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Performance evaluation of waste fried vegetable oil in a medium grade low heat rejection diesel engine

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Combustion characteristics.

Abstract

Investigations were carried out to evaluate the performance of direct injection diesel engine with medium grade low heat rejection (LHR) combustion chamber and 3 mm air gap insulated piston, 3 mm air gap insulated liner, and ceramic coated cylinder head [ceramic coating with the thickness of 500 μ was made on the inside portion of the cylinder head].The engine had different operating conditions [normal temperature and pre–heated temperature] of crude waste fried vegetable oil (WFVO) which was collected from restaurants, hotels, etc. with varied injector opening pressure and injection timing. Performance parameters and exhaust emissions were evaluated at various values of brake mean effective pressure of the engine, while combustion parameters were determined at full load operation of the engine using special pressure–crank angle software package. Comparative studies were performed between vegetable oil operation and diesel operation in the engine with both versions of the combustion chamber with varied injection timing and injector opening pressure. Conventional engine (CE) showed deteriorated performance, while the engine with medium grade LHR combustion chamber had improved performance with waste fried vegetable oil operation at the recommended injection timing and pressure. Performance of both versions of the combustion chamber improved with advanced injection timing and at higher injector opening pressure compared with CE with pure diesel operation. The optimum injection timing was 32^o bTDC (before top dead centre) with conventional engine, while it was 30^o bTDC for the engine with LHR combustion chamber and vegetable oil operation. Compared with pure diesel operation in the conventional engine, at manufacturer's recommended injection timing of 27^o bTDC, peak brake thermal efficiency increased by 9% at full load operation, brake specific energy consumption decreased by 2%, volumetric efficiency decreased by 13%, smoke levels decreased by 10%, and nitrogen oxide (NO_x) levels increased by 44% with waste fried vegetable oil operation in the engine with LHR combustion chamber.

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Nomenclature

ρ_a =Density of air, kg/m^3
ρ_d=Density of fuel, gm/cc
η_d=Efficiency of dynamometer, 0.85
a=Area of the orifice flow meter, m^2
BP=Brake power of the engine, kW
BMEP=Brake mean effective pressure, bar
C=Number of carbon atoms in fuel composition
C_d =Coefficient of discharge, 0.65
Cp=Specific heat of water, kJ/kg. K°
D=Bore of the cylinder, 80 mm
D=Diameter of the orifice flow meter, 20 mm
DF=Diesel fuel
H=Number of hydrogen atoms in fuel
HSU=Hartridge smoke unit
I=Ammeter reading, ampere
H=Difference of water level in U-tube water manometer in cm of water column
IT=Injection timing, degree bTDC
K=Number of cylinders, 01
L=Stroke of the engine, 110 mm
m_a=Mass of air inducted in the engine, kg/h
m_f=Mass of fuel, kg/h
m_w=Mass flow rate of coolant (water), kg/s
n=Power cycles per minute, N/2,
N=Speed of the engine, 1500 rpm
NT=Normal temperature, degree centigrade
P_a=Atmosphere pressure in mm of mercury
Pt=Preheated temperature
R=Gas constant for air, 287 J/kg. K°
T=Time taken for collecting 10 cc of fuel, s
T_a=Room temperature, C°
T_I=Inlet temperature of water, C°
T_o=Outlet temperature of water, C°
V=Voltmeter reading, volt
V_s=Stroke volume, m^3

1. Introduction

Alternate fuel research has been the topic with the highest priority in the context of depletion of fossil fuels at alarming rate and increase of pollution levels of the engines with conventional fuels. High consumption of diesel fuel in not only agriculture sector but also transport sector compels the substitution of diesel fuel with suitable, renewable energy resources. Alcohols and vegetable oils are the major alternate fuels for diesel fuel, as they are renewable in nature. Alcohols have low cetane number and hence engine modification is necessary if they are to be used as fuels in diesel engines. Moreover, most of the alcohol produced in India is diverted for petro–chemical industries. On the other hand, vegetable oils have high cetane number, which is compatible to diesel fuel.

Rudolph Diesel, the inventor of the engine that bears his name, experimented with fuels ranging from powdered coal to peanut oil [1]. Several Studies have experimented the use of vegetable oils as fuel in the conventional engines (CE) and reported that its poor performance while citing the problems of high viscosity, low volatility, and polyunsaturated character [2–7]. Not only that, the common problems of crude vegetable oils in diesel engines include the formation of carbon deposits, oil ring sticking, and thickening and gelling of lubricating oil as a result of contamination by vegetable oils. The drawbacks associated with the use of vegetable oils in conventional diesel engines call for hot combustion chamber provided by the engine with low heat rejection (LHR) combustion chamber, which has significant characteristics of higher operating temperature, maximum heat release, higher brake thermal efficiency, and ability to handle lower calorific value and high viscous fuel.

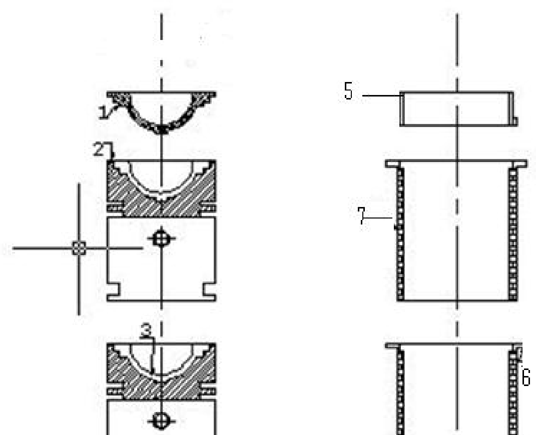
The concept of engine with LHR combustion chamber is to prevent heat flow to the coolant by providing thermal insulation on its path. Engines with LHR combustion chambers are classified depending on the degree of insulation as low grade, medium grade, and high grade insulated engines. Low grade insulated combustion chamber provides thermal resistance by means of coatings with low thermal conductivity materials on piston, liner, and cylinder head, while medium grade LHR combustion chamber provides an air gap in the piston and other components with low thermal conductivity materials like superni (an alloy of nickel), cast iron, and mild steel. The combination of low and medium grade LHR combustion chamber results in high grade LHR combustion chamber.

