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## Influence of Injector Opening Pressure on Performance and Exhaust Emissions in DI Diesel Engine with Air Gap Insulated Piston and Air Gap Insulated Liner with Diesel Operation

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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 pure diesel operation 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  $NO_x$  analyzer. Engine with LHR combustion chamber improved performance at 80% of the full load operation 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 conservation of diesel fuel 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.

The concept of LHR combustion chamber is to reduce coolant losses by providing thermal resistance in the path of heat flow to the coolant, thereby 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 superalloy, 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.[3-5] 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.

The technique of providing an air gap in the piston involved the complications of joining two different metals. Investigations were carried out on LHR-2 combustion chamber- with air gap insulated piston with pure diesel.[6]. However, the bolted design employed by them could not provide complete sealing of air in the air gap. 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% [7].

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Sound levels determine the phenomena of combustion in engine whether the performance was improving or deteriorating. Studies were made on sound levels with convention engine with vegetable oils and it was reported from the studies, that performance deteriorated with vegetable oil operation on conventional engine leading to produce high sound levels.[8-9]

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.[8-9]

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 diesel fuel with varied injector opening pressure. Comparative performance studies were made with medium grade LHR combustion chamber with conventional engine CE with diesel operation.

## 2. MATERIAL AND METHOD

The physical-chemical properties of the diesel fuel are presented in Table-1. Fuels with flash point above 52° C are considered as safe. Thus diesel is extremely safe fuel to handle. Diesel has high cetane number, which shows efficient combustion in compression ignition engine. Diesel fuel has moderate viscosity. Hence there are no injection problems.

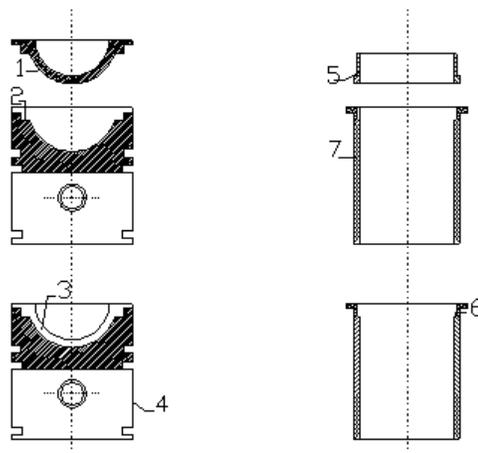
**Table.1.** Properties of Diesel Fuel

Property	Units	Diesel
Carbon chain	--	C <sub>8</sub> -C <sub>28</sub>
Cetane Number		55
Density	gm/cc	0.84
Bulk modulus @ 20Mpa	Mpa	1475
Kinematic viscosity @ 40°C	cSt	2.25
Flash point (Open cup)	°C	66
Molecular weight	--	226
Colour	--	Light yellow

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 [7]. 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°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

4 Body of piston

5 Superni insert with threads

6 Air gap

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

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orifice flow meter and U-tube water manometer). The naturally aspirated engine was provided with water-cooling system in which inlet temperature of water was maintained at 80°C by adjusting the water flow rate. Engine oil was provided with a pressure feed system. No temperature control was incorporated, for measuring the lube oil temperature. Injector opening pressure was changed 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 and coolant water outlet temperatures were measured with thermocouples made of iron and iron-constantan connected to temperature indicators. .

**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°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

## Operating conditions:

Test fuel used in the experimentation were diesel. 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). The results were tabulated and comparative studies of performance parameters, exhaust emissions were reported at different operating conditions of the compression ignition engine.

## Nomenclature

$\rho_a$  = density of air, kg/m<sup>3</sup>  
 $\rho_d$  = density of fuel, gm/cc  
 $\eta_d$  = efficiency of dynamometer, 0.85  
 $a$  = area of the orifice flow meter in m<sup>2</sup>  
 BP = brake power of the engine, kW  
 $C_d$  = coefficient of discharge, 0.65  
 $C_p$  = 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  
 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, °C  
 $T_i$  = inlet temperature of water, °C  
 $T_o$  = outlet temperature of water, °C  
 $V$  = voltmeter reading, volt  
 $V_s$  = stroke volume, m<sup>3</sup>  
 VE = Volumetric efficiency, %

