International Journal of Thermal Technologies
E-ISSN 2277 – 4114
Available at http://inpressco.com/category/ijtt/

Research Article

Effect of Injection Timing on Performance Parameters of Direct Injection Diesel Engine with Air Gap Insulation

N. Janardhan†, M.V.S. Murali Krishna†*, Ch. Kesava Reddy‡ and N.Durga Prasad Rao†

†Mechanical Engineering Department, ChaitanyaBharathi Institute of Technology,Gandipet, Hyderabad 500 075, Telangana State, India,
‡Mechatronics Engineering Department, Mahatma Gandhi Institute Technology,Gandipet, Telangana, India
†Bharat Dynamics Limited, Hyderabad

Accepted 12 Feb 2015, Available online 01 March2015, **Vol.5, No.1 (March 2015)**

Abstract

Experiments were carried out to evaluate the performance 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. Performance parameters [brake thermal efficiency, exhaust gas temperature, coolant load, volumetric efficiency and sound 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. 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 deteriorated performance at manufacturer's recommended injection timing of 27°bTDC, and the performance 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.*

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.(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,C*et al*,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.

Wallace *et al*. also studied the performance of the insulated piston engine in which air gap thickness was maintained at 2-mm.(Wallace,F.J*et al*,1983). The major finding was drastic reduction in volumetric efficiency, which resulted in very high combustion temperatures attained in view of the decreased air fuel ratios from 18.27 to astonishingly small 12.76, which was inadmissible in practice. The drastic increase in exhaust gas temperature from 640°C of the conventional engine to 810°C in the insulated piston engine was also rather abnormal for a diesel engine exhaust. The coolant load had also drastically increased from 3.3 kW for conventional engine to 6.2 kW, when the heat barrier piston was used and this is much against the expectation. This was attributed to enhance heat transfer rates from gas to cylinder wall because of

*Corresponding author: **M.V.S. Murali Krishna**

drastic increase of gas temperatures throughout the cycle.

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.(Karthikeyan,S*et al*, 1985). They reported the deterioration in the performance of the engine at all loads, when compared to neat diesel operation on conventional engine. This was due to higher exhaust gas temperatures. The nomonic 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.

JabezDhinagar*et al*. conducted experiments on LHR engine, with an air gap insulated piston, air gap insulated liner and ceramic coated cylinder head.(JabezDhinagar. S *et al*,1989). The piston with nomonic 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 with pure diesel operation, at recommended injection timing. Hence the injection timing was retarded to achieve better 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, D.A. *et al*,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 and reported brake specific fuel consumption was improved by 3%.(Rama Mohan, K. *et al*,1999).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, C.M. *et al*,2000; Murali Krishna, M.V.S *et al*,2004; Chennakesava Reddy *et al*,2011;Janardhan, N *et al*,2012;MuraliKrishna,M.V.S*et al*,2013;Srikanth, D. *et al*,2013). It was reported from their investigations that engine with LHR-2 combustion chamber increased brake thermal efficiency by 8-10% in comparison with neat diesel operation on CE. The performance was further 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.

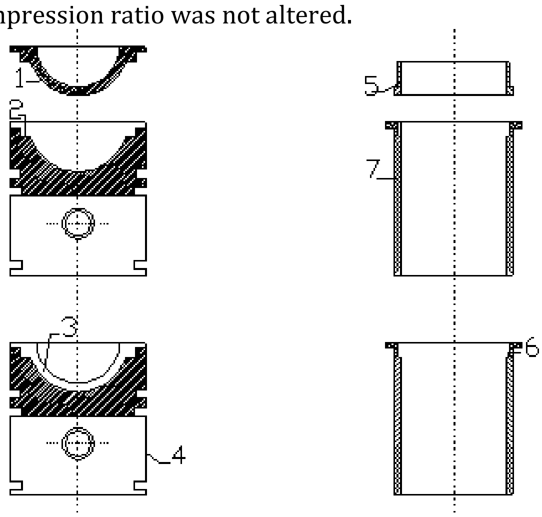
Materials and Methods

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.

Table1 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 poin (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. The height of the piston was maintained such that compression ratio was not altered.



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

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.

