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Effect of injection timing on exhaust emissions and combustion characteristics of direct injection diesel engine with high grade insulated combustion chamber

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ABSTRACT

Experiments were carried out to study exhaust emissions and combustion characteristics of diesel engine with high grade heat rejection (LHR–3) 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 and ceramic coated cylinder head with neat diesel with varied injection timing. Combustion diagnosis was carried out by using miniature Piezo electric transducer, TDC (top dead centre) and special pressure-crank angle software package at full load operation of the engine. 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–3 combustion chamber and compared with neat diesel operation on conventional engine (CE) at similar operating conditions. The optimum injection timing was found to be 31^o bTDC (before top dead centre) with conventional engine, while it was 28^o bTDC for engine with LHR–3combustion chamber with diesel operation. Engine with LHR–3 combustion chamber with neat diesel operation showed increased particulate emissions and NO_x levels at manufacturer's recommended injection timing of 27^o bTDC, and the exhaust emissions and combustion characteristics improved marginally with advanced injection timing of 28^o bTDC in comparison with same version of the combustion chamber at 27^{o}bTDC.

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

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 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 (an alloy of nickel), 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. Hot combustion chamber was more suitable for burning high viscous vegetable oils. Investigations were carried out on single cylinder four-stroke water cooled diesel engine of 3.68 brake power at a speed of 1500 rpm at a compression ratio of 16:1 with engine with LHR–3 combustion chamber consisting of air gap insulated piston with superni crown, air gap insulated liner with superni insert and ceramic coated cylinder head with crude vegetable oils as alternative fuels with varied injection timing and pressure. [4–6]. Engine with LHR–3combustion chamber decreased particulate emissions by 10–15% and increased NO_x levels by 40–45% with crude vegetable oils in comparison with CE with mineral diesel operation. Exhaust emissions and combustion characteristics were further improved with an increase of injection pressure and advanced injection timing. Crude vegetable oils were converted to biodiesel by esterification in order to reduce viscosity and improve cetane value. Experiments were conducted on same configuration of the engine as specified in Ref [4–6] with biodiesel with varied injection timing and injection pressure. Particulate emissions decreased by 25–30% and NO_x levels increased by 45–50% with biodiesel operation with LHR–3 combustion chamber.[7–9]. Experiments were conducted with engine as specified in Ref [4–6] with different combustion chambers of LHR–1, LH–2 and LHR–3 with crude vegetable oils and biodiesel with varied injection pressure at injection timing of 27^o bTDC [10–12;13]. It was reported from their studies that particulate emissions decreased, while increasing NO_x levels with increase of degree of insulation and further improved with increase of injection pressure. However, no systematic investigations were reported on comparative performance of the engine with LHR–3 combustion chamber with mineral diesel with varied injection timing. The present paper attempted to study exhaust emissions and combustion characteristics of high grade LHR combustion chamber, which consisted of air gap insulated piston, air gap insulated liner and ceramic coated cylinder head fuelled with mineral diesel with varied injection timing. Comparative performance studies were made on engine with LHR–3 combustion chamber with conventional engine with diesel operation.

2.MATERIALS AND METHODS

The physical-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^{o}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)	^{o}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.

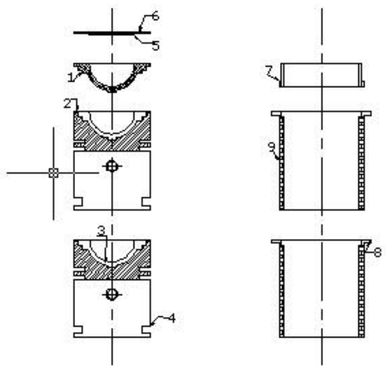

Figure.1. Schematic diagram of assembly the insulated piston, insulated liner and ceramic coated cylinder head of the engine with LHR–3 combustion chamber .

