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Research Article

Experimental Investigations on Performance Parameters with Low Heat Rejection Diesel Engine with Varied Air Gap Thickness

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Abstract

Conservation of fossil fuels is gaining momentum along with adapting alternative fuel technology methods for the researchers and manufacturers involved in combustion research. The concept of low heat rejection (LHR) engine is to minimize heat flow to the coolant by providing thermal resistance in the path of heat flow to the coolant and thus increase thermal efficiency. It has significant characteristics of higher operating temperature, maximum heat release, and ability to handle low calorific value fuel. Investigations were carried out to evaluate the performance of diesel engine with air gap insulated low heat rejection (LHR–3) engine consisting of air gap insulated piston with superni crown , air gap insulated liner with superni insert and ceramic coated cylinder head with neat diesel with varied air gap thickness and injection timing. Performance parameters of brake thermal efficiency, brake specific fuel consumption, exhaust gas temperature, coolant load and volumetric efficiency were determined at various values of brake power. The optimum air gap thickness was found to be 2.8 mm with LHR–3engine with diesel operation. LHR engine with neat diesel operation showed deteriorated performance at manufacturer's recommended injection timing of 27^o bTDC (before top dead center) and the performance improved marginally with advanced injection timing of 28.5^o bTDC in comparison with conventional engine (CE) at 27^{o}bTDC.

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

1. Introduction

In the scenario of i) increase of vehicle population at an alarming rate due to advancement of civilization, ii) use of diesel fuel in not only transport sector but also in agriculture sector leading to fast depletion of diesel fuels and iii) increase of fuel prices in International market leading to burden on economic sector of Govt. of India, the conservation of diesel fuel has become pertinent for the engine manufacturers, users and researchers involved in the combustion research. [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 *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 (an alloy of nickel), cast iron and mild steel etc.

LHR combustion chambers were classified as ceramic coated (LHR–1), air gap insulated (LHR–2) and combination of ceramic coated and air gap insulated engines(LHR–3) combustion chambers depending on degree of insulations. Experiments were conducted with neat diesel operation with ceramic coated diesel engine [Paralak *et a*l, 2005; Ekrem *et al,* 2006; Ciniviz *et al,* 2008; Janardhan et al, 2014; Janardhan et al, 2015]. They reported that brake specific fuel consumption decreased by 3-4% with ceramic coated diesel engine in comparison with conventional engine. Creating an air gap in the piston involved the complications of joining two different metals. Investigations were carried out on air gap insulated piston with neat diesel operation [Parker et al, 1987].

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However, the bolted design employed by them could not provide complete sealing of air in the air gap. It was made a successful attempt of 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 [Ramamohan et al, 1999; Janardhan *et al,* 2015]. The optimum injection timing was found to be 29.5^o bTDC. BSFC reduced 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. The optimum air gap thickness was found to be 3 mm with air gap insulated piston with diesel as fuel [Ramamohan et al, 1999].

It should be expected that the thickness of the air gap play an important role on the insulation effect in low heat rejection engines. It was also studied the performance of the insulated piston engine in which air gap thickness was maintained at 2-mm[Wallace *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^{o}C of the conventional engine to 810^{o}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 drastic increase of gas temperatures throughout the cycle.

Air gap thickness of 2-mm in LHR engines was employed on which the performance studies on alcohols and neat diesel were taken [Karthikeyan *et al.* 1985]. It was found that the peak pressures increased over and above that of the conventional piston engine and BSFC also increased at all the loads over and above the conventional piston.

Investigations were carried out with an air gap of 4-mm in the low heat rejection engine [Alkidas *et al,* 1986]. The engine employed ethylene glycol at 120^o C as coolant. In comparison to the conventional configuration of the engine, the low heat configuration resulted in a small increase in brake thermal efficiency for light load conditions, reduction in volumetric efficiency, increase in exhaust energy and an increase in the heat rejection to the lubricating oil. Heat release analysis performed on the two engines showed overall fuel burning in the LHR engine consequently shorter combustion duration in the LHR engine than in the conventional engine. This has caused higher nitric oxide emissions, higher hydrocarbon emissions but slightly lower particulate emission. The increase in hydrocarbon emissions was attributed to the increase in the contribution of lubricating oil resulted because of higher cylinder liner temperatures in the LHR engines.