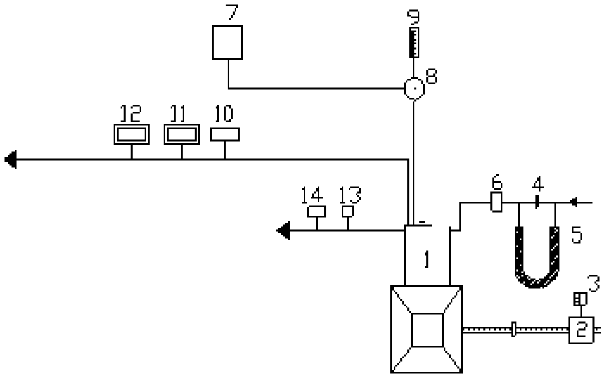
The test fuel used in the experimentation was neat diesel. The schematic diagram of the experimental setup with diesel operation is shown in Figure 2. The specifications of the experimental engine are shown in Table-2.

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

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°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 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

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) Since exhaust emissions were not measured in the experiment, part No.11 and Part No.12 were not in use. 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.

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

Nomenclature

ρ_a=density of air, kg/m^3
ρ_d=density of fuel, gm/cc
η_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-2= Insulated combustion chamber with air gap insulated piston and air gap insulated liner
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, ^oC
T_I=inlet temperature of water, ^oC
T_o=outlet temperature of water, ^oC
V=voltmeter reading, volt
V_s=stroke volume, m^3

VE=Volumetric efficiency, %

Definitions of used values:

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

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

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

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

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

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

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

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

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

3. Results and Discussion

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.

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^{o}bTDC in CE. Performance deteriorated if the injection timing was greater than 31^{o}bTDC. This was because of increase of ignition delay.

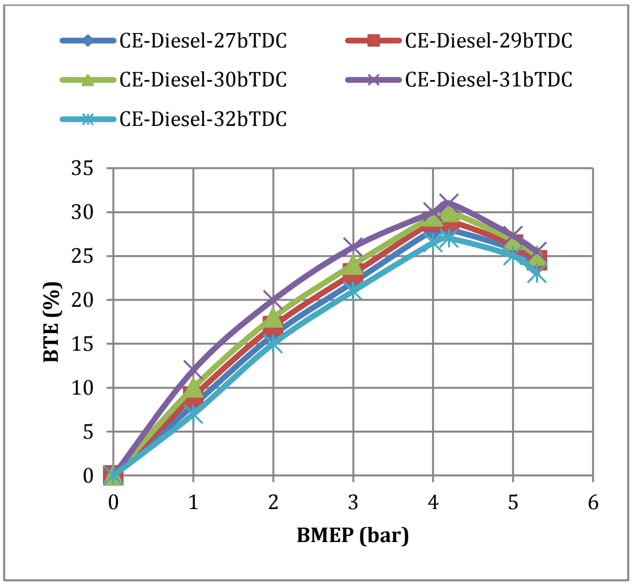


Fig.3 variation of brake thermal efficiency (BTE) with brake mean effective pressure (BMEP) in the conventional engine with neat diesel, at various injection timings at an injector opening pressure of 190 bar.

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

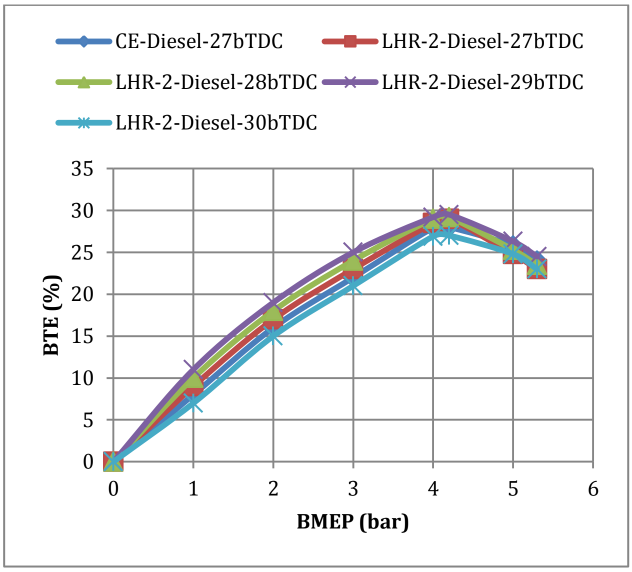


Fig.3 variation of brake thermal efficiency (BTE) with brake mean effective pressure (BMEP) in the engine with LHR–2 combustion chamber with neat diesel, at various injection timings at an injector opening pressure of 190 bar.

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.

Curves in Fig.5 indicate that at all loads, BTE was observed to be higher with CE at the optimum injection timing when compared with engine with LHR-2 combustion chamber. This was due to higher advanced injection timing with CE than engine with LHR-2 combustion chamber.

