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EFFECT OF INJECTION TIMING ON EXHAUST EMISSIONS AND COMBUSTION CHARACTERISTICS OF DIRECT INJECTION DIESEL ENGINE WITH AIR GAP INSULATION

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***Abstract**:* Experiments were carried out to study exhaust emissions of diesel engine with air gap insulated low heat rejection (LHR–2) combustion chamber consisting of air gap insulated piston with 3 mm air gap, with superni (an alloy of nickel) crown, air gap insulated liner with superni insert with neat diesel with varied injection timing. Exhaust emissions of particulate emissions and nitrogen oxide (NO_x) levels were determined at various values of brake mean effective pressure (BMEP) of the LHR-2 combustion chamber and compared with neat diesel operation on conventional engine (CE) at similar operating conditions. Combustion diagnosis was carried out using miniature Piezo electric pressure transducer, TDC (top dead centre) and special pressure-crank angle software package at full load operation. The optimum injection timing was found to be 31°bTDC (before top dead centre) with conventional engine, while it was 29° bTDC for engine with LHR–2 combustion chamber with diesel operation. Engine with LHR–2 combustion chamber with neat diesel operation showed increased particulate emissions and NOx levels at manufacturer's recommended injection timing of 27° bTDC, and the they improved marginally with advanced injection timing of 31°bTDC in comparison with CE at 27°bTDC.

***Keywords*:** Conservation of diesel, conventional engine, LHR combustion chamber, Performance.

I.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 for his remarkable invention of diesel engine. Compression ignition (CI) engines, due to their excellent fuel efficiency and durability, have become popular power plants for automotive applications. This is globally the most accepted type of internal combustion engine used for powering agricultural implements, industrial applications, and construction equipment along with marine propulsion. [2–3].

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 ceramic coated (LHR–1), air gap insulated (LHR–2) and combination of ceramic coated and air gap insulated engines(LHR-3) combustion chambers depending on degree of insulations.

Wallace et al. also studied the performance of the insulated piston engine in which air gap thickness was maintained at 2-mm. [4]. The major finding was increase of particulate emissions due to reduction of air–fuel ratios from 18.27 to astonishingly small 12.76, which was inadmissible in practice.

Karthikeyan et al. studied the performance of a diesel engine by insulating engine parts employing 2-mm air gap in the piston and the liner, thus attaining a semi-adiabatic condition. [5]. The nimonic piston with 2-mm air gap was studded with the body of the piston. Mild steel sleeve, provided with 2-mm air gap was fitted with the total length of the liner. They reported increase of particulate emissions at all loads, when compared to neat diesel operation on conventional engine. This was due to higher exhaust gas temperatures.

Jabez Dhinagar et al. conducted experiments on LHR engine, with an air gap insulated piston, air gap insulated liner and ceramic coated cylinder head. [6]. The piston with nimonic crown with 2 mm air gap was fitted with the body of the piston by stud design.

Mild steel sleeve was provided with 2 mm air gap and it was fitted with the 50 mm length of the liner. The performance was deteriorated with this engine and increase of particulate emissions at full load operation with neat diesel operation, at recommended injection timing. Hence the injection timing was retarded to improve performance and pollution levels.

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.[7]. However, the bolted design employed by them could not provide complete sealing of air in the air gap. Investigations were carried out with engine with LHR–2 combustion chamber with air gap insulated piston with nimonic crown threaded with the body of the piston fuelled with neat diesel with varied injection timing. It was reported from their investigations that particulate emissions and NO_x levels decreased and improved combustion characteristics with advanced injection timing of 29.5° bTDC Engine with LHR combustion chamber was more suitable for vegetable oil operation, as hot combustion chamber was maintained by it in burning high viscous vegetable oils. Experiments were conducted on engine with LHR–2 combustion chamber with varied injection timing and injection pressure. [9–14]. It was reported from their investigations that engine with LHR-2 combustion chamber decreased particulate emissions by 8-10% in comparison with neat diesel operation on CE. Exhaust emissions and combustion characteristics were improved with advanced injection timing.

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–2 combustion chamber was fuelled with diesel fuel with varied injection timing. Comparative performance studies were made on engine with LHR–2 combustion chamber with conventional engine with diesel operation.

MATERIALS AND METHODS

This part deals with fabrication of air gap insulated piston and air gap insulated liner, brief description of experimental set-up, specification of experimental engine, operating conditions and definitions of used values.

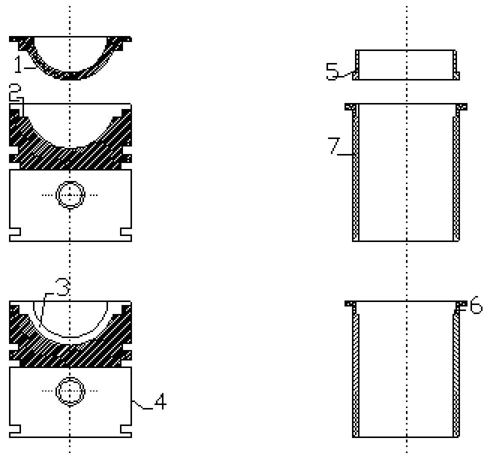
The physic-chemical properties of the diesel fuel are presented in Table-1.

