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Studies on Direct Injection Diesel Engine with Air Gap Insulated Low Heat Rejection Combustion Chamber with Tyre Oil

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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 in square meter
BP = brake power of the engine, kW
C = number of carbon atoms in fuel composition
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
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 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
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, seconds
T_a = room temperature, degree centigrade
T_I = inlet temperature of water, degree centigrade
T_o = outlet temperature of water, degree centigrade
V = voltmeter reading, volt
V_s = stroke volume, m^3

ABSTRACT: *In the context of fast depletion of fossil fuels, ever increase of fuel prices in International Market, causing economic burden, increase of pollution levels with fossil fuels, the search for alternative fuels has become pertinent. On the other hand, diesel is being consumed in agricultural sector and transport sector due to rapid growth of automotive vehicles. Also, the disposal of used tyres from automotive vehicles becomes inexhaustible. Though many disposal methods are available to dispose the waste automobile tyres, the problem still persists. Crude rubber oil, derived by slow pyrolysis of waste rubber products, can be used as fuel in diesel engine, as its properties are comparable to diesel fuel. However, the disadvantages associated with these oils of high viscosity and low volatility call for engine with low heat rejection (LHR) combustion chamber, with significant characteristics of higher operating temperature, maximum heat release, higher brake thermal efficiency and ability to handle the lower calorific value fuel. Investigations were carried out to evaluate the performance of diesel engine with LHR combustion chamber consisting of air gap insulated piston and air gap insulated liner with different operating conditions (normal temperature and preheated temperature) of tyre oil with varied injector opening pressure. Performance parameters [brake thermal efficiency, exhaust gas temperature, sound levels, coolant load and volumetric efficiency] were determined at various values of brake mean effective pressure (BMEP) of the engine and compared with conventional engine with diesel operation at similar operating conditions. Tyre oil showed deteriorated performance with conventional engine (CE), while LHR combustion chamber improved the performance in comparison with pure diesel operation at similar operating conditions. Engine with LHR combustion chamber with tyre oil operation increased peak brake thermal efficiency by 7%, at full load operation– decreased brake specific energy consumption by 8%, decreased exhaust gas temperature by 5%, decreased coolant load by 20%, decreased sound levels by 5%, decreased volumetric efficiency by 4% and smoke levels were comparable in comparison with pure diesel operation at similar operating conditions.*

Keywords*: Rubber oil, LHR combustion chamber, Fuel performance and smoke levels.*

1. INTRODUCTION

The resources of petroleum as fuel are dwindling day by day and increasing demand of fuels, as well as increasingly stringent regulations, pose a challenge to science and technology [1].

Alcohols (ethanol and methanol) are important substitutes for diesel fuel in diesel engine. Alcohols have good volatility and low C/H ratio. However, they have 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 comparable to diesel fuel. Rudolph Diesel, the inventor of the diesel engine that bears his name, experimented with fuels ranging from powdered coal to peanut oil and hinted that vegetable oil would be the future oil in diesel engine [2]. 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 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 [3–5].

Experiments were conducted on preheated vegetable oils [temperature at which viscosity of the vegetable oils were matched to that of diesel fuel] and it was reported that performance improved marginally with preheated vegetable oils [6–9]. Increased injector opening pressures may also result in efficient combustion in compression ignition engine [3–5].

Disposal of rubber products, generated from worn or damaged automotive tyres and industrial conveyor belts is becoming an environmental challenge in many developing countries due to their non-bio degradability characteristics. Tyre oil was used in diesel engine as blended fuel with diesel [10–12]. Results showed that the brake thermal efficiency of the engine fueled with blends increased with an increase in blend concentration. Exhaust emissions of nitrogen oxides, hydro carbon levels and carbon mono oxide levels and smoke were found to be higher at higher loads due to the high aromatic content and longer ignition delay. Peak pressure increased and ignition delay were longer than diesel fuel. However, It was concluded that it is possible to use tyre pyrolysis oil in diesel engines as an alternate fuel in the future.

However, the disadvantages associated with use of tyre oil of high viscosity and low volatility 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 i) ceramic coated combustion chambers (LHR–1 combustion problem) by providing low thermal conductivity material on cylinder head, crown surface of piston and inner portion of liner and ii) air gap insulated combustion chambers (LHR–2 combustion chamber), where air gap is created in the piston and other components with low-thermal conductivity materials like superni, cast iron and mild steel.