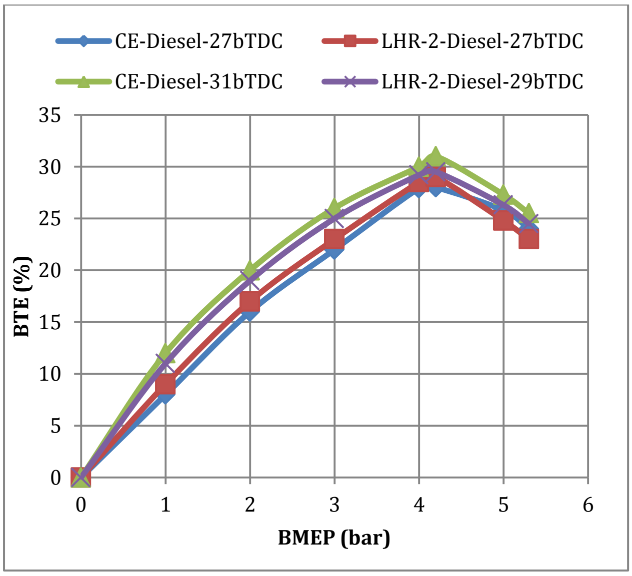


Fig.5 Variation of brake thermal efficiency with brake mean effective pressure effective pressure (BMEP) with conventional engine (CE) and engine with LHR–2 combustion chamber at recommended injection timing and optimum injection timing.

Fig.6 shows that engine with LHR-2 combustion chamber increased peak BTE by 4% at 27°bTDC, while decreasing it by 5% at 29°bTDC when compared with CE at 27°bTDC and 31°bTDC.

When engine with different versions of the combustion chamber is to be tested, then brake specific fuel consumption (BSFC) at full load is to be determined in order to compare the performance of the engine.Fig.6 indicates that engine with LHR-2 combustion chamber increased BSFC at full load operation by 3% at 27°bTDC and 6% at 29°bTDC when compared with CE at 27°bTDC and 31°bTDC. This was due reduction of ignition delay.

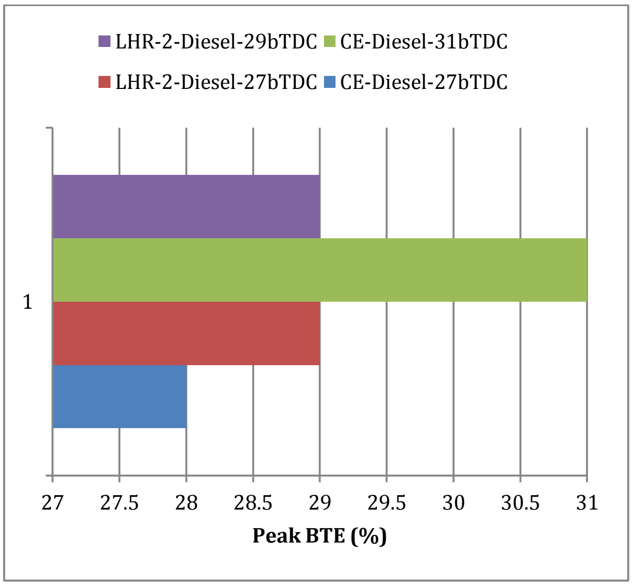


Fig.6 Bar charts showing the variation of peak brake thermal efficiency (%) with conventional engine (CE) and engine with LHR-2 combustion chamber at recommended injection timing and optimized injection timing.

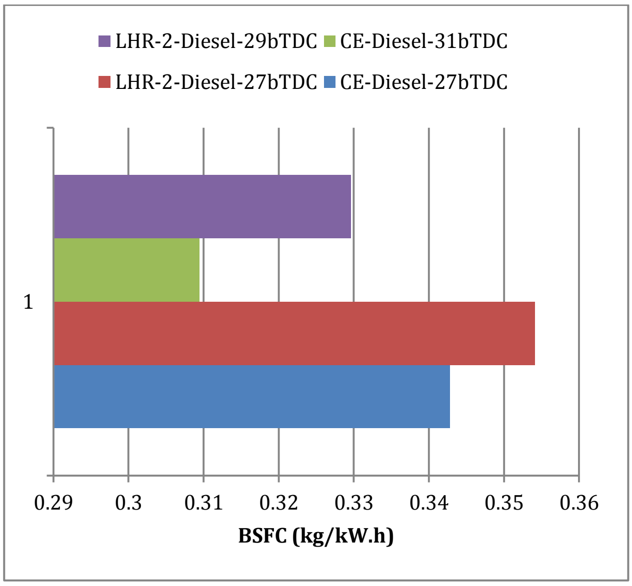


Fig.6 Bar charts showing the variation of brake specific fuel consumption (BSFC) at full load operation with conventional engine (CE) and engine with LHR–2 combustion chamber at recommended injection timing and optimized injection timing.