Table.1. Properties Of Diesel

Property	Units	Diesel
Carbon chain	--	C_8-C_{28}
Cetane Number		55
Density	gm/cc	0.84
Bulk modulus @ 20Mpa	Mpa	1475
Kinematic viscosity @ 40°C	cSt	2.25
Sulfur	%	0.25
Oxygen	%	0.3
Air fuel ratio (stochiometric)	--	14.86
Lower calorific value	kJ/kg	44800
Flash point (Open cup)	°C	68
Molecular weight	--	226
Colour	--	Light yellow

LHR-2 combustion chamber (Fig.1) contained a two-part piston; the top crown made of low thermal conductivity material, superni–90 (an alloy of nickel) screwed to aluminum body of the piston, providing a 3 mm air gap in between the crown and the body of the piston. The optimum thickness of air gap in the air gap piston was found to be 3-mm for improved performance of the engine with diesel as fuel. [8]. The height of the piston was maintained such that compression ratio was not altered.

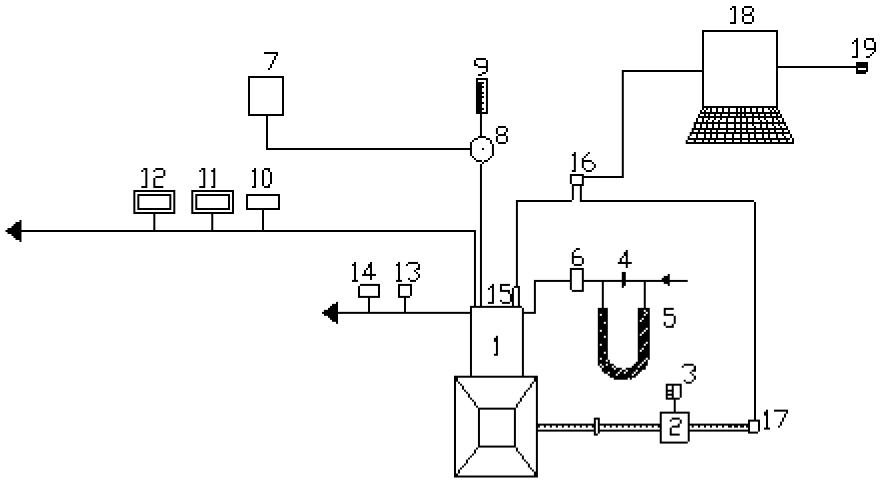
A superni-90 insert was screwed to the top portion of the liner in such a manner that an air gap of 3-mm was maintained between the insert and the liner body. At 500°C the thermal conductivity of superni-90 and air are 20.92 and 0.057 W/m-K.



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

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

The test fuel used in the experimentation was neat diesel. The schematic diagram of the experimental setup with diesel operation is shown in Figure 2. The specifications of the experimental engine are shown in Table-2. Experimental setup used for study of exhaust emissions on low grade LHR diesel engine with cottonseed biodiesel in Fig.3 The specification of the experimental engine (Part No.1) is shown in Table.2 The engine was connected to an electric dynamometer (Part No.2. Kirloskar make) for measuring its brake power. Dynamometer was loaded by loading rheostat (Part No.3). The combustion chamber consisted of a direct injection type with no special arrangement for swirling motion of air. Burette (Part No.9) method was used for finding fuel consumption of the engine with the help of fuel tank (Part No7) and three way valve (Part No.8). Air-consumption of the engine was measured by air-box method consisting of an orifice meter (Part No.4), U-tube water manometer (Part No.5) and air box (Part No.6) assembly.



1.Engine, 2.Electical Dynamometer, 3.Load Box, 4.Orifice flow meter, 5.U-tube water manometer, 6.Air box, 7.Fuel tank, 8, Three way valve 9.Burette, 10. Exhaust gas temperature indicator, 11.Smoke opacity meter, 12.NO_x Analyzer, 13.Outlet–jacket water temperature indicator, 14. Outlet-jacket water flow meter, 15. Piezo-electric pressure transducer, 16.Console, 17.TDC encoder, 18.Personal Computer and 19. Printer.

Fig.2 Schematic diagram of experimental set–up

The naturally aspirated engine was provided with water-cooling system in which outlet temperature of water is maintained at 80ºC 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.