Investigations were carried out by various researchers on engines with low degree LHR combustion chambers-(ceramic coated engines) with pure diesel operation [13–15]. It was reported from their investigations that brake specific fuel consumption (BSFC) improved in the range 5-9% and pollution levels decreased with ceramic coated combustion chamber.

Sound levels determine the phenomena of combustion in engine whether the performance was improving or deteriorating. Studies were made on sound levels with low grade LHR combustion chamber with vegetable oils [3–5]. It was reported from the studies, that performance deteriorated with vegetable oil operation on conventional engine leading to produce high sound levels and improved with LHR combustion chamber causing low sound levels.

Little literature was available on studies of tyre oil with air gap insulated LHR combustion chamber. Hence it was attempted here to evaluate the performance of the engine with air gap insulated LHR combustion chamber with tyre oil at different injector opening pressure and compared with engine with conventional engine. Comparative studies were also made with diesel operation at similar operating conditions.

Property	Units	Diesel	Crude Tyre Oil (CTO)
Cetane Number		55	45
Density	gm/cc	0.84	0.92
Bulk modulus @ 20Mpa	Mpa	1475	1850
Kinematic viscosity @ 40^{o}C	cSt	2.25	9.0
Total Sulfur	Ppm	498	9106
Low calorific value	kJ/kg	42 000	40000
Flash point (Open cup)	^{o}C	66	94
Colour	--	Light yellow	Light orange
Preheated temperature	^{o}C	--	125

Table-1
Physical-Chemical Properties of Test Fuels

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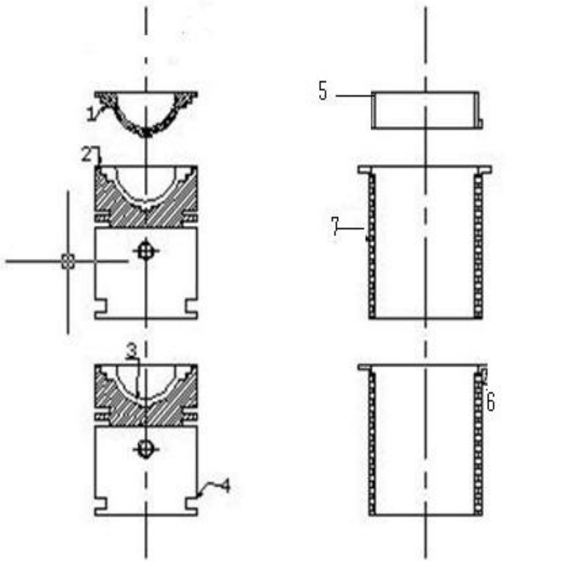
2. MATERIALS AND METHODS

2.1 Manufacturing of Tyre Oil

Crude rubber oil is derived by slow pyrolysis of waste rubber products, where rubber containing material is heated in an oxygen free reactor at temperatures between 300–350^{o} C and the resulting gasses are condensed into liquid crude rubber oil with excellent and consistent fuel properties with a high calorific value, thus maybe used directly as fuel or blended with other fuels [11]. Table–1 shows physical-chemical properties of test fuels.

2.2 Fabrication Of Engine with LHR Combustion Chamber

Figure.1 shows the details of insulated piston and insulated liner employed in the experiment.



1. Crown with threads
2. Gasket
3. Air gap
4. Body

Insulated piston

5. Insert with threads
6. Air gap
7. Liner

Insulated liner

Figure 1: Assembly details of air gap insulated piston and air gap insulated liner

The engine with LHR combustion chamber contains a two–part piston the top crown made of low thermal conductivity material, superni–90 was screwed to aluminum body of the piston, providing a 3mm air gap in between the crown and the body of the piston. The optimum thickness of air gap in the air gap piston is found to be 3 mm for improved performance of the engine with superni inserts with diesel as fuel. A superni–90 insert was screwed to the top portion of the liner in such a manner that an air gap of 3mm is maintained between the insert and the liner body. The properties and composition of superni–90 material are shown in Table-2 and Table-3.