1.Superni crown with threads, 2. Superni gasket, 3.Air gap in piston, 4. Body of the piston, 5. Ceramic coating on inside portion of cylinder head, 6. Cylinder head, 7. Superni insert with threads, 8. Air gap in liner and 9. Body of liner 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. Partially stabilized zirconium (PSZ) of thickness 500 microns was coated on inside portion of cylinder head by means plasma arc coating. The combination of low thermal conductivity materials of superni, air and PSZ offers thermal resistance in the path of coolant. At 500^{o}C thermal conductivities of superni-90, air and PSZ are 20.92,0.057 and 2.01 W/m-K The test fuel used in the experimentation was neat diesel. The schematic diagram of the experimental setup with diesel operation is shown in Fig.2

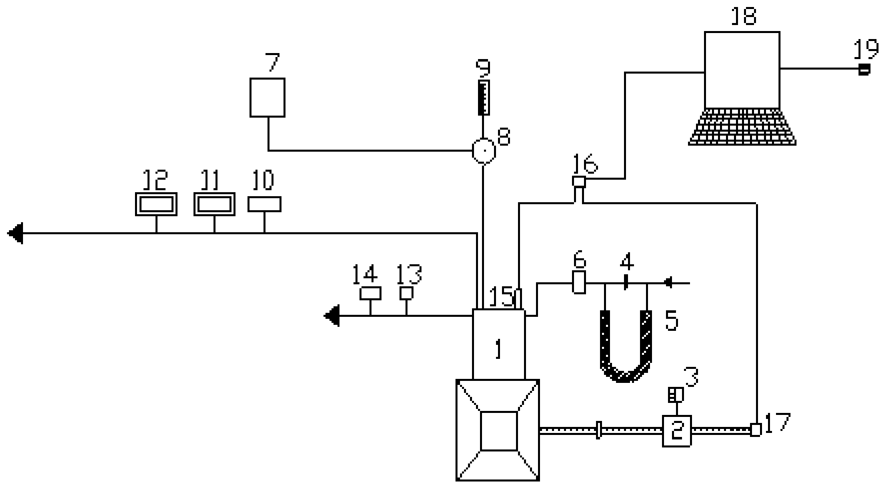

Figure.2. Schematic diagram of experimental set–up

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 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).

Table.2. Specifications of the Test Engine

Description	Specification
Engine make and model	Kirloskar (India) AV1
Maximum power output at a speed of 1500 rpm	3.68 kW
Number of cylinders ×cylinder position× stroke	One × Vertical position × four-stroke
Bore × stroke	80 mm × 110 mm
Method of cooling	Water cooled
Rated speed (constant)	1500 rpm
Fuel injection system	In-line and direct injection
Compression ratio	16:1
BMEP @ 1500 rpm	5.31 bar
Manufacturer's recommended injection timing and pressure	27^{o}bTDC × 190 bar
Dynamometer	Electrical dynamometer
Number of holes of injector and size	Three × 0.25 mm
Type of combustion chamber	Direct injection type
Fuel injection nozzle	Make: MICO-BOSCH No- 0431-202-120/HB
Fuel injection pump	Make: BOSCH: NO- 8085587/1

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. The naturally aspirated engine was provided with water-cooling system in which outlet temperature of water is 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. The naturally aspirated engine was provided with water-cooling system in which outlet temperature of water is maintained at 80°C 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 performance of the engine was studied. Exhaust emissions of particulate matter and nitrogen oxides (NO_x) were recorded by smoke opacity meter (AVLIndia, 437; Part No.11) and NO_x Analyzer (Netel India ;4000 VM; Part No.12) at various values of brake mean effective pressure 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; Part No.15), fitted on the cylinder head to measure pressure in the combustion chamber was connected to a console (Part No.16), which in turn was connected to Pentium personal computer. TDC (top dead centre) encoder (AVL Austria: 365x; Part No.17) 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 (Part No.18). 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.

