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*Research Article*

# Experimental Investigations on Di Diesel Engine with High Grade Insulated Combustion Chamber with Varied Injection Timing

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**Abstract**

*Investigations were carried out to evaluate the performance of diesel engine with air gap insulated low heat rejection (high grade LHR or 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. Determined the Performance parameters [brake thermal efficiency, exhaust gas temperature, coolant load, volumetric efficiency and sound levels,]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°bTDC (before top dead centre) with conventional engine, while it was 28° bTDC for engine with LHR–3combustion chamber with diesel operation. Engine with LHR–3 combustion chamber with neat diesel operation showed deteriorated performance at manufacturer's recommended injection timing of 27° bTDC, and the performance improved marginally with advanced injection timing of 28° bTDC in comparison with CE at 27°bTDC.*

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

**Nomenclature**

$\rho_a$ =density of air, kg/m$^3$
$\rho_d$ =density of fuel, gm/cc
$\eta_d$ =efficiency of dynamometer, 0.85
$a$= area of the orifice flow meter, m$^2$
BP=brake power of the engine, kW
$C_d$=coefficient of discharge, 0.65
$Cp$=specific heat of water in kJ/kg K
$D$=bore of the cylinder, 80 mm
$d$=diameter of the orifice flow meter, 20 mm
DI=diesel injection
$I$=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
LHR-3= Insulated combustion chamber with air gap insulated piston, air gap insulated liner and ceramic coated cylinder head
$m_a$=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$^3$
VE=Volumetric efficiency, %

## 1. Introduction

In view of increasing population of vehicles at a speed rate and use of diesel fuel in transport sector agriculture sector etc., leading to fast depletion of diesel fuels. Increase of fuel prices 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. [Matthias Lamping *et al*, 2008].
Dr. Diesel made a mark for his invention of diesel engine. Compression ignition (CI) engines is used in applications like power plants for automotive applications as it is having their excellent fuel efficiency and durability. The most accepted type of engine is internal combustion engine which is used for powering agricultural implements, industrial

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applications, and construction equipment along with marine propulsion. [Cummins *et al,* 1993; Avinash Kumar Agarwal *et al,* 2013].

LHR combustion chamber concept is to reduce heat losses by providing thermal resistance in the path of heat flow to the coolant, there by gaining thermal efficiency. Various other techniques can be used for achieving LHR ceramic coated engines inside the head 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.

Ceramic coating inside the cylinder head are said to be low grade LHR or LHR–1, air gap insulated piston and air gap insulated liner are said to be medium grade LHR or LHR–2 and combination of low grade LHR and medium grade LHR said to be high grade LHR or LHR–3. combustion chambers depending on degree of insulations. Authors conducted with neat diesel operation with ceramic coated diesel engine [Paralak *et al*, 2005; Ekrem *et al,* 2006; Ciniviz *et al,* 2008; Janardhan *et al*, 2014; Janardhan *et al*, 2015]. They revealed that brake specific fuel consumption decreased by 3-4% with ceramic coated diesel engine in comparison with conventional engine. Keeping air gap in the piston given the complications of joining two different metals. Experiments were conducted on air gap insulated piston with neat diesel operation [Parker *et al*, 1987]. Air gap in the piston with bolted design by them could not provide complete sealing of air in the air gap. Screwing the crown made of low thermal conductivity material, superni to the body of the piston, by keeping a gasket, made of superni in between these two parts made them success. [Ramamohan *et al*, 1999; Janardhan *et al,* 2015]. The optimum injection timing was found to be 29.5$^o$ bTDC. BSFC decreased by 12% at part–load and 4% at full load at an injection timing of 29.5° bTDC with the optimized insulated piston engine in comparison with CE operating at an injection timing of 27° bTDC. Investigations were carried out engine with air gap insulated piston, air gap insulated liner and ceramic coated cylinder head(LHR-3) with varied injection timing and injection pressure to study pollution levels of particulate emissions and nitrogen oxide levels. [Janardhan *et al,* 2013]. It is revealed from their investigations that smoke levels were almost negligible and drastically increased $NO_x$ levels. It is realized that burning high viscous vegetable oils, hot combustion chamber is more suitable. Experiments were conducted on single cylinder four-stroke water cooled diesel engine of 3.68 brake power with a speed of 1500 rpm at a compression ratio of 16:1 and 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. [Kesava Reddy *et al*, 2012; Janardhan *et al,* 2012; Chowdary *et al,* 2012]. Using LHR–3combustion chamber improved brake thermal efficiency by 6–8% with crude vegetable oils in comparison with CE with diesel operation. Improvement in the performance was found with an increase of injection pressure and advanced injection timing.

