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Studies on Performance and Exhaust Emissions in DI Diesel Engine with Air Gap Insulated Piston and Air Gap Insulated Liner with Waste Fried Vegetable Oil

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Abstract: *Investigations were carried out to evaluate the performance of a engine with low heat rejection (LHR) combustion chamber with air gap insulated piston and air gap insulated liner with waste fried vegetable oil with varied injector opening pressure. Performance parameters (brake thermal efficiency, exhaust gas temperature, coolant load, volumetric efficiency, sound levels) and exhaust emissions (particulate emissions and oxides of nitrogen) were determined at various values of brake mean effective pressure (BMEP of the engine. Particulate emissions were measured by AVL Smoke meter, while* NO_x *by Netel Chromatograph NOx analyzer. Engine with LHR combustion chamber improved at all loads and it increased further with an increase of injector opening pressure.*

Keywords: Conventional engine, LHR combustion chamber, Performance, Exhaust emissions.

1. INTRODUCTION

In the scenario of i) increase of vehicle population at an alarming rate due to advancement of civilization, ii) use of diesel fuel in not only transport sector but also in agriculture sector leading to fast depletion of diesel fuels and iii) increase of fuel prices in International market leading to burden on economic sector of Govt. of India, the search for alternative fuels has become pertinent for the engine manufacturers, users and researchers involved in the combustion research. [1]

The nation should pay gratitude towards Dr. Diesel [2], for his remarkable invention of diesel engine as the civilization of any country is linked with number of vehicles used by its public.

Vegetable oils and alcohols are promising substitute fuels for diesel fuels as they are renewable in nature. Alcohols (ethanol and methanol) have good volatility but low cetane number. That too, most of alcohol produced in India is diverted for Petro–chemical industries. On the other hand vegetable oils have cetane number comparable to diesel fuel. Hence modification of the engine is not necessary if they are to be used as fuels in diesel engine. Several researchers 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 poly unsaturated character. [3–7]. The disadvantages associated with use of vegetable oils in diesel engine call for engine with LHR 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 (LHR-1), medium grade (LHR-2) and high grade (LHR-3) combustion chambers depending on degree of insulations.

Investigations were carried out by various researchers on LHR-1 combustion chambers- ceramic coated engines with pure diesel operation [8–10]. It was reported from their investigations that brake specific fuel consumption (BSFC) improved in the range 5-9% and smoke levels decreased with ceramic coated combustion chamber. Investigations were extended to vegetable oils with engine with ceramic coated combustion chamber. [11–13]. Performance marginally improved and smoke levels decreased by 10% with ceramic coated combustion chamber with vegetable oil operation.

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Investigations were carried out with LHR-2 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 brake specific fuel consumption improved by 5%[14]. Investigations were carried out with engine with LHR-2 combustion chamber with vegetable oil operation and reported that vegetable oil improved performance, decreased smoke levels by 15% and increased nitrogen oxide levels. [15–17]
By controlling the injector opening pressure and the injection rate, the spray cone angle is found to depend on injector opening pressure. Few investigators reported that injector opening pressure has a significance effect on the performance and formation of pollutants inside the direct injection diesel engine.[5–7]. Preheating of vegetable oils reduce viscosity and improved the spray pattern of the fuel and improve performance and reduce pollution levels of smoke and NOx levels.[5–7]
The present paper attempted to evaluate the performance of medium grade LHR combustion chamber, which consisted of air gap insulated piston and air gap insulated liner. This medium grade LHR combustion chamber was fuelled with waste fried vegetable oil collected from hotels and restaurants, with varied injector opening pressure. Comparative performance studies were made with medium grade LHR combustion chamber with conventional engine CE with waste fried vegetable oil operation.

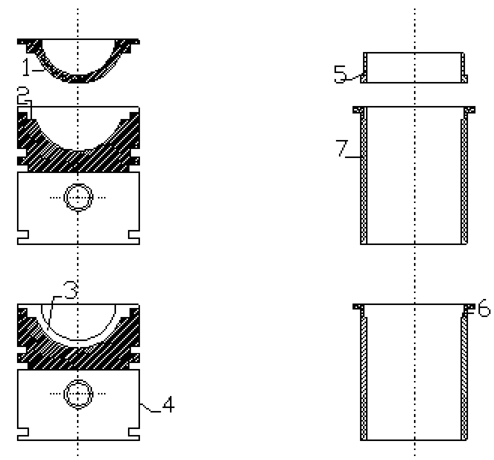
2. MATERIAL AND METHOD

The physical-chemical properties of the diesel fuel and palm oil based waste fried vegetable oil (WFVO) are presented in Table-1.