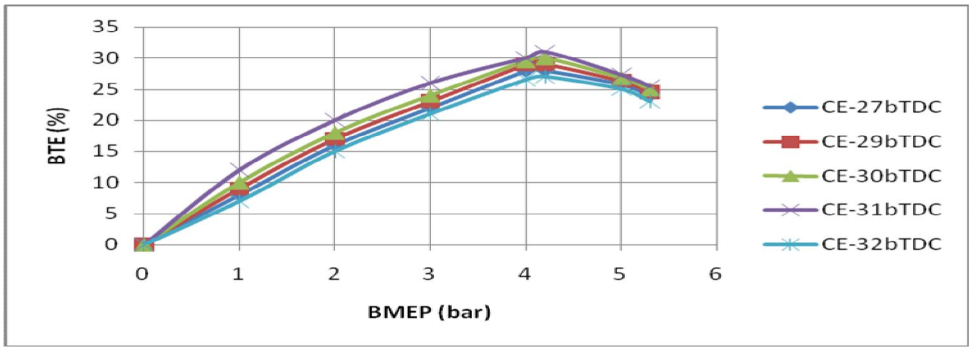

Figure.3 variation of brake thermal efficiency (BTE) with brake mean effective pressure (BMEP) in the conventional engine with neat diesel, at various injection timings at an injector opening pressure of 190 bar.

BTE increased with an increase of BMEP up to 80% of the full load, and beyond that load, it decreased with neat diesel operation. This was because increased fuel conversion efficiency and volumetric efficiency up to 80% of the full load,. Decrease of fuel conversion efficiency, mechanical efficiency and oxygen–fuel ratios were responsible for deterioration of the performance beyond 80% of the full load. BTE increased at all loads with advanced injection timings in the conventional engine, due to early initiation of combustion and increase of contact period of fuel with air leading to improve oxygen– 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. Fig.4, indicates that engine with LHR–3 combustion chamber with mineral diesel showed deteriorated performance at all loads, when compared with CE at an injection timing of in27° bTDC. This was because of reduction of ignition delay, which reduced pre–mixed combustion as a result of which, less time was available for proper mixing of diesel and air and diesel fuel leading to incomplete combustion. More over at full load, increased diffusion combustion and friction resulted from reduced ignition delay.

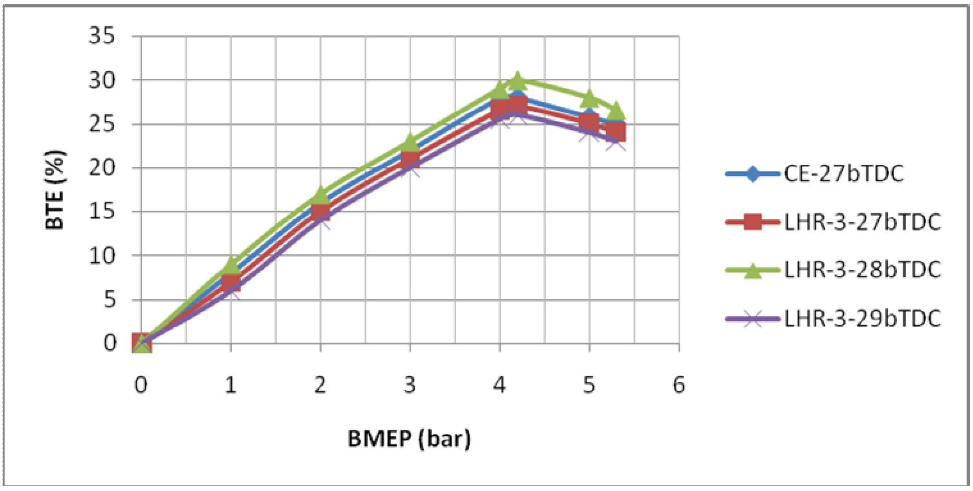

Figure.4. Variation of brake thermal efficiency (BTE with engine with LHR–3 combustion chamber with neat diesel, at various injection timings at an injector opening pressure of 190 bar.