Table-2
Properties of superni-90 material

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

Table-3
Composition of superni-90 material

Cobalt -- 2.0 %, Chromium---2.93 %, Aluminum-- 1.5 %, Titanium – 2.5 %,Carbon—0.07%, Iron – 1 % and Nickel – Balance.

2.3 Description of the Experimental Set–Up [16]

Schematic diagram of experimental set–up used for the investigations on compression ignition diesel engine and LHR combustion chamber with tyre oil is shown in Figure.2.

The specifications of the experimental engine are shown in Table–4. 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 with output signals of current and voltage. The accuracy obtained with loading of dynamometer is ±1%. The fuel consumption was registered with the aid of fuel measuring device (Burette and stop watch) and then mass flow rate of fuel was determined by knowing the density of the fuel. Density of fuel was determined by hydrometer. Percentage error obtained with measurement of fuel flow rate assuming laminar film in the burette was within the limit. The accuracy of determination of brake thermal efficiency obtained is ±2%. The speed of the engine was measured with digital tachometer with accuracy ±1%. 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). Air–box was provided with damper to damp out oscillations. 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. 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 using nozzle testing device 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, coolant outlet temperature was measured with thermocouples made of iron and iron–constantan connected to analogue temperature indicators. The accuracy with these temperature indicators are ±1%.

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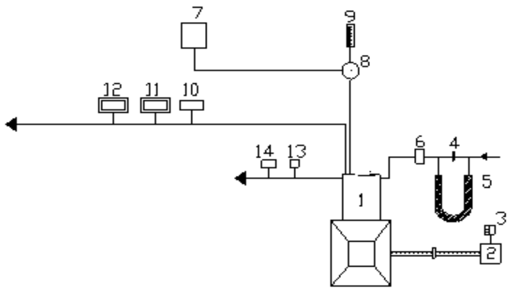

1.Engine, 2.Electical Dynamo meter, 3.Load Box, 4.Orifice meter, 5.U-tube water manometer, 6.Air box, 7.Fuel tank, 8,Three-way valve, 9.Burette, 10. Exhaust gas temperature indicator, 11.AVL Smoke meter, 12.Netel Chromatograph NOx Analyzer, 13.Outlet jacket water temperature indicator, 14. Outlet-jacket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

Figure 2: Schematic diagram of experimental set–up

Table-4
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

2.4 Operating conditions

Various test fuels used in experimentation were pure diesel and tyre oil. 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.

2.5 Definitions of used values

$m_f= \frac{10\times\rho_d\times3600}{t\times1000}$--(1)

BP= $\frac{V\times I}{\eta_d\times1000}$ (2)

BTE= $\frac{BP\times3600}{m_f\times CV}$ (3)

BSEC=$\frac{1}{BTE}$ ----(4)

BP= $\frac{BMEP\times10^5\times L\times A\times n\times k}{60000}$ ---(5)

CL= $m_w \times c_p \times (T_o - T_i)$ **(7)**

$$m_a = C_d \times a \times \sqrt{2\times10\times g\times h\times\rho_a} \times 3600 \quad (6)$$

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

$\eta_v = \frac{m_a\times2}{60\times\rho_a\times N\times V_s}$ **(8)**

$\rho_a = \frac{P_a\times10^5}{750\times R\times T_a}$ (9)

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3. RESULTS AND DICUSSIONS

3.1 Performance Parameters

Data of pure diesel with CE was taken from reference [4]. Data of pure tyre oil with CE was taken from reference [17]. Curves in Figure.3, indicate brake thermal efficiency increased up to 80% of the full load and beyond that load it decreased in conventional engine with test fuels. This was due to increase of fuel conversion up to 80% of full load. Beyond 80% of peak load, air fuel ratios got reduced as oxygen was completely used up. Conventional engine operated with crude tyre oil (CTO) showed deteriorated performance at all loads when compared with the pure diesel operation on conventional engine at 27^{o} bTDC. This was due to higher viscosity and accumulation of carbon on nozzle tip with cotton seed oil. In addition, less air entrainment by the fuel spay suggested that the fuel spray penetration might increase and resulted in more fuel reaching the combustion chamber walls. Furthermore droplet mean diameters (expressed as Sauter mean) were larger for tyre oil leading to reduce the rate of heat release as compared with diesel fuel. This also, contributed the higher ignition (chemical) delay of the tyre oil due to lower cetane number. According to the qualitative image of the combustion under the crude tyre oil operation with CE, the lower BTE was attributed to the relatively retarded and lower heat release rates. Curves in same figure indicate that LHR version of combustion chamber with tyre oil operation at recommended injection timing showed improvement in the performance for the entire load range compared with conventional engine with pure diesel.