Table 1. Properties of Diesel Fuel

Test Fuel	Viscosity at 25^{o}C (Centi–poise)	Specific gravity at 25^o C
Diesel	12.5	0.84
WFVO	80	0.90

Engine with LHR combustion chamber (figure.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 3mm air gap in between the crown and the body of the piston. The optimum thickness of air gap in the air gap piston was found to be 3-mm for improved performance of the engine with diesel as fuel [14]. The height of the piston was maintained such that compression ratio was not altered. A superni–90 insert was screwed to the top portion of the liner in such a manner that an air gap of 3-mm was maintained between the insert and the liner body. At 500^{o}C the thermal conductivity of superni-90 and air are 20.92 and 0.057 W/m–K respectively



1 Superni crown with threads 2 Superni gasket 3 Air gap in piston, 4 Body of piston, 5 Superni insert with threads 6 Air gap in liner and 7 Body of liner

Figure 1: Assembly Details of Air Gap Insulated Piston and Air Gap Insulated Liner

Schematic diagram of experimental setup used for the investigations on compression ignition diesel engine with diesel operation is shown in figure 2. The test fuel used in the experimentation was pure diesel. 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^o C by adjusting the water flow rate.

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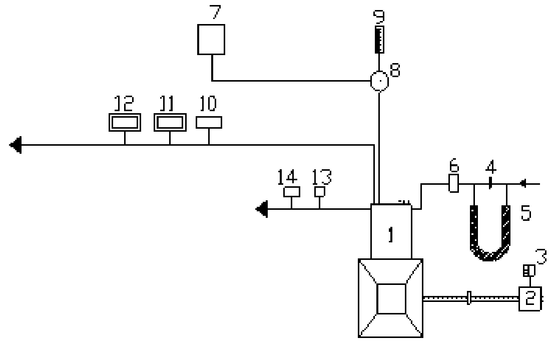

1.Engine, 2.Electical Dynamo meter, 3.Load Box, 4.Orifice meter, 5.U-tube water manometer, 6.Air box, 7.Fuel tank, 8, Three way valve, 9.Burette, 10. Exhaust gas temperature indicator, 11.AVL Smoke meter, 12.Netel Chromatograph NOx Analyzer, 13.Outlet jacket water temperature indicator and 14. Outlet-jacket water flow meter

Figure 2: Schematic diagram of experimental set–up

Engine oil was provided with a pressure feed system. No temperature control was incorporated, for measuring the lube oil temperature. Injector opening pressure was changed from 190 bar to 270 bar (in steps of 40 bar) using nozzle testing device.

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
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

The maximum injector opening pressure was restricted to 270 bar due to practical difficulties involved. Exhaust gas temperature and coolant water outlet temperatures were measured with thermocouples made of iron and iron-constantan connected to temperature indicators. .

Operating conditions:

Test fuel used in the experimentation was diesel and waste fried vegetable oil. Different injector opening pressures attempted in this experiment were 190, 230 and 270 bar. The various combustion chambers used in experiment were conventional combustion chamber and medium grade LHR combustion chamber with air gap insulated piston and air gap insulated liner.

The engine was started with diesel fuel and allowed to have a warm up for about 15 minutes. Each test was repeated ten times to ensure the reproducibility of data according to the procedure adopted in error analysis. (Minimum number of trials must be not less than ten).

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 in m^2
BP=brake power of the engine, kW
C_d=coefficient of discharge, 0.65
Cp=specific heat of water in kJ/kg K
D=bore of the cylinder, 80 mm
D=diameter of the orifice flow meter, 20 mm
DI=diesel injection
DF=Diesel fuel
HSU=Hartridge smoke unit
I=ammeter reading, ampere
H=difference of water level in U–tube water manometer in cm of water column
K=number of cylinders, 01
L=stroke of the engine, 110 mm
m_a=mass of air inducted in engine, kg/h
m_f=mass of fuel, kg/h
m_w=mass flow rate of coolant, g/s
n=power cycles per minute, N/2,
N=speed of the engine, 1500 rpm
P_a=atmosphere pressure in mm of mercury
R=gas constant for air, 287 J/kg K
T=time taken for collecting 10 cc of fuel, second
T_a=room temperature, ^{o}C
T_I=inlet temperature of water, ^{o}C
T_o=outlet temperature of water, ^{o}C
V=voltmeter reading, volt
V_s=stroke volume, m^3
VE=Volumetric efficiency, %