Increased radiation losses were one of the reasons for the deterioration. BTE increased with advanced injection timing at all loads with mineral diesel with engine with LHR–3 combustion chamber. This was because of increase of atomization of fuel with advanced injection timing. Peak BTE was increased by 6% at its optimum injection timing of 28^{o} bTDC, in comparison with CE at 27^{o} bTDC. Earlier researcher on this aspects made similar observations [7].

3.2 Exhaust Emissions

Particulate emissions and nitrogen oxide (NO_x) 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. [14–16]. 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. 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 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 particulate emissions was due to less available oxygen, causing the decrease of air-fuel ratio, leading to incomplete combustion, producing more soot density. The variation of particulate emissions 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. At 27 ^{o} bTDC, engine with LHR–3 combustion chamber increased particulate emissions at all laods, when compared with CE. This was due to fuel cracking at higher temperatures prevailing in hot combustion chamber provided by LHR engine. 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.

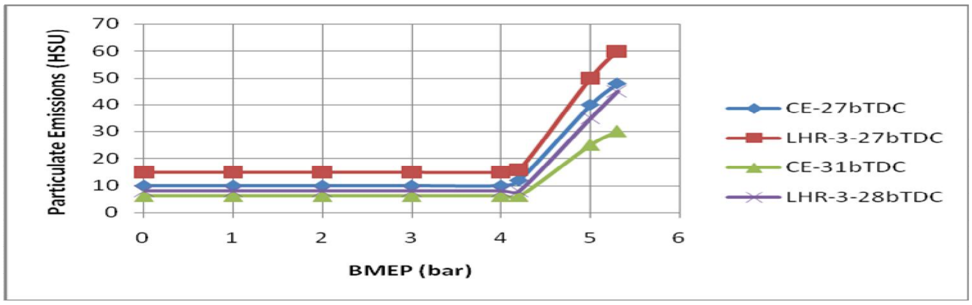

Figure.5. Variation of particulate emissions in Hartridge Smoke Unit with brake mean effective pressure (BMEP) with engine with both versions of the combustion chamber at recommended and optimized injection timing.

Fig.6 indicates that engine with LHR–3 combustion chamber increased particulate emissions at full load by 25% at 27^{o} bTDC and 50% at 28^{o} bTDC, when compared with CE at 27^{o} bTDC and 31^{o} bTDC. This was due to higher injection advance with CE than engine with LHR–3 combustion chamber. This was also due to reduction of volumetric efficiency with heating of air with insulated components of LHR–3 combustion chamber.

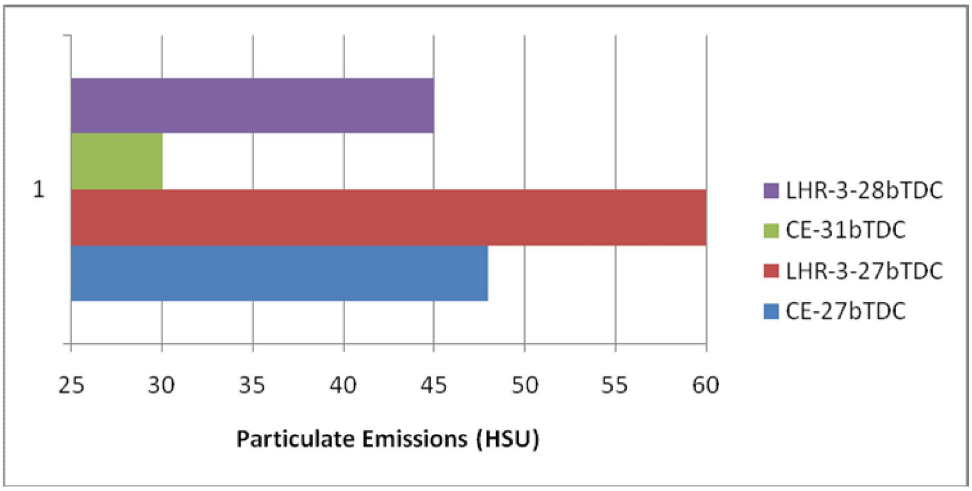

Figure.6. Bar charts showing the variation of particulate emissions at full load with injection timing with both versions of the combustion chamber