Fig.7 indicates that exhaust gas temperatures (EGT) increased with an increase of BMEP with both versions of the combustion chamber. This was due to increase of fuel consumption with load. Engine with LHR–2 combustion chamber decreased EGT up to 80% of the full load and beyond that load it increased EGT in comparison with CE. This was due to improved combustion with provision of thermal insulation and hence improved air fuel ratios up to 80% of the full load. Increase of EGT beyond that load was decrease of ignition delay. EGT at all loads decreased 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 because, when 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.

The trends of these variations observed by the other researchers were in good agreement with the trend observed by the authors. Wallace *et al*. noticed 26.5% increase in value of EGT. Karthikeyan*et al*. observed an increase in exhaust gas temperature from 450° C of conventional engine to 550°C, in the LHR combustion chamber working out to percentage increase of 22%, JabezDhinagarreported that low heat rejection diesel engine at the injection timing of 25°bTDC showed a clear overall increase of 50°C to 200°C in exhaust gas temperature over the entire load range compared to the base diesel.

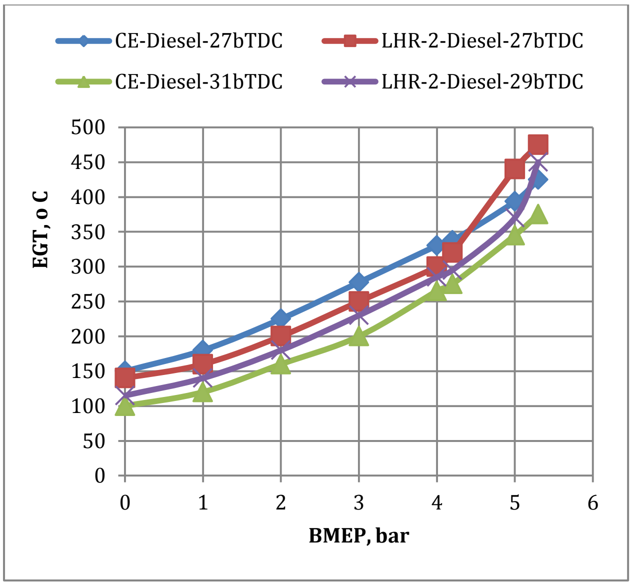


Fig.7 Variation of exhaust gas temperature (EGT) with brake mean effective pressure effective pressure (BMEP) with conventional engine (CE) and engine with LHR–2 combustion chamber at recommended injection timing and optimum injection timing.

Fig.8 indicates that engine with LHR-2 combustion chamber increased EGT at full load operation by 6% at 27°bTDC and 20% at 29°bTDC when compared with CE at 27°bTDC and 31°bTDC. This was due reduction of ignition delay. This was also due to higher injection advance with CE.

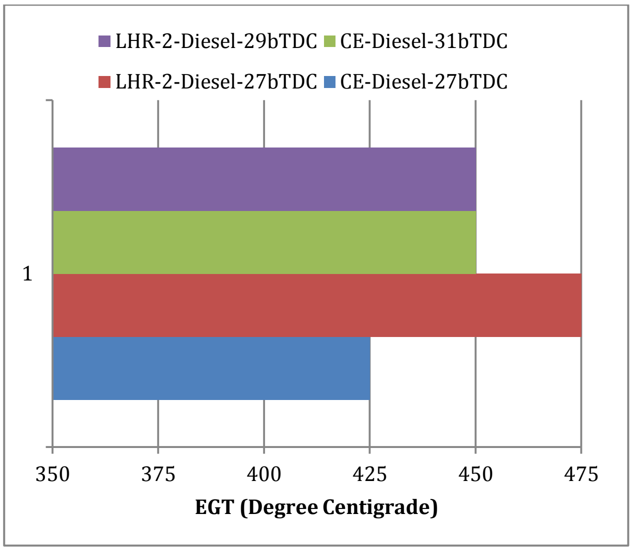


Fig.8 Bar charts showing variation of the exhaust gas temperature (EGT) at full load operation with conventional engine (CE) and engine with LHR–2 combustion chamber at recommended injection timing and optimized injection timing.

