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Comparative Studies On Exhaust Emissions and Combustion Characteristics Of Direct Injection Diesel Engine With Different Combustion Chambers With Rice Brawn Oil Based Biodiesel

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Abstract*—Experiments were carried out to determine exhaust emissions and combustion characteristics of diesel engine with high grade low heat rejection (LHR) combustion chamber 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 (normal temperature and preheated temperature) of rice brawn oil based biodiesel (ERBO) with varied injection timing and injector opening pressure. Exhaust emissions [smoke levels and oxides of nitrogen [NOx]] and combustion characteristics [peak pressure, maximum rate of pressure rise and time of occurrence of peak pressure] were determined at full load operation of the engine and compared with conventional engine at similar operating conditions. Biodiesel decreased smoke levels and increased NOx emissions with LHR combustion chamber when compared with conventional engine at similar operating conditions.

Index Terms—**Need for Alternate Fuels, Vegetable Oil, Biodiesel, LHR combustion chamber, Performance.**

I. INTRODUCTION

This section deals with need and necessity of alternative fuels and various alternative fuels. Investigations carried out by various researchers on crude vegetable oils and biodiesel at normal temperature and preheated temperature in compression ignition engine were mentioned. Conclusions from their investigations were given. Objectives of the investigations were given at the end of the section The world is presently confronted with the twin crises of fossil fuel depletion and environmental degradation. The fuels of bio origin can provide a feasible solution of this worldwide petroleum crisis (1-2). Vegetable oils and alcohols are prominent substitutes for diesel fuel. Alcohols have good volatility and low cetane number. Hence engine modification is necessary for use them as fuel in diesel engines. That too, most of the alcohol produced is diverted for Petro-chemical industries in India. Vegetable oils which are renewable in nature have properties compatible to diesel fuel. Rudolph Diesel, the inventor of the diesel engine that bears his name, experimented [3] with fuels ranging from powdered coal to peanut oil. Several researchers [4-6] 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 different fatty acids present in the vegetable oil are palmic, steric, lingoceric, oleic, linoleic and fatty acids [6]. These fatty acids increase smoke emissions and also lead to incomplete combustion due to improper air-fuel mixing. Experiments were conducted [7-10] on preheated vegetable oils [temperature at which viscosity of the vegetable oils were matched to that of diesel fuel] and it was reported that preheated vegetable oils improved the performance marginally. The problems of crude vegetable oils can be solved, if these oils are chemically modified to bio-diesel. The U.S. Department of Energy has stated [11] that, “Raw or refined vegetable oil, 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 vii) dilution of lubricating oil, viii) reduced engine life, ix) increased maintenance cost, xi) stress on fuel injection system. The above mentioned problems are reduced if crude vegetable oils are converted [11] into biodiesel, which have low molecular weight, low dense and low viscosity when compared with crude vegetable oils. Investigations were carried out [12-16] with biodiesel with CE and reported that biodiesel operation

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Volume 3, Issue 6, December 2013

improved the performance, reduced smoke emissions and increased NOx emissions. The drawbacks associated with biodiesel and crude vegetable oils for use in diesel engines call for LHR hot combustion chamber. The concept of LHR combustion chamber is to reduce coolant losses by providing thermal resistance in the path of heat flow to the coolant, there by gaining thermal efficiency. Several methods adopted for achieving LHR to the coolant are ceramic coated engines and air gap insulated engines with creating air gap in the piston and other components with low-thermal conductivity materials like superni, cast iron and mild steel etc. LHR combustion chambers were classified as low degree, medium grade and high grade LHR combustion chambers depending on degree of insulations. Investigations were carried out by various researchers [17-19] on low degree LHR combustion chambers- ceramic coated engines with pure diesel operation. It was reported from their investigations that pollution levels decreased with ceramic coated combustion chamber. Studies were also made [20-22] ceramic coated LHR combustion chamber with biodiesel operation. It was reported from their investigations that ceramic coated LHR combustion chambers decreased smoke levels and increased NOx emissions with biodiesel operation. The technique of providing an air gap in the piston involved the complications of joining two different metals. Investigations were carried out [23] on medium grade LHR combustion chambers- with air gap insulated piston with pure diesel. However, the bolted design employed by them could not provide complete sealing of air in the air gap. Investigations [24] were carried out with LHR combustion chamber with air gap insulated piston with nimonic crown threaded with the body of the piston fuelled with pure diesel with varied injection timing and reported that marginal decrease of smoke levels by 20% and increase of NOx emissions by 40%. Experiments were conducted with medium grade LHR combustion chamber- air gap insulated piston with superni crown and air gap insulated liner with superni insert with varied injection timing and injector opening pressure with different alternate fuels like crude vegetable oils [25-27] and biodiesel [28]. It was reported that medium grade LHR combustion chamber decreased smoke levels and increased NOx emissions. Investigations were carried out [29-30] on high degree of insulation-with air gap insulated piston, air gap insulated liner and ceramic coated cylinder head with biodiesel varied injection timing and injector opening pressure. It was reported that biodiesel improved smoke levels and drastically increased NOx emissions. The present paper attempted to determine exhaust emissions and combustion characteristics of high grade LHR combustion chamber, which consisted of air gap insulated piston, air gap insulated liner and ceramic coated cylinder head. This high grade LHR combustion chamber was fuelled with rice brawn oil based biodiesel (ERBO) with varied vied injector opening pressure and injection timing. Comparative performance studies were made on high grade LHR combustion chamber and CE with biodiesel operation.