Investigations were carried out on the engine with low grade LHR combustion chamber with pure diesel and vegetable oils. Ceramic coatings were reported to provide adequate insulation,

improved brake specific fuel consumption (BSFC), and reduced smoke emissions [8-13].
Investigations were carried out on the engine with medium grade LHR combustion chamber with varied injection timing and injector opening pressure with vegetable oils. The results reported that thermal efficiency increased by 10%, volumetric efficiency decreased by 6%, smoke levels decreased by 8% with the engine with LHR combustion chamber at 27^{o} bTDC and at an injector opening pressure of 190 bar when compared with pure diesel operation in the conventional engine [14-18]. Performance improved with advanced injection timing and higher injector opening pressure.
Also, the engine with high grade LHR combustion chamber with air gap insulated piston, air gap insulated liner, and ceramic coated cylinder head with vegetable oil operation was studied and the results revealed that the engine with LHR combustion chamber improved performance with vegetable oil operation in comparison with pure diesel operation in the conventional engine [19-20]. However, the engine with high grade LHR combustion chamber with vegetable oil operation drastically increased NO_x levels by 59% compared with the conventional engine with diesel operation.
Increased injector opening pressure may also result in efficient combustion in compression ignition engine [5–7]. It has a significant effect on the performance and formation of pollutants inside the direct injection diesel engine combustion. Performance of the engine was reported to be improved with advanced injection timing [4–6]. Also, it increased NO_x levels and decreased particulate emissions.
The present paper attempted to evaluate the performance of palm oil based waste fried vegetable oil with an engine with medium grade LHR combustion chamber, which contained an air gap insulated piston and air gap insulated liner with different operating conditions of vegetable oil with varied injector opening pressure and injection timing. Then, it was compared with the conventional engine in similar operating conditions.

2. Materials and method

It is observed in Fig. 1 that the engine with medium grade LHR combustion chamber contained a two part piston – the top crown made of low thermal conductivity material, superni–90 was screwed to aluminum body of the piston and provided a 3mm air gap in between the crown and body of the piston. The optimum thickness of air gap in the air gap piston was found to be 3 mm for the improved performance of the engine with superni insert with diesel as fuel [15]. A superni–90 insert was screwed to the top portion of the liner in such a manner that a 3 mm air gap was maintained between the insert and liner body. Properties and composition of superni-90 are shown in Tables 1 and 2. Plates 1 and 2 demonstrate photographic views of insulated piston and insulated liner.



1.Superni crown, 2. Superni gasket, 3.Air gap in the piston, 4.Aluminium body of the piston, 5.Superniinsert, 6. Air gap in the liner, 7.Body of the liner.

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

Table 1. Properties of superni–90.

Thermal conductivity at 500^0 C	21 W/m–K
Melting Point	1400^0C
Young's modulus at 500^0C	1328 N/m^2
Mean coefficient of Thermal expansion	14.1×10^{-6}
Electrical resistivity at room temperature	1ohm m^2 /m

Table 2. Composition of supreni–90.

Cobalt	2.0 %
Chromium	2.93 %,
Aluminum	1.5 %,
Titanium	2.5 %
Carbon	0.07%
Iron	1 %
Nickel	Balance

Fig. 2. Photographic view of air gap insulated piston.

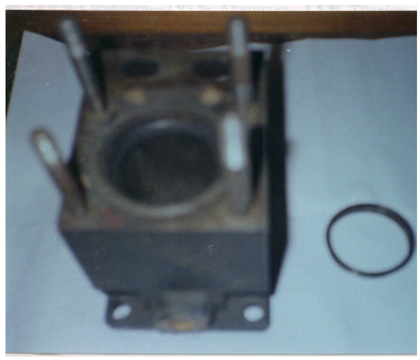
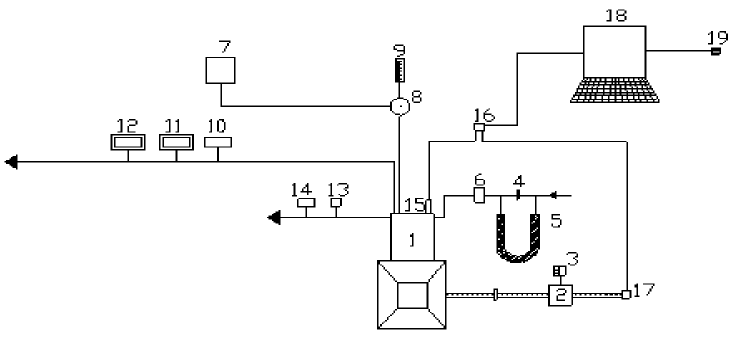


Fig. 3. Photographic view of air gap insulated liner.



1. Engine, 2.Electical dynamometer, 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. 4. Schematic diagram of experimental set–up.