The temperature and availability of oxygen are the reasons for the formation of NO_x. For both versions of the combustion chamber, Fig.7 indicates that NO_x concentrations raised steadily as the fuel/air ratio increased with increasing BP/BMEP, at constant injection timing. At part load, NO_x concentrations were less in both versions of the engine. This was due to the availability of excess oxygen. At remaining loads, 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). At 27^o bTDC, engine with LHR–3 combustion chamber increased NO_x levels at all loads in comparison with CE. This was due to increased heat release rate with insulated engine. NO_x 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 NO_x levels in the exhaust of conventional engine at its optimum injection timing. However, NO_x levels decreased with advanced injection timing with engine with LHR–3 combustion chamber with diesel. This was due to decrease of combustion temperatures with improved air fuel ratios

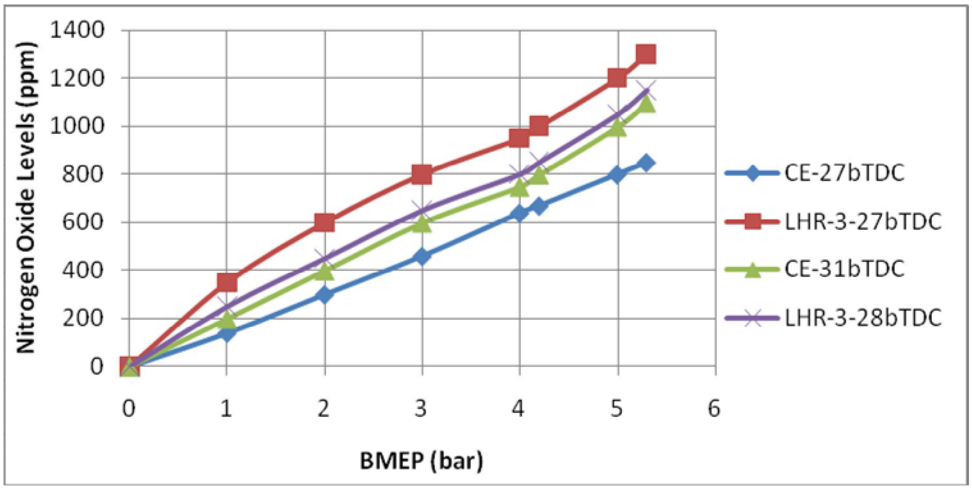

Figure.7. Variation of nitrogen oxide levels with brake mean effective pressure (BMEP) with engine with both versions of the combustion chamber at recommended and optimized injection timing.

Fig.8 indicates that engine with LHR–3 combustion chamber increased NO_x levels by 53% at 27^o bTDC and 5% at 28^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–3 combustion chamber at 27^o bTDC and increased injection advance or resident time with CE.