II. METHODOLGY

This part deals with preparation of biodiesel, properties of biodiesel along with diesel fuels, fabrication of air gap insulated piston, air gap insulated liner and ceramic coated cylinder head, brief description of experimental set-up, specification of experimental engine, operating conditions and definitions of used values. Due to very high free fatty acid, rice bran oil was converted into methyl ester by the two stage process [31]. In the first stage rice bran oil was reacted with CH3OH in presence of an acid catalyst (H2SO4) to convert free fatty acid into fatty ester. A specified amount 1000g of rice bran oil was taken in a round bottom flask and heated up to 60-65°C. In a separate flasks CH3OH (950 g) and H2SO4 (22 g) were taken and properly mixed and then stirred for 4 h and maintained at 60°C.It was allowed to cool overnight without stirring. When acid number of the mixture reaches to less than 1, the second stage was started. During this stage, a mixture 1000g obtained from the first stage was taken in around bottom flask and heated up to 60°C. Methanol (200ml) and KOH 4.5g were properly mixed in other flask and then introduced into the round bottom flask containing the mixture from first stage. The mixture stirred vigorously for 2h and then allowed to cool overnight. Glycerol was separated by adding warm water at 60°C to the mixture. Glycerol and soap formed during the process settled down the bottom. Top layer containing rice bran oil methyl ester 91% was removed with the help of a separating funnel and wasted two times with water and dried. The physic-chemical properties of the crude vegetable oil and biodiesel in comparison to ASTM biodiesel standards are presented in Table-1.

Table I. Properties of Test Fuels

Property	Units	Diesel	Biodiesel (ERBO)	ASTM D 6751-02
Carbon chain	--	C_8-C_{28}	C_{16}-C_{24}	C_{12}-C_{22}
Cetane Number		55	52	48-70
Density	gm/cc	0.84	0.86	0.87-0.89
Bulk modulus @ 20Mpa	Mpa	1475	1800	NA
Kinematic viscosity				

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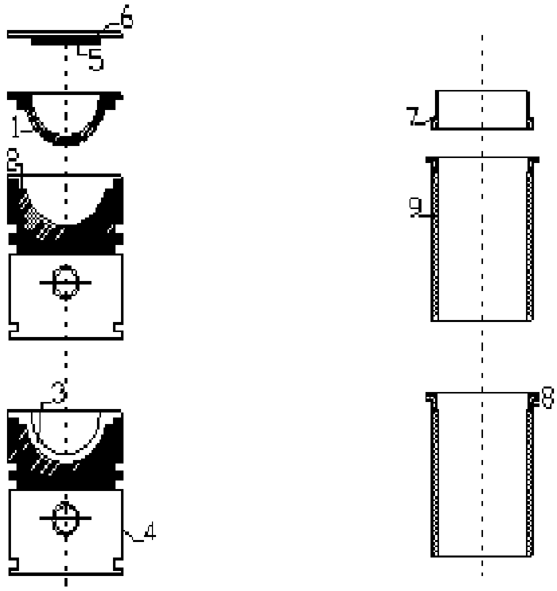
Volume 3, Issue 6, December 2013

