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Experimental Investigations on exhaust emissions with an air gap insulated engine

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ABSTRACT

Experiments were carried out to study exhaust emissions of diesel engine withair gap insulated low heat rejection (LHR–2) combustion chamber consisting of air gap insulated piston with 3mm air gap, with superni (an alloy of nickel) crown and air gap insulated liner with superni insert with neat diesel with varied injection timing. Exhaust emissions of particulate emissions, nitrogen oxide (NO_x) levels, carbon monoxide (CO) and un-burnt hydrocarbons (UBHC) were determined at various values of brake mean effective pressure (BMEP) of the LHR-2 combustion chamberand compared with neat diesel operation on conventional engine (CE) at similar operating conditions. 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 pollution levels at manufacturer's recommended injection timing of 27°bTDC, and they improved marginally with advanced injection timing of 29°bTDC in comparison with CE at 27°bTDC. The exhaust emissions from diesel engine cause health hazards and also environmental disorders. Hence, controlling of these emissions in an important and urgent step.

Key words: *Conservation of diesel, Conventional engine, LHR combustion chamber, Performance and exhaust emissions.*

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, conservation of diesel fuel has become pertinent for the engine manufacturers, users and researchers involved in the combustion research (Matthias Lamping *et al.,* 2008).

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 (Cummins and Jr. Lyle, 1993; Avinash Kumar Agarwal *et al*., 2013).

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.

Investigations were carried out to determine air gap thickness for improved performance with diesel fuel in which air gap thickness was maintained at 2-mm (Wallace, *et al.*, 1983). 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.

Experiments were conducted to evaluate 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 (Karthikeyan, *et al.*, 1985). Thenimonic 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.

Experiments were conducted experiments on LHR engine, with an air gap insulated piston, air gap insulated liner and ceramic coated cylinder head (Jabez Dhinagar, *et al.*, 1989). 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 (Parker and Dennison, 1987). 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 (Rama Mohan, *et al.*, 1999). They reported from their investigations that particulate emissions and NO_x levels decreased and improved combustion characteristics with advanced injection timing of 29.5^{o}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 (Vara Prasad, *et al.*, 2000; Srikanth, *et al.*, 2013). They 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 study exhaust emissions of medium grade LHR or LHR-2 engine, 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 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 physico-chemical properties of the diesel fuel are presented in Table 1.

Table 1. Properties of Diesel (Courtesy from IICT, Hyderabad

Property	Units	Diesel
Carbon chain	—	C_8-C_{28}
Cetane Number		55
Density	g/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 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 (Ramamohan *et al.*, 1999). The height of the piston was maintained such that compression ratio was not altered.

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

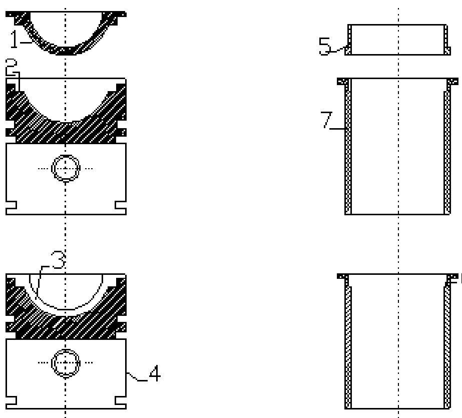
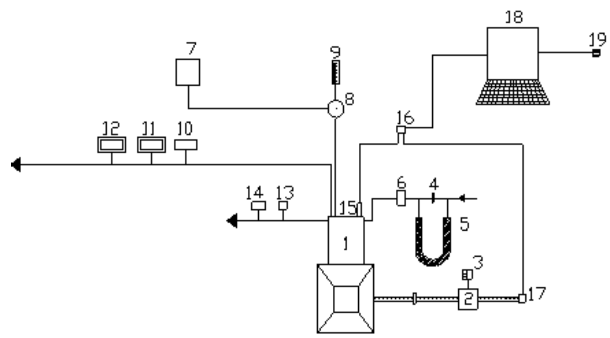


Fig. 1. Assembly details of air gap insulated piston and air gap insulated liner

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

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.