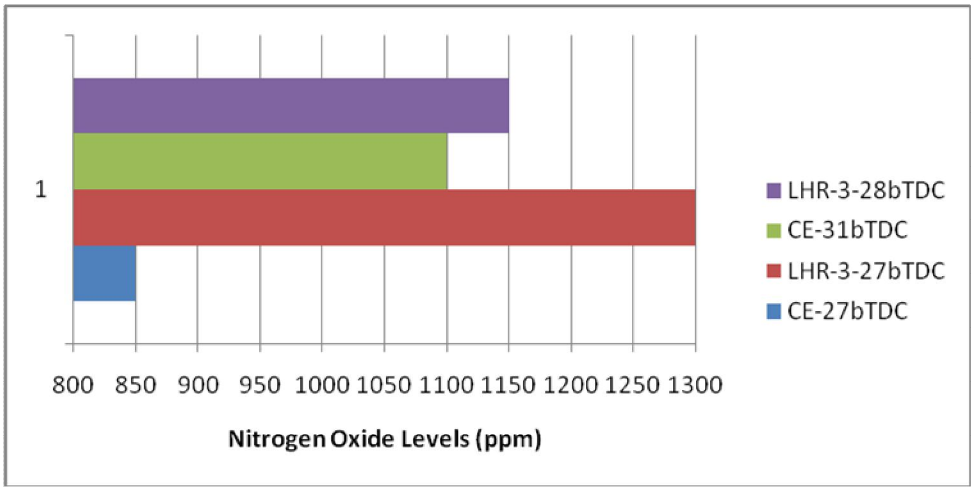

Figure.8. Bar charts showing the variation of nitrogen oxide levels at full load with injection timing with both versions of the combustion chamber

3.3 Combustion Characteristics

From Fig. 9, it is observed that peak pressure at full load operation increased with engine with LHR–3 combustion chamber at 27^o bTDC in comparison with CE.

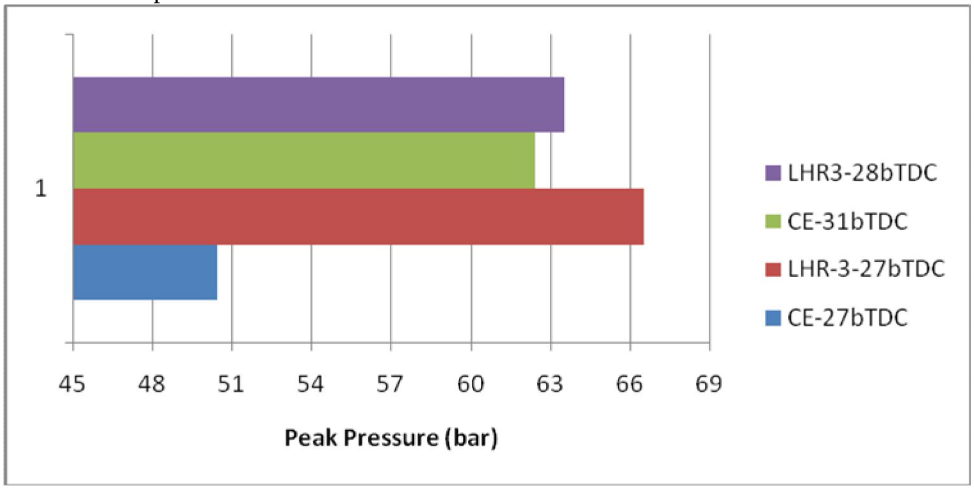

Figure.9 Bar charts showing the variation of peak pressure at full load with injection timing with both versions of the combustion chamber

This was due to high explosion of charge in hot environment provided by LHR combustion chamber. This was also because the LHR–3 combustion 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-1 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–3 version of the combustion chamber. Increase of NO_x emissions with CE and decrease the same with engine with LHR–3 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–3 combustion chamber increased peak pressure at full load by 32% at 27^o bTDC and 2% at 28^o bTDC when compared with CE at 27^o bTDC and at 31^o bTDC. Fig.10 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. Engine with LHR–3 combustion chamber increased MRPR at full load by 41% at 27^o bTDC and 16% at 28^o bTDC when compared with CE at 27^o bTDC and at 31^o bTDC. This was due to reduction of ignition delay with insulated engine.