From Fig.9, it is observed that coolant load increased with the increase of BMEP in the conventional engine and LHR-2 combustion chamber. The LHR–2 combustion chamber gave lower coolant load up to 80% of the full load, when compared to convectional engine. Air being a bad conductor offers thermal resistance for heat flow through the piston and liner.

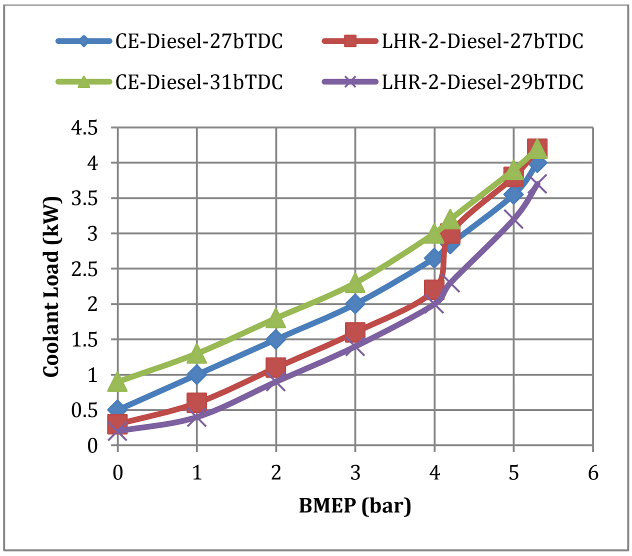


Fig.9 Variation of coolant load with brake mean effective pressure effective pressure (BMEP) with conventional engine (CE) and engine with LHR–2 combustion chamber at recommended injection timing and optimum injection timing.

It was therefore evident that thermal barrier provided in the piston and liner resulted in reduction of coolant load up to 80% of the full load. Beyond 80% of the full load, coolant load in the LHR-2 combustion chamber increased over and above that of the convectional engine, with which efficiency was deteriorated at full load of the LHR–2 combustion chamber, when compared to the convectional engine. This was because in cylinder, the heat rejection at full load was primarily due to un-burnt fuel concentration near the combustion chamber walls.

The air-fuel ratio got reduced to a reasonably low value at this load confirming the above trend. However, when heat rejection calculations of coolant were made, the heat lost to lubricant should also be considered. As in the present investigations the lubricant heat loss was not considered, this aspect was not depicted in coolant load calculations. This was also due to the fact that nearer peak load; a greater amount of heat was being transferred through the un-insulated cylinder head, where higher temperatures exist. Wallace *et al*. and Ramamohan also observed the same trend at full load operation. Wallace *et al* reported a drastic increase in coolant load from 3.3 kW in conventional engine to 6.2 kW in LHR engine amounting to percentage increase of 88% and Rama Mohan reported the percentage increase of coolant load of 8.5%, while the author observed a percentage increase of 12.5% at full load operation. This was due to the different degree of insulations and configurations employed by the author and other researchers.

Coolant load reduced in the LHR–2 combustion chamber with advanced injection timing This was due to decrease of combustion temperatures in the LHR-2 combustion chamber with which heat flow to the coolant also reduced. In case of conventional engine, un-burnt fuel concentration reduced with effective utilization of energy, released from the combustion, coolant load increased marginally at all loads due to increase of gas temperatures, 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–2 combustion chamber was due to recovery from coolant load at their respective optimum injection timings. Ramamohannoticed the similar trend as observed by the author at optimum injection timing with his LHR combustion chamber.

Fig.10 indicates that engine with LHR–2 combustion chamber increased coolant load at full load operation by 5% at 27^o bTDC, while decreasing it by 14% at 29^{o}bTDC when compared with CE at 27^{o}bTDC and 31^{o}bTDC. This was due reduction of ignition delay with engine with LHR–2 combustion chamber at 27^{o}bTDC and increase of gas temperatures with CE at 31^{o}bTDC and decrease the same with engine with LHR–2 combustion chamber at 29^o bTDC.

From the curves in Fig.11, it is noticed that volumetric efficiency decreased with the increase of BMEP in both versions of the combustion chamber.