Figures 2 and 3 represent the photographic views of the air gap insulated piston and air gap insulated liner.
The experimental setup used for the investigations on the engine with medium grade LHR combustion chamber with crude waste fried vegetable oil (WFVO) is shown in Fig. 4.
Figure 5 represents the photographic view of the experimental setup used for investigations on the engine with LHR combustion chamber with waste fried vegetable oil. Specifications of the experimental engine are shown in Table 3. The engine tests were carried out with a single cylinder, four strokes, naturally aspirated, compression ignition engine. The engine was operated at the rated constant speed of 1500 rev/min. The combustion chamber consisted of a direct injection type with no special arrangement for swirling air motion. It was connected to an electric dynamometer for measuring its brake power. The dynamometer was loaded by a loading rheostat. Brake power at different percentages of load was calculated by knowing the values of the output signals (voltmeter reading and ammeter reading) of the dynamometer and speed of the engine. Accuracy of the engine load was ±0.2kW andits speed was measured by a digital tachometer with the accuracy of ±1%. The fuel consumption was registered using a fuel measuring device (Burette and stop watch) and then mass flow rate of the fuel was determined by knowing the fuel density that was determined by a hydrometer. Percentage error obtained through the measurement of the fuel flow rate, assuming laminar film in the burette, was within the limit. Accuracy of the determination of the obtained brake thermal efficiency was ±2%. Waste fried vegetable oil was injected into the engine through the conventional injection system. Fuel tank and glass burette arrangements were made for measuring vegetable oil consumption using a stop watch.
Air consumption of the engine was obtained by an air box, orifice flow meter, and U–tube water manometer assembly. By means of orifice flow meter and U-tube water manometer, air discharge was calculated, by which mass flow rate of air was calculated. Percentage error, which was obtained by measuring the difference of water levels in U–tube water manometer while assuming a laminar film in the manometer, was within the limit. Air
box with diaphragm was used to damp out the pulsations produced by the engine in order to ensure a steady flow of air through the intake manifold. The naturally aspirated engine was provided with a water-cooling system in which the inlet temperature of water was maintained at 80^o C by adjusting the water flow rate. The water flow rate was measured by means of an analogue water flow meter with the measurement accuracy of ±1%. Engine oil was provided by a pressure feed system. No temperature control was incorporated for measuring temperature of the lube oil. Copper shims of suitable size were provided in between the pump body and engine frame to vary the injection timing. Then, its effect on the performance of the engine was studied along with the change of injector opening pressure from 190 to 270 bar (in steps of 40 bar) using a nozzle testing device. The maximum injector opening pressure was restricted to 270 bars due to practical difficulties. Exhaust gas temperature was measured by employing the iron and iron-constantan thermocouples connected to a temperature indicator. Accuracy of the analogue exhaust gas indicator was ±1%.

Fig. 5. Photographic view of the experimental set–up.

Table 3. Specifications of the tested 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°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

2. 1. Measuring exhaust emissions

Exhaust emissions of particulate matter and NO_x were recorded by AVL (a company trade name) particulate matter analyzer and Netel Chromatograph (a company trade name) NO_x analyzer at full load operation of the engine. Specifications of the analyzers are given in Table 4. Accuracy of these analyzers was ±1%.

2. 2. Determining combustion characteristics

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 a 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), and maximum rate of pressure rise (MRPR) from the signals of pressure and crank angle at the full load operation of the engine. Accuracy of the instrumentation was±1%.

Table 4. Specifications of analyzers.

Name of the analyzer	Measuring Range	Precision	Resolution	Accuracy of Measurement
AVL Particulate Matter Analyzer	0-100 HSU	1 HSU	1 HSU	±1%
Netel Chromatograph NO_x analyzer	0-2000 ppm	1 ppm	1 ppm	±1%

Table 5. Properties of test fuels.

Test Fuel	Viscosity at 25°C (Centi-poise)	Specific gravity at 25° C	Cetane number	Low Calorific value (kJ/kg)
Diesel	12.5	0.84	55	42000
Crude waste fried vegetable oil	80	0.90	48	36000
Standard	ASTM D 445	ASTM D 4809	ASTM D 613	ASTM D 4809

2. 3. Properties of tested fuels

Properties of the tested fuels of diesel and crude vegetable oil used in this work are presented in Table 5. The low heating value of crude vegetable oil was approximately 14% less than that of diesel fuel. However, specific gravity of crude vegetable oil was approximately 7% higher than that of diesel fuel.

2. 4. Operating conditions

Tested fuels used in the experimentation were diesel and crude waste fried vegetable oil. Different injector opening pressures which were used in this experiment were 190, 230, and 270 bar. Various injection timings of this investigation were 27–34° bTDC. Various combustion chambers used in the experiment included conventional combustion chamber and medium grade LHR combustion chamber. The engine was started with diesel fuel and allowed to have a warm up for about 15 min. Each test was repeated ten times to ensure data reproducibility according to the procedure adopted in error analysis (minimum number of trials must not be less than 10). Results were tabulated and comparative studies of Performance parameters exhaust emissions, and combustion characteristics were reported in different operating conditions of the compression ignition engine. Waste fried vegetable oil was heated to some temperature (preheated temperature, 90° C) where its viscosity matched that of diesel fuel at room temperature.

Definitions of the applied values

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

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

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

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

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

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

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

$$a = \frac{\pi \times d^2}{4} \tag{8}$$

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

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

Optimum injection timing: It is the injection timing at which maximum thermal efficiency of the engine is obtained at all loads.

2. 5. Methodology adopted with crude vegetable oil operation

Mass of vegetable oil at various loads was determined at full load operation using Eq. (1) by knowing the fuel density and time taken for 10 cc of fuel flow into the engine measured by a stop watch. Brake power at different loads of the engine was determined using Eq. (2) by knowing the output signals of an electric dynamometer, current, and voltage. Brake thermal efficiency of the engine was calculated using Eq. (3) by knowing brake power, mass of consumed fuel, and calorific value of the tested fuel. Brake specific energy consumption at full load operation was determined using Eq. (4) by knowing brake thermal efficiency. Brake mean effective pressure of the engine was determined by Eq. (5) through knowing engine cylinder dimensions and speed. Coolant load was calculated using Eq. (6) by knowing coolant flow rate, inlet and outlet temperatures of coolant, and specific heat of coolant. Volumetric efficiency was determined using Eqs. (7- 10).

3. Performance parameters

Curves in Fig. 6 indicate that brake thermal efficiency increased up to 80% of the full load operation due to increase in fuel efficiency. Beyond this load, it decreased due to increase of friction power and decrease of air–fuel ratios with the tested fuels at the recommended injection timing. Conventional engine with vegetable oil showed performance deterioration for the entire load range when compared with the pure diesel operation in CE, which was due to high viscosity and low calorific value of the vegetable oil. In addition, less air entrainment by the fuel spay suggested that the fuel spray penetration might increase and result in the entry of more fuel into the combustion chamber walls. Furthermore, droplet mean diameters (expressed as Sauter mean) were larger for vegetable oil, which led to reduced rate of heat release compared with diesel fuel. This issue also contributed to higher ignition (chemical) delay of the vegetable oil due to lower cetane number. BTE increased with the advanced injection timing with CE with the vegetable oil at all loads when compared with CE at the recommended injection timing and pressure. This issue was owing to the initiation of combustion at an earlier period and efficient combustion with increased oxygen entrainment in fuel spray and peak pressures which resulted in higher BTE. BTE increased at all loads when the injection timing was advanced to 32^{o} bTDC in the conventional engine at the normal temperature of vegetable oil. Increased BTE at optimum injection timing over the recommended injection timing with vegetable oil with CE could be attributed to its longer ignition delay and combustion duration. BTE increased at all loads when the injection timing was advanced to 32^{o} bTDC in CE at the preheated temperature of WFVO .Performance was further improved with CE with the preheated vegetable oil for the entire load range when compared with the normal vegetable oil. Preheating of the vegetable oil reduced viscosity, which improved the spray characteristics of the oil and reduced the impingement of the fuel spray on combustion chamber walls and caused efficient combustion and thus improved BTE.