Biodiesel of converted crude vegetable oil by esterification in order to reduce viscosity and improve cetane value. Investigations were carried out on same configuration of the engine as specified in Ref [[Kesava Reddy *et al*, 2012; Janardhan *et al,* 2012; Chowdary *et al,* 2012]with crude vegetable oil based biodiesel with varied injection timing and injection pressure. Improvement was found in the performance with biodiesel operation with LHR–3 combustion chamber.[Krishna Murthy, 2010; Venkateswara Rao *et al,* 2013; Subba Rao *et al,* 2013]. Investigation were carried out with engine as specified in Ref [Kesava Reddy *et al*, 2012; Janardhan *et al,* 2012; Chowdary *et al,* 2012] with different combustion chambers of LHR–1, LHR–2 and LHR–3 with crude vegetable oils and biodiesel with varied injection pressure at injection timing of 27$^o$ bTDC [Muraii Krishna 2004; Kesava Reddy *et al*. 2012; Murali Krishna *et al,* 2012; Ratna Reddy *et al,* 2012].They found improved performance with increasing order of degree of insulation and further improved with increase of injection pressure.

No systematic investigations were reported on comparative performance of the engine with LHR–3 combustion chamber with diesel with varied injection timing.

An attempt is made in the present paper to evaluate the performance of high grade LHR combustion chamber, which consisted of air gap insulated piston, air gap insulated liner and ceramic coated cylinder head fuelled with diesel with varied injection timing. Performance studies were made in comparison with engine with LHR–3 combustion chamber with conventional engine with diesel operation.

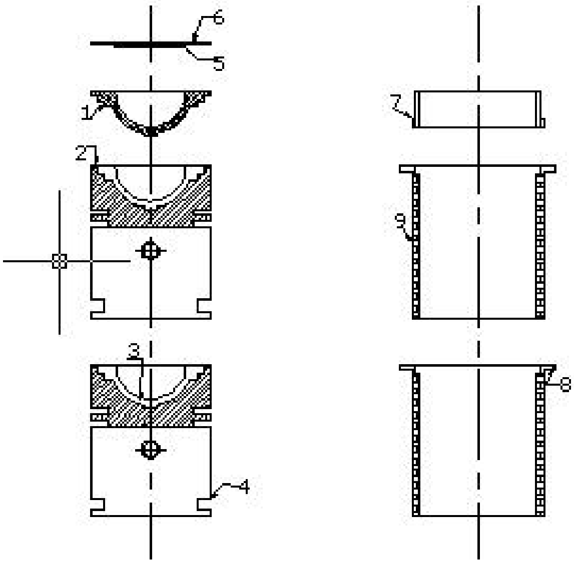
## 2. Materials and Methods

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–3 combustion chamber (Fig.1) consists of 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.



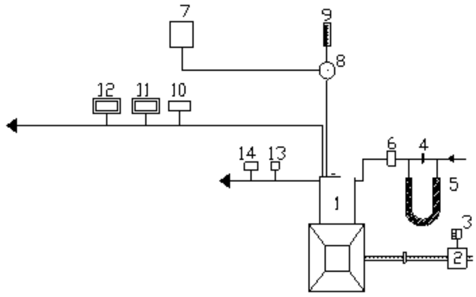

1.Superni crown with threads, 2. Superni gasket, 3.Air gap in piston, 4. Body of the piston, 5. Ceramic coating on inside portion of cylinder head, 6. Cylinder head, 7. Superni insert with threads, 8. Air gap in liner and 9. Body of liner

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

The liner is fitted with superni-90 insert, which was screwed to the top portion liner to maintain 3mm gap between the insert and the liner body.

The inside portion of the cylinder head coated with Partially Stabilized Zirconium (PSZ) of thickness 500 microns was coated by means plasma arc procedure. The combination of low thermal conductivity materials of superni, air and PSZ offers thermal resistance in the path of coolant. At 500°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. The experimental engine specifications are given in Table-2. An electric dynamometer (Part No.2. Kirloskar make) connected to engine for measuring its brake power. Rheostat (Part No.3) is used load the Dynamometer. A direct injection type combustion chamber consisted with no special arrangement for swirling motion of air. Fuel consumption of the engine can be measured by Burette (Part No.9) method 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 Dynamo meter, 3.Load Box, 4.Orifice meter, 5.U-tube water manometer, 6.Air box, 7.Fuel tank, 8, Three way valve, 9.Burette, 10. Exhaust gas temperature indicator, 11.AVL Smoke meter, 12.Netel Chromatograph NOx Analyzer, 13.Outlet jacket water temperature indicator and 14. Outlet-jacket water flow meter.