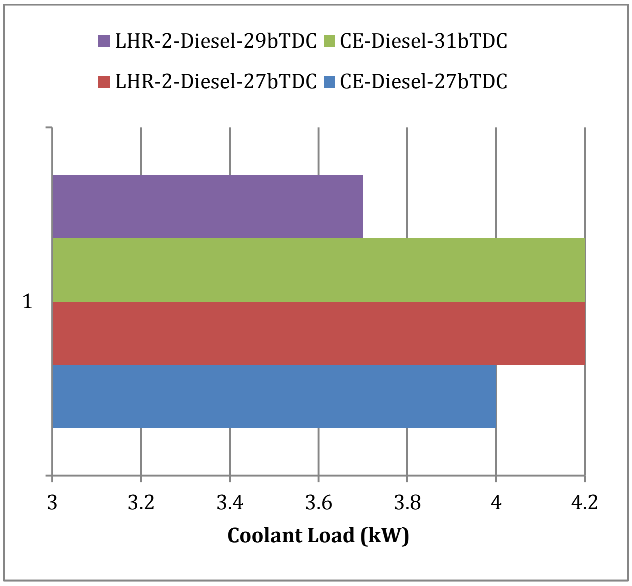


Fig.10 Bar charts showing the variation of coolant load at full load operation with conventional engine (CE) and engine with LHR–2 combustion chamber at recommended injection timing and optimized injection timing.

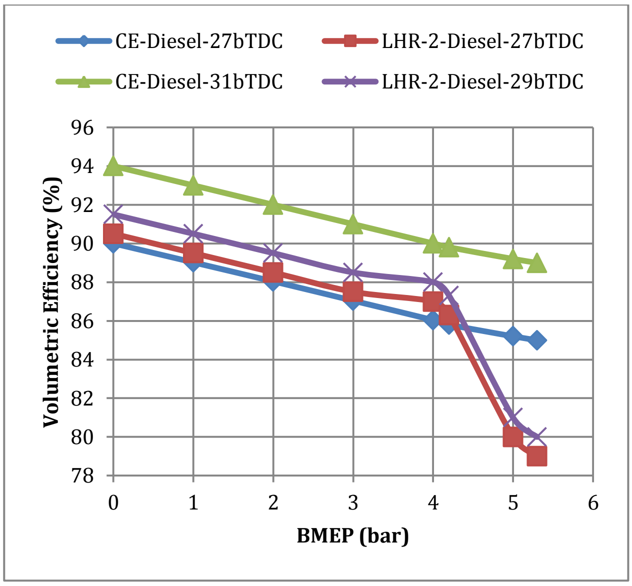


Fig.11 Variation of volumetric efficiency with brake mean effective pressure effective pressure (BMEP) with conventional engine (CE) and engine with LHR–2 combustion chamber at recommended injection timing and optimum injection timing.

This was due to increase of gas temperature with the load. At the recommended injection timing, volumetric efficiency in the LHR-2 combustion chamber increased up to 80% of the full load operation and beyond that load, it decreased when compared to the conventional engine. This was due improved air fuel ratios with the improved combustion with engine with LHR-2 combustion chamber up to 80% of the full load. Beyond that load, it was because of increase of temperature of incoming charge in the hot environment created with the provision of insulation, causing reduction in the density and hence the quantity of air. However, this variation in volumetric efficiency is very small between these two versions of the engine, as volumetric efficiency mainly depends on speed of the engine, valve area, valve lift, timing of the opening or closing of valves and residual gas fraction rather than on load variation.

Ramamohanalso observed the similar trends in the magnitude of volumetric efficiency. He reported a drop of 8% in volumetric efficiency with his LHR combustion chamber at peak load when compared to the conventional engine. The authors observed reduction in the value of the volumetric efficiency was high, when compared with the similar data of Ramamohan. This was due to the different degree of insulations adopted by the authors and Ramamohan. Volumetric efficiency increased in the conventional engine and decreased in LHR-2 combustion chamber at their optimum injection timings, compared to the conventional engine at the recommended injection timing. This was due to decrease of un-burnt fuel fraction in the cylinder leading to increase in volumetric efficiency in the conventional engine while heating effect of the air causes to decrease the volumetric efficiency in the LHR-2 combustion chamber. However, the volumetric efficiency was marginally more with LHR-2 combustion chamber engine at its optimized injection timing compared to the same version of the engine at the recommended injection timing. This was due to reduction of combustion wall temperatures which in turn depend on exhaust gas temperatures.