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.



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 80 ^{o}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). Injection timing was varied with an electronic sensor; to evaluate its effect on the exhaust emissions of the engine was investigated.

Exhaust emissions of particulate matter were recorded by smoke opacity meter (AVL India, 437) and CO, UBHC and NO_x were determined by Netel Chromatograph Multi gas analyzer (Netel VM 4000) at various values of brake mean effective pres-

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-BOSCHNo- 0431-202-120/HB
Fuel injection pump	Make: BOSCH: NO- 8085587/1

sure (BMEP) of the engine 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, carbon monoxide levels, un-burnt hydrocarbons and nitrogen oxide 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.

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

Results and Dicussion

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.

BTE increased up to 80% of the full load and beyond that load, it decreased with diesel operation with both versions of the engine at various injection timings. This is due to increase of fuel conversion efficiency, mechanical efficiency and oxygen-fuel ratio. Beyond 80% of the load, BTE decreased due to reduction of the same. BTE increased with the advanced injection timings in the conventional engine at all loads, due to early initiation of combustion and

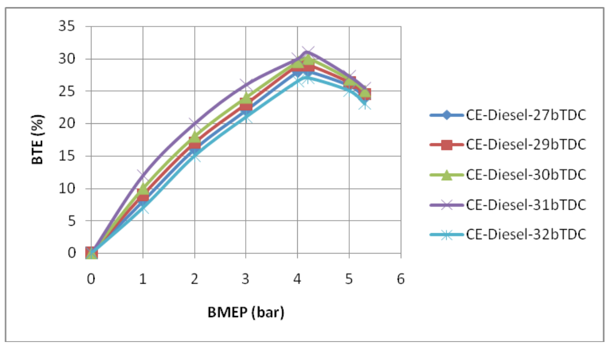


Fig. 3. Variation of brake thermal efficiency (BTE) with brake mean effective pressure (BMEP)

Table 3. Range and accuracy of Analyzers

S.No	Name of the Analyzer	Principle adopted	Range	Accuracy
1	AVL Smoke Analyzer	Opacity	0-100 HSU (Hartridge Smoke Unit)	±1 HSU
2	Netel Chromatograph CO analyzer	Infrared absorption spectrograph	0-10%	± 0.1%
3	Netel Chromatograph UBHC analyzer	NDIR	0-1000 ppm	±5 ppm
4	Netel Chromatograph NO_x analyzer	Chemiluminiscence	0-5000pm	±5 ppm

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.

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.

Exhaust Emissions

Exhaust from diesel engine cause health hazards like inhaling of these pollutants cause severe headache, tuberculosis, lung cancer, nausea, respiratory problems, skin cancer, hemorrhage, etc. (Fulekar, 2004; Khopkar, 2012). 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 shows the variation of particulate emissions with brake mean effective pressure (BMEP) at recommended injection timing (RIT) and optimum injection timing (OIT) with both versions of the engine. This figure 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.

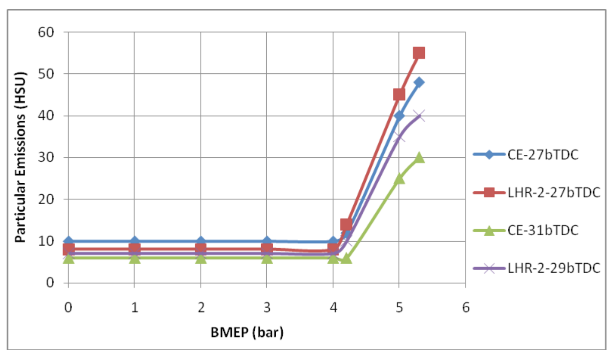


Fig. 4. Variation of particular emissions in Hartridge Smoke Unit (HSU) with brake mean effective pressure (BMEP)

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°bTDC. Similar observations were made by other researcher (Rama Mohan, *et al.*, 1999).