High cylinder temperatures helped in improved evaporation and faster combustion of the fuel injected into the combustion chamber. Reduction of ignition delay of the crude tyre oil in the hot environment of the engine with LHR–2 combustion chamber improved heat release rates and efficient energy utilization.

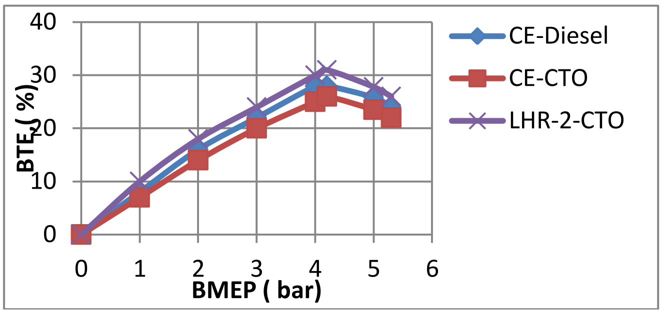

Figure 3: Variation of brake thermal efficiency with brake mean effective pressure (BMEP) with tyre oil operation in conventional engine and engine with LHR–2 combustion chamber at an injection timing of 27^{o} bTDC and injector opening pressure of 190 bar.

Injector opening pressure was varied from 190 bars to 270 bar to improve the spray characteristics and atomization of the tyre oil with conventional engine and engine with LHR–2 combustion chamber.

From Table-5, it is observed that that peak brake thermal efficiency increased with increase in injector opening pressure in both versions of the combustion chamber at different operating conditions of the tyre oil. The improvement in brake thermal efficiency at higher injector opening pressure was due to improved fuel spray characteristics. Preheated tyre oil showed marginally higher BTE than normal tyre oil. This was due to improvement in spray characteristics with improved air fuel rations.

CE with pure diesel operation showed improved BTE, while engine with LHR–2 combustion chamber showed improved BTE with tyre oil operation. Hence engine with LHR–2 combustion chamber was more suitable for crude tyre oil operation, while CE was more suitable for diesel operation.

Table-5
Data of Peak BTE

Test Fuel	Peak BTE (%)											
	Conventional Engine (CE)						Engine with LHR–2 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
Diesel	28	--	29	---	30	--	29	--	30	--	30.5	--
CTO	26	27	27	28	28	29	31	31.5	31.5	32	32	32.5

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Brake specific fuel consumption (BSFC), is not used to compare the two different fuels, because their calorific value, density, chemical and physical parameters are different. Brake specific energy consumption (BSEC) defined as energy consumed by the engine in producing I kW brake power. From Table-6, BSEC at full load operation decreased with increase of injector opening pressure with both versions of the combustion chamber with different operating conditions of tyre oil. This was due to efficient combustion with improved air fuel ratios giving lower value of brake specific energy consumption. Bulk modulus of the fuel increased with increase of injector opening pressure leading to generate higher peak pressure leading to reduce the value of brake specific energy consumption. BSEC was lower with preheated tyre oil than normal tyre oil. Bulk modulus and hence compressibility of the fuel also change with preheating. That shows lower energy substitution and effective energy utilization of tyre oil, which could replace 100% diesel fuel. BSEC was higher with tyre oil with conventional engine due to due to higher viscosity, poor volatility and reduction in heating value of tyre oil because of its poor atomization and combustion characteristics. BSEC with engine with LHR–2 combustion chamber with pure diesel operation was marginally lower when compared with conventional engine. This was due to improved combustion with improved air fuel ratios. BSEC was lower with conventional engine with diesel operation, while it was lower with engine with LHR–2 combustion chamber with tyre oil. This once again established the fact that engine with LHR–2 combustion chamber more suitable for tyre oil operation.