## Definitions of used values:

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

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

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

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

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

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

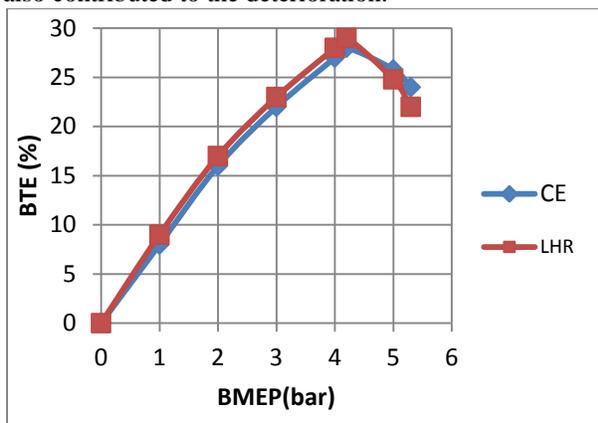
$$a = \frac{\pi \times d^2}{4} \quad \text{equation (7)}$$

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

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

### 3. RESULTS AND DISCUSSION

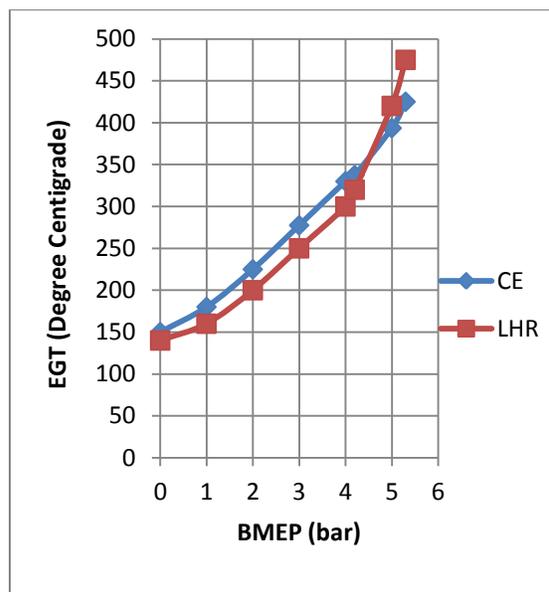
Curves in figure.2 indicates that BTE increased up to 80% of the full load in the engine with LHR-2 combustion chamber at the recommended injection timing and beyond this load, it decreased when compared with conventional engine with pure diesel operation. The increase of BTE was due to improved air fuel ratios and improved combustion. As the combustion chamber was insulated to greater extent, it was expected that high combustion temperatures would be prevalent in engine with LHR-2 engine. It tends to decrease the ignition delay thereby reducing pre-mixed combustion as a result of which, less time was available for proper mixing of air and fuel in the combustion chamber leading to incomplete combustion, with which BTE decreased beyond 80% of the full load. More over at this load, friction and increased diffusion combustion resulted from reduced ignition delay. Increased radiation losses might have also contributed to the deterioration.



**Figure 2:** Variation of brake thermal efficiency (BTE) 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.

Figure.3 indicates that exhaust gas temperature (EGT) increased with increase of BMEP for both versions of the combustion chamber. This was because of increased fuel consumption with which gas temperatures increased and hence EGT. The engine with LHR combustion chamber at the recommended injection timing recorded lower EGT up to 80% of the load and beyond that load it

increased compared to conventional engine at the recommended injection timing. This indicated that heat rejection was restricted through the piston and liner, thus maintaining the hot combustion chamber as result of which the exhaust gas temperature increased.