It was studied the performance of diesel engine under partial insulation in which liner alone was insulated using ceramic material of 1-mm thickness and full insulation where liner, cylinder head and valves were coated with PSZ coating in addition to providing an air gap of 2-mm extending over the major portion of the piston surface in which the Nimonic crown and the associated spacer ring were attached to the aluminium body of the piston [Baluswamy *et al.* 1989]. However the method of attachment was not mentioned. In the case of partial insulated configuration, exhaust gas temperature increased along with brake thermal efficiency and coolant load over that of the conventional engine. However, no explanation was offered as to how exhaust energy and coolant-load along with brake thermal efficiency could be increased. In the case of fully insulated configuration, exhaust gas temperatures increase further and coolant loads as well as brake thermal efficiency decreased in comparison with conventional engine. Of course, volumetric efficiency decreased marginally for partially insulated condition and drastically in fully insulated situation. Peak pressures and rates of pressure rise increased in the LHR engine in comparison with the conventional engine, whereas ignition delay decreased in the fully insulated LHR engine. Smoke densities also decreased in both partially insulated and fully insulated configurations of LHR engines over that of conventional engines

Experiments were conducted with an air gap of 2-mm of the LHR engines where special study on the variations of the volumetric efficiency was made [Jabez Dhinagar *et al*. 1989]. It was found that volumetric efficiency drastically decreased with the air gap insulated piston because of the hot environment of induction air. This was compensated, by supercharging the engine for avoiding deteriorated performance. As the exhaust temperature in the insulated piston was more, it was felt that a turbocharger could be effectively used using the high enthalpy exhaust gases. On the basis of simulated turbocharger, supercharge conditions were fixed which compensated by the air gap insulated piston engine.

Investigations were conducted on low heat rejection engines in which the air gap thickness was maintained at 1-mm, 2-mm, 3-mm, and 3.8-mm and it was concluded that 3mm air gap gave minimum BFSC with decreased coolant loads and lower smoke densities up to 80% of full load when injection timing was maintained at 27^o bTDC recommended by the manufacturer for the conventional engine [Rama Mohan *et al.* 1999]. However, at full load there was marginal increase of BSFC in the air gap insulated piston over that of the conventional engine. Peak pressure and rates of pressure rise reduced marginally with increasing air gap thickness.

However, no systematic investigations were reported on comparative performance of the LHR-3 engine with diesel with varied air gap thickness.

The present paper attempted to evaluate the performance of LHR-3 engine, which consisted of air gap insulated piston, air gap insulated liner and ceramic coated cylinder head fuelled with diesel with varied air gap thickness and injection timing. Comparative performance studies were made with LHR-3 engine with conventional engine with diesel operation.

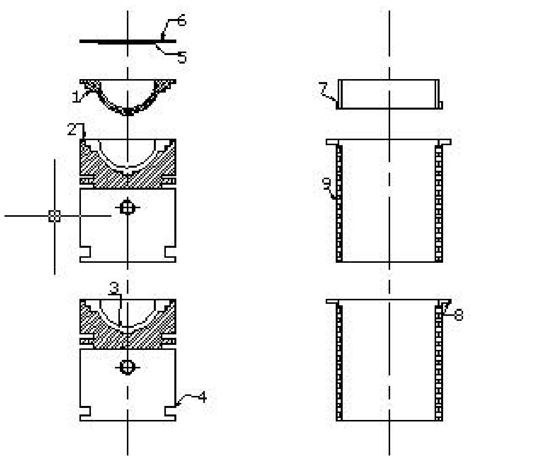
2. Materials and Methods

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^oC	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)	^oC	68
Molecular weight	--	226
Colour	--	Light yellow

LHR–3 engine (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 optimum air gap in between the crown and the body of the piston. The optimum thickness of air gap in the air gap piston was discussed in article Results and Discussion.



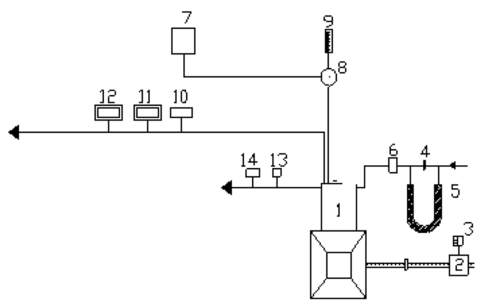
1.Superni crown with threads, 2. Superni gasket, 3.Air gap in piston, 4. Body of the piston, 5. Ceramic coating on inside portion of cylinder head, 6. Cylinder head, 7. Superni insert with threads, 8. Air gap in liner and 9. Body of liner

Fig.1 Schematic diagram of assembly the insulated piston, insulated liner and ceramic coated cylinder head of the LHR-3 engine

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 optimum air gap thickness was maintained between the insert and the liner body.