Table-6
Data of brake specific energy consumption (BSEC) at full load Operation

Test Fuel	Brake Specific Energy Consumption (kW.h) at full load operation											
	Conventional Engine (CE)						Engine with LHR–2 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
Diesel	4		3.96		3.92		4.2		4.1		4.0	
CTO	4.3	4.1	4.0	3.96	3.96	3.94	3.88	3.84	3.84	3.80	3.80	3.76

From the Figure.4, it is observed that conventional engine with crude tyre oil oil operation recorded drastically higher value of EGT at all loads compared with CE with pure diesel operation. Though calorific value (or heat of combustion) of fossil diesel is more than that of crude tyre oil, its density is less in comparison with tyre oil. Therefore lesser the heat is released in the combustion chamber leading to generate lower temperature with diesel operation on conventional engine. Also, there is an advanced combustion of crude tyre oil due to its higher bulk modulus. However its cetane number is less when compared to fossil diesel. Hence there is no effect of bulk modulus on injection timing (advance or retardation) and heat release. Crude tyre oil operation on conventional engine exhausted more amount of heat in comparison with pure diesel operation on CE. Lower heat release rates and retarded heat release associated with high specific energy consumption caused increase in exhaust gas temperature in conventional engine. Ignition delay in the conventional engine with different operating conditions of tyre oil increased the duration of the burning phase. At recommended injection timing, with tyre oil operation, engine with LHR–2 combustion chamber recorded lower value of exhaust gas temperature when compared with conventional engine. This was due to reduction of ignition delay in the hot environment with the provision of the insulation in the engine with LHR–2 combustion chamber, which caused the gases expanded in the cylinder giving higher work output and lower heat rejection. This showed that the performance improved with engine with LHR–2 combustion chamber over CE with tyre oil operation.

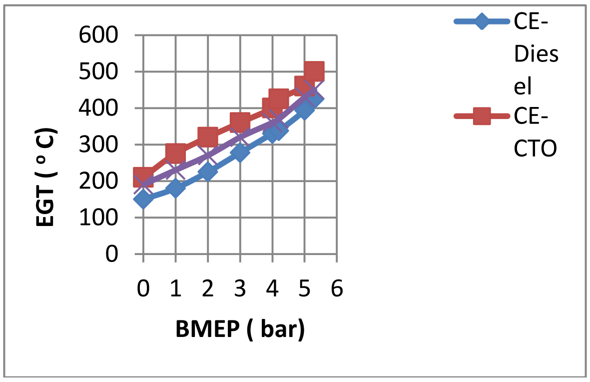

Figure 4: Variation of exhaust gas temperature (EGT) with brake mean effective pressure (BMEP) with tyre oil operation in conventional engine and engine with LHR–2 combustion chamber at an injection timing of 27° bTDC and injector opening pressure of 190 bar.

From Table-7, it is noticed that exhaust gas temperature decreased with increase of injector opening pressure with both versions of the combustion chamber with tyre oil, which confirmed that performance increased with increase of injector opening pressure. This was due to improved spray characteristics of the fuel with improved air fuel ratios. From same table, it is noticed that the exhaust gas temperatures of preheated tyre oil were higher than that of normal tyre oil in conventional engine, which indicates the increase of diffused combustion due to high rate of evaporation and improved mixing between fuel and air. Therefore, as the fuel temperature increased, the ignition delay decreased and the main combustion phase (that is, diffusion controlled combustion) increased, which in turn raised the temperature of exhaust gases.

Table-7
Data of exhaust gas temperature at full load operation

Injection Timing (° bTDC)	Test Fuel	Exhaust Gas Temperature at peak load operation (degree Centigrade)											
		Conventional Engine (CE)						Engine with LHR–2 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	
	CTO	500	550	450	500	425	450	450	425	425	4000	400	375

It can be observed from Figure.5, that volumetric efficiency decreased with an increase of brake mean effective pressure in both versions of the combustion chamber with tyre oil operation. 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 tyre oil operation decreased at all loads when compared with conventional engine with pure diesel operation. Volumetric efficiency mainly depends on speed of the engine, valve area, valve lift, timing of the opening or closing of valves and residual gas fraction rather than on load variation. Hence with tyre oil operation with conventional engine, volumetric efficiency decreased in comparison with pure diesel operation on conventional engine, as residual gas fraction increased**. T**his was due to increase of deposits with tyre oil operation with conventional engine. This was also due to increase of exhaust gas temperatures with conventional engine with tyre oil operation which in turn increased combustion chamber wall temperature. The reduction of volumetric efficiency with engine with LHR–2 combustion chamber 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–2 combustion chamber.