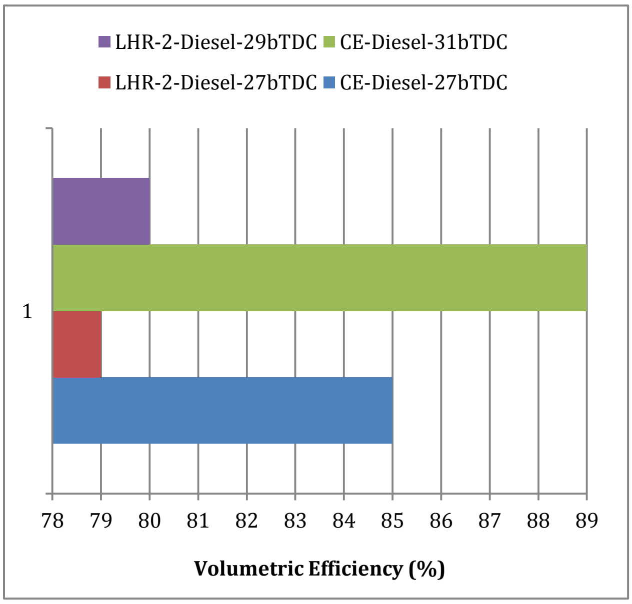


Fig.12 Bar charts showing the variation of volumetric efficiency at full load operation with conventional engine (CE) and engine with LHR–2 combustion chamber at recommended injection timing and optimized injection timing.

Dhinagar*et al*. reported that a drop of volumetric efficiency of the order 10% at part load and 15% at peak load for LHR version of the combustion chamber, at 25°bTDC, with respect to the conventional engine. This was mainly due to preheating of the air in his experimentation. This was also due to the difference in the degree of insulation adopted by the Dhinagar and authors.

Fig.12 indicates that engine with LHR-2 combustion chamber decreased volumetric efficiency at full load operation by 7% at 27°bTDC and 10% at 29°bTDC when compared with CE at 27°bTDC and 31°bTDC. This was due heating of air with insulated components of engine with LHR-2 combustion chamber. This was due to lower EGT with CE.

From the curves of Fig.13 it is noticed that sound intensity decreased with LHR-2 combustion chamber up to 80% of the full load operation and beyond that it increased over and above the conventional engine.

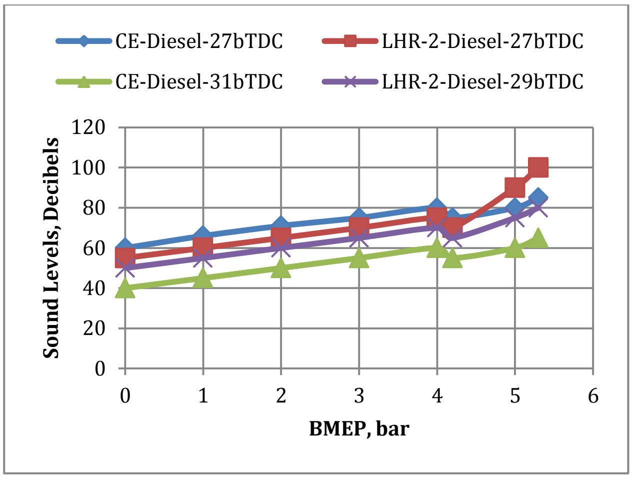


Fig.13 Variation of sound levels with brake mean effective pressure effective pressure (BMEP) with conventional engine (CE) and engine with LHR–2 combustion chamber at recommended injection timing and optimum injection timing.

This was because of improved combustion with improved air fuel ratios up to 80% of the full load and beyond that load performance deteriorated with reduction of ignition delay. Ignition delay got reduced with LHR-2 combustion chamber and less time was available for complete combustion giving rise noise levels. Sound levels improved with advanced injection timing with both versions of the combustion chamber. This was because of improved combustion with early start of combustion leading to increase ignition delay.

Fig.14 indicates that engine with LHR-2 combustion chamber increased sound levels at full load operation by 18% at 27°bTDC and 23% at 29°bTDC when compared with CE at 27°bTDC and 31°bTDC. This was due to deterioration of combustion at full load operation with LHR-2 combustion chamber leading to increase of sound levels.

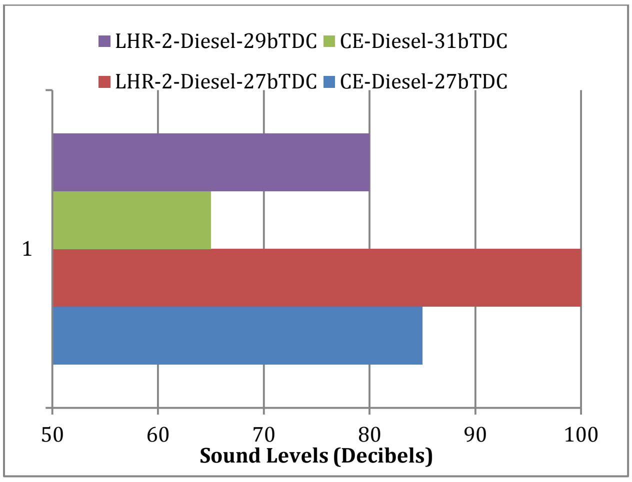


Fig.14 Bar charts showing the variation of sound levels at full load operation with conventional engine (CE) and engine with LHR-2 combustion chamber at recommended injection timing and optimized injection timing.