**Fig.2** Schematic diagram of experimental set–up

**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 |

Water-cooling system is used to cool the engine 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) and the pressure feed system is used to pump the engine oil for lubricating the engine. No temperature control was incorporated, for measuring the lube oil temperature. Iron and iron-constantan thermocouples attached for measuring the exhaust gas temperature and coolant water outlet temperatures to the exhaust gas temperature indicator (Part No.10) and outlet jacket temperature indicator (Part No.13). Since exhaust emissions were not measured in the experiment, part No.11 and Part No.12 were not in use. To vary the injection timing copper shims of suitable size were inserted in between the pump body and the engine frame and its effect on the performance of the engine was studied.

*Operating Conditions*

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

*Definitions of used values:*

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

$$BP = \frac{V \times I}{\eta_d \times 1000} \quad (2)$$

$$BTE = \frac{BP \times 3600}{m_f \times CV} \quad (3)$$

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

$$CL = m_w \times c_p \times (T_o - T_i) \quad (5)$$

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

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

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

$$\rho_a = \frac{P_a \times 10^5}{750 \times R \times T_a} \quad (9)$$

## 3. Results and Discussions

*3.1 Performance Parameters*

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

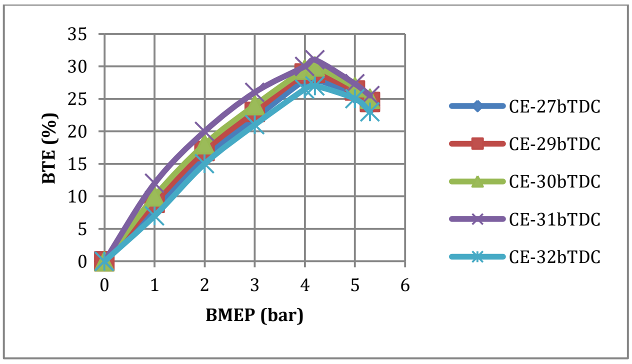



**Fig.3** Brake thermal efficiency (BTE) variation with brake mean effective pressure (BMEP) in the conventional engine with neat diesel, at different injection timings at an injector opening pressure of 190 bar

BTE is raised with an increase of BMEP up to 80% of the full load, and beyond that load, it come down with neat diesel operation. This is due to fuel conversion efficiency and volumetric efficiency increased up to 80% of the full load. Fuel conversion efficiency and mechanical efficiency decreased and oxygen–fuel ratios were responsible for deterioration of the performance beyond 80% of the full load. BTE was improved at all loads with advanced injection timings in the conventional engine as it was early initiation of combustion and increase of contact period of fuel with air leading to improve oxygen– fuel ratios period. Based on maximum brake thermal efficiency the optimum injection timing was obtained. The optimum injection timing was found to be 31°bTDC in CE as it was given Maximum BTE. Performance deteriorated if the injection timing was greater than 31°bTDC as it was increases of ignition delay.

Fig.4, shows that performance was deteriorated engine with LHR–3 combustion chamber at all loads with diesel fuel, when compared with CE at an injection timing of 27° bTDC. Due to reduction of ignition delay, less time was available for proper mixing of air and diesel, leading to incomplete combustion. More over at full load, increased diffusion combustion and friction resulted from reduced ignition delay.

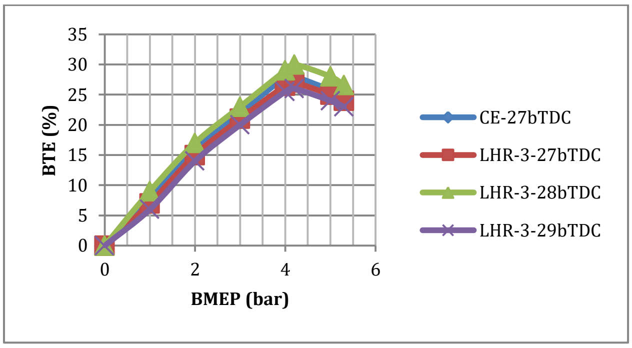



**Fig.4** Variation of brake thermal efficiency (BTE with engine with LHR–3 combustion chamber with neat diesel, at various injection timings at an injector opening pressure of 190 bar.