Figure 7 indicates that LHR version of the combustion chamber showed improvement in the performance for the entire load range compared with CE with pure diesel operation. High cylinder temperatures helped in better evaporation and faster combustion of the fuel injected into the combustion chamber. Reduction of ignition delay of the vegetable oil in the hot environment of the LHR engine improved heat release rates and efficient energy utilization.

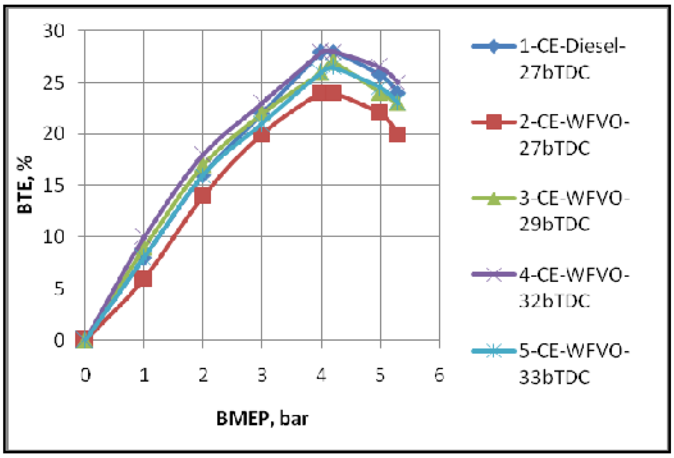


Fig. 6. Variation of brake thermal efficiency (BTE) with brake mean effective pressure (BMEP) in conventional engine (CE) at different injection timings with waste fried vegetable oil (WFVO) operation.

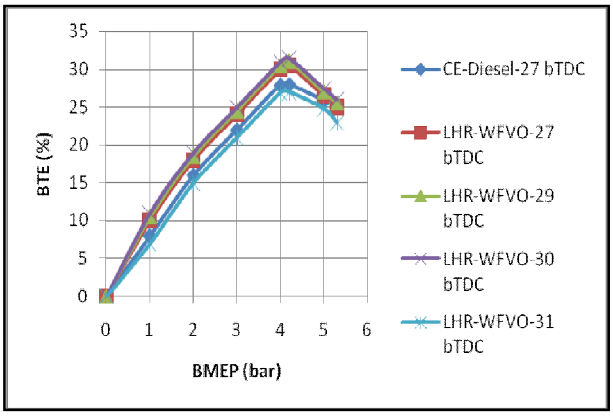


Fig. 7. Variation of brake thermal efficiency (BTE) with brake means effective pressure (BMEP) in engine with LHR combustion chamber at different injection timings with WFVO**.**

Preheating of vegetable oil further improved performance in the engine with LHR combustion chamber. The optimum injection timing was found to be 30^{o} bTDC with the engine with LHR combustion chamber with normal WFVO. The reason was that the hot combustion chamber of LHR engine reduced ignition delay and combustion duration and hence the optimum injection timing was obtained earlier with the engine with LHR combustion chamber when compared with CE with the vegetable oil operation.

According to Fig. 8, the engine with LHR combustion chamber at optimum injection timing showed higher efficiency than CE at its optimum injection timing at all loads.

This issue was due to the conversion of exhaust enthalpy into useful work of the engine with LHR combustion chamber.

Injector opening pressure was varied from 190 bar to 270 bar to improve the spray characteristics and atomization of the vegetable oil and injection timing was advanced from 27 to 34^{o} bTDC for CE and the engine with LHR combustion chamber.

Data of pure diesel operation in the engine with medium grade LHR combustion chamber were obtained from [15]. The optimum injection timing for the conventional engine was

31°bTDC, while it was 29°bTDC with pure diesel operation. Since cetane number of diesel was higher, the optimum injection timing was also closer to TDC for both versions of the combustion chamber with diesel operation compared with vegetable oil operation.

It can be observed in Table 6 that peak BTE increased with increase in the injector opening pressure of both versions of the combustion chamber in different operating conditions of the vegetable oil, which was due to the reduction of mean diameter of fuel particle with increased injector opening pressure causing effective combustion which increased peak BTE with the tested fuels. Diesel operation in the conventional engine showed higher peak BTE that was due to higher calorific value of diesel fuel compared with vegetable oil. Vegetable oil operation in the engine with LHR combustion chamber showed a higher value of peak BTE compared with diesel operation, because ignition delay with the diesel fuel was reduced at high temperatures. Hence, the engine with LHR combustion chamber was more suitable for vegetable oil operation than diesel operation.

Based on Table 7, brake specific energy consumption (BSEC), which is an effective performance parameter, was used for evaluating the performance of different fuels with different properties. It is defined as energy consumed by an engine in producing unit brake power. Conventional engine with vegetable oil operation increased BSEC at full load operation when compared with diesel fuel because of high viscosity, poor volatility, and reduced heating value of vegetable oil leading to their poor atomization and combustion characteristics. Similar trends were noticed in [6].

BSEC at full load operation decreased with the increase of injector opening pressure and advancing of the injection timing in different operating conditions of the vegetable oil. This point showed higher energy substitution and effective energy utilization of vegetable oil, which could replace 100% diesel fuel. BSEC at full load operation showed a lower value with the engine with LHR combustion chamber with vegetable oil operation than the diesel operation.

According to Table 8, exhaust gas temperature (EGT) decreased with increase in injector opening pressure and injection timing with both versions of the engine, which confirmed that performance increased with the increase of injector opening pressure. Preheating of vegetable oil increased EGT in both versions of the combustion chamber.