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Definitions of used values:

$m_f = \frac{10 \times \rho_d \times 3600}{t \times 1000}$ equation (1)

$BP = \frac{V \times I}{\eta_d \times 1000}$ equation (2)

$BTE = \frac{BP \times 3600}{m_f \times CV}$ equation (3)

$BP = \frac{BMEP \times 10^5 \times L \times A \times n \times k}{60000}$ ---equation (4)

$CL = m_w \times c_p \times (T_o - T_i)$ equation (5)

$m_a = C_d \times a \times \sqrt{2 \times 10 \times g \times h \times \rho_a} \times 3600$ (6)

$a = \frac{\pi \times d^2}{4}$ equation (7)

$\eta_v = \frac{m_a \times 2}{60 \times \rho_a \times N \times V_s}$ equation (8)

$\rho_a = \frac{P_a \times 10^5}{750 \times R \times T_a}$ equation (10)

3. RESULTS AND DISCUSSION

Brake thermal efficiency was calculated from equations (1), (2) and (3). Brake mean effective pressure was calculated from equation (4). Curves from figure 3 indicate that BTE increased up to 80% of the full load operation with test fuels due to increase in fuel efficiency and beyond this load it decreased due to increase of friction power, decrease of volumetric efficiency and air fuel ratio. At recommended injection timing, CE at with vegetable oil showed the deterioration in the performance at all loads when compared with the pure diesel operation on CE. This was due to high viscosity and low calorific value of the vegetable oil In addition, less air entrainment by the fuel spray suggested that the fuel spray penetration might increase and resulted in more fuel reaching the combustion chamber walls.

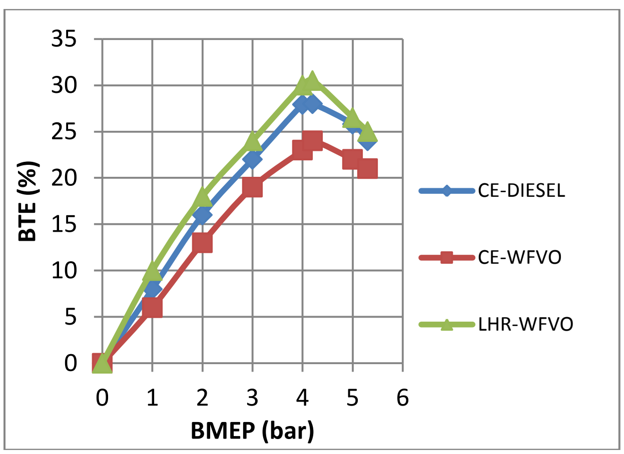

Figure 3: Variation of brake thermal efficiency (BMEP) with brake mean effective pressure (BMEP) with pure diesel operation in conventional engine and engine with air gap insulated piston and air gap insulated liner (LHR) at an injection timing of 27^o bTDC and injector opening pressure of 190 bar.

Furthermore droplet mean diameters (expressed as Sauter mean) were larger for vegetable oil leading to reduce the rate of heat release as compared with diesel fuel. This also, contributed the higher ignition (chemical) delay of the vegetable oil due to lower cetane number. Engine with LHR combustion chamber showed improved thermal efficiency at all loads when compared with pure diesel operation on CE. High cylinder temperatures helped in better evaporation and faster combustion of the fuel injected into the combustion chamber with engine with LHR 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.

From fig. 4, it is noticed that CE with WFVO at the recommended injection timing recorded higher EGT at all loads compared with CE with pure diesel operation. Lower heat release rates and retarded heat release associated with high specific energy consumption caused increase in EGT in CE. Ignition delay in the CE with different operating conditions of vegetable oil increased the duration of the burning phase. Though the calorific value (or heat of combustion) of fossil diesel is more than that of vegetable oil, density of vegetable oil is higher therefore greater amount of heat was released in the combustion chamber leading to higher exhaust gas temperature with CE with vegetable oil operation. Engine with LHR combustion chamber recorded lower value of EGT when compared with CE with vegetable oil operation. This was due to reduction of ignition delay in the hot environment with the provision of the insulation in the engine with LHR combustion chamber, which caused the gases expanded in the cylinder giving higher work output and lower heat rejection. This showed that the performance was improved with LHR engine over CE with vegetable oil operation.