Fig. 5 presents bar chart showing the several of particulate levels with both versions of the engine at full load at recommended injection timing (RIT) and optimum injection timing (OIT).

It indicates that engine with LHR-2 combustion chamber increased particulate emissions at full load by 25% at 27°b TDC and 33% at 29°b TDC in comparison with CE at 27°b TDC and 31°b TDC. This was due to reduction of ignition delay with engine with LHR-2 combustion chamber at 27°b TDC and increased injection timing advance with CE in comparison with insulated engine.

Fig. 6 shows the variation of nitrogen oxide levels (NO_x) particulate emissions with brake mean effective pressure (BMEP) at recommended injection timing (RIT) and an optimum injection timing (OIT) with both versions of the engine. The temperature and availability of oxygen are the reasons for the formation of NO_x Fig. 6 indicates that NO_x concentrations raised steadily as the fuel/air ratio increased with increasing BP/BMEP, at constant injection timing.

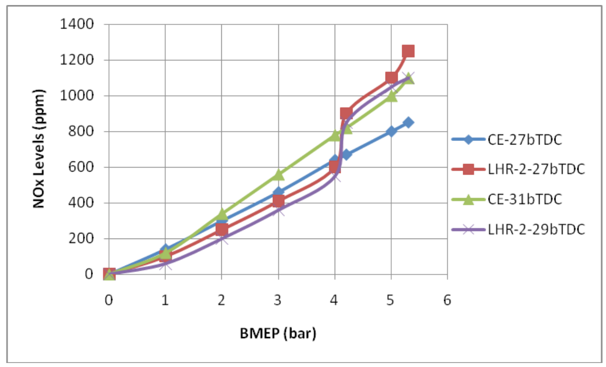


Fig. 6. Variation of nitrogen oxide (NO_x) levels with brake mean effective pressure (BMEP)

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

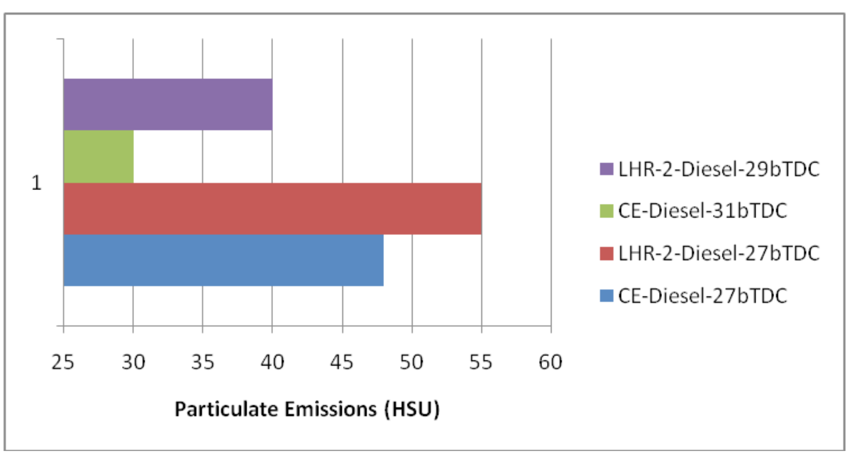


Fig. 5. Bar charts showing the variation of particulate emissions at full load

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 NO_x 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 (Ramamohan *et al*., 1999).

Fig. 7 presents bar chart showing the several of nitrogen oxide levels (NO_x) levels with both versions of the engine at full load at recommended injection timing (RIT) and optimum injection timing (OIT).