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	27ºbTDC × 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 naturally aspirated engine was provided with water-cooling system in which outlet temperature of water is maintained at 80ºC by adjusting the water flow rate, which was measured by water flow meter (Part No.14). Exhaust gas temperature (EGT) and coolant water outlet temperatures were measured with thermocouples made of iron and iron-constantan attached to the exhaust gas temperature indicator (Part No.10) and outlet jacket temperature indicator (Part No.13). Copper shims of suitable size were provided in between the pump body and the engine frame, to vary the injection timing and its effect on the exhaust emissions and combustion characteristics of the engine was studied

Exhaust emissions of particulate matter and nitrogen oxides (NO_x) were recorded by smoke opacity meter (AVL India, 437) and NO_x Analyzer (Netel India ;4000 VM) at full load operation of the engine. Table 3 shows the measurement principle, accuracy and repeatability of raw exhaust gas emission analyzers/ measuring equipment for particulate emissions and NO_x levels. Analyzers were allowed to adjust their zero point before each measurement. To ensure that accuracy of measured values was high, the gas analyzers were calibrated before each measurement using reference gases.

Table.3

Specifications of the

Smoke Opacimeter (AVL, India, 437). And NO_x Analyzer (Netel India, (4000 VM))

Pollutant	Measuring Principle	Range	Least Count	Repeatability
Particulate Emissions	Light extinction	1–100%	0.1% of Full Scale (FS)	0.1% for 30 minutes
NO_x	Chemiluminiscence	1–5000 ppm	0.5% of FS	≤0.5% F.S

Water cooled Piezo electric transducer(AVL Austria: QC34D)., fitted on the cylinder head to measure pressure in the combustion chamber was connected to a console, which in turn was connected to Pentium personal computer. TDC (top dead centre) encoder (AVL Austria: 365x) with a crank angle (CA) resolution of 0.5 crank angle degrees (CAD) provided at the extended shaft of the dynamometer was connected to the console to determine the crankshaft position. A special pressure-crank angle (P–θ) software package evaluated the combustion characteristics such as peak pressure (PP), time of occurrence of peak pressure (TOPP) and maximum rate of pressure rise (MRPR) from the signals of pressure and crank angle at the peak load operation of the engine. Pressure-crank angle diagram was obtained on the screen of the personal computer.

Operating Conditions: Fuel used in experiment was neat diesel. Various injection timings attempted in the investigations were 27–34°bTDC.

3. RESULTS AND DICUSSION

3.1.Performance Parameters

The variation of brake thermal efficiency (BTE) with brake mean effective pressure (BMEP) in the conventional engine (CE) with pure diesel, at various injection timings at an injector opening pressure of 190 bar, is shown in Fig. 3.

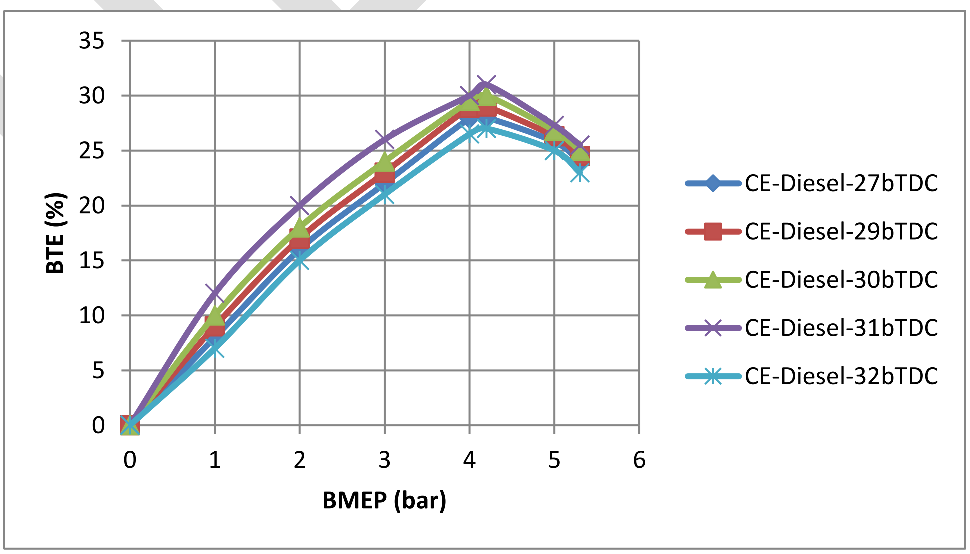


Fig.3 variation of brake thermal efficiency (BTE) with brake mean effective pressure (BMEP) in the conventional engine with neat diesel, at various injection timings at an injector opening pressure of 190 bar.

BTE increased with the advanced injection timings in the conventional engine at all loads, due to early initiation of combustion and increase of contact period of fuel with air leading to improve air fuel ratios period. The optimum injection timing was obtained by based on maximum brake thermal efficiency. Maximum BTE was observed when the injection timing was advanced to 31°bTDC in CE. Performance deteriorated if the injection timing was greater than 31°bTDC. This was because of increase of ignition delay.