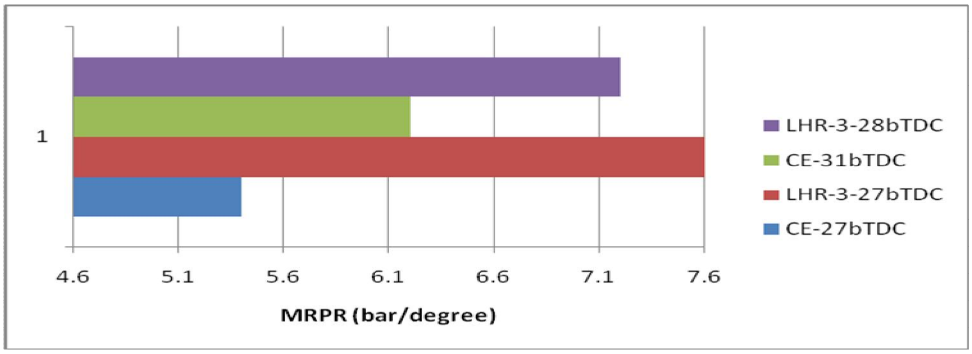

Figure.10 Bar charts showing the variation of maximum rate of pressure rise (MRPR) at full load with injection timing with both versions of the combustion chamber

From Fig.11, 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

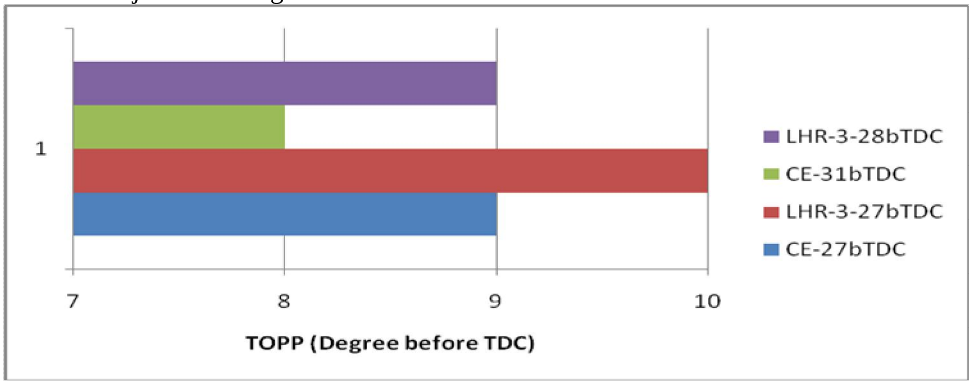

Figure.11 Bar charts showing the variation of time of occurrence of peak pressure (TOPP) at full load with injection timing with both versions of the combustion chamber

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–3 combustion chamber increased TOPP at full load by 11% at 27^o bTDC and 12% at 28^o bTDC when compared with CE at 27^o bTDC and at 31^o bTDC. This was due to continuation of combustion with hot insulated components of LHR–3combustion chamber giving TOPP away

4. CONCLUSIONS

from TDC in comparison with CE.

1. Engine with LHR–3 combustion chamber showed increased pollution levels of particulate emissions and nitrogen oxide levels at 27 ^{o} bTDC in comparison with conventional engine at 27 ^{o} bTDC.
2. Engine with LHR–3 combustion chamber showed increased peak pressure, maximum rate of pressure rise and increased time of occurrence of peak pressure at the full load operation at 27 ^{o} bTDC in comparison with conventional engine at 27 ^{o} bTDC.
3. At full load, engine with LHR–3 combustion chamber at 28^o bTDC, decreased particulate emissions by 25%, NOx levels by 12%, peak pressure by 5%, maximum rate of pressure rise by 5% and time of occurrence peak pressure by 10% in comparison with same configuration of combustion chamber at an injection timing of 27 ^{o} bTDC.

AT full load operation, conventional engine at 31^o bTDC, decreased particulate emissions by 38%, increased NO_x 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 and Suggestions

Comparative studies on performance parameters 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–3 combustion chamber with diesel operation is necessary. Studies on exhaust emissions with varied injection timing and injection pressure with neat diesel operation on engine with LHR–3 combustion chamber can be taken up.

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