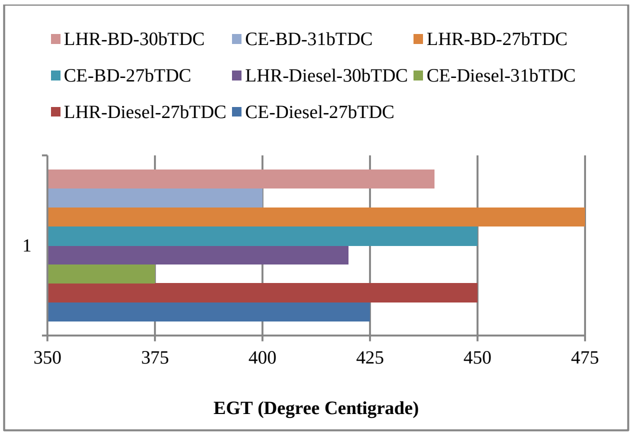


Fig.6 Bar charts showing the variation of exhaust gas temperature (EGT) at full load operation with test fuels at recommended and optimized injection timings at an injector opening pressure of 190 bar in conventional engine and LHR combustion chamber.

From Fig.6, CE with biodiesel operation increased EGT at full load by 6% at 27^o bTDC and 7% at 31^o bTDC when compared with neat diesel operation on CE. Though calorific value of biodiesel is less, its density is high giving rise to higher heat input and hence higher EGT than mineral diesel operation. This was also due to retarded heat release rate of biodiesel due to its high duration of combustion with its high viscosity. Engine with LHR

Table 3 Data of Peak Brake Thermal Efficiency, brake specific energy consumption and exhaust gas temperature at full load operation

Injection Timing (^{o}bTDC)	Test Fuel	Peak BTE (%)				BSEC at full load operation (kW.h)				EGT at full load operation (Deg. Centigrade)			
		Injection Pressure (Bar)				Injection Pressure (Bar)				Injection Pressure (Bar)			
		190		270		190		270		190		270	
		NT	PT	NT	PT	NT	PT	NT	PT	NT	PT	NT	PT
27(CE)	DF	28	--	30	--	4.0	--	3.92	--	425	--	395	--
	BD	27	28	29	30	4.08	4.04	4.0	3.96	450	475	400	425
27(LHR)	DF	27.5	--	29	--	4.1	--	3.96	--	450	--	410	--
	BD	28	28.5	29	29.5	3.98	3.94	3.90	3.86	475	450	425	400
30(LHR)	DF	29		30		3.80		3.72		420	--	380	--
	BD	30	31	32	32.5	3.76	3.72	3.68	3.64	440	420	400	380
31(CE)	DF	31		32		3.6	--	3.8	---	375	---	325	--
	BD	29	29.5	30.5	31	3.80	3.76	3.72	3.68	400	425	350	375

combustion chamber with biodiesel operation increased EGT at full load by 6% at 27^{o} bTDC and 5% at 30^{o} bTDC when compared with neat diesel operation on same configuration of the combustion chamber. However, engine with LHR combustion chamber with biodiesel operation increased EGT at full load by 6% at 27^{o} bTDC and 10% at 30^{o} bTDC in comparison with CE at 27^{o} bTDC and at 31^{o} bTDC. This indicated that heat rejection was restricted through cylinder head, thus maintaining the hot combustion chamber as result of which the exhaust gas temperature increased. EGT decreased with advanced injection timing with test fuels as seen from Figure. This was because, when the injection timing was advanced, the work transfer from the piston to the gases in the cylinder at the end of the compression stroke was too large, leading to reduce in the value of EGT.

Though the calorific value (or heat of combustion) of fossil diesel is more than that of biodiesel; density of biodiesel was higher ,therefore greater amount of heat was released in the combustion chamber leading to higher exhaust gas temperature with conventional engine, which confirmed that performance was comparable with CE with biodiesel operation in comparison with neat diesel operation.

Injector opening pressure was varied from 190 bar to 270 bar to improve the spray characteristics and atomization of the test fuels and injection timing is advanced from 27 to 34^{o} bTDC for CE and LHR combustion chamber. As it is observed from Table.3, peak brake thermal efficiency increased with increase in injector opening pressure at different operating conditions of the biodiesel. For the same physical properties, as injector opening pressure increased droplet diameter decreased influencing the atomization quality, and more dispersion of fuel particle, resulting in turn in better vaporization, leads to improved air-fuel mixing rate, as extensively reported in the literature [Venkateswara Rao, N., Murali Krishna, M.V.S. and Murthy, P.V.K.*etal*,2013; Srikanth, D., Murali Krishna, M.V.S., Ushasri, P. and Krishna Murthy, P.V. *etal*, 2013;Anirudh Gautam and Avinash Kumar Agarwal. *etal*,2013]. In addition, improved combustion leads to less fuel consumption.