The variation of BTE with BMEP in the LHR–2 combustion chamber with neat diesel at various injection timings at an injector opening pressure of 190 bar, is shown in Fig. 4

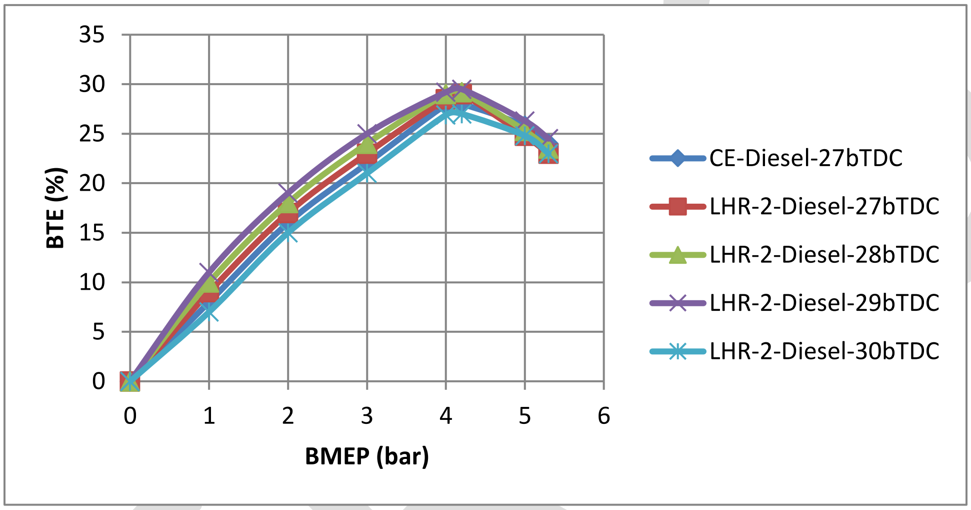


Fig.3 variation of brake thermal efficiency (BTE) with brake mean effective pressure (BMEP) in the engine with LHR–2 combustion chamber with neat diesel, at various injection timings at an injector opening pressure of 190 bar.

BTE increased up to 80% of the full load in the LHR–2 combustion chamber at the recommended injection timing and beyond this load, it decreased over and above that of the conventional engine. As the combustion chamber was insulated to greater extent, it was expected that high combustion temperatures would be prevalent in LHR 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. Higher value of BTE at all loads including 100% full load was observed when the injection timing was advanced to 29° bTDC in the LHR–2 combustion chamber. Further advancing of the injection timing resulted in increase in fuel consumption due to longer ignition delay. Hence it was concluded that the optimized performance of the LHR-2 combustion chamber was achieved at an injection timing of 29°bTDC.

3.2 Exhaust Emissions

Particulate emissions and nitrogen oxide (NOx) levels are the emissions from diesel engine cause health hazards like inhaling of these pollutants cause severe headache, tuberculosis, lung cancer, nausea, respiratory problems, skin cancer, hemorrhage, etc. [15–17]. The contaminated air containing carbon dioxide released from automobiles reaches ocean in the form of acid rain, there by polluting water. Hence control of these emissions is an immediate task and important.

A rich fuel–air mixture resulted in higher smoke because of the availability of oxygen was less. Fig.4 indicates that particulate emissions increased from no load to full load in both versions of the combustion chamber. During the first part, the smoke level was 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 more soot density. The variation of smoke levels with the BMEP, typically showed a inverted L-shaped 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, marginal reduction of particulate emissions was observed in the engine with LHR–2 combustion chamber, when compared to the conventional engine. This was due to the increased oxidation rate of soot in relation to soot formation. Higher surface temperatures of engine with LHR–2 combustion chamber aided this process. Soot formation and buildup in the engine cylinder was also a very important consideration. Soot is formed during combustion in low oxygen regions of the flames. Engine with LHR–2 combustion chamber shorten the delay period, which curbs thermal cracking, responsible for soot formation.

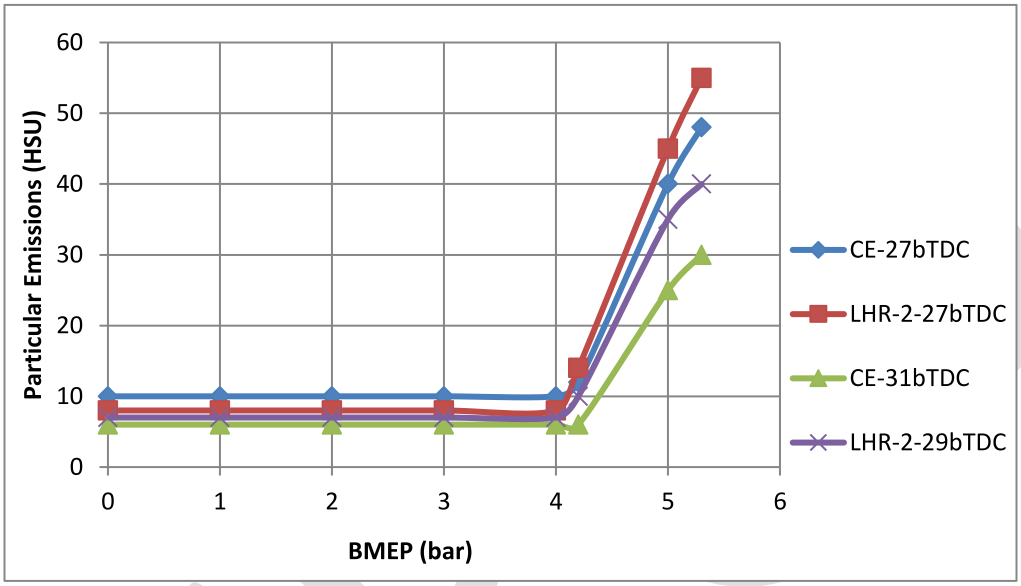


Fig.4. Variation of particular emissions in Hartridge Smoke Unit (HSU) with brake mean effective pressure (BMEP) in conventional engine (CE) and engine with LHR-2 combustion chamber at recommended injection timing and at optimized injection timing at an injector opening pressure of 190 bar.