Increased radiation losses were one of the reasons for the deterioration. Advancement of injection timing increased the BTE at all loads with diesel with LHR–3 combustion chamber. This is due to increase of atomization of fuel with advanced injection timing. Peak BTE was found to be increased by 6% at 28° bTDC optimum injection timing, in comparison with CE at 27° bTDC. Earlier researcher on this aspects made similar observations.

Curves in Fig.5 indicate that optimum injection timing was obtained earlier with engine with LHR-3 combustion chamber at 28bTDC. This is due to hot combustion chamber of LHR engine decreases the ignition delay and combustion duration. It is found that BTE to be higher with CE at the optimum injection timing when compared with engine with LHR-3 combustion chamber. This was due to higher advanced injection timing with CE than engine with LHR-2 combustion chamber.

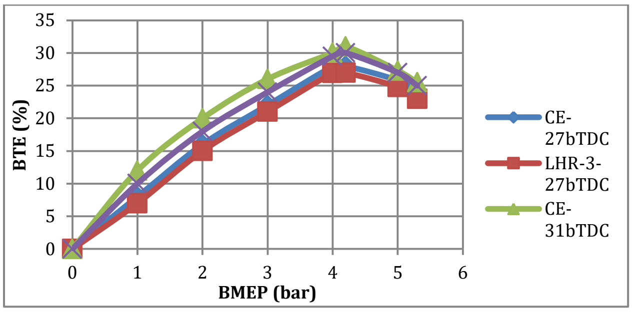



**Fig.5** Brake thermal efficiency variation with brake mean effective pressure effective pressure (BMEP) with conventional engine (CE) and engine with LHR–2 combustion chamber at recommended injection timing and optimum injection timing.

Fig.6 shows that engine with LHR–3 combustion chamber, BTE was decreased peak BTE by 4% at 27°bTDC and 3% at 28° bTDC when compared with CE at 27°bTDC and 31°bTDC. Due to reduction of ignition delay with engine with LHR–3 combustion chamber.

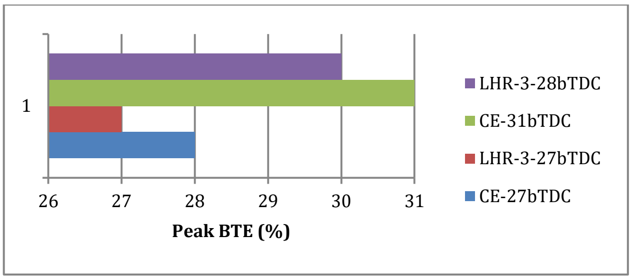



**Fig.6** Bar charts showing the peak brake thermal efficiency variation (%) with conventional engine (CE) and engine with LHR–3 combustion chamber at recommended injection timing and optimized injection timing

To compare the performance of the engine with different versions of the combustion chamber, then brake specific fuel consumption (BSFC) at full load is to be determined. Fig.7 shows that engine with LHR–3 combustion chamber increased BSFC at full load operation by 4% at 27°bTDC and 4% at 28°bTDC when compared with CE at 27°bTDC and 31°bTDC. This was due reduction of ignition delay.

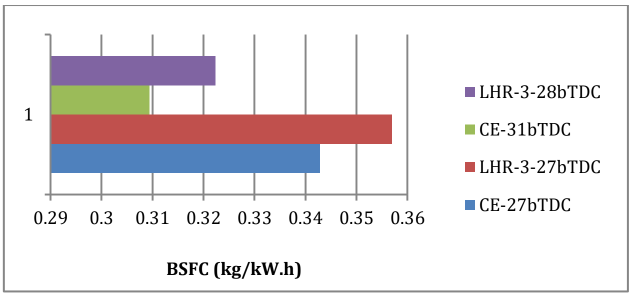



**Fig.7** Bar charts showing the brake specific fuel consumption (BSFC) variation at full load operation with conventional engine (CE) and engine with LHR–3 combustion chamber at recommended injection timing and optimized injection timing

Fig.8 indicates that exhaust gas temperatures (EGT) was found to be increased with an increase of BMEP with both versions of the combustion chamber LHR-3 and CE. The main reason due to increase of fuel consumption with load. EGT was found to be higher with engine with LHR–3 combustion chamber at all loads in comparison with CE. This shows that hot insulated combustion chamber restricted the heat and more amount of heat will be utilized in converting into useful work. EGT was decreased at all loads with advanced injection timing with both versions of the combustion chamber due to improved atomization of fuel, and more time available for gases to expand. This was also due the injection timing was advanced, the work transfer from the piston to the gases in the cylinder at the end of the compression stroke was too large, leading to reduce in EGT.