Performance improved further with the preheated biodiesel when compared with normal biodiesel. This was due to reduction in viscosity of the fuel. Preheating of the biodiesel reduced the viscosity, which improved the spray characteristics of the oil causing efficient combustion thus improving brake thermal efficiency. The cumulative heat release was more for preheated biodiesel than that of biodiesel and this indicated that there was a significant increase of combustion in diffusion mode. This increase in heat release was mainly due to better mixing and evaporation of preheated biodiesel, which leads to complete burning.

From Table.3, it is noticed that BSEC at full load operation decreased with increase of injector opening pressure with different operating conditions of the test fuels. This was due to increase of air entrainment in fuel spray giving lower BSEC. BSEC decreased with the preheated biodiesel at full load operation when compared with normal biodiesel. Preheating of the biodiesel reduced the viscosity, which improved the spray characteristics of the oil.From same Table, it is noticed that EGT at full load operation of preheated biodiesel was higher than that of normal biodiesel, which indicates the increase of diffused combustion due to high rate of evaporation and improved mixing between methyl ester and air. Therefore, as the fuel temperature increased, the ignition delay decreased and the main combustion phase(that is, diffusion controlled combustion) increased, which in turn raised the temperature of exhaust gases. The value of exhaust gas temperature decreased with increase in injector opening pressure with test fuels as it is evident from Table. This was due to improved spray characteristics of the fuel with increase of injector opening pressure.

Fig.7 indicates CE with biodiesel operation increased coolant load at full load by 3% at 27^{o} bTDC and 5% at 31^{o} bTDC when compared with neat diesel operation on CE. This was due to un-burnt fuel concentration at combustion chamber walls. Engine with LHR combustion chamber with biodiesel operation decreased coolant load at full load by 5% at 27^{o} bTDC and 6% at 30^{o} bTDC when compared with neat diesel operation on same configuration of the combustion chamber. This was due to improved combustion eliminating deposits at near combustion chamber walls. Coolant load at full load operation increased with CE while decreasing with engine with LHR combustion chamber with advanced injection timing with test fuels.In case of CE, un-burnt fuel concentration reduced with effective utilization of energy, released from

Table 4 Data of Coolant Load and Volumetric Efficiency at full load operation

Injection Timing (°bTDC)	Test Fuel	Coolant Load (kW)				Volumetric Efficiency (%)			
		Injection Pressure (Bar)				Injection Pressure (Bar)			
		190		270		190		270	
		NT	PT	NT	PT	NT	PT	NT	PT
27(CE)	DF	4.0	---	4.4	---	85	--	87	--
	BD	4.1	3.9	4.5	4.3	83	82	85	84
27(LHR)	DF	3.8	--	3.4	--	80		82	
	BD	3.6	3.4	3.2	3.0	79	80	81	82
30(LHR)	DF	3.6		4.0		81		83	
	BD	3.4	3.2	3.0	2.8	80	81	82	83
31(CE)	DF	4.2	--	4.6	---	89	--	91	--
	BD	4.4	4.2	4.8	4.6	87	88	90	89

the combustion, coolant load with test fuels increased marginally at full load operation, due to un-burnt fuel concentration reduced with effective utilization of energy, released from the combustion, with increase of gas temperatures, when the injection timing was advanced to the optimum value. However, the improvement in the performance of CE was due to heat addition at higher temperatures and rejection at lower temperatures, while the improvement in the efficiency of the engine with LHR combustion chamber was due to recovery from coolant load at their optimum injection timings with test fuels.

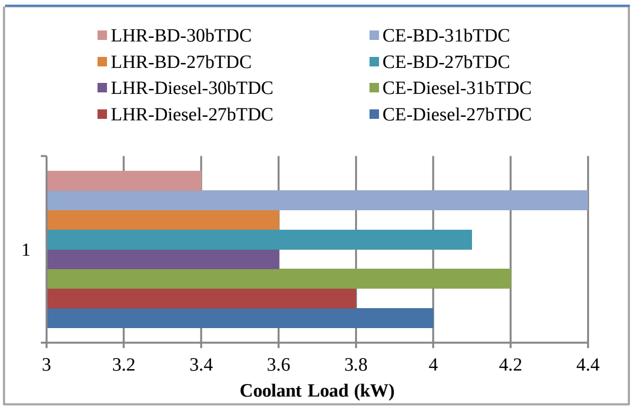


Fig.7 Bar charts showing the variation of coolant load at full load operation with test fuels at recommended and optimized injection timings at an injector opening pressure of 190 bar in conventional engine and LHR combustion chamber.

Volumetric efficiency depends on density of the charge which intern depends on temperature of combustion chamber wall. Fig. 8 denotes that engine with LHR combustion chamber with mineral diesel decreased volumetric efficiency at full load operation by 6% at 27° bTDC and 9% at 30° bTDC in comparison with CE at 27° bTDC and at 31° bTDC.