Beyond 80% of full load, marginal and slight increase of particulate emissions was observed in the LHR-2 combustion chamber, when compared to conventional engine. This was due to fuel cracking at higher temperature, leading to increase in smoke density. Higher temperature of LHR–2 combustion chamber produced increased rates of both soot formation and burn up. The reduction in volumetric efficiency and air-fuel ratio were responsible factors for increasing particulate emissions in the engine with LHR–2 combustion chamber near the full load operation of the engine. As expected, smoke increased in the LHR–2 combustion chamber because of higher temperatures and improper utilization of the fuel consequent upon predominant diffusion combustion. Particulate emissions decreased with advanced injection timing at all loads with engine with both versions of the combustion chamber. This was due to increase of contact period with fuel with air and thus improving atomization characteristics in both versions of the combustion chamber. Higher combustion temperatures are also conducive for reducing particulate emissions. Fuel cracking reactions were eliminated with LHR–2 combustion chamber due to low combustion temperatures. This confirmed improvement in fuel utilization with the injection timing of 29^{o} bTDC. Rama Mohan also observed the similar trends.[7]

Fig.5 indicates that engine with LHR-2 combustion chamber increased particulate emissions at full load by 25% at 27^{o} bTDC and 33% at 29^{o} bTDC in comparison with CE at 27^{o} bTDC and 31^{o} bTDC. This was due to reduction of ignition delay with engine with LHR–2 combustion chamber at 27^{o} bTDC and increased injection timing advance with CE in comparison with insulated engine.

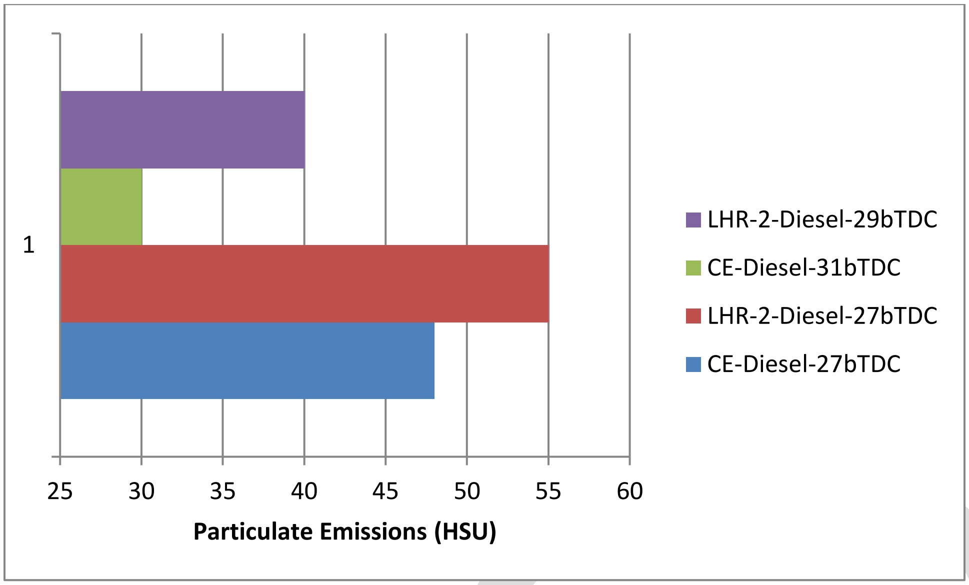


Fig.5. Bar charts showing the variation of particulate emissions at full load with injection timing with both versions of the combustion chamber

The temperature and availability of oxygen are the reasons for the formation of NO_x. For both versions of the combustion chamber, Fig.6 indicates that NO_x concentrations raised steadily as the fuel/air ratio increased with increasing BP/BMEP, at constant injection timing.