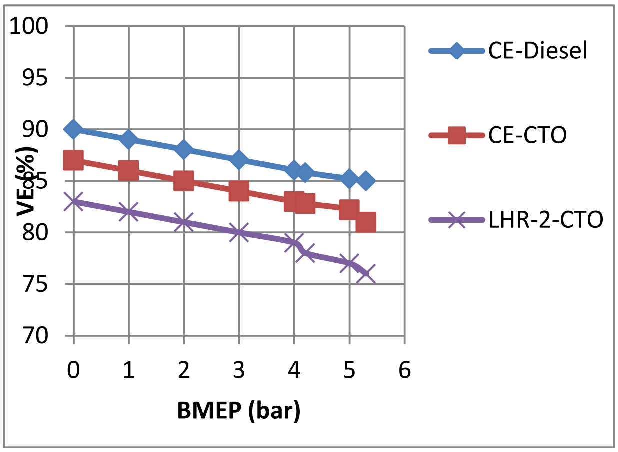

Figure 5: Variation of volumetric efficiency with brake mean effective pressure (BMEP) with tyre oil operation in conventional engine and engine with LHR–2 combustion chamber at an injection timing of 27° bTDC and injector opening pressure of 190 bar.

From Table-8, it is observed that volumetric efficiency increased with increase of injector opening pressure in both versions of the combustion chamber with tyre oil. This was also due to improved fuel spray characteristics and evaporation at higher injector opening pressures leading to marginal increase of volumetric efficiency. This was also due to the reduction of residual fraction of the fuel, with the increase of injector opening pressure. Increase of volumetric efficiency depends on combustion chamber wall temperature, which in turn depends on exhaust gas temperatures. With increase of injector opening pressure, exhaust gas temperatures decreased and hence volumetric efficiency increased. Preheating of the crude tyre oil marginally decreased volumetric efficiency in conventional engine, when compared with the normal temperature of crude tyre oil, because of reduction of bulk modulus, density of the fuel and increase of exhaust gas temperatures.

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Table-8
Data of Volumetric efficiency at full load operation

Injection Timing (° bTDC)	Test Fuel	Volumetric Efficiency (%) at peak load operation											
		Conventional Engine (CE)						Engine with LHR–2 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	
	CTO	81	80	82	81	83	82	76	77	77	78	78	79

Coolant load increased with BMEP for test fuels with both versions of the combustion chamber as noticed from Figure-6. This was due to increase of gas temperatures. Cooling load was higher with tyre operation with conventional engine. This was due to un-burnt fuel concentration at combustion chamber walls. Coolant load decreased with engine with LHR-1 combustion chamber with tyre oil operation. This was due to not only insulation provided with LHR–2 combustion chamber, but also due to improved combustion with the provision of insulation. This was also because of improved air fuel ratio with which gas temperatures decreased.

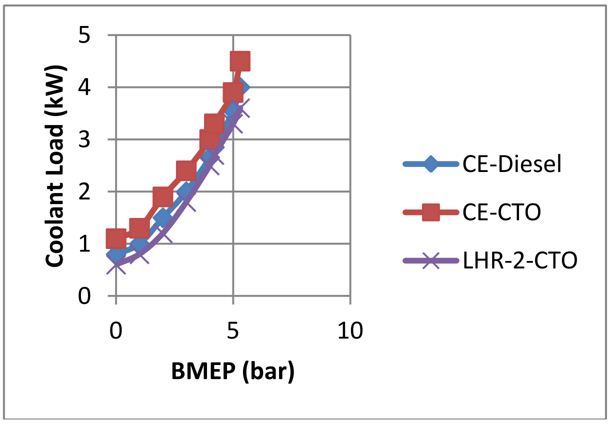

Figure 6: Variation of coolant load with brake mean effective pressure (BMEP) with tyre oil operation in conventional engine and engine with LHR–2 combustion chamber at an injection timing of 27^{o} bTDC and injector opening pressure of 190 bar.