**Figure 3:** 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° bTDC and injector opening pressure of 190 bar.

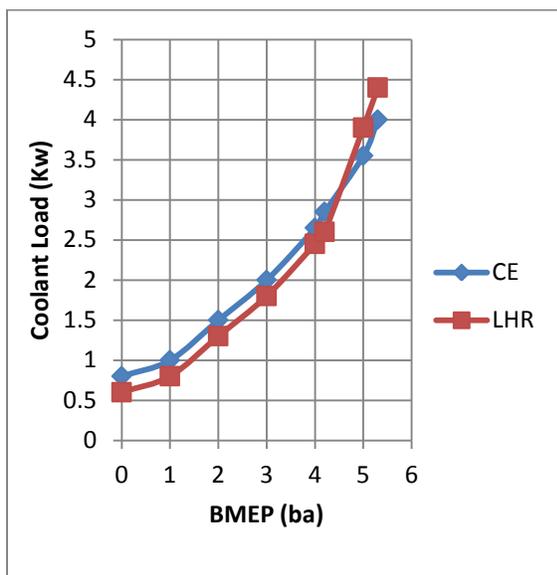
Curves in figure.4, indicates that coolant load increased with increase of BMEP for both versions of the combustion chamber. This was due to increase of combustion temperatures due to consumption of fuel. The engine with LHR combustion chamber gave lesser coolant load up to 80% of the full load, when compared to conventional engine. Air being a bad conductor offers thermal resistance for heat flow through the piston and liner. It was therefore evident that thermal barrier provided in the piston and liner resulted in reduction of coolant load up to 80% of the full load. Beyond 80% of the full load, coolant load in the engine with LHR-2 combustion chamber increased over and above that of the conventional

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engine, with which efficiency was deteriorated at full load of the engine with LHR-2 combustion chamber, when compared to the convectional engine. This was because in cylinder, the heat rejection at full load was primarily due to un-burnt fuel concentration near the combustion chamber walls.

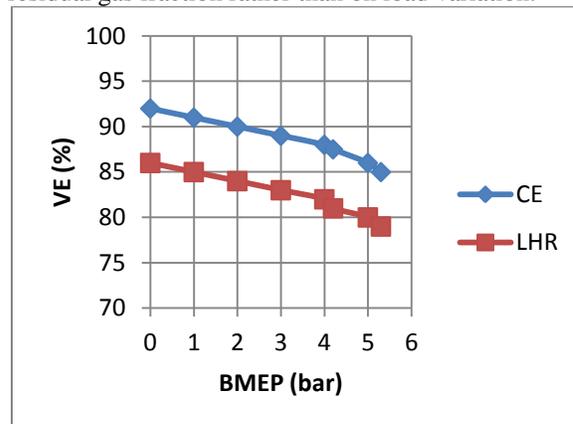
The air-fuel ratio got reduced to a reasonably low value at this load confirming the above trend. However, when heat rejection calculations of coolant were made, the heat lost to lubricant should also be considered. As in the present investigations the lubricant heat loss was not considered, this aspect was not depicted in coolant load calculations. This was also due to the fact that nearer full load; a greater amount of heat was being transferred through the un-insulated cylinder head, where higher temperatures exist.



**Figure 4:** Variation of coolant load 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.

Curves in figure.5 indicate that volumetric efficiency decreased with the increase of brake mean effective pressure (BMEP) in 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 engine with LHR-2 combustion chamber decreased at all loads when compared to the conventional engine. This was due increase of temperature of incoming charge in the hot

environment created with the provision of insulation, causing reduction in the density and hence the quantity of air. However, this variation in volumetric efficiency is very small between these two versions of the engine, as volumetric efficiency mainly depends on speed of the engine, valve area, valve lift, timing of the opening or closing of valves and residual gas fraction rather than on load variation.

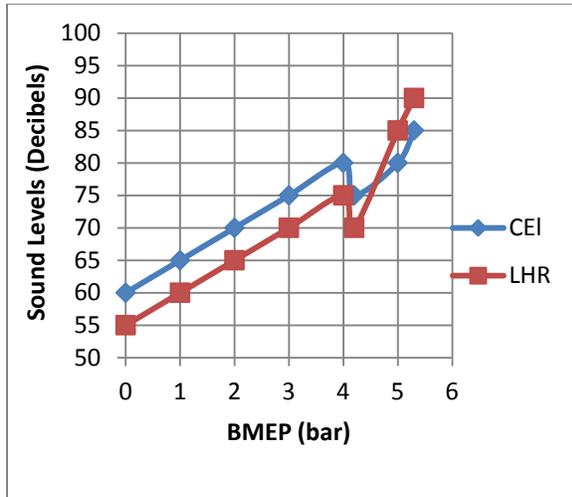


**Figure 5:** Variation of volumetric efficiency (VE) with brake mean effective pressure(BMEP) with pure diesel operation in conventional engine and engine with air gap insulated piston and air gap insulated liner (LHR) at an injection timing of 27°bTDC and injector opening pressure of 190 bar.