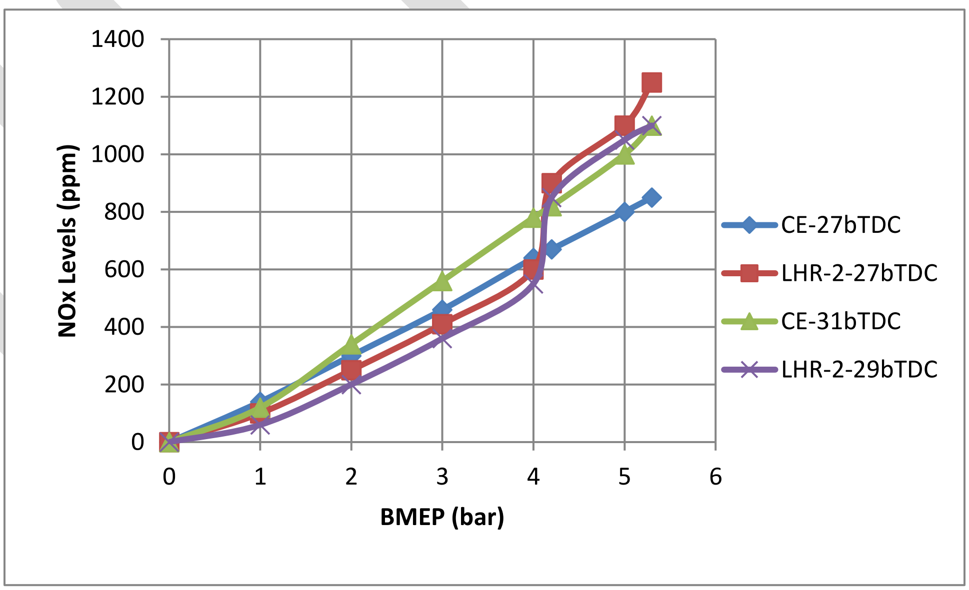


Fig.6. Variation of nitrogen oxide (NO_x) levels with brake mean effective pressure (BMEP) in conventional engine (CE) and engine with LHR-2 combustion chamber at recommended injection timing and at optimized injection timing at an injector opening pressure of 190 bar.

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, NOx 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 NO_x emissions should be roughly proportional to the mass of fuel injected (provided burned gas pressures and temperature do not change greatly).

The LHR-2 combustion chamber recorded lower NOx levels up to 80% of the full load, and beyond that load it produced higher NOx levels compared to conventional engine. As the air-fuel ratios were higher in the LHR-2 combustion chamber, causing more dilution, due to mixing with the excess air, leading to produce less NOx concentrations, up to 80% of the full load, when compared to CE. Beyond 80% of full load, due to the reduction of fuel-air equivalence ratio with LHR–2 combustion chamber, which was approaching to the stoichiometric ratio, causing higher value of NOx levels. NOx emissions increased with advanced injection timing with CE. Increasing the injection advance resulted in higher combustion temperatures and increase of resident time leading to produce higher value of NOx levels in the exhaust of conventional engine at its optimum injection timing. However, NOx levels decreased with advanced injection timing with engine with LHR-2 combustion chamber with diesel. This was due to decrease of combustion temperatures with improved air fuel ratios. Rama Mohan reported the similar trend with NO_x emissions in the LHR engine at the recommended and optimum injection timings.[8].

Fig.7 indicates that engine with LHR-2 combustion chamber increased NO_x levels by 45% at 27^o bTDC and comparable at 29^o bTDC when compared with CE at 27^o bTDC and at 31^o bTDC. This was due to increase of peak pressures in the LHR-2 combustion chamber at 27^o bTDC and resident time with CE.

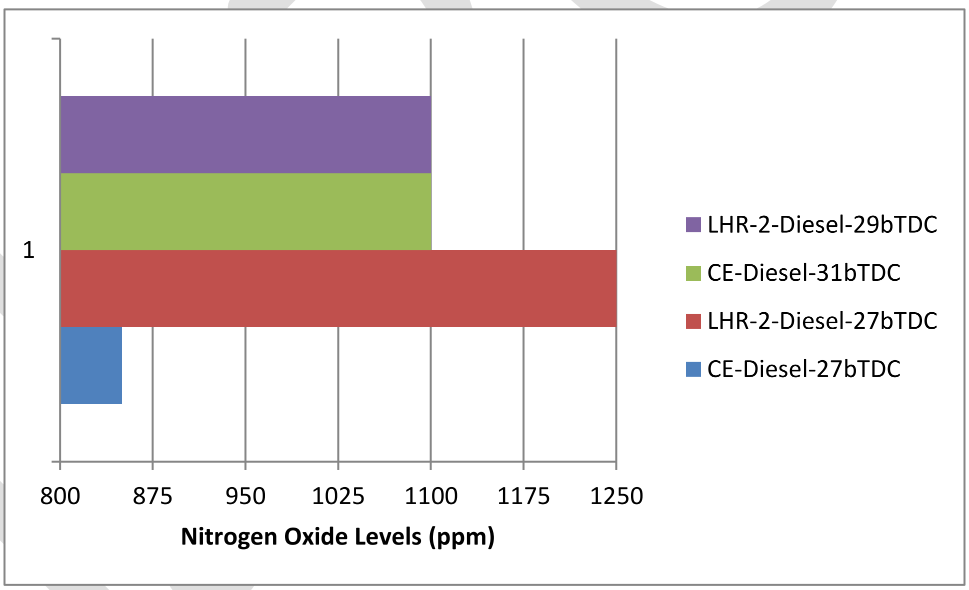


Fig.7 Bar charts showing the variation of nitrogen oxide levels (NO_x) at full load with injection timing with both versions of the combustion chamber