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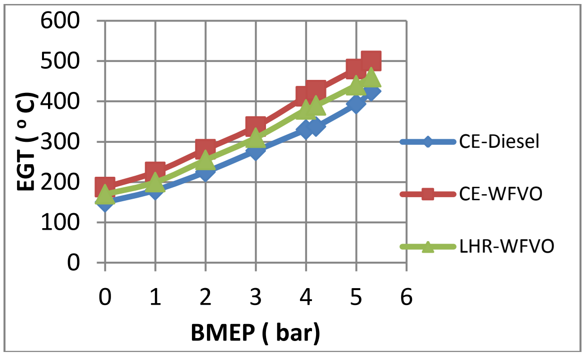

Figure 4: Variation of exhaust gas temperature (EGT) with brake mean effective pressure(BMEP)with pure diesel operation in conventional engine and engine with air gap insulated piston and air gap insulated liner (LHR) at an injection timing of 27^obTDC and injector opening pressure of 190 bar.

Coolant load was calculated from equation (5). Curves in figure.5, indicates that coolant load increased with increase of BMEP for both versions of the combustion chamber this was due to increase of combustion temperatures due to consumption of fuel. Coolant load was marginally higher with CE with vegetable oil operation when compared with diesel operation on CE. This was due to un-burnt fuel concentration at the walls of combustion chamber. The air-fuel ratio got reduced to a reasonably low value at this load confirming the above trend. However, when heat rejection calculations of coolant were made, the heat lost to lubricant should also be considered. As in the present investigations the lubricant heat loss was not considered, this aspect was not depicted in coolant load calculations. Coolant load reduced with engine with LHR 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 coolant decreased with effective thermal insulation with engine with LHR combustion chamber.

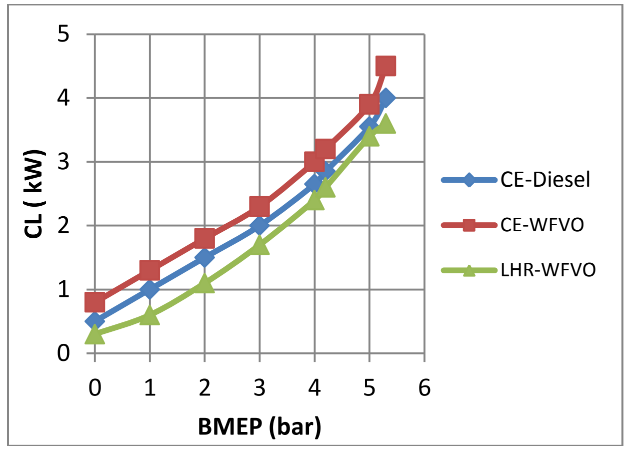

Figure 5: Variation of coolant load (CL) with brake mean effective pressure(BMEP) with pure diesel operation in conventional engine and engine with air gap insulated piston and air gap insulated liner (LHR) at an injection timing of 27^obTDC and injector opening pressure of 190 bar.

Volumetric efficiency was calculated from equations (6–9). From figure. 6, it is evident that volumetric efficiency (VE) decreased with an increase of BMEP in engine with both versions of the combustion chamber. This was due to increase of gas temperature with the load. At the recommended injection timing, volumetric efficiency in the both versions of the combustion chamber with WFVO operation decreased at all loads when compared with CE with pure diesel operation. This was due increase of temperature of incoming charge in the hot environment created with the provision of insulation, causing reduction in the density and hence the quantity of air with engine with LHR combustion chamber. Volumetric efficiency depends on temperature of combustion chamber walls which in turn depends on exhaust gas temperature. As exhaust gas temperatures were higher with CE with vegetable oil operation, volumetric efficiency was observe to be lower with CE.