Figure.6 shows that sound intensity increased up to 80% of the full load and decreased at 80% of the full load and beyond it increased again for both versions of the combustion chamber. This was due to increase of thermal efficiency at 80% of the full load. Beyond that load, thermal efficiency decreased due to reduction of volumetric efficiency. Sound levels decreased with engine with LHR combustion chamber up to 80% of the full load when compared with conventional engine.

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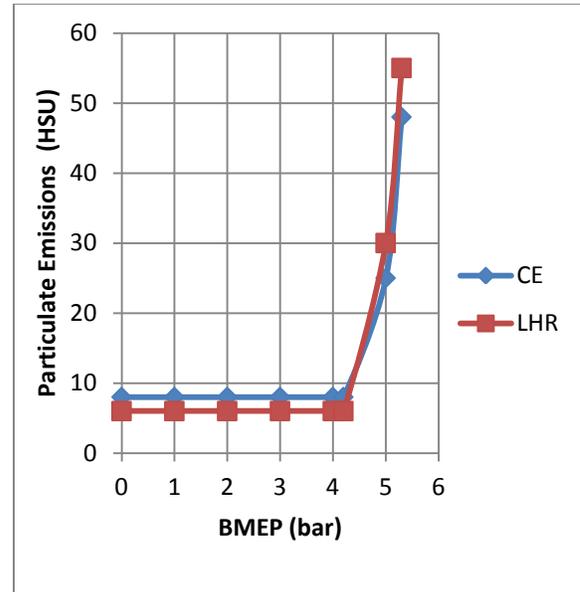


**Figure 6:** Variation of sound levels with brake mean effective pressure(BMEP) with pure diesel operation in conventional engine and engine with air gap insulated piston and air gap insulated liner (LHR) at an injection timing of 27°bTDC and injector opening pressure of 190 bar.

This showed that combustion improved with engine with LHR combustion chamber up to 80% of the load. However, sound intensity increased over and above conventional engine. This was due to deterioration of the performance of the engine with LHR combustion chamber beyond 80% of the full load.

## B. 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. Up to 80% of full load, drastic reduction of smoke intensity was observed in the engine with LHR combustion chamber, when compared to the conventional engine. This was due to the increased oxidation rate of particulate matter in relation to formation of particulate emissions. Higher surface temperatures of the engine with LHR combustion chamber aided this process.



**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.

Particulate emissions are formed during combustion in low oxygen regions of the flames. Engine with LHR combustion chamber shorten the delay period, which curbs thermal cracking, responsible for particulate emissions. Beyond 80% of full load, marginal and slight increase of particulate emissions was observed in the engine with LHR combustion chamber, when compared to conventional engine. This was due to fuel cracking at higher temperature, leading to increase in particulate emissions. Higher temperature of engine with LHR combustion chamber produced increased rates of both particulate emissions and burn up. The reduction in volumetric efficiency and air-fuel ratios were responsible factors for increasing particulate emissions in the engine with LHR combustion chamber at near full load operation of the engine. As expected, particulate emissions increased in the engine with LHR combustion chamber, because of higher temperatures and improper utilization of the fuel consequent upon predominant diffusion combustion.

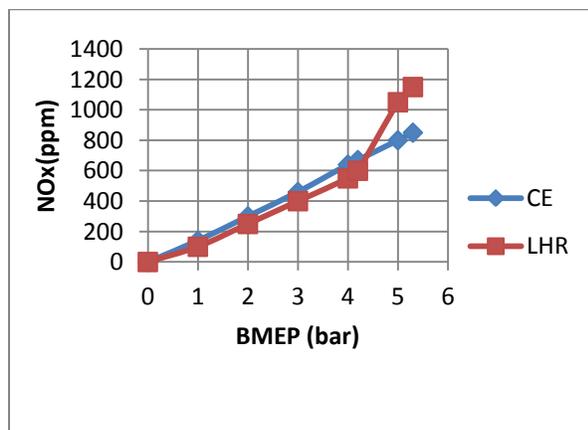
The temperature and availability of oxygen are the reasons for the formation of NO<sub>x</sub>. For both versions of the combustion chamber, NO<sub>x</sub> concentrations raised steadily as the fuel/air ratio increased (Figure.8) with increasing BP/BMEP, at constant injection timing. At part load, NO<sub>x</sub> concentrations

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were less in both versions of the engine. This was due to the availability of excess oxygen. At remaining loads, NO<sub>x</sub> concentrations steadily increased with the load in both versions of the combustion chamber. This was because, local NO<sub>x</sub> 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<sub>x</sub> 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 NO<sub>x</sub> emissions should be roughly proportional to the mass of fuel injected (provided burned gas pressures and temperature do not change greatly).