Partially stabilized zirconium (PSZ) of thickness 500 microns was coated on inside portion of cylinder head by means plasma arc coating. The combination of low thermal conductivity materials of superni, air and PSZ offers thermal resistance in the path of coolant. At 500^oC 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 specifications of the experimental engine are shown in Table-2. Experimental setup used for study of exhaust emissions on low grade LHR diesel engine with cottonseed biodiesel in Fig.3 The specification of the experimental engine (Part No.1) is shown in Table.2 The engine was connected to an electric dynamometer (Part No.2. Kirloskar make) for measuring its brake power. Dynamometer was loaded by loading rheostat (Part No.3). The combustion chamber consisted of a direct injection type with no special arrangement for swirling motion of air. Burette (Part No.9) method was used for finding fuel consumption of the engine with the help of fuel tank (Part No7) and three way valve (Part No.8). Air-consumption of the engine was measured by air-box method consisting of an orifice meter (Part No.4), U-tube water manometer (Part No.5) and air box (Part No.6) assembly.



1.Engine, 2.Electical 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	27ºbTDC × 190 bar
Dynamometer	Electrical dynamometer
Number of holes of injector and size	Three × 0.25 mm
Type of combustion chamber	Direct injection type
Fuel injection nozzle	Make: MICO-BOSCH No- 0431-202-120/HB
Fuel injection pump	Make: BOSCH: NO- 8085587/1

The naturally aspirated engine was provided with water-cooling system in which outlet temperature of water is maintained at 80ºC by adjusting the water flow rate.Engine oil was provided with a pressure feed system. No temperature control was incorporated, for measuring the lube oil temperature The naturally aspirated engine was provided with water-cooling system in which outlet temperature of water is maintained at 80ºC by adjusting the water flow rate, which was measured by water flow meter (Part No.14). Exhaust gas temperature (EGT) and coolant water outlet temperatures were measured with thermocouples made of iron and iron-constantan attached to the exhaust gas temperature indicator (Part No.10) and outlet jacket temperature indicator (Part No.13) 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 27–34ºbTDC.

3. Results and Discussions

3.1 Performance Parameters

As mentioned earlier, air gap thickness can be varied with varying gasket thickness for air gap insulated piston and thickness of insert for air gap insulated liner. Fig.3 shows variation of brake specific fuel consumption (BSFC) ith air gap thickness for LHR-3 engine at recommended injection timing of 27º bTDC at full load. BSFC decreased with an increase of air gap thickness of LHR-3 engine. Improved combustion with improved oxygen-fuel ratios might have caused efficient combustion. However, beyond certain limit (2.8 mm), BSFC increased with an increase of exhaust gas enthalpy. Reduction in volumetric efficiency and mechanical efficiency were responsible for deterioration of the performance of the engine, for air gap thickness greater than 2.8 mm. Hence the optimum air gap thickness was observed to be 2.8 mm for LHR engine.

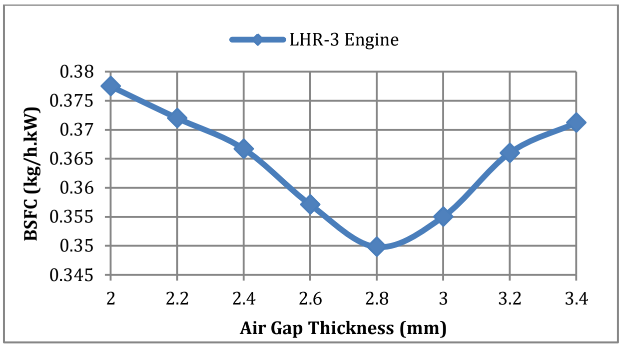


Fig.3 Variation of brake specific fuel consumption with air gap thickness of LHR-3 engine at recommended injection timing of 27 º bTDC at full load

As mentioned earlier, injection timing can be varied with inserting of copper shims between pump body and engine frame Fig.4 shows the variation of brake thermal efficiency (BTE) with brake mean effective pressure (BMEP) in the conventional engine (CE) with neat diesel, at various injection timings at an injector opening pressure of 190 bar.