@ 40°C	cSt	2.25	3.5	1.9-6.0
Sulfur	%	0.25	0.0	0.05
Oxygen	%	0.3	11	11
Air fuel ratio (stochiometric)	--	14.86	13.8	13.8
Lower calorific value	kJ/kg	42 000	38500	37 518
Flash point (Open cup)	°C	66	174	130
Molecular weight	--	226	261	292
Preheated temperature	°C	--	65	--
Colour	--	Light yellow	Yellowish orange	---

LHR diesel engine (Fig.1) 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. The optimum thickness of air gap in the air gap piston was found to be 3-mm [24], for improved performance of the engine with diesel as fuel. The height of the piston was maintained such that compression ratio was not altered. Partially stabilized zirconium of thickness 500 microns was applied on inner side of cylinder head by plasma technique. 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°C the thermal conductivity of superni-90 and air are 20.92 and 0.057 W/m-K respectively



1. Crown 2.Gasket 3.Air Gap 4.Body 5. Ceramic Coating 6. Cylinder Head 7 Insert 8. Air Gap 9. Liner

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

Insulated piston Insulated liner Ceramic coated cylinder head

Schematic diagram of experimental setup used for the investigations on compression ignition diesel engine and LHR combustion chamber with biodiesel (ERBO) is shown in Fig.2. The test fuels used in the experimentation were pure diesel and rice bran oil based biodiesel. The schematic diagram of the experimental setup with test fuels is shown in Figure 2. The specifications of the experimental engine are shown in Table-2. The combustion chamber consisted of a direct injection type with no special arrangement for swirling motion of air. The engine was connected to an 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 an air-box method (Air box was provided with an orifice flow meter and U-tube water manometer). The naturally aspirated engine was provided with water-cooling system in which inlet temperature of water was maintained at 80°C by adjusting the water flow rate. Engine oil was provided with a pressure feed system. No temperature control was incorporated, for measuring the lube oil temperature. Copper shims of suitable size were provided in between the pump body and the engine frame, to vary the injection timing and its effect on the performance of the engine was studied, along with the change of injector opening pressure from 190 bar to 270 bar (in steps of 40 bar) using nozzle testing device. The maximum injector opening pressure was restricted to 270 bar due to practical difficulties involved. Exhaust gas temperature was measured with thermocouples made of iron and iron-constantan. Exhaust emissions of smoke and NO_x were recorded by AVL (A company trade name) smoke meter and Netel Chromatograph (A company trade name) NOx analyzer respectively at full load operation of the engine. The specifications of the analyzers were given in Table-3.

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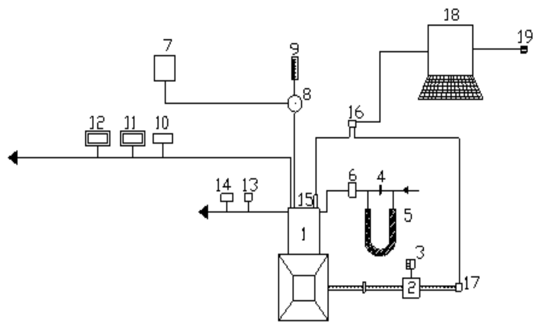
Volume 3, Issue 6, December 2013

Table II. Specifications of the Test Engine

Description	Specification
Engine make and model	Kirloskar (India) AV1
Maximum power output at a speed of 1500 rpm	3.68 kW
Number of cylinders ×cylinder position× stroke	One × Vertical position × four-stroke
Bore × stroke	80 mm × 110 mm
Method of cooling	Water cooled
Rated speed (constant)	1500 rpm
Fuel injection system	In-line and direct injection
Compression ratio	16:1
BMEP @ 1500 rpm	5.31 bar
Manufacturer's recommended injection timing and pressure	27^{o}bTDC × 190 bar
Dynamometer	Electrical dynamometer
Number of holes of injector and size	Three × 0.25 mm
Type of combustion chamber	Direct injection type
Fuel injection nozzle	Make: MICO-BOSCH No- 0431-202-120/HB
Fuel injection pump	Make: BOSCH: NO- 8085587/1

Table III. Specifications of Analyzers

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



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. Schematic Diagram of Experimental Set-up

Various test fuels used in experimentation were pure diesel and rice bran oil based biodiesel. Different operating conditions of the biodiesel were normal temperature and preheated temperature. Different injector opening injector opening pressures attempted in this experimentation were 190 bar, 230 bar and 270 bar. Various injection timings attempted in the investigations were 27-34^{o}bTDC.