3.3 Combustion Characteristics

From Fig. 8, it is observed that peak pressure at full load operation increased with engine with LHR-2 combustion chamber at 27^{o}bTDC in comparison with CE. This was due to high explosion of charge in hot environment provided by LHR combustion chamber. This was also because the LHR-2combustion chamber exhibited higher temperatures of combustion chamber walls leading to continuation of combustion, giving rise higher peak pressures. PP increased with CE, while decreasing the same with engine with LHR-2 combustion chamber with advanced injection timings. This was due to explosion of accumulated charge with increase of ignition delay with CE, and improved combustion with improved air fuel ratios with which gas temperatures and peak pressures decreased in LHR-2 version of the combustion chamber. Increase of NO_x emissions with CE and decrease the same with engine with LHR-2 combustion chamber with advanced injection timings established the fact that PP at full load operation increased with CE, while decreasing the same with insulated engine with advanced injection timing. Engine with LHR-2 combustion chamber increased peak pressure at full load by 20% at 27° bTDC and 7% at 29° bTDC when compared with CE at 27° bTDC and at 31° bTDC.

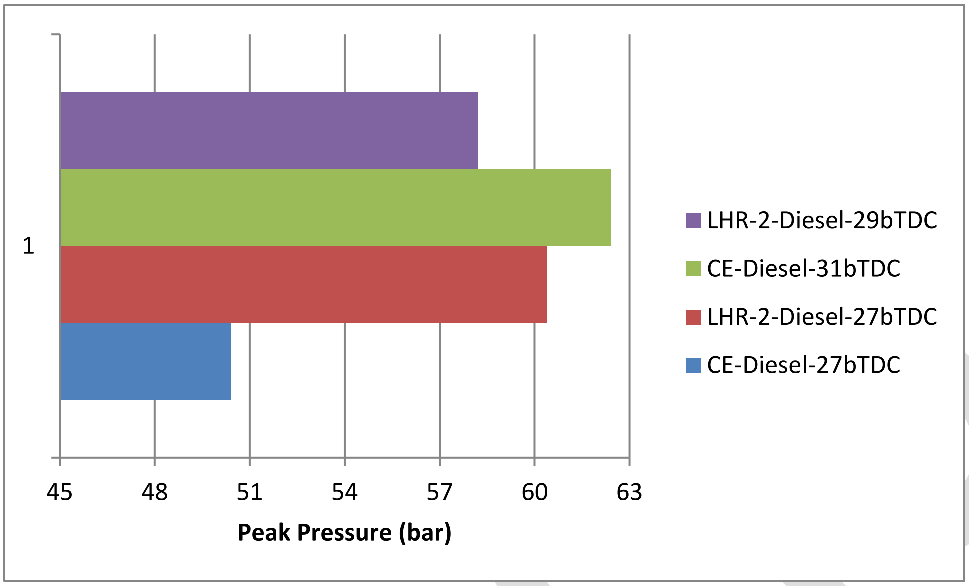


Fig.8 Bar charts showing the variation of peak pressure at full load with injection timing with both versions of the combustion chamber

Fig.9 indicates that Maximum rate of pressure raise (MRPR) at full load followed the similar trends with peak pressure in both versions of the combustion chamber. The trends observed by the authors on the aspect of MRPR in LHR-2 combustion chamber agreed well with the findings of Rama Mohan at the recommended injection timing.[8]. Engine with LHR-2 combustion chamber increased MRPR at full load by 33% at 27° bTDC and 13% at 29° bTDC when compared with CE at 27° bTDC and at 31° bTDC. This was due to reduction of ignition delay with insulated engine.