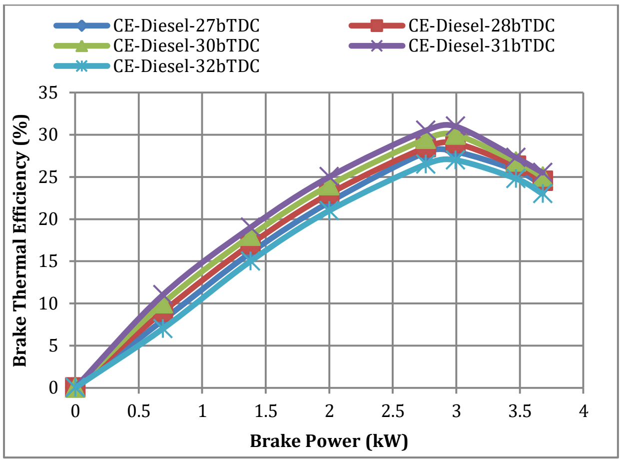


Fig.4 variation of brake thermal efficiency (BTE) with brake mean effective pressure (BMEP) in the conventional engine with neat diesel, at various injection timings at an injector opening pressure of 190 bar

BTE increased with an increase of BMEP up to 80% of the full load, and beyond that load, it decreased with neat diesel operation. This was because increased fuel conversion efficiency and volumetric efficiency up to 80% of the full load,. Decrease of fuel conversion efficiency, mechanical efficiency and oxygen–fuel ratios were responsible for deterioration of the performance beyond 80% of the full load. BTE increased at all loads with advanced injection timings in the conventional engine, due to early initiation of combustion and increase of contact period of fuel with air leading to improve oxygen– fuel ratios period. The optimum injection timing was obtained by based on maximum brake thermal efficiency. Maximum BTE was observed when the injection timing was advanced to 31ºbTDC in

CE. Performance deteriorated if the injection timing was greater than 31°bTDC. This was because of increase of ignition delay.

Fig.5 shows variation of brake thermal efficiency with brake power with varied injection timing with neat diesel at an injection pressure of 190 bar. Fig indicates that LHR engine with diesel showed deteriorated performance at all loads, when compared with CE at an injection timing of 27° bTDC. This was because of reduction of ignition delay, which reduced pre–mixed combustion as a result of which, less time was available for proper mixing of diesel and air and diesel fuel leading to incomplete combustion.

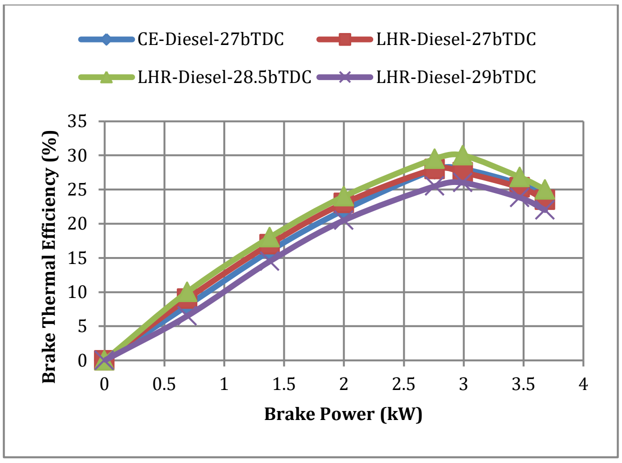


Fig.5 Variation of brake thermal efficiency (BTE with engine with LHR–3 combustion chamber with neat diesel, at various injection timings at an injector opening pressure of 190 bar

More over at full load, increased diffusion combustion and friction resulted from reduced ignition delay.

Increased radiation losses were one of the reasons for the deterioration. BTE increased with advanced injection timing at all loads with diesel with LHR engine. This was because of increase of atomization of fuel with advanced injection timing. Peak BTE increased by 6% at its optimum injection timing of 28.5° bTDC, in comparison with CE at 27° bTDC.

Curves in Fig.4 and Fig. 5 indicate that optimum injection timing was obtained earlier withLHR-3 engine. This is because hot combustion chamber of LHR engine decreased ignition delay and combustion duration. BTE was observed to be higher with CE at the optimum injection timing when compared with LHR engine. This was due to higher advanced injection timing with CE than engine with LHR-3 combustion chamber. 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 LHR-3 engine 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.