Exhaust gas temperatures of the preheated vegetable oil were marginally higher than those of normal biodiesel, which indicated the increase of diffused combustion due to high rate of evaporation and improved mixing between fuel and oxygen. Therefore, as the fuel temperature increased, the ignition delay decreased and the main combustion phase (i.e. diffusion controlled combustion) increased, which in turn raised the temperature of exhaust gases. The exhaust gas temperature decreased with increase in injector opening pressure with the tested fuels, as is evident in the same table.

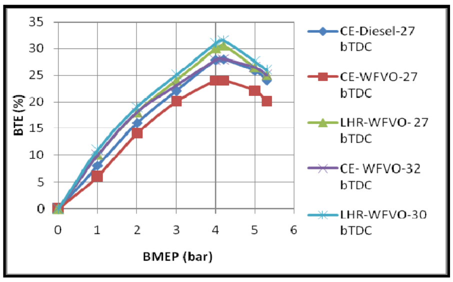


Fig. 8. Variation of brake thermal efficiency (BTE) with brake mean effective pressure (BMEP) with both versions of the combustion chamber at their optimum injection timings at the injector opening pressure of 190 bar.

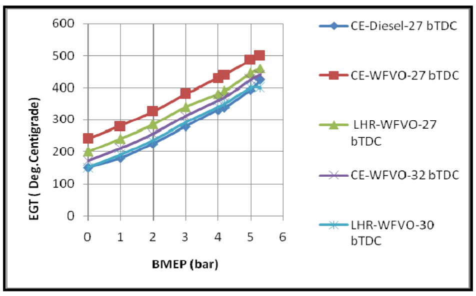


Fig. 9. Variation of exhaust gas temperature (EGT) with brake mean effective pressure (BMEP) in conventional engine (CE) and LHR engine at the recommended injection timing and optimized injection timings with waste fried vegetable oil (WFVO) operation.

Table 6. Data of Peak Brake Thermal Efficiency (BTE).

Injection Timing (^{o}bTDC)	Test Fuel	Peak BTE (%)											
		Conventional Engine (CE)						LHR Engine					
		Injector Opening Pressure (Bar)						Injector Opening Pressure (Bar)					
		190		230		270		190		230		270	
		NT	PT	NT	PT	NT	PT	NT	PT	NT	PT	NT	PT
27	DF	28	--	29	---	30	--	29	--	30	--	30.5	--
	WFVO	24	25	25	26	26	27	30.5	31	31	31.5	31.5	32
29	DF	--	--	--	--	--		29.5	--	30	--	30.5	--
30	WFVO	--	--	--	--	--	-	31.5	32	32	32.5	32.5	33
31	DF	31	--	30.5	--	30	--	--	--	--	--	--	--
32	WFVO	28	29	29	30	30	31	--	--	--	--	--	--

Table7. Data of Brake specific energy consumption (BSEC) at full load operation.

Injection Timing (^{O} bTDC)	Test Fuel	BSEC (kW. h)											
		Conventional Engine						Engine with LHR Combustion chamber					
		Injector Opening Pressure (Bar)						Injector Opening Pressure (Bar)					
		190		230		270		190		230		270	
		NT	PT	NT	PT	NT	PT	NT	PT	NT	PT	NT	PT
27	DF	4.00	--	3.92	--	3.84	--	4.16	---	4.08	--	4.00	--
	WFVO	4.98	4.70	4.78	4.68	4.65	4.60	3.92	3.90	3.88	3.86	3.84	3.82
29	DF	--	-	---	--	--	--	3.92		3.88		3.84	
30	WFVO	--	--	--	--	--	--	3.88	3.86	3.86	3.84	3.82	3.80
31	DF	3.77	--	3.80	--	3.84	--	--	--	--	--	--	--
32	WFVO	3.98	3.94	3.94	3.90	3.90	3.86	-	---	--	--	-	----

Table 8. Data of Exhaust gas temperature (EGT) at full load operation.

Injection timing (^{o} b TDC)	Test Fuel	EGT at full load (^{o}C)											
		Conventional Engine						Engine with LHR Combustion chamber					
		Injector Opening Pressure (Bar)						Injector Opening Pressure (Bar)					
		190		230		270		190		230		270	
		NT	PT	NT	PT	NT	PT	NT	PT	NT	PT	NT	PT
27	DF	425	--	410	---	395	--	475	---	450	--	425	--
	WFVO	500	525	475	500	460	485	460	500	440	480	420	460
29	DF	--	--	--	--	--	--	450	--	425	--	400	--
30	WFVO	--	--	--	--	--	---	400	440	380	420	360	400
31	DF	375	---	400	--	425	---	--	--	--	--	--	-
32	WFVO	440	465	420	445	410	435						--

versions of the combustion chamber, which was due to increased gas temperature with the load. At the recommended injection timing, volumetric efficiency in both versions of the combustion chamber with WFVO operation decreased at all loads when compared with CE with pure diesel operation that was due to increased temperature of incoming charge in the hot environment made by providing insulation, which caused reduction in density and hence air quantity with the engine with LHR combustion chamber. Volumetric efficiency increased marginally in CE and the engine with LHR combustion chamber at optimized injection timings when compared with the recommended injection timings with WFVO.

This issue was due to decrease in the reduction of exhaust gas temperatures, as volumetric efficiency depends on combustion wall temperature due to reduced un–burnt fuel fraction in the cylinder leading to increased volumetric efficiency of CE and reduced gas temperatures with the engine with LHR combustion chamber when injection timing was advanced. Similar trends were observed in [16-18].

According to Table 9, volumetric efficiency marginally increased with the advancing of the injection timing and increasing of injector opening pressure in both versions of the combustion chamber due to better fuel spray characteristics and evaporation at higher injector opening pressure, which caused the marginal increase of VE. This point was also related to the reduction of residual fraction of the fuel with the increase of injector opening pressure. Preheating of the vegetable oil marginally decreased volumetric efficiency in both versions of the combustion chamber because of the increased exhaust gas temperatures and in turn combustion wall temperatures. Similar observations were made in [15].