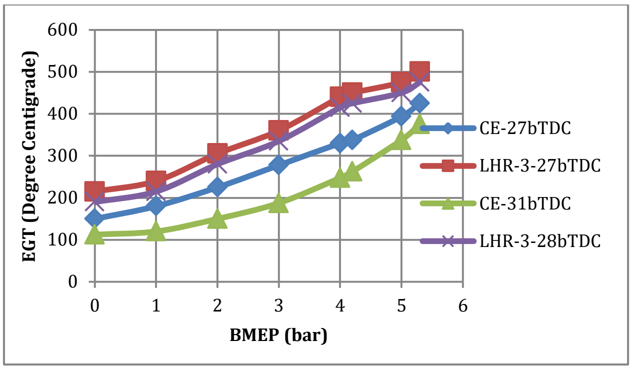



**Fig.8** Exhaust gas temperature (EGT) variation with brake mean effective pressure effective pressure (BMEP) with conventional engine (CE) and engine with LHR–3 combustion chamber at recommended injection timing and optimum injection timing

Fig.9 shows that engine with LHR-3 combustion chamber raised the EGT at full load operation by 18% at 27°bTDC and 20% at 28°bTDC when compared with CE at 27°bTDC and 31°bTDC. It is the due reduction of ignition delay. This was also due to higher injection advance with CE.

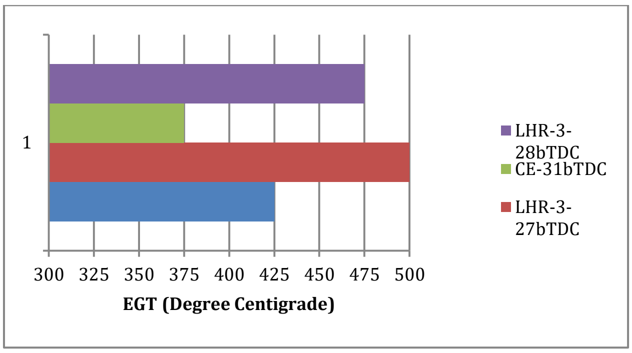



**Fig.9** Bar charts showing the exhaust gas temperature (EGT) variation at full load operation with conventional engine (CE) and engine with LHR–3 combustion chamber at recommended injection timing and optimized injection timing

From Fig.10, it is observed that coolant load increased with the increase of BMEP in the conventional engine and LHR-2 combustion chamber. Coolant load was found to be lower with the LHR–3 combustion chamber at all loads, when compared to conventional engine. Because of the resistance being existed through the piston and liner as Air gap maintained in them.

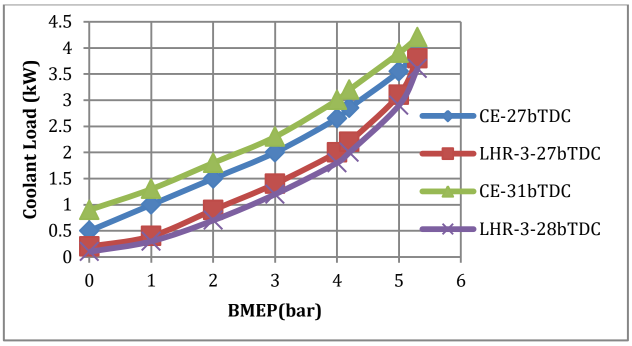



**Fig.10** Coolant load variation with brake mean effective pressure effective pressure (BMEP) with conventional engine (CE) and engine with LHR–3 combustion chamber at recommended injection timing and optimum injection timing

Thermal barrier provided in all three possible way of heat escaping resulted reduction of coolant load. As advancing the injection timing in the LHR-3 combustion chamber, Coolant load found to be reduced. Because decrease of combustion temperatures in the LHR-3 combustion chamber with which heat flow to the coolant also reduced. In case of conventional engine, energy effective utilization, un-burnt fuel concentration reduced, released from the combustion, increase gas temperatures ,coolant load increased marginally at all loads, when the injection timing was advanced to the optimum value. However, the improvement in the performance of the conventional engine was due to heat addition at higher temperatures and rejection at lower temperatures, while the improvement in the efficiency of the LHR–3 combustion chamber was due to recovery from coolant load at their respective optimum injection timings.