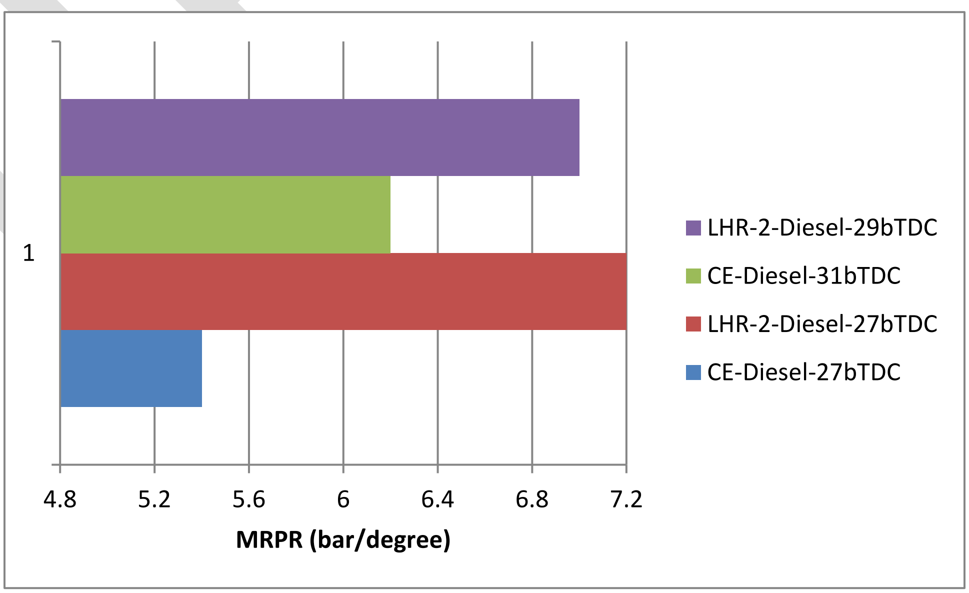


Fig.9 Bar charts showing the variation of maximum rate of pressure rise (MRPR) at full load with injection timing with both versions of the combustion chamber

From Fig.10, it is observed that time of occurrence of peak pressure (TOPP) at full load decreased (shifted towards TDC) with the advanced injection timing and in both versions of the combustion chamber. This was confirmed that both versions of the combustion chamber showed improvement in performance, when the injection timings were advanced to their optimum values. Engine with LHR-2 combustion chamber increased TOPP at full load by 11% at 27º bTDC and 12% at 29º bTDC when compared with CE at 27º bTDC and at 31º bTDC. This was due to continuation of combustion with hot insulated components of LHR-2 combustion chamber giving TOPP away from TDC in comparison with CE. .

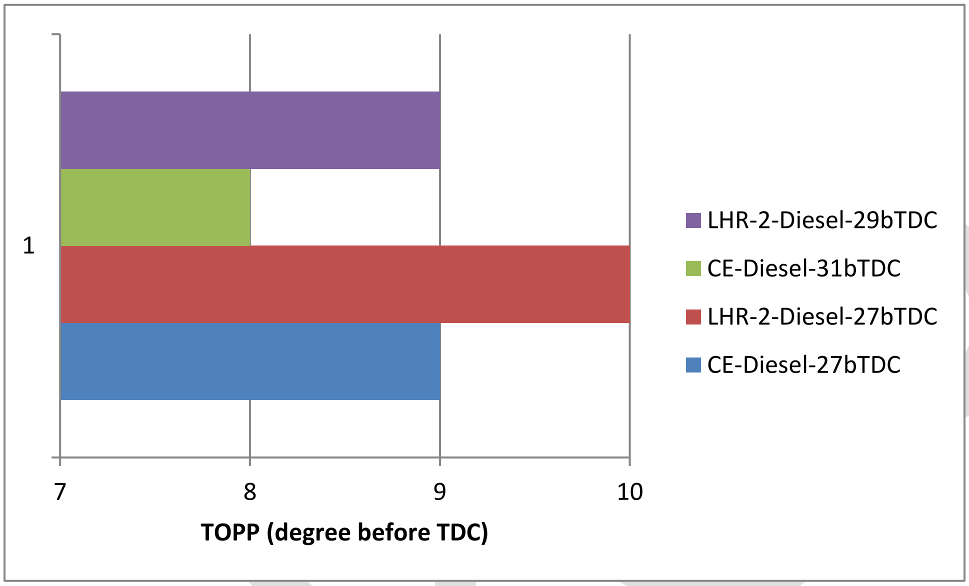


Fig.10 Bar charts showing the variation of time of occurrence of peak pressure (TOPP) at full load with injection timing with both versions of the combustion chamber

4. CONCLUSIONS

4. **Engine with LHR-2 combustion chamber showed improved exhaust emissions of particulate emissions and NO_x levels at 80% of the full load operation at 27 º bTDC in comparison with conventional engine at 27 º bTDC.**
5. **Engine with LHR-2 combustion chamber showed increased particulate emissions and NOx levels at the full load operation at 27 º bTDC in comparison with conventional engine at 27 º bTDC.**
6. **At full load operation, engine with LHR-2 combustion chamber at 29º bTDC, decreased particulate emissions by 27%, NOx levels by 12% , decreased peak pressure by 3%, MRPR by 3% and TOPP by 10% in comparison with same configuration of combustion chamber at an injection timing of 27 º bTDC.**
7. **AT full load operation, conventional engine at 31º bTDC, decreased particulate emissions by 38%, NOx levels by 29%, peak pressure by 24%, MRPR by 15% and decreased TOPP by 11% in comparison with CE at an injection timing of 27 º bTDC.**

4.1 Research Findings

Comparative studies on exhaust emissions and combustion characteristic with direct injection diesel engine with LHR–2 combustion chamber and conventional combustion chamber were determined at varied injection timing with neat diesel operation.

4.2 Future Scope of Work

Hence further work on the effect of injector opening on pressure with engine with LHR–2 combustion chamber with diesel operation is necessary. Studies on performance parameters with varied injection timing and injection pressure with neat diesel operation on engine with LHR-2 combustion chamber can be taken up.

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