Definitions of used values:

Recommended injection timing: It is the injection timing of the engine with maximum efficiency of the engine with minimum pollution levels. Optimum injection timing: It is injection timing at which maximum thermal efficiency was obtained at all loads and beyond this injection timing, efficiency of the engine decreased.

III. RESULTS AND DICUSSION

This section deals with determination of optimum injection timing with engine with conventional combustion chamber with biodiesel operation, determination of optimum injection timing with engine with LHR combustion chamber with biodiesel operation, determination of exhaust emissions with CE and LHR combustion chamber with biodiesel operation and determination of combustion characteristics with LHR combustion chamber Data of pure diesel was taken from reference [29]. The optimum injection timing with conventional engine was 31^{o}bTDC, while with LHR combustion chamber it was 28^{o}bTDC.

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Volume 3, Issue 6, December 2013

A. Performance Parameters

Figure 3 indicates that BTE increased up to 80% of the full load operation (BMEP=4.2 bar) due to conversion of increase of fuel efficiency and beyond that load it decreased due to decrease of air fuel ratios [26] as oxygen was completely used up with both test fuels. Curves from Figure 3 indicate that CE with bio-diesel showed the compatible 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 compatible performance of CE with bio-diesel operation.

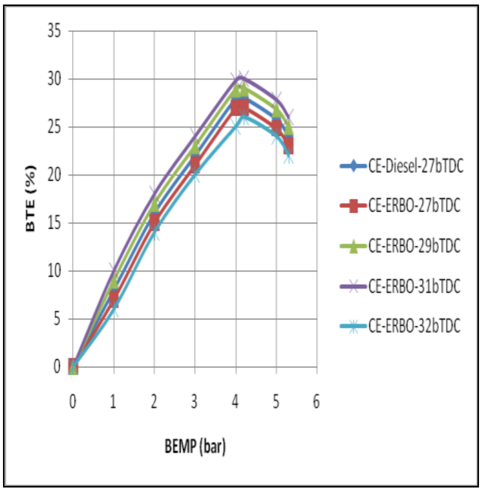

Fig.3. Variation Of Brake Thermal Efficiency (BTE) With Brake Mean Effective Pressure (BMEP) in Conventional Engine (CE) at Various Injection Timings With Biodiesel (ERBO) Operation at an Injector Opening Pressure Of 190 Bar.

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 was due to initiation of combustion at earlier period and efficient combustion with increase of air entrainment [26] in fuel spray giving higher BTE. BTE increased at all loads when the injection timing was advanced to 31^{o}bTDC in CE at the normal temperature of bio-diesel. Similar trends were observed with preheated biodiesel also. Preheating of the biodiesel reduced the viscosity, which improved the spray characteristics of the oil. From Fig.4, it is observed that LHR combustion chamber with biodiesel showed the improved performance for the entire load range compared with CE with pure diesel operation. This was because of efficient combustion of biodiesel in the hot environment provided by LHR combustion chamber. The optimum injection timing was found to be 28^{o}bTDC with LHR combustion chamber with normal bio-diesel and preheated biodiesel operations. 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 with CE with the biodiesel operation.

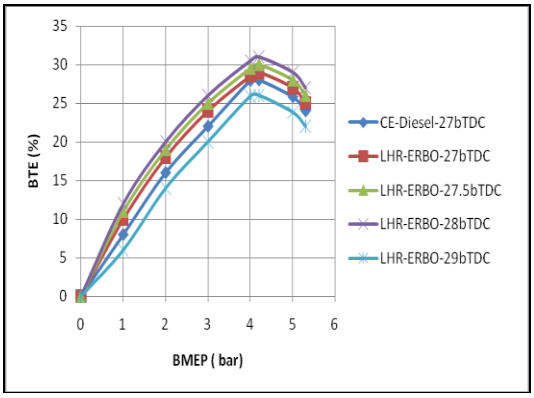

Fig.4 Variation Of Brake Thermal Efficiency (BTE) With Brake Mean Effective Pressure (BMEP) In LHR Combustion Chamber At Various Injection Timings With Biodiesel (ERBO) Operation An Injector Opening Pressure Of 190 Bar.