It is observed from Table-9, coolant load increased marginally in the conventional engine while it decreased in the engine with LHR–2 combustion chamber with increasing of the injector opening pressure with tyre oil. This was due to the fact with increase of injector opening pressure with conventional engine, increased nominal fuel spray velocity resulting in better fuel-air mixing with which gas temperatures increased. The reduction of coolant load in the LHR engine was not only due to the provision of the insulation but also it was due to better fuel spray characteristics and increase of air-fuel ratios causing decrease of gas temperatures and hence the coolant load. Coolant load decreased marginally with preheating of tyre this was due to improved air fuel ratios with improved spray characteristics leading to reduction of gas temperatures.

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Table-9
Data of coolant load at peak load operation

Injection Timing (° bTDC)	Test Fuel	Data of Coolant Load (kW) at peak load operation.											
		Conventional Engine (CE)						Engine with LHR–2 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	---	4.2	--	3.8	---
	CTO	4.5	4.3	4.2	4.0	4.0	3.8	3.6	3.4	3.4	3.2	3.2	3.0

This indicates at recommended injection timing, sound intensities marginally increased in CE with tyre oil operation in comparison with CE with pure diesel operation as noticed from Figure.7. Higher viscosity, bulk modulus, duration of combustion and poor volatility caused moderate combustion of tyre oil leading to generate higher sound levels. The engine with LHR–2 combustion chamber decreased sound intensity when compared with pure diesel operation on CE. This was because of hot environment in engine with LHR–2 combustion chamber improved the combustion of tyre oil. This was also due to decrease of density and bulk modulus of fuel at higher temperatures leading to produce lower levels of sound with LHR engine.

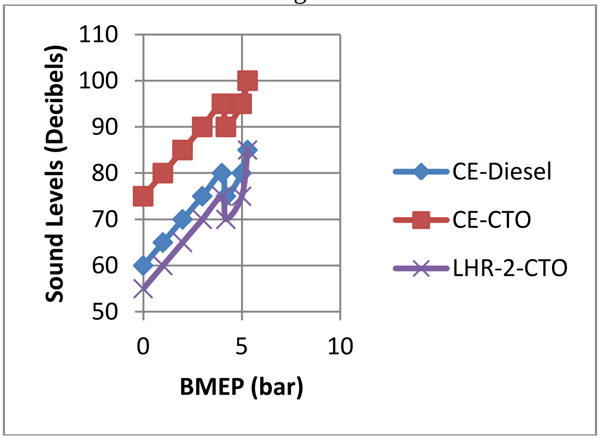

Figure 7: Variation of sound levels with brake mean effective pressure (BMEP) with tyre oil operation in conventional engine and engine with LHR–2 combustion chamber at an injection timing of 27° bTDC and injector opening pressure of 190 bar.

It is observed that sound intensity marginally decreased with increase of injector opening pressure for both versions of the engine with the tyre oil as noticed from Table-10. This was because of improved combustion with increased air fuel ratios. This was also due to simultaneous increase of bulk modulus and density. This was due to improved spray characteristic of the fuel, with which there was no impingement of the fuel on the walls of the combustion chamber leading to produce efficient combustion. Sound intensities were lower at preheated condition of tyre oil, when compared with their normal condition. This was due to improved spray characteristics, decrease of density and reduction of bulk modulus of the fuel.

Table-10
Data of sound intensity at full load operation

Injection Timing (° bTDC)	Test Fuel	Data of Sound Intensity (decibels) at peak load operation											
		Conventional Engine (CE)						Engine with LHR–2 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	Diesel	85	--	80	--	75	--	90	--	85	--	80	--
	CTO	100	95	95	90	90	85	85	80	80	75	75	70

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3.2 Pollution Levels

Curves in Figure.8 indicate that drastic increase of smoke levels at all loads with CE fuelled with tyre oil was observed when compared with pure diesel operation on CE. This was due to the higher value of ratio of C/H tyre oil, when compared with pure diesel. The increase of smoke levels was also due to decrease of air-fuel ratios and volumetric efficiency. Smoke levels were related to the density of the fuel. Smoke levels were higher with tyre oil due to its high density. However, smoke levels were comparable with engine with LHR–2 combustion chamber with tyre oil operation, due to efficient combustion and less amount of fuel accumulation on the hot combustion chamber walls of the LHR engine at different operating conditions of tyre oil.