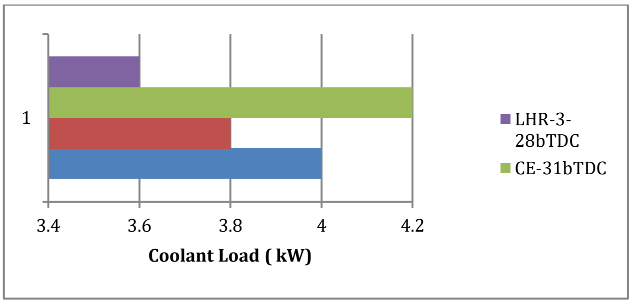



**Fig.11** Bar charts showing the coolant load variation at full load operation with conventional engine (CE) and engine with LHR–3 combustion chamber at recommended injection timing and optimized injection timing

Fig.11 indicates that engine with LHR–3 combustion chamber decreased coolant load at full load operation by 5% at 27° bTDC and 14% at 28°bTDC when compared with CE at 27°bTDC and 31°bTDC. This was due increase of gas temperatures with CE at 31°bTDC and decrease the same with engine with LHR–3 combustion chamber at 28° bTDC.

From the curves in Fig.12, it is observed with the increase of BMEP, volumetric efficiency decreased with in both versions of the combustion chamber.

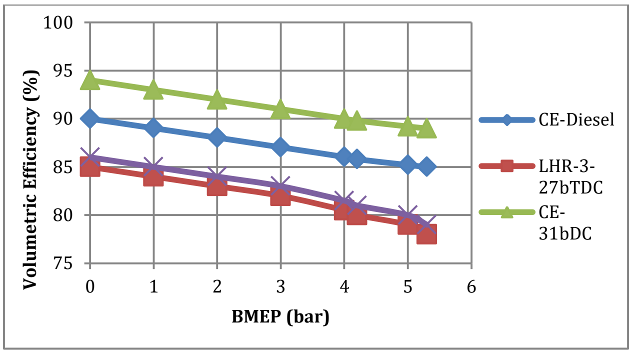



**Fig.12** Volumetric efficiency variation with brake mean effective pressure effective pressure (BMEP) with conventional engine (CE) and engine with LHR–3 combustion chamber at recommended injection timing and optimum injection timing

Engine with LHR-3 combustion chamber found lower volumetric efficiency at all loads when compared with CE, due to increase of gas temperature with the load. Increase of temperature of incoming charge with hot insulated components of the engine causing reduction in the density and hence the quantity of air. Volumetric efficiency was found to be increased at all loads marginally with advanced injection timing with both versions of the combustion chamber. Due to reduction of combustion chamber wall temperature, which in turn depends on EGT. Volumetric efficiency variation between these two versions of the engine is very small. However, 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.

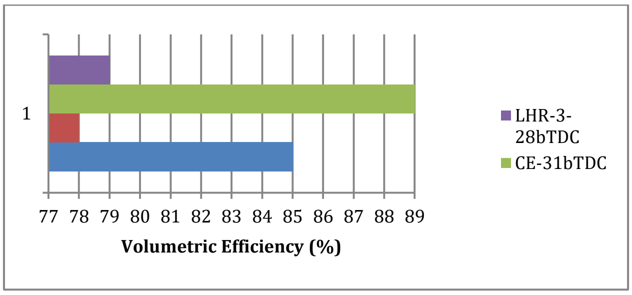



**Fig.13** Bar charts showing the volumetric efficiency variation at full load operation with conventional engine (CE) and engine with LHR–3combustion chamber at recommended injection timing and optimized injection timing

Fig.13 shows that engine with LHR–3 combustion chamber decreased volumetric efficiency at full load operation by 8% at 27ºbTDC and 11% at 28º bTDC when compared with CE at 27ºbTDC and 31ºbTDC. This was due heating of air with insulated components of engine with LHR–3 combustion chamber. This was due to lower EGT with CE.

From the curves in Fig.14, it is observed that sound levels were found to be increased up to 80% of the full load with both versions of the combustion chamber as due fuel consumption increases. It is found that 80% of the full load, they decreased initially and later increased with both versions of the combustion chamber. Because of the increase in thermal efficiency and attains peak thermal efficiency at 80% of full load. Beyond 80% of the load deteriorate engine performance and increase in sound levels as decrease of mechanical efficiency and fuel conversion efficiency. When compared with CE, the engine with LHR–3 combustion chamber increased sound levels at all loads, which confirmed that performance deteriorated with engine with LHR-3 combustion chamber.