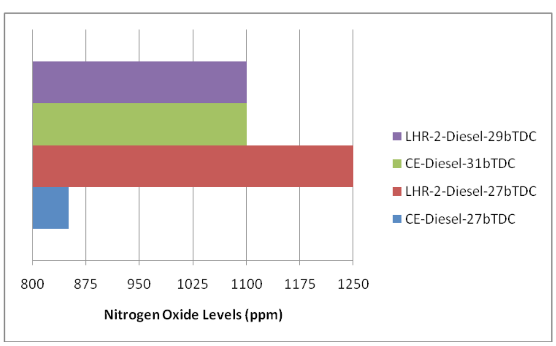


Fig. 7. Bar charts showing the variation of nitrogen oxide levels (NO_x) at full load

Fig.7 indicates that engine with LHR-2 combustion chamber increased NO_x levels by 45% at 27°b TDC and comparable at 29°b TDC when compared with CE at 27°b TDC and at 31°b TDC. This was due to increase of peak pressures in the LHR-2 combustion chamberat 27°b TDC and resident time with CE. NOx levels can be controlled by exhaust gas recirculation (EGR) or selective catalytic reduction technique (SCRT).

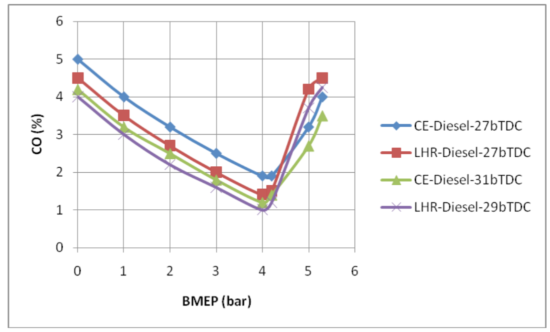


Fig. 8. Variation of carbon monoxide (CO) levels with brake mean effective pressure (BMEP)

Fig. 8 shows the variation of carbon monoxide (CO) levels with brake mean effective pressure (BMEP) at recommended injection timing (RIT) and an optimum injection timing (OIT) with both versions of the engine. CO levels were observed to be high at low loads and at full loads. At low loads, the charge is diluted by residual gases, and hence fuel is enriched at idling of the engine. At full load, in order to develop more power, fuel is enriched. At 80% of the full load, CO emissions were found to be lower, due to improved thermal efficiency. CO emissions decreased with both versions of the engine with advanced injection timing due to improved atomization characteristics of the fuel. LHR engine showed reduction of CO emissions up to 80% of the full load and beyond that load it increased CO emissions than CE with diesel operation because of oxygen-fuel ratio.

Fig.9 presents bar chart showing the several of CO levels with both versions of the engine at full load at recommended injection timing (RIT) and optimum injection timing (OIT). LHR engine increased CO levels by 8% at RIT, and 2% at OIT in comparison with CE with neat diesel operation. This is due increase of heat release rate and faster rate of combustion of fuel. Reduction of ignition delay also contributed an increase of CO levels at full load operation.

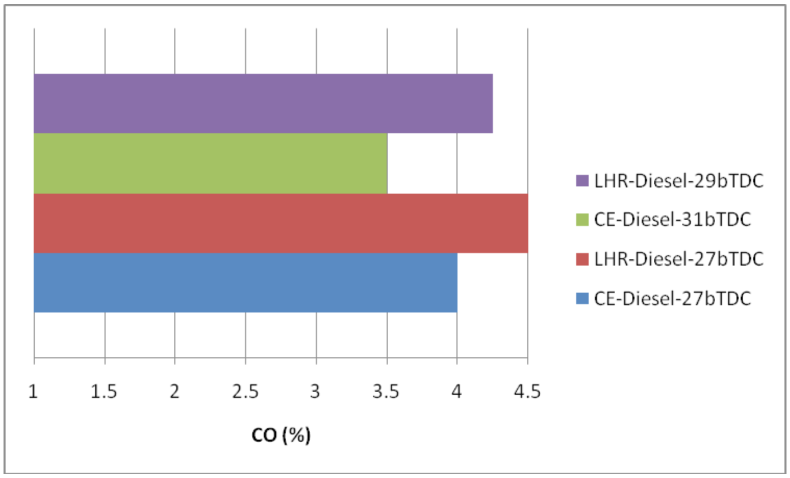


Fig. 9. Bar charts showing the variation of CO levels at full load with injection timing