The engine with LHR combustion chamber recorded lower NO<sub>x</sub> levels up to 80% of the full load, and beyond that load it produced higher NO<sub>x</sub> levels compared to conventional engine. As the air-fuel ratios were higher in the LHR combustion chamber, causing more dilution, due to mixing with the excess air, leading to produce less NO<sub>x</sub> concentrations, up to 80% of the full load, when compared to conventional engine. Beyond 80% of full load, due to the reduction of fuel-air equivalence ratio with LHR combustion chamber, which was approaching to the stoichiometric ratio, causing higher value of NO<sub>x</sub> levels.



**Figure 8:** Variation of nitrogen oxide (NO<sub>x</sub>) 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.

## C. 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 engine, at the recommended injection timing. This was due to improved fuel spray 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 fuel consumption (BSFC) 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 burning. BSFC of the conventional engine was improved at a higher injector opening pressure of 270 bar, when compared to the same version of the engine at the recommended injection pressure of 190 bar. This was due to improved atomization of the fuel at higher injector opening pressure.

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.

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.

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.

Sound levels decreased marginally with increase of injector opening pressure in both versions of the combustion chamber at full load operation. This showed that combustion improved with increase of injector opening pressure. This was because of increase of fuel atomization characteristics.

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**Table.3** Variation of Performance Parameters with Injector Opening Pressure with Diesel Operation

Parameter /Unit	Conventional Engine			Engine with LHR combustion chamber		
	190	230	270	190	230	270
Peak BTE(%)	28	29	30	29	30	30.5
BSFC (kg/kW.h)	0.34	0.33	0.32	0.35	0.34	0.33
EGT (°C)	425	410	395	475	450	425
Coolant Load (kW)	4.0	4.2	4.4	4.5	4.2	3.8
VE (%)	85	86	87	79	80	81
Sound Levels (Decibels)	85	80	75	90	85	80

Table 4 shows variation of pollution levels with injector opening pressure for conventional engine and engine with LHR combustion chamber with pure diesel operation.

Particulate emissions decreased marginally with increase of injector opening pressure in both versions of the combustion chamber at full load operation. This was due to improved spray characteristics of the fuel. NO<sub>x</sub> emissions increased with the increase of injector opening pressure in the CE due to increase of fuel-air mixing rate, heat release rate during the premixed- combustion and mixing-controlled combustion phases. However, NO<sub>x</sub> emissions decreased with LHR combustion chamber. This was because of decrease of gas temperatures in the LHR combustion chamber and increase of the same in the CE with the increase of injector opening pressure.

**Table.4** Variation of Pollution Levels with Injector Opening Pressure with Diesel Operation

Parameter /Unit	Conventional Engine			Engine with LHR combustion chamber		
	190	230	270	190	230	270
Particulate Emissions (HSU)	48	38	34	55	50	45
NO <sub>x</sub> emissions (ppm)	850	900	950	1150	1100	1050

## 4. CONCLUSIONS

In comparison with conventional engine, engine with LHR combustion chamber with diesel operation increased peak brake thermal efficiency by 3%, brake

specific fuel consumption by 3%, exhaust gas temperature by 12%, coolant load by 12%, decreased volumetric efficiency by 7%, increased sound levels by 6%, particulate emissions by 14% and NO<sub>x</sub> levels by 29% at an injector opening pressure of 190 bar.

However, performance parameters and pollution levels improved with engine with LHR combustion chamber at an injector opening pressure of 270 bar in comparison with 190 bar.

### 4.1 Research Findings.

Engine with LHR combustion chamber with air gap insulated piston and air gap insulated liner improved performance parameters and pollution levels at 80% of the full load operation when compared with conventional engine with diesel operation.

### 4.2 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 diesel fuel like vegetable oils, biodiesel and alcohols.

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