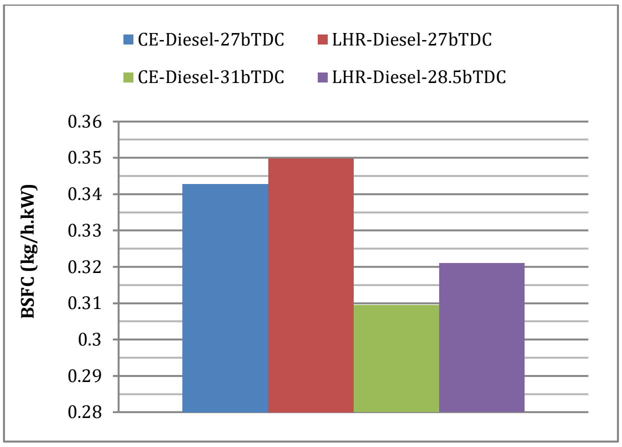


Fig.6 Bar charts showing the variation of brake specific fuel consumption (BSFC) at full load operation with conventional engine (CE) and LHR engine at recommended injection timing and optimized injection timing

Fig.7 indicates that exhaust gas temperatures (EGT) increased with an increase of brake power with both versions of the combustion chamber. Increase of fuel consumption with load might have increased EGT. EGT was observed to be higher with LHR–3 engine at all loads in comparison with CE. This shows that heat rejection was confined in hot insulated combustion chamber, More amount of heat was utilized in converting into useful work by CE when compared with LHR-3 engine leading to reject more amount of heat by insulated engine. 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.

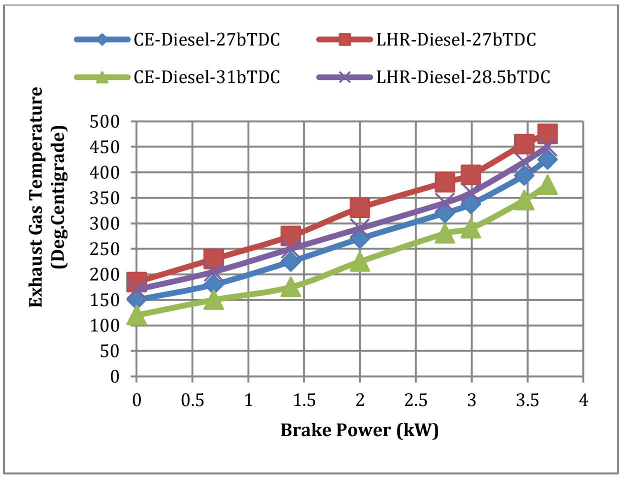


Fig.8 Variation of exhaust gas temperature (EGT) with brake power with conventional engine (CE) and LHR-3 engine at recommended injection timing and optimum injection timing

LHR-3 engine increased 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. This was due reduction of ignition delay. This was also due to higher injection advance with CE.

Fig.9 shows variation of coolant load with brake power with both versions of the engine at recommended injection timing and optimum injection timing with neat diesel operation. From Fig.9, it is observed that coolant load increased with the increase of brake power in both versions of the engine. The LHR–3 engine gave lower coolant load at all loads, when compared to conventional engine. Air being a bad conductor offers thermal resistance for heat flow through the piston and liner apart from providing insulation in the path of heat flow to the coolant.

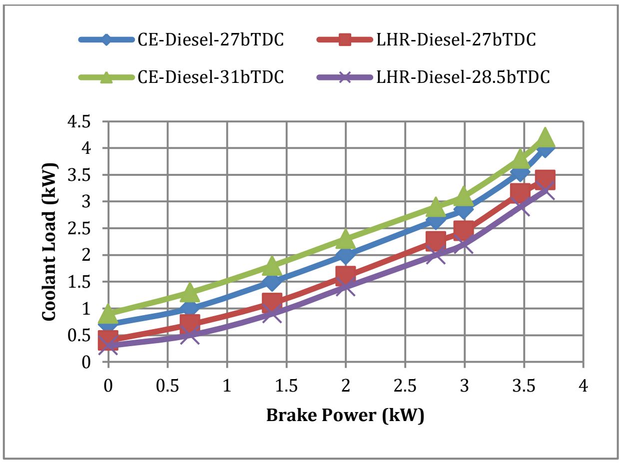


Fig.9 Variation of coolant load with brake power with conventional engine (CE) and LHR-3 engine at recommended injection timing and optimum injection timing

It was therefore evident that thermal barrier provided in the piston, liner and cylinder head resulted in reduction of coolant load. Coolant load reduced in the LHR–3 engine with advanced injection timing This was due to decrease of combustion temperatures in the LHR-3 engine 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–3 engine was due to recovery from coolant load at their respective optimum injection timings.

LHR-3 engine 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 LHR- engine at 28° bTDC.

Fig.10 shows variation of volumetric efficiency with brake power with both versions of the engine at recommended injection timing and optimum injection timing with neat diesel operation From the curves in Fig.10, it is noticed that volumetric efficiency decreased with the increase of brake power in both versions of the combustion chamber.