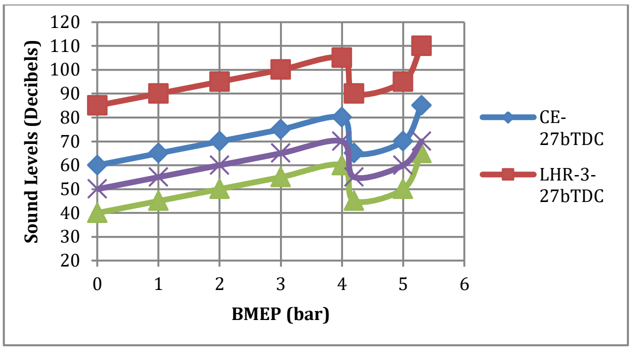



Fig.14 Sound levels variation with brake mean effective pressure effective pressure (BMEP) with conventional engine (CE) and engine with LHR–3 combustion chamber at recommended injection timing and optimum injection timing

As advancing the injection timing, sound levels improved with both versions of the combustion chamber. This was because of improved combustion with early start of combustion leading to improve atomization characteristics.

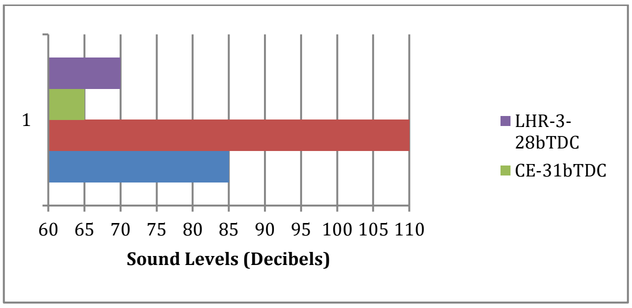



**Fig.15** Bar charts showing the sound levels variation at full load operation with conventional engine (CE) and engine with LHR–3 combustion chamber at recommended injection timing and optimized injection timing

Fig.15 indicates that engine with LHR–3 combustion chamber increased sound levels at full load operation by 29% at 27ºbTDC and 8% at 28ºbTDC when compared with CE at 27ºbTDC and 31ºbTDC. This was due to deterioration of combustion at full load operation with LHR–3 combustion chamber at 27ºbTDC and improved combustion at 28ºbTDC.

## Conclusions

1) Engine with LHR–3 combustion chamber showed deteriorate performance at the full load operation in terms of brake thermal efficiency, exhaust gas temperature, volumetric efficiency and sound levels at 27 º bTDC in comparison with conventional engine at 27 º bTDC.
2) Engine with LHR–3 combustion chamber at 28º bTDC, increased brake thermal efficiency by 11%,at full load–decreased BSFC by 10%, exhaust gas temperature by 5%, coolant load by 5%, increased volumetric efficiency by 1%and decreased sound levels by 36% in comparison with same configuration of combustion chamber at an injection timing of 27 º bTDC.
3) Conventional engine increased brake thermal efficiency by 11%, at full load–decreased BSFC by 10%, exhaust gas temperature by 12%, increased coolant load by 5%, volumetric efficiency by 5% and decreased sound levels by 24% with advanced injection timing of 31 º bTDC.

*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.

*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.

## Acknowledgments

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## References

Matthias Lamping, Thomas Körfer, Thorsten Schnorbus, Stefan Pischinger, Yunji Chen: Tomorrows Diesel Fuel Diversity – Challenges and Solutions, SAE 2008-01—1731

Cummins, C. and Jr. Lyle Diesel's Engine, Volume 1: From Conception To 1918". Wilsonville, OR, USA: Carnot Press, ISBN 978-0-917308-03-1, 1993.

Avinash Kumar Agarwal,Dhananjay Kumar Srivastava , Atul Dhar, Rakesh Kumar Maurya,

Pravesh Chandra Shukla, Akhilendra Pratap Singh.(2013), Effect of fuel injection timing and pressure on combustion, emissions and performance characteristics of a single cylinder diesel engine, Fuel, 111, pp 374–383.

Parlak, A., Yasar, H., ldogan O. (2005).The effect of thermal barrier coating on a turbocharged Diesel engine performance and exergy potential of the exhaust gas. Energy Conversion and Management, ISSN: 0196-8904, 46(3), 489–499.

Ekrem, B., Tahsin, E., Muhammet, C. (2006). Effects of thermal barrier coating on gas emissions and performance of a LHR engine with different injection timings and valve adjustments. Journal of Energy Conversion and Management, ISSN: 0196-8904, 47,1298-1310.

Ciniviz, M., Hasimoglu, C., Sahin, F., Salman, M. S. (2008). Impact of thermal barrier coating application on the performance and emissions of a turbocharged diesel engine. Proceedings of The Institution of Mechanical Engineers Part D-Journal Of Automobile Eng,222 (D12), ISSN: : 0954-4070, 2447–2455.