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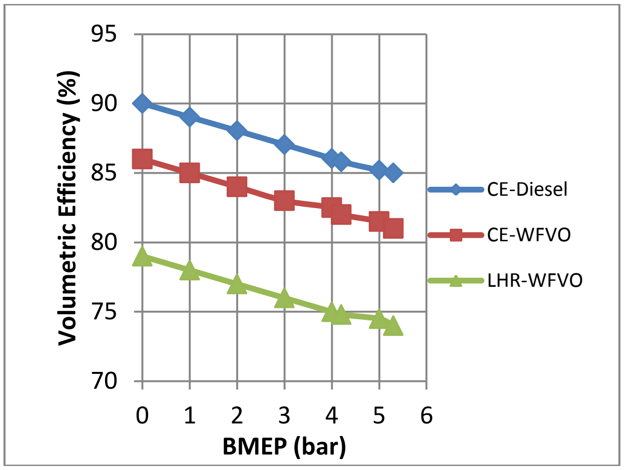

Figure 6: Variation of volumetric efficiency (VE) with brake mean effective pressure(BMEP) with pure diesel operation in conventional engine and engine with air gap insulated piston and air gap insulated liner (LHR) at an injection timing of 27° bTDC and injector opening pressure of 190 bar.

Pollution Levels

Figure.7 indicates that particulate emissions increased from no load to full load in both versions of the combustion chamber. During the first part, the particulate emissions were more or less constant, as there was always excess air present. However, in the higher load range there was an abrupt rise in smoke levels due to less available oxygen, causing the decrease of air-fuel ratio, leading to incomplete combustion, producing higher particulate emissions. The variation of particulate emissions with the BMEP typically showed an inverted L behavior due to the pre-dominance of hydrocarbons in their composition at light load and of carbon at high load.

It is observed from figure, drastic increase of particulate emissions was observed at the full load operation in CE at normal temperature of the vegetable oil, compared with pure diesel operation on CE. This was due to the higher value of the ratio of C/H of crude vegetable oil (0.6) when compared with pure diesel (0.45). The increase of particulate emissions was also due to decrease of air-fuel ratios and VE with vegetable oil compared with pure diesel operation. Particulate emissions were related to the density of the fuel. Since vegetable oil has higher density compared to diesel fuels, particulate emissions were higher with vegetable oil. However, engine with LHR combustion chamber marginally reduced particulate emissions due to efficient combustion and less amount of fuel accumulation on the hot combustion chamber walls of the engine with LHR combustion chamber at different operating conditions of the vegetable oil compared with the CE.

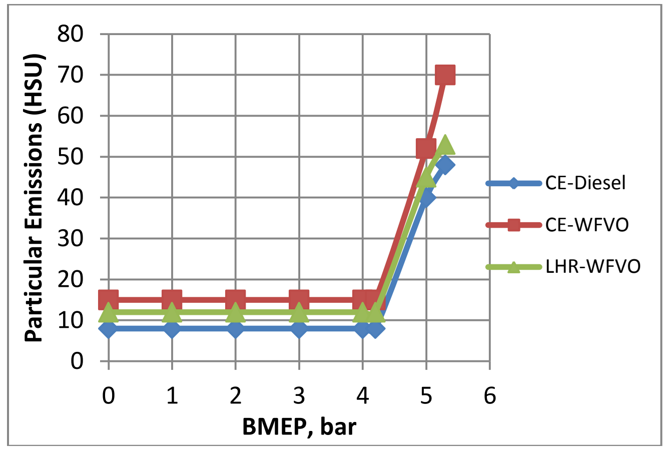

Figure 7: Variation of particulate emissions with brake mean effective pressure(BMEP) with pure diesel operation in conventional engine and engine with air gap insulated piston and air gap insulated liner (LHR) at an injection timing of 27° bTDC and injector opening pressure of 190 bar.

The temperature and availability of oxygen are the reasons for the formation of NO_x. For both versions of the combustion chamber, NO_x concentrations raised steadily as the fuel/air ratio increased (Figure.8) with increasing BP/BMEP, at constant injection timing. At part load, NO_x concentrations were less in both versions of the engine. This was due to the availability of excess oxygen. At remaining loads, NO_x concentrations steadily increased with the load in both versions of the combustion chamber. This was because, local NO_x concentrations raised from the residual gas value following the start of combustion, to a peak at the point where the local burned gas equivalence ratio changed from lean to rich. At full load, with higher peak pressures, and hence temperatures, and larger regions of close-to-stoichiometric burned gas, NO_x levels increased in both versions of the engine. Though amount of fuel injected decreased proportionally as the overall equivalence ratio was decreased, much of the fuel still burns close to stoichiometric. Thus NOx emissions should be roughly proportional to the mass of fuel injected (provided burned gas pressures and temperature do not change greatly). From fig.8, it is evident that NOx levels were lower in CE while they were higher in engine with LHR combustion chamber with vegetable oil at all loads when compared with diesel operation.