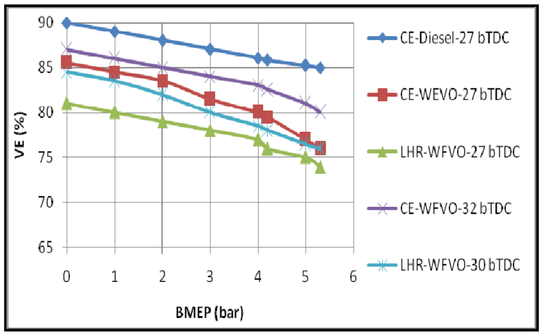


Fig. 10. Variation of volumetric efficiency (VE) with brake mean effective pressure (BMEP) in the conventional engine (CE) and engine with LHR combustion chamber at the recommended injection timing and optimized injection timings with waste fried vegetable oil (WFVO) operation.

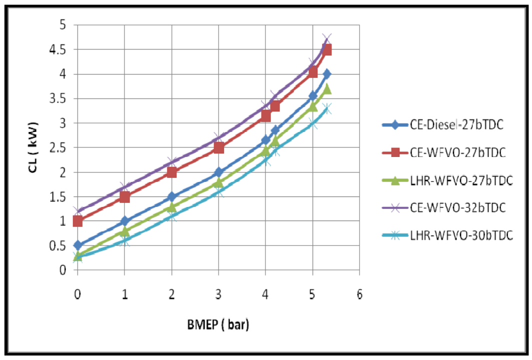


Fig. 11. Variation of coolant load (CL) with brake mean effective pressure (BMEP) in conventional engine (CE) and engine with LHR combustion chamber at the recommended injection timing and optimized injection timings with waste fried vegetable oil (WFVO) operation.

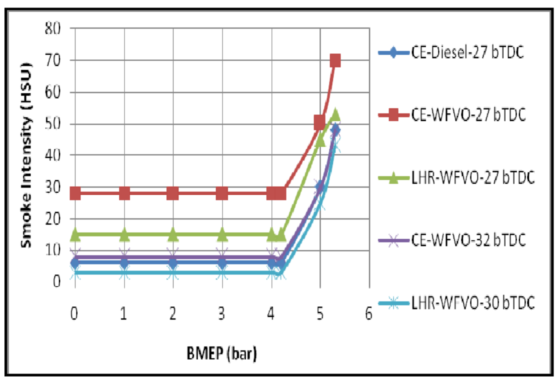


Fig. 12. Variation of smoke intensity in Hartridge smoke unit (HSU) with brake mean effective pressure (BMEP) in the conventional engine (CE) and engine with LHR combustion chamber at the recommended injection timing and optimized injection timings with crude WFVO.

Table 9. Data of Volumetric Efficiency at full load operation.

Injection timing (° bTDC)	Test Fuel	Volumetric efficiency at full load (%)											
		Conventional Engine						Engine with LHR combustion chamber					
		Injector Opening Pressure (Bar)						Injector Opening Pressure (Bar)					
		190		230		270		190		230		270	
		NT	PT	NT	PT	NT	PT	NT	PT	NT	PT	NT	PT
27	DF	85	--	86	--	87	--	79	--	80	--	81	--
	WFVO	76	75	78	77	80	79	74	73	76	75	77	76
29	DF	--	--	--	--	--	--	80		81		82	
30	WFVO	--	--	--	--	--	--	76	75	77	76	79	78
31	DF	89	--	88	--	87	--						
32	WFVO	80	79	82	81	83	82	-	--	-	--	--	-

Figure 11 indicates that the conventional engine with vegetable oil had a higher value of coolant load when compared with pure diesel operation in the same configuration of the engine, which was due to the increase of un–burnt fuel concentration at the combustion chamber walls with the conventional engine leading to produce a high value of coolant load. In the case of conventional engine, un–burnt fuel concentration reduced with effective utilization of energy, released from the combustion, coolant load with the tested fuels increased marginally at full load operation, with increase of gas temperatures, when the injection timing was advanced to the optimum value. However, improvement in the performance of the conventional engine was due to heat addition at higher temperatures and rejection at lower temperatures, while improvement in the efficiency of the engine with LHR combustion chamber was due to recovery from coolant load at their optimum injection timings with the tested fuels.

Coolant load reduced with LHR version of the combustion chamber with vegetable oil operation when compared with CE with pure diesel operation at all loads. Heat output was properly utilized and hence efficiency increased and heat loss to the coolant decreased with effective thermal insulation with the engine with LHR combustion chamber. Similar trends have been observed in other in terms of coolant load [16-18].

Coolant load decreased preheating of the vegetable oil in both versions of the combustion chamber, as observed in Table10.

This issue was due to reduced gas temperatures. From the same table, the coolant load marginally increased in the conventional engine, while it decreased in the engine with LHR combustion chamber with increasing the injector opening pressure with the tested fuels. This issue was related to the fact that, with increase of injector opening pressure with the conventional engine, increased nominal fuel spray velocity resulting in improved fuel–air mixing, with which gas temperatures increased. Reduction of coolant load in the engine with LHR combustion chamber was due to not only the provision of the insulation but also improved fuel spray characteristics and increased oxygen–fuel ratios, which caused decrease of gas temperatures and hence the coolant load.

Table 10. Data of Coolant load at full load operation.

Injection timing (^{o}bTDC)	Test Fuel	Coolant Load (k W)											
		Conventional Engine						Engine with LHR Combustion chamber					
		Injector Opening Pressure (Bar)						Injector Opening Pressure (Bar)					
		190		230		270		190		230		270	
		NT	PT	NT	PT	NT	PT	NT	PT	NT	PT	NT	PT
27	DF	4.0		3.8		3.6		4.5		4.3		4.1	
	WFVO	4.5	4.3	4.7	4.5	4.9	4.7	3.7	3.5	3.5	3.3	3.3	3.1
29	DF	--	--	--	--	--	--	3.6		3.4		3.2	
30	WFVO	--	--	--	--	--	--	3.3	3.1	3.1	2.9	2.9	2.7
31	DF	4.2		4.4		4.6							
32	WFVO	4.7	4.9	4.5	4.7	4.7	4.5						

4. Exhaust emissions

According to Fig. 12, lower levels of smoke was observed with both versions of the engine with the tested fuels up to 80% of the full load operation and drastically increased beyond 80% of the full load operation.