Janardhan, N., Murali Krishna, M.V.S., Kesava Reddy, Ch. and Durga Prasada Rao. N. (2014).Effect of injection timing on performance parameters of DI diesel engine with ceramic coated cylinder head, International Journal of Scientific and Engineering Research, 5(12),1596–1607.

N. Janardhan, M.V.S. Murali Krishna, Ch.Kesava Reddy and N.Durga Prasada Rao. (2015), Effect of injection timing on exhaust emissions and combustion characteristics of direct injection diesel engine with ceramic coated cylinder head, International Journal of Current Engineering Technology, February, 5(1),46–52.

Parker, D.A. and Dennison, G.M. (1987). The development of an air gap insulated piston. SAE Paper No. 870652, 1987.

Rama Mohan, K., Vara Prasad, C.M., Murali Krishna, M.V.S. (1999). Performance of a low heat rejection diesel engine with air gap insulated piston, ASME Journal of Engineering for Gas Turbines and Power, 121(3),530-540.

Janardhan, N., Murali Krishna, M.V.S., Kesava Reddy, Ch. and Durga Prasada Rao, (2015).Effect of injection timing on performance parameters of DI diesel engine with air gap insulation, International Journal of Thermal Technologies, 5(1),9–17.

Janardhan, N., Murali Krishna, M.V.S., Kesava Reddy, Ch. and Durga Prasada Rao, N. (2014),Effect of injection timing on exhaust emissions and combustion characteristics of direct injection diesel engine with high grade insulated combustion chamber, International Journal of Application or Innovation in Engineering and Management, 3(12), 213–221.

Kesava Reddy, Ch., Murali Krishna, M.V.S., Murthy, P.V.K. and Ratna Reddy, T. (2012) Performance evaluation of a high grade low heat rejection diesel engine with crude pongamia oil. International Journal of Engineering Research and Applications, 2(5), pp 1505-1516.

Janardhan, N., Murali Krishna, M.V.S., Ushasri, P. and Murthy, P.V.K. (2012)., Performance of a low heat rejection diesel engine with jatropha, . International Journal of Engineering Inventions, 1(2), 23–35.

Chowdary, R.P., Murali Krishna, M.V.S., Reddy, T.K.K. and Murthy,P.V.K. (2012). Performance evaluation of a high grade low heat rejection diesel engine with waste fried vegetable oil. International Journal of Engg & Technology, 2(3), pp 440-450

Krishna Murthy, P.V. (2010). Studies on biodiesel with low heat rejection diesel engine. PhD Thesis, J. N. T. University, Hyderabad, India.

Venkateswara Rao, N., Murali Krishna, M.V.S. and Murthy, P.V.K. (2013). Effect of injector opening pressure and injection timing on performance parameters of high grade low heat rejection diesel engine with tobacco seed oil based biodiesel. International Journal of Current Engineering & Technology, 3(4),pp 1401-1411.

Subba Rao, B., Ramjee, E., Murthy, P.V.K. and Murali Krishna, M.V.S. (2013). Studies on exhaust emissions and combustion characteristics of tobacco seed oil in crude form and biodiesel from a high grade low heat rejection diesel engine. International Journal of Industrial Engineering and Technology, 3(1), pp 27-36.

Murali Krishna, M.V.S., Janardhan, N., Murthy, P.V.K., Ushasri, P. and Nagasarada. (2012). A comparative study of the performance of a low heat rejection diesel engine with three different levels of insulation with vegetable oil operation. Archive of Mechanical Engineering (Poland), LIX (1), pp 101-128.

Kesava Reddy, Ch., Murali Krishna, M.V.S., Murthy, P.V.K. and Ratna Reddy, T. (2012). A comparative study of the performance evaluation of a low heat rejection engine with three different levels of insulation with crude pongamia oil operation. Canadian Journal on Mechanical Sciences & Engineering, 3(3), pp 59-71.

Murali Krishna, M.V.S., Chowdary, R.P., Reddy, T.K.K. and Murthy, P.V.K. (2012). A comparative study of the performance of a low heat rejection diesel engine with three different levels of insulation with waste fried vegetable oil operation. International Journal of Science & Technology (Australia), 2(6),pp 358-371.

Ratna Reddy, T., Murali Krishna, M.V.S., Kesava Reddy, Ch. and Murthy, P.V.K. (2012). Comparative performance of different versions of the low heat rejection diesel engine with mohr oil based bio-diesel. International Journal of Research & Reviews in Applied Sciences, 1(1), pp 73-87.