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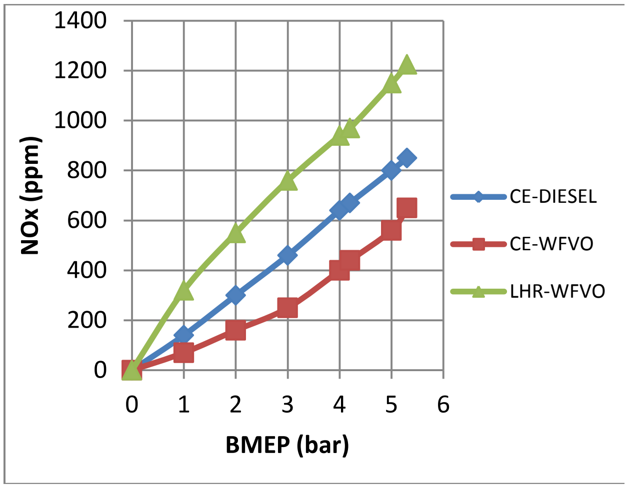

Figure 8: Variation of nitrogen oxide (NO_x) levels with brake mean effective pressure(BMEP) with pure diesel operation in conventional engine and engine with air gap insulated piston and air gap insulated liner (LHR) at an injection timing of 27°bTDC and injector opening pressure of 190 bar.

This was due to lower heat release rate because of high duration of combustion causing lower gas temperatures with the vegetable oil operation on CE, which reduced NOx levels. Increase of combustion temperatures with the faster combustion and improved heat release rates in engine with LHR combustion chamber caused higher NOx levels.

Effect of Injector Opening Pressure

Table.3 shows variation of performance parameters and pollution levels with injector opening pressure for conventional engine and engine with LHR combustion chamber with pure diesel operation.

Peak BTE increased with increase of injector opening pressure marginally in both versions of the combustion chamber with test fuels, at the recommended injection timing. This was due to improved fuel spay characteristics of the fuel at the increased injection pressure. Poor performance at lower injection pressures indicated slow mixing probably because of insufficient spray penetration with consequent slow mixing during diffusion burning.

Brake specific energy consumption (BSEC) at full load operation fuel consumption decreased with increase of injector opening pressure with both versions of the combustion chamber. This was due to increase of fuel spray characteristics. Poor performance at lower injection pressures indicated slow mixing probably because of insufficient spray penetration with consequent slow mixing during diffusion BSEC was higher with CE with vegetable oil operation as vegetable oil has high viscosity and low calorific value and hence more energy was required for combustion of the fuel. BSEC was lower with preheated vegetable oil than normal vegetable oil due to improved spray characteristic. BSEC was lower with engine with LHR combustion chamber with vegetable oil operation. This showed that vegetable oil was more suitable for engine with LHR combustion chamber.

Exhaust gas temperature (EGT) decreased marginally with increase of injector opening pressure with diesel operation. This was due to improved air fuel ratios with improved spray characteristics. EGT was higher with preheated vegetable oil than normal vegetable oil in both versions of the combustion chamber, which indicates the increase of diffused combustion due to high rate of evaporation and improved mixing between vegetable oil and oxygen. 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.

Coolant load increased with increase of injector opening pressure with conventional engine, while it decreased with engine with LHR combustion chamber. This was due to decrease of gas temperatures with improved air fuel ratios with engine with LHR combustion chamber, while gas temperatures increased in conventional engine due to efficient combustion with improved spray characteristics of the fuel. Coolant load decreased with preheated condition of vegetable oil in comparison with normal vegetable oil. This was because of reduction of gas temperatures and improved spray characteristics

Volumetric efficiency increased marginally with increase of injector opening pressure in conventional engine and engine with LHR combustion chamber. This was due to improved spray characteristics leading to reduce gas temperatures. This was also reduction of exhaust gas temperatures with increase of injector opening pressure as volumetric efficiency depends on combustion chamber wall temperature.