Conclusions

1. Engine with LHR-2 combustion chamber showed improved performance at 80% of the full load operation in terms of brake thermal efficiency, exhaust gas temperature, coolant load, volumetric efficiency, and sound levels at 27 ° bTDC in comparison with conventional engine at 27 ° bTDC.
2. Engine with LHR-2 combustion chamber showed deteriorate performance at the full load operation at 27 ° bTDC in comparison with conventional engine at 27 ° bTDC.
3. Engine with LHR-2 combustion chamber at 29° bTDC, increased brake thermal efficiency by 2%,at full load–decreased BSFC by 7%, exhaust gas temperature by 5%, coolant load by 12%, increased volumetric efficiency by 1%and decreased sound levels by 20% in comparison with same configuration of combustion chamber at an injection timing of 27 ° bTDC.
4. 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 ° 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–2 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-2 combustion chamber can be taken up.

Acknowledgments

Authors thank authorities of ChaitanyaBharathi Institute of Technology, Hyderabad for providing facilities for carrying out this research work. Financial assistance provided by All India Council for Technical Education (AICTE), New Delhi, is greatly acknowledged.

References

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

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

Avinash Kumar Agarwal,Dhananjay Kumar Srivastava , AtulDhar, Rakesh Kumar Maurya, Pravesh Chandra Shukla, AkhilendraPratap 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.

Wallace, F.J., Kao, T.K., Alexander, W.D., Cole, A. and Tarabad, M. (1983), Thermal barrier piston and their effect on the performance of compound diesel engine cycles, SAE Paper No. 830312, 1983.

Karthikeyan, S., Arunachalam, M., SrinivasanRao, P. and GopalaKrishanan, K.V. (1985), Performance of an alcohol, diesel oil dual-fuel engine with insulated engine parts", Proceedings of 9th National Conference of I.C. Engines and Combustion, pp 19-22, Indian Institute of Petroleum, Dehradun.

JabezDhinagar, S., Nagalingam, B. and Gopala Krishnan, K.V., " Use of alcohol as the sole fuel in an adiabatic diesel engine", Proceedings of XI National Conference on IC Engines and Combustion, pp: 277-289, I.I.T., Madras, 1989.

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. and 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), pp 530–540.

Vara Prasad, C.M, Murali Krishna, M.V.S., Prabhakar Reddy, C. and Rama Mohan, K. (2000).Performance evaluation of non edible vegetable oils as substitute fuels in low heat rejection diesel engine. *Institute of Engineers (London), 214(2),Part-D, Journal of Automobile Engineering,*pp 181-187.

Murali Krishna, M.V.S. (2004). Performance evaluation of low heat rejection diesel engine with alternate fuels.PhD Thesis, J. N. T. University, Hyderabad.

Chennakesava Reddy, Murali Krishna, M.V.S., Murthy, P.V.K., and RatnaReddy,T. (2011).Potential of low heat rejection diesel engine with crude pongamia oil. *International Journal of Modern Engineering Research,* 1(1), pp*210-224.*

Janardhan, N., Murali Krishna, M.V.S., Ushasri, P. and Murthy, P.V.K. (2012).Potential of a medium low heat rejection diesel engine with crude jatropha oil.*International Journal of Automotive Engineering and Technologies, 1(2), pp 1-16*

Murali Krishna, M.V.S., DurgaPrasadaRao, N., Anjeneya Prasad, A. and Murthy, P.V.K. (2013).Improving of emissions and performance of rice brawn oil in medium grade low heat rejection diesel engine.*International Journal of Renewable Energy Research,* 3(1), pp 98-108.

Srikanth, D., Murali Krishna, M.V.S., Ushasri, P. and Krishna Murthy, P.V.(2013), Comparative studies on medium grade low heat rejection diesel engine and conventional diesel engine with crude cotton seed oil, *International Journal of Innovative Research in Science, Engineering and Technology, 2(10), pp 5809-5228.*