Injector opening pressure was varied from 190 bars to 270 bar to improve the spray characteristics and atomization of the vegetable oils and injection timing was advanced from 27 to 34^{o}bTDC for CE and LHR engine. Part load variations were very small and minute for the exhaust emissions and combustion characteristics. 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 was discussed with the help of Tables.

B. Exhaust Emissions

From Fig.5, it is observed that drastic increase of smoke levels at peak load operation with biodiesel operation was observed compared with pure diesel operation. This was due to the higher value of ratio of C/H (C= Number of carbon atoms and H= Number of hydrogen atoms in fuel composition ($C_{10}H_{14}N_2$), higher the value of this ratio means, number of carbon atoms are higher leading to produce more carbon dioxide and more carbon monoxide and hence higher smoke levels) of crude vegetable oil **(0.7)** when compared with pure diesel (0.45). The increase of smoke levels was also due to decrease of air-fuel ratios [29] and volumetric efficiency [29] with biodiesel compared with pure diesel operation. Smoke levels were related to the density of the fuel. Since biodiesel has higher density compared to diesel fuel, smoke levels were higher with biodiesel. Smoke levels decreased [29] at the respective optimum injection timing

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International Journal of Engineering and Innovative Technology (IJEIT)
Volume 3, Issue 6, December 2013

with test fuels. The inherent oxygen of biodiesel might have provided some useful interactions between air and fuel, particularly in the fuel-rich region. Certainly, it is evident proof of the oxygen content of biodiesels enhanced the oxidation of hydrocarbon reactions thus reducing smoke levels. Smoke levels were lower with pure diesel operation followed by biodiesel. This was due to lower value of C/H ratio, density, air fuel ratio of the diesel fuel. The data from Table 4 shows a decrease in smoke levels with increase of injector opening pressure, with different operating conditions of the biodiesel. This was due to improvement in the fuel spray characteristics at higher

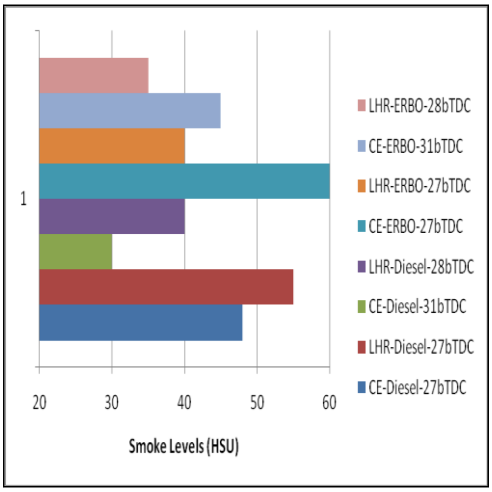

Fig.5 Bar charts showing the variation of smoke levels in Hartridge smoke unit (HSU) at full load operation with test fuels at recommended and optimized injection timings at an injector opening pressure of 190 bar.

Injector opening pressure and increase of air entrainment, at the advanced injection timings, causing lower smoke levels. Even though viscosity of biodiesel was higher than diesel, high injector opening pressure improves spray characteristics, hence leading to a shorter physical delay period. The improved spray also leads to better mixing of fuel and air resulting in turn in fast combustion. This will enhance the performance [29]. Preheating of the biodiesel (Table.4) reduced smoke levels, when compared with normal temperature of the biodiesel. This was due to i) the reduction of density of the biodiesel, as density was directly related to smoke levels, ii) the reduction of the diffusion combustion proportion 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 directed into the combustion chamber**.** By the esterification process, the viscosity of the biodiesel was brought down many times. Since there was drop in the viscosity, naturally the density of the esterified oil was also dropped at the room temperature. Volatility of the biodiesel also increased with the esterification process. Smoke levels were lower with diesel operation followed by biodiesel. This was due to the nature of the composition of the diesel fuel. Diesel fuel has lower value of C/H, lower viscous and lower theoretical air fuel ratio.