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. However, this variation in volumetric efficiency is very small between these two versions of the engine, as 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. However, engine with LHR combustion chamber with biodiesel decreased volumetric efficiency at full load operation by 5% at 27° bTDC and 8% at 30° bTDC in comparison with CE at 27° bTDC and at 31° bTDC with biodiesel operation.

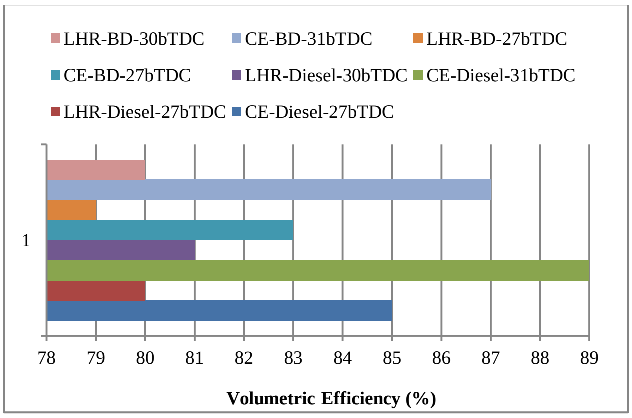


Fig.8 Bar charts showing the variation of volumetric efficiency at full load operation with test fuels at recommended and optimized injection timings at an injector opening pressure of 190 bar in conventional engine and LHR combustion chamber.

Volumetric efficiency was higher with neat diesel operation at recommended and optimized injection timing with both versions of the combustion chamber in comparison with biodiesel operation. This was due to increase of combustion chamber wall temperatures with biodiesel operation due to accumulation of un-burnt fuel concentration. This was also because of increase of combustion chamber wall temperature as exhaust gas temperatures increased with biodiesel operation in comparison with neat diesel operation.

Volumetric efficiency increased marginally with both versions of the engine with test fuels with advanced injection timing. This was due to decrease of combustion chamber wall temperatures with improved air fuel ratios.

From Table.4, it is observed that coolant load decreased marginally with preheating of biodiesel. This was due to improved air fuel ratios with improved spray characteristics. From same Table, it is seen that coolant load increased marginally in CE, while decreasing it in engine with LHR combustion chamber with increase of the injector opening pressure with test fuels. This was due to the fact with increase of injector opening pressure with conventional engine, increased nominal fuel spray velocity resulting in improved fuel-air mixing with which gas temperatures increased. The reduction of coolant load in the LHR combustion chamber was not only due to the provision of the insulation but also it was due to better fuel spray characteristics and increase of air-fuel ratios causing decrease of gas temperatures and hence the coolant load.

From Table.4, it is evident that preheating of the biodiesel marginally decreased volumetric efficiency, when compared with the normal temperature of biodiesel, because of reduction of bulk modulus, density of the fuel and increase of exhaust gas temperatures.

Volumetric efficiency at full load operation increased with increase of injector opening pressure with test fuels. This was due to improved fuel spray characteristics and evaporation at higher injection pressures leading to marginal increase of volumetric efficiency. This was also because of decrease of exhaust gas temperatures and hence combustion chamber wall temperatures. This was also due to the reduction of residual fraction of the fuel, with the increase of injector opening pressure.

4. Summary

Advanced injection timing and increase of injector opening pressure improved performance with biodiesel operation on engine with LHR combustion chamber. Preheated biodiesel further improved performance in both versions of the combustion chamber.

Comparison with CE with biodiesel*:* Engine with low grade LHR combustion chamber with cottonseed biodiesel improved performance over CE on the aspect of peak brake thermal efficiency, brake specific energy consumption, coolant load. However, it increased exhaust gas temperatures marginally and reduced volumetric efficiency in comparison with CE. Comparison with mineral diesel operation*:*

Conventional engine with biodiesel operation showed comparable performance, while engine with LHR combustion chamber improved performance when compared with mineral diesel operation. Hence it can be conveniently said that that engine with LHR combustion chamber is more suitable for biodiesel operation.

Research Findings

Performance was evaluated with engine with LHR combustion chamber consisting of ceramic coated combustion chamber with varied injector opening pressure and injection timing at different operating conditions of cottonseed biodiesel.

Future Scope of studies

Engine with low grade LHR combustion chamber gave lower volumetric efficiency at full load operation. The reduction of volumetric efficiency can be reduced by super charging.

Scientific Significance

Change of injection timing and injection pressure were attempted to evaluate the performance of the engine with change of configuration of combustion chamber with different operating conditions of the biodiesel.

Social Significance

Use of renewable fuels will strengthen agricultural economy, which curbs crude petroleum imports, saves foreign exchange and provides energy security besides addressing the environmental concerns and socio-economic issues.

Novelty

Change of injection timing of the engine was accomplished by inserting copper shims between pump body and engine frame. Performance of engine was evaluated with the simultaneous change of configuration of combustion chamber, change of fuel composition, change of operating temperature of fuel, change of engine parameters like injection timing and injection pressure and compared with mineral diesel operation.

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