During the first part, the smoke level was more or less constant, as there was always excess air. However, in the higher load range, there was an abrupt rise in smoke levels due to less available oxygen, causing decreased air–fuel ratio which led to incomplete combustion and producing more soot density. Variation of smoke levels with BMEP typically showed an inverted U-shaped behavior due to the pre-dominance of hydrocarbons in their composition at light load and of carbon at high load. It is observed in the figure that the drastic increase of smoke levels was observed at the full load operation in the conventional engine with vegetable oil operation compared with pure diesel operation in the conventional engine. Similar trends have been found in [16-18]. This point was due to the higher value of C/H of WFVO (0.57, (fuel composition $C_{55}H_{96}O_6$) when compared with pure diesel (0.45). The increase of smoke levels was also due to decreased oxygen–fuel ratios and volumetric efficiency with vegetable oil compared with pure diesel operation. Smoke levels were related to the density of the fuel. Since vegetable oil has higher density than diesel fuels, smoke levels were higher with vegetable oil. However, the engine with LHR combustion chamber marginally reduced smoke levels due to efficient combustion and less amount of fuel accumulation on the hot combustion chamber walls of the engine with LHR combustion chamber in different operating conditions of the vegetable oil compared with the CE. Density influences the fuel injection system. Decreasing the fuel density tended to increase spray dispersion and spray penetration. From Table 11, preheating of the vegetable oils reduced smoke levels in both versions of the engine when compared with normal temperature of the vegetable oil, which was due to i) the reduction of density of the vegetable oils, as density is directly proportional to smoke levels, ii) the reduction of the diffusion combustion proportion in CE with the preheated vegetable oil, iii) the reduction of the viscosity of the vegetable oil; using this oil the fuel spray did not impinge on the combustion chamber walls with lower temperatures; rather; it was directed into the combustion. It was noticed that smoke levels decreased with increase of injection timing and injector opening pressure in both versions of the combustion chamber with different operating conditions of the vegetable oil, due to improvement in the fuel spray characteristics at higher injector opening pressure and increased air entrainment at the advanced injection timings, which caused lower smoke levels. Similar trends have been observed in [16-18].

Table 11. Data of Smoke levels in Hartridge Smoke Unit (HSU) at full load operation.

Injection Timing (^{o} bTDC)	Test Fuel	Smoke Levels at Full Load Operation (HSU)											
		Conventional Engine						Engine with LHR Combustion Chamber					
		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	--
	WFVO	70	65	65	60	60	58	53	48	48	43	43	38
29	DF							40		35		30	
30	DF	--	--	--	--	--	--	43	38	38	33	33	28
31	DF	30	---	30	--	35	--	--	--	--	--	--	-
32	DF	48	45	45	40	40	35	--	--	--	---	--	--

This finding was due to lower heat release rate because of high duration of combustion, causing lower gas temperatures with the vegetable oil operation in CE, which reduced NO_x levels. Increase of combustion temperatures with faster combustion and improved heat release rates in the engine with LHR combustion chamber caused higher NO_x levels. NO_x levels increased in CE and decreased in the engine with LHR combustion chamber with advanced injection timings, which was due to the increase of resident time and gas temperatures with CE and decrease of gas temperatures with the engine with LHR combustion chamber at optimum injection timings when compared at the recommended injection timing.

NO_x levels decreased with preheating the vegetable oil in both versions of the combustion chamber, as noticed in Table 12. As fuel temperature increased, there was an improvement in the ignition quality, which caused shortening of ignition delay. A short ignition delay period reduced the peak combustion temperature which suppressed NO_x formation.

From the same table, it can be noted that NO_x levels increased with increase of injector opening pressure in the conventional engine and decreased in the engine with LHR combustion chamber with different operating conditions of vegetable oil that was due to enhanced spray characteristics; thus, fuel air mixture preparation and evaporation process were improved in the conventional engine and combustion was enhanced with the improved oxygen–fuel ratios in the engine with LHR combustion chamber. Similar observations were made in terms of nitrogen oxides levels in [16-18].

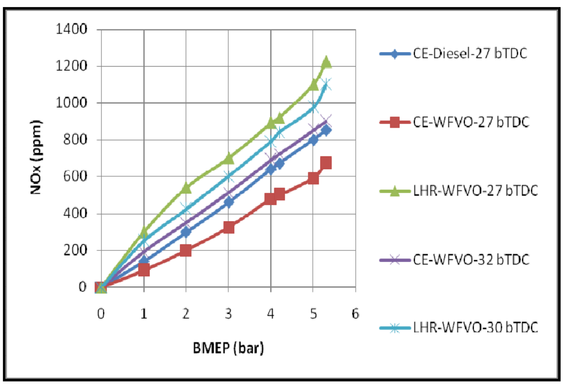


Fig. 13. Variation of nitrogen oxide levels (NO_x levels with brake mean effective pressure (BMEP) in the conventional engine and engine with LHR combustion chamber at the recommended injection timing and optimized injection timings with crude waste fried vegetable oil (WFVO) operation.

5. Combustion characteristics

Using Table 13, it could be noticed that peak pressures were lower in CE, while they were higher in the engine with LHR combustion chamber with vegetable oil operation at the recommended injection timing and pressure, when compared with pure diesel operation in CE. This issue was due to increased ignition delay, as vegetable oils require large duration of combustion. Meanwhile, the piston started making downward motion, thus increasing volume, when combustion took place in CE. The engine with LHR combustion chamber increased the mass-burning rate of the fuel in the hot environment, leading to the production of higher peak pressures. The advantage of using the engine with LHR combustion chamber for vegetable oil is obvious, as it could burn low cetane and high viscous fuels.

Peak pressures increased with the increase of injector opening pressure and advancing of the injection timing in CE with the tested fuels. Higher injector opening 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, a more amount of the fuel was accumulated in the combustion chamber due to increase of ignition delay, as the fuel spray found the air at lower pressure and temperature in the combustion chamber.