Fig.10 shows the variation of un burnt hydro carbons (UBHC) levels with brake mean effective pressure (BMEP) at recommended injection timing (RIT) and an optimum injection timing (OIT) with both versions of the engine.

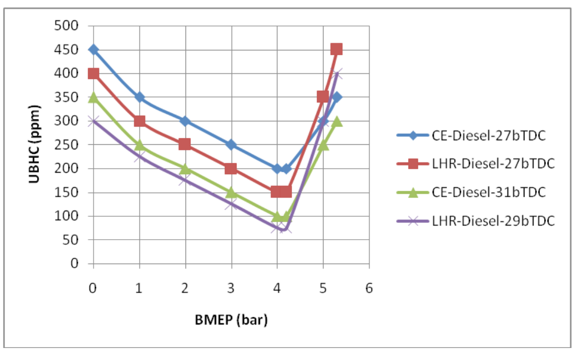


Fig. 10. Variation of un-burnt hydro carbons with brake mean effective pressure (BMEP)

UBHC followed similar trends with CO with both versions of the engine at RIT and OIT. UBHC emissions improved with advanced injection timing with both versions of the engine. This is due to more time available with fuel to react with oxygen leading to improve atomization characteristics of the fuel. LHR engine decreased UBHC emissions up to 80% of the full load and beyond that load, it increased than CE. This is due to improved oxygen-fuel ratio up to 80% of the full load and deterioration of the same beyond 80% of the full load.

Fig. 11 presents bar chart showing the various UBHC levels with both versions of the engine at full load at recommended injection timing (RIT) and optimum injection timing (OIT).LHR engine increased UBHC levels by 33% at RIT, and 33% at OIT in comparison with CE with neat diesel operation. CO is formed due to incomplete combustion, because of improper oxygen-fuel ratio, while UBHC is formed due to accumulation of fuel in crevice volume. LHR engine increased UBHC emissions at full load because of reduction of ignition delay of the fuel. Control of UBHC emissions was provided by a catalytic converter with air injection.

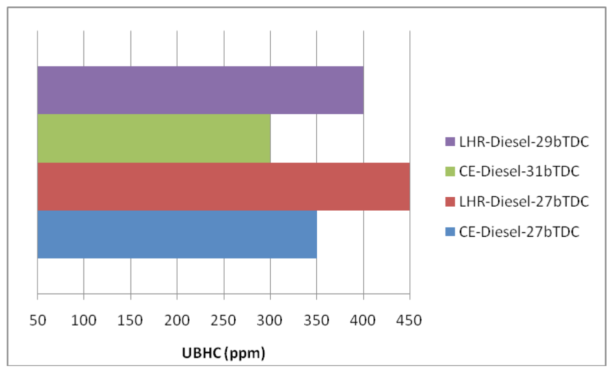


Fig. 11. Bar charts showing the variation of UBHC levels at full load

Conclusion

1. Engine with LHR-2 combustion chamber showed improved pollutants at 80% of the full load operation at 27°b TDC in comparison with conventional engine at 27°TDC.
2. Engine with LHR-2 combustion chamber showed increased particulate emissions, NO_x levels, CO levels and UBHC levels at the full load operation at 27°bTDC and optimum injec-

tion timing in comparison with conventional engine at 27°b TDC.

3. SCRT (Selective catalytic reduction technique) and exhaust gas recirculation (EGR) were suggested to control NO_x levels.
4. Catalytic converter was suggested to control CO emissions and UBHC emissions

Research Findings

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

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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