Preheating of the waste fried vegetable oil marginally improved volumetric efficiency, because of reduction of un-burnt fuel concentration with efficient combustion, when compared with the normal temperature of the test fuels.

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Table 3: Variation of Performance Parameters & Pollution levels with Injector Opening Pressure with Vegetable Oil Operation

Parameter /Unit	Fuel	Operating Condition	Conventional Engine			Engine with LHR combustion chamber		
			190	230	270	190	230	270
Peak BTE(%)	DF	--	28	29	30	29	30	30.5
	WFVO	NT	24	25	26	30.5	31	31.5
		PT	25	26	27	31	31.5	32
BSEC (kW.h)	DF	--	4.0	3.92	3.84	4.16	4.08	4.0
	WFVO	NT	4.98	4.78	4.65	3.92	3.88	3.84
		PT	4.70	4.68	4.60	3.90	3.86	3.82
EGT (^{o}C)	DF	---	425	410	395	475	450	425
	WFVO	NT	500	475	460	460	440	420
		PT	525	500	480	420	400	380
Coolant Load (kW)	DF		4.0	4.2	4.4	4.5	4.2	3.8
	WFVO	NT	4.5	4.3	4.1	3.7	3.5	3.3
		PT	4.3	4.1	3.9	3.5	3.3	3.1
Volumetric Efficiency (%)	DF		85	86	87	79	80	81
	WFVO	NT	81	82	83	74	76	77
		PT	82	83	84	75	77	78
Particulate Emissions (HSU)	DF	--	48	38	34	55	50	45
	WFVO	NT	70	65	60	53	48	43
		PT	65	60	58	48	43	38
NOx Levels (ppm)	DF	-	850	900	950	1150	1100	1050
	WFVO	NT	675	650	625	1225	1175	1125
		PT	650	625	600	1175	1125	1075

Data from Table 3 shows a decrease in particulate emissions with increase of injector opening pressure, with different operating conditions of the crude vegetable oil. This was due to improvement in the fuel spray characteristics at higher injector opening pressure and increase of air entrainment, at the advanced injection timings, causing lower smoke levels. The improved spray also leads to better mixing of fuel and air resulting in turn in fast combustion. This will enhance the performance. Preheating of the vegetable oil reduced particulate emissions , when compared with normal temperature of the vegetable oil. This was due to i) the reduction of density of the vegetable oil (s), as density was directly related to particulate emissions, ii) the reduction of the diffusion combustion proportion with the preheated vegetable oil, iii) the reduction of the viscosity of the vegetable oil, with which the fuel spray does not impinge on the combustion chamber walls of lower temperatures rather than it directed into the combustion chamber

As seen from Table.3, NO_x slightly increased with test fuels in CE, while they decreased in engine with LHR combustion chamber as injector opening pressure increased. This was due to increase of gas temperatures with CE, while they decreased in engine with LHR combustion chamber. NO_x levels decreased with preheating of the vegetable oil as noticed from Table.3. The fuel spray properties may be altered due to differences in viscosity and surface tension. 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 vegetable oil

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lead to reduction in NO_x formation. As fuel temperature increased, there was an improvement in the ignition quality, which will cause shortening of ignition period which lowers NOx formation.

4. CONCLUSIONS

In comparison with conventional engine, engine with LHR combustion chamber with vegetable oil operation increased peak BTE by 25%, at full load operation- brake specific energy consumption decreased by 21%, exhaust gas temperature decreased by 8%, volumetric efficiency decreased by 9%, coolant load decreased by 18%, particular emissions decreased by 24% and NOx levels increased by 81%.

4.1 Research Findings

Engine with LHR combustion chamber with air gap insulated piston and air gap insulated liner with waste fried vegetable oil operation improved performance parameters and decreased particulate emissions and increased NOx levels

4.2 Social Significance

Waste fried vegetable oil causes many health disorders particularly lever disorders. Instead of throwing this oil, this oil can be significantly used in diesel engine so as to reduce economic burden but also health hazards.

4.3 Future Scope of Work

Injection timing can also be varied along with change of injector opening pressure. Investigations can be extended to alternate fuels for biodiesel and alcohols. Ethanol or methanol can be inducted during suction stroke and vegetable oil can be injected at the near end of compression stroke to reduce NOx emissions with engine with LHR combustion chamber.

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