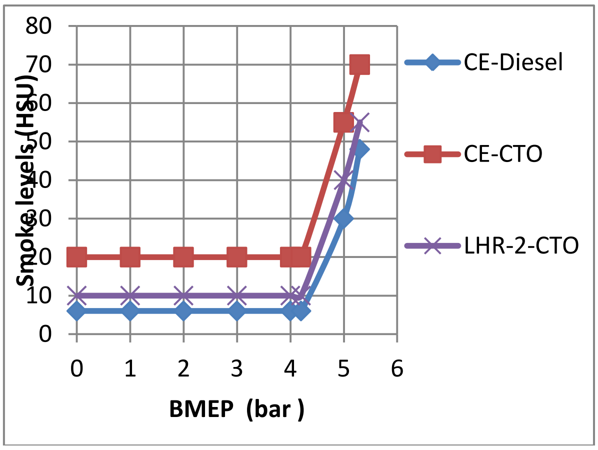

Figure 8: Variation of smoke levels in Hartridge Smoke Unit (HSU) with brake mean effective pressure (BMEP) with tyre oil operation in conventional engine and engine with LHR–2 combustion chamber at an injection timing of 27^{o} bTDC and injector opening pressure of 190 bar.

Data from Table-11 shows that smoke levels decreased with increase of injection timing in both versions of the combustion chamber, with different operating conditions of the tyre oil. This was due to improvement in the fuel spray characteristics with higher injector opening pressures, causing lower smoke levels. Preheating of the tyre oil decreased smoke levels in both versions of the engine, when compared with normal temperature of tyre oil. This was due to i) the reduction of density of the tyre oil, as density was directly related to smoke levels, ii) the reduction of the diffusion combustion proportion in conventional engine with the preheated tyre oil, iii) reduction of the viscosity of the tyre oil, with which the fuel spray does not impinge on the combustion chamber walls of lower temperatures rather than it was directed into the combustion chamber**.**

Table-11
Data of Smoke levels at full load operation

Injection timing (^{o} bTDC)	Test Fuel	Smoke levels (Hartridge Smoke Unit, HSU) at full load operation											
		CE						Engine with LHR–2 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	Diesel fuel	48	--	38	--	34	--	55	--	50	--	45	--
	CTO	70	65	65	60	60	55	55	50	50	45	45	40

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4. SUMMARY

Engine with LHR combustion chamber with tyre oil operation increased peak brake thermal efficiency by 19%, at full load operation– decreased brake specific energy consumption by 10%, decreased exhaust gas temperature by 10%, decreased coolant load by 20%, decreased sound levels by 15%, decreased volumetric efficiency by 6% and decreased smoke levels by 21% in comparison with conventional engine at similar operating conditions in comparison with conventional engine. Conventional engine with tyre oil at normal temperature decreased peak BTE by 3%, at full load operation-increased brake specific energy consumption by 7%, exhaust gas temperature increased by 2%, decreased volumetric efficiency by 5%, increased coolant load by 13%, increased sound levels by 18% and increased smoke levels by 46% in comparison with conventional engine with pure diesel operation. Performance parameters and pollution levels improved with an increase of injector opening pressure with engine with both versions of the combustion chamber. With preheating of tyre oil- Peak brake thermal efficiency increased, at full load operation-brake specific energy consumption decreased, exhaust gas temperature increased, volumetric efficiency decreased, coolant load decreased, sound levels decreased and smoke levels decreased when compared with normal temperature of the tyre oil.

4.1 Research Findings

Comparative studies on performance parameters and pollution levels with direct injection diesel engine with air gap insulated combustion chamber and engine with conventional combustion chamber were made at varied injector opening pressure with different operating conditions of tyre oil. Experimental results were compared with pure diesel operation at similar operating conditions.

4.2 Future Scope of Work

Hence further work on the effect of injection timing on performance parameters, exhaust emissions and combustion characteristics with air gap insulated LHR combustion chamber with tyre oil operation is necessary.

ACKNOWLEDGMENTS

Authors thank authorities of Chaitanya Bharathi Institute of Technology, Hyderabad for providing facilities for carrying out research work. Financial assistance provided by All India Council for Technical Education (AICTE), New Delhi is greatly acknowledged.

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