When the fuel-air mixture burns, it produces more combustion temperatures and pressures due to increase of the mass of the fuel.

The engine with LHR combustion chamber marginally decreased peak pressure with advanced injection timing and increased injector opening pressure due improved evaporation rate of fuel and instantaneous burning of fuel-air mixture in the hot environment of LHR combustion chamber. Value of TOPP decreased with the advancing of the injection timing and increase of injector opening pressure in both versions of the combustion chamber with different operating conditions of vegetable oils.

TOPP was higher with different operating conditions of vegetable oils in CE when compared with pure diesel operation in CE. This issue was also due to higher ignition delay with the vegetable oil when compared with pure diesel fuel, which once again established the fact by observing lower peak pressures and higher TOPP that CE with vegetable oil operation showed some deterioration in the performance when compared with pure diesel operation in CE.

Preheating of the vegetable oil showed lower TOPP compared with vegetable oil at normal temperature in both versions of the combustion chamber.

Once again, this issue was confirmed by observing lower TOPP and higher PP; performance of both versions of the engine improved with the preheated vegetable oil compared with the normal vegetable oil. Maximum rate of pressure rise (MRPR) followed similar trends with peak pressure in both versions of the combustion chamber; this trend of increased MRPR and decreased TOMRPR indicated better and faster energy substitution and utilization by vegetable oils, which could replace 100% diesel fuel. However, these combustion characters were within the limits; hence, the vegetable oils could be effectively substituted for diesel fuel. Similar trends have been observed in terms of combustion characteristics with the engine with medium grade LHR combustion chamber in [16-18].

6. Conclusions

Vegetable oil operation at 27^{o}bTDC in CE showed deterioration in the performance, while the engine with LHR combustion chamber demonstrated improved performance when compared with pure diesel operation in CE. In comparison with the conventional engine with pure diesel operation, the conventional engine with vegetable oil operation decreased peak BTE by 14% at the recommended injection timing, while decreasing by 10% at optimum injection timing at full load operation–increased brake specific energy consumption by 24% and 6%, increased exhaust gas temperature by 18% and 17%, decreased volumetric efficiency by 10% and 10%, increased coolant load by 12% and 12%,

Table 12. Data of NO_x levels at full load operation.

Injection timing (° b TDC)	Test Fuel	NOx levels (ppm)											
		Conventional Engine						Engine with LHR Combustion chamber					
		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	----	810	----	770	---	1150	--	1100	--	1050	--
	WFVO	675	650	650	625	625	600	1225	1150	1210	1160	1160	1110
29	DF	--	--	--	--	--	--	1100		1050		1000	
30	WFVO	--	--	--	--	--	--	1100	1050	1050	1000	1000	950
31	DF	1100	---	1150	---	1200	---	--	--	--	--	--	--
32	WFVO	900	860	860	820	820	780	--	--	--	--	--	--

Table 13. Data of PP, MRPR and TOPP at full load operation

Injection timing (°bTDC)/ Test fuel	Combustion chamber version	Peak Pressure (PP)(bar)				Maximum Rate of Pressure Rise (MRPR) (Bar/deg)				Time of Occurrence of Peak Pressure Rise (TOPP) (Deg)			
		Injector Opening Pressure (Bar)				Injector Opening Pressure (Bar)				Injector Opening Pressure (Bar)			
		190		270		190		270		190		270	
		NT	PT	NT	PT	NT	PT	NT	PT	NT	PT	NT	PT
27/Diesel	CE	50.4	--	53.5	---	5.4	---	6.0	--	9	-	8	--
	LHR	60.4	--	56.7	--	7.2	--	6.4	--	10	--	9	--
27/ WFVO	CE	46.1	48.9	49.7	51.7	4.0	4.2	4.6	4.8	11	11	11	11
	LHR	59.2	57.2	57.5	55.5	6.8	6.6	6.5	6.3	10	9	9	9
29/Diesel	LHR	58.2	--	54.4	--	6.8	---	6.0	--	10	-	9	-
30/WFVO	LHR	57.2	55.5	55.3	53.4	6.4	6.0	6.0	5.6	10	9	9	9
31/Diesel	CE	62.4	--	64.5	--	6.2	--	6.8	---	8	--	8	--
32/WFVO	CE	51.3	52.6	55.5	54.6	4.8	5.0	5.2	5.4	9	9	9	9

increased smoke levels by 46% and 60%, decreased NOx levels by 20% and 18%, decreased peak pressure by 8% and 18%, and decreased maximum rate of pressure rise by 26% and 22% . When compared with the engine with LHR combustion chamber with pure diesel operation, the engine with LHR combustion chamber with vegetable oil operation increased peak BTE by 5% at the recommended injection timing, while increasing by 7% at optimum injection timing at full load operation decreased brake specific energy consumption by 15% and 1%, decreased exhaust gas temperature by 3% and 11%, increased volumetric efficiency by 6% and 5%, decreased coolant load by 18% and comparable, smoke levels were comparable, increased NOx levels by 7% and comparable , peak pressures comparable and maximum rate of pressure rise were comparable.

- Preheating the vegetable oil improved performance and decreased the pollution levels when compared with normal vegetable oil in both versions of the combustion chamber. Improvement in the performance was observed with the advancing of the injection timing and increase of injector opening pressure with the vegetable oil operation in both versions of the combustion chamber.

6. 1. Research findings

Investigations were carried out on the engine with medium grade LHR combustion chamber with different operating conditions of crude waste fried vegetable oil with varied injection timing and injector opening pressure.

6. 2. Social significance

Waste fried vegetable oil can be successfully used as a fuel for compression ignition engine. Instead of throwing waste away, economic burden on the government can be reduced if vegetable oil is substituted for diesel in diesel engines. Health hazards for people who consume waste fried vegetable oil can be also reduced.

6. 3. Future scope of work

Although the engine with LHR combustion chamber increased thermal efficiency with vegetable oil, it drastically increased nitrogen oxide levels compared with the conventional engine with diesel operation. NO_x levels can be reduced by an appropriate catalytic converter which employs zeolite as the catalyst [21].

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