Table IV. Data of Smoke Levels at Full Load Operation

Injection Timing (bTDC)	Test Fuel	Peak BTE (%)											
		Conventional Engine						LHR Engine					
		Injector opening pressure (Bar)						Injector opening pressure (Bar)					
		190		230		270		190		230		270	
		NT	PT	NT	PT	NT	PT	NT	PT	NT	PT	NT	PT
27	DF	48	--	38	--	34	--	55	--	50	--	45	--
	ERBO	60	55	55	50	50	45	40	35	35	30	30	25
28	DF	--	--	--	--	--	--	40		35		30	
	ERBO	--	--	--	--	--	--	35	30	30	25	25	20
31	DF	30	--	35	--	40	--	-	-	-	-	-	-
	ERBO	45	40	50	45	55	50	-	-	-	-	-	-

DF-Diesel Fuel, ERBO- Biodiesel, NT- Normal or Room Temperature, PT- Preheat Temperature

In Fig.6 NOx are the precursor pollutants which can combine to form photochemical smog. These irritate the eyes and throat, reduces the ability of blood to carry oxygen to the brain and can cause headaches, and pass deep into the lungs causing respiratory problems for the human beings. Long-term exposure has been linked with leukemia. Therefore, the major challenge for the existing and future diesel engines is meeting the very tough emission targets at affordable cost, while improving fuel economy. Temperature and availability of oxygen are two

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Volume 3, Issue 6, December 2013

favorable conditions to form NOx levels. At peak load, NOx levels increased with test fuels at recommended injection timing due to higher peak pressures, temperatures as larger regions of gas burned at close-to-stoichiometric ratios.

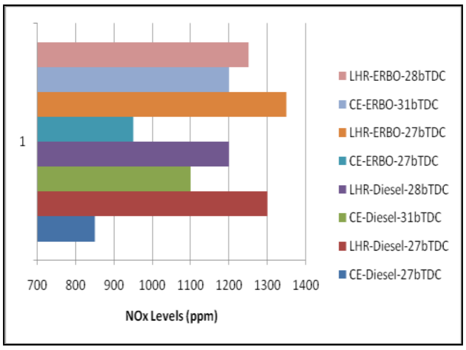

Fig.6 Bar charts showing the variation of NOx levels at peak load operation with test fuels at recommended and optimized injection timings at an injector opening pressure of 190 bar.

The rice bran oil based biodiesel having long carbon chain (C_{20}-C_{32}) [31] recorded more NOx than that of fossil diesel having both medium (C_8-C_{14}) as well as long chain (C_{16}-C_{28}). The increase in NOx emission might be an inherent characteristic of biodiesel due to the presence of 54.9% of mono-unsaturated fatty acids (MUFA) and 18% of poly-unsaturated fatty acids (PUFA). That means, the long chain unsaturated fatty acids (MUFA and FUPA) such as oleic C18:1 and linoliec C18:2 fatty acids are mainly responsible for higher levels of NOx emission [32]. Another reason for higher NOx levels is the oxygen (10%) present in the methyl ester. The presence of oxygen in normal biodiesel leads to improvement in oxidation of the nitrogen available during combustion. This will raise the combustion bulk temperature responsible for thermal NOx formation Many researchers reported that oxygen [32] and nitrogen [33] content of biodiesel can cause an increase in NO*x* emissions. The production of higher NOx with biodiesel fueling is also attributable to an inadvertent advance of fuel injection timing due to higher bulk modulus of compressibility, with the in-line fuel injection system. Residence time and availability of oxygen had increased, when the injection timing was advanced with test fuels, which caused higher NOx levels. From the Table 5, it is noted that these levels increased with increase of injector opening pressure with different operating conditions of biodiesel as it is noticed from the Table.5. NO*x* slightly increased with test fuels as injector opening pressure increased. The increase in peak brake thermal efficiency [29] was proportional to increase in injector opening pressure. Normally, improved combustion causes higher peak brake thermal efficiency due to higher combustion chamber pressure (Table. 5) and temperature and leads to higher NO*x* formation. This is an evident proof of enhanced spray characteristics, thus improving fuel air mixture preparation and evaporation process. NOx levels decreased with preheating of the biodiesel as noticed from the Table.5. The fuel spray properties may be altered due to differences in viscosity and surface tension. The spray properties affected may include droplet size, droplet momentum, and degree of mixing, penetration, and evaporation. The change in any of these properties may lead to different relative duration of premixed and diffusive combustion regimes. Since the two burning processes (premixed and diffused) have different emission formation characteristics, the change in spray properties due to preheating of the biodiesel are lead to reduction in NOx formation. As fuel temperature increased, there was an improvement in the ignition quality, which will cause shortening of ignition delay. A short ignition delay period lowers the peak combustion temperature which suppresses NOx formation [32]. Lower levels of NOx is also attributed to retarded injection, improved evaporation, and well mixing of preheated biodiesel due to their viscosity at preheated temperatures. Biodiesel has higher value of NOx emissions followed by diesel. This was because of inherent nature of biodiesel as it has oxygen molecule in its composition.

Table V. Data of NOx Levels at Full Load Operation

Injection Timing (bTDC)	Test Fuel	Peak BTE (%)											
		Conventional Engine						LHR Engine					
		Injector opening pressure (Bar)						Injector opening pressure (Bar)					
		190		230		270		190		230		270	
		NT	PT	NT	PT	NT	PT	NT	PT	NT	PT	NT	PT
27	DF	850	----	890	----	930	---	1300	--	1280	--	1260	--
	ERBO	950	900	1000	950	1050	1000	1350	1300	1300	1250	1250	1200
28	DF	--	--	--	--	--	--	1200		1150		1100	
	ERBO	--	--	--	--	--	--	1250	1200	1200	1150	1150	1100
31	DF	1100	-	1150	-	1200	--	-	-	-	-	-	-
	ERBO	1200	1150	1250	1200	1300	1250	-	-	-	-	-	-

DF-Diesel Fuel, ERBO- Biodiesel, NT- Normal or Room Temperature, PT- Preheat Temperature

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C. Combustion Characteristics

From the Figure 6, it is noticed that peak pressure for normal biodiesel was slightly higher than that of diesel fuel; even though biodiesel was having lower value of lower calorific value. Biodiesel advanced the peak pressure position as compared to fossil diesel because of its higher bulk modulus and cetane number. This shift is mainly due to advancement of injection due to higher density and earlier combustion due to shorter ignition delay caused by higher cetane number of biodiesel. When, a high density (or high bulk modulus) fuel is injected, the pressure wave travels faster from pump end to nozzle end, through a high pressure in-line tube [32]. This causes early lift of needle in the nozzle, causing advanced injection. Hence, the combustion takes place very close to TDC (lower value of time of occurrence of peak pressure) and the peak pressure slightly high due to existence of smaller cylinder volume near TDC. However, the peak pressures of preheated methyl ester were less than that of normal biodiesel (Table.6). When the engine is running on preheated biodiesel the fuel injection was slightly delayed, due to decrease in bulk modulus of biodiesel with the increase in fuel temperature. The reasons for lower peak pressures of preheated biodiesel was also attributed to earlier combustion caused by short ignition delay (due to faster evaporation of the fuel) at their preheated temperatures. Peak pressures increased with the increase of injector opening pressure and with the advancing of the injection timing with the test fuels. Peak pressure increased as injector opening increased. This may be due to smaller sauter mean diameter [32-33] shorter breakup length, better dispersion, and better spray and atomization characteristics. This improves combustion rate in the premixed combustion phase.

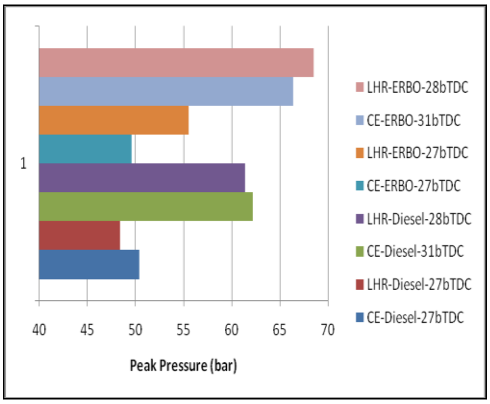

Fig.7 Bar charts showing the variation of peak pressure at peak load operation with test fuels at recommended and optimized injection timings at an injector opening pressure of 190 bar.

Maximum rate of pressure rise (MRPR) was highest (Fig.8) for normal diesel followed by the biodiesel. With biodiesel, as injector opening pressure increased, spray characteristic improved and in turn burned fuel increased again and in turn combustion rate increased in the premixed combustion phase [33].

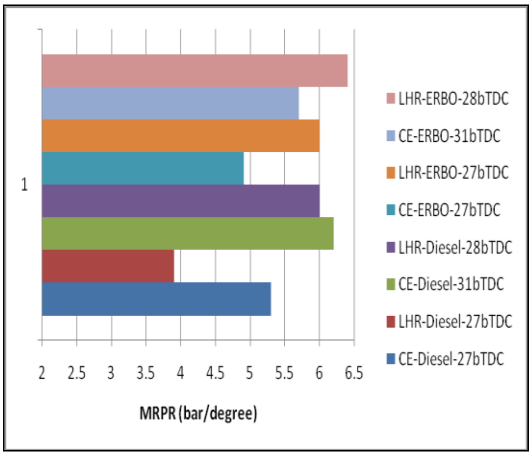

Fig.8 Bar charts showing the variation of maximum rate of pressure rise at peak load operation with test fuels at recommended and optimized injection timings at an injector opening pressure of 190 bar.

When the engine is operated under the full load condition, the mechanical loading is at the maximum level. The differences [32] in maximum rate of pressure rise, the peak cylinder pressure, and the occurrence of peak cylinder pressure during the maximum mechanical loading may cause performance loses. In the present work both peak cylinder pressure and maximum rate of pressure rise of biodiesel was lower and occurrence of peak cylinder pressure slightly deviated away when compared to normal diesel [33]. However it decreased as injector opening pressure of biodiesel increased. The value of time of occurrence of peak pressure (TOPP) decreased with the advancing of the injection timing and with increase of injector opening pressure at different operating conditions of the test fuels. Preheating of the biodiesel showed lower TOPP, compared with biodiesel at normal temperature. This once again confirmed by observing the lower TOPP, the performance of the engine improved with the preheated biodiesel compared with the normal biodiesel. This trend of increase of maximum rate of pressure rise indicated improved and faster energy substitution and utilization by biodiesel in engine, which could replace 100% diesel fuel. That too, all these combustion characters were within the limits hence the biodiesel can be effectively substituted for diesel fuel.

Table VI. Data of Combustion Characteristics at Peak Load Operation

Injection Timing (° bTDC)	Test Fuel	PP (bar)				MRPR (bar/deg)				TOPP (deg)			
		Injector opening pressure				Injector opening pressure				Injector opening pressure			
		190		270		190		270		190		270	
		NT	PT	NT	PT								
27(CE)	DF	50.4	--	53.5	---	5.4	--	6.0	--	10		9	
	ERBO	49.6	48.4	52.5	51.5	5.0	4.0	5.3	4.3	11	10	10	9
27(LHR)	DF	48.4	--	51.2	--	3.8	3.2	4.5	3.8	11	10	10	9
27(LHR)	ERBO	55.5	54.2	57.6	55.4	6.3	5.7	6.8	6.3	10	9	10	9
28(LHR)	ERBO	68.5	67.3	70.4	69.6	6.8	5.6	7.2	6.0	8	8	8	8
28(LHR)	DF	61.4	-	62.4	--	6.0		6.2		8		8	
31(CE)	DF	62.2	--	61.9	--	6.2	--	6.8.	--	8		8	
	ERBO	66.4	64.1	68.5	65.5	5.7	4.5	6.1	4.9	8	8	8	8

IV. CONCLUSION

LHR combustion chamber with biodiesel operation decreased smoke emissions by 33% and 22% respectively at recommended and optimized injection timings when compared with CE. LHR combustion chamber with biodiesel operation increased NOx emissions by 42% and 4% respectively at recommended and optimized injection timings when compared with CE. LHR combustion chamber with biodiesel operation increased peak pressure by 4% and 3% respectively at recommended and optimized injection timings when compared with CE. LHR combustion chamber with biodiesel operation increased maximum rate of pressure rise by 20% and 20% respectively at recommended and optimized injection timings when compared with CE.

A. Research Findings and Suggestions

Comparative studies on performance parameters with direct injection diesel engine with high grade low heat rejection combustion chamber and conventional combustion chamber were determined at varied injector opening pressure and injection timing with different operating conditions of the biodiesel. Experimental results were compared with pure diesel operation at similar operating conditions. Hence further work on other grades of LHR combustion chamber with biodiesel operation is necessary. Reduction of NOx emissions from LHR combustion chamber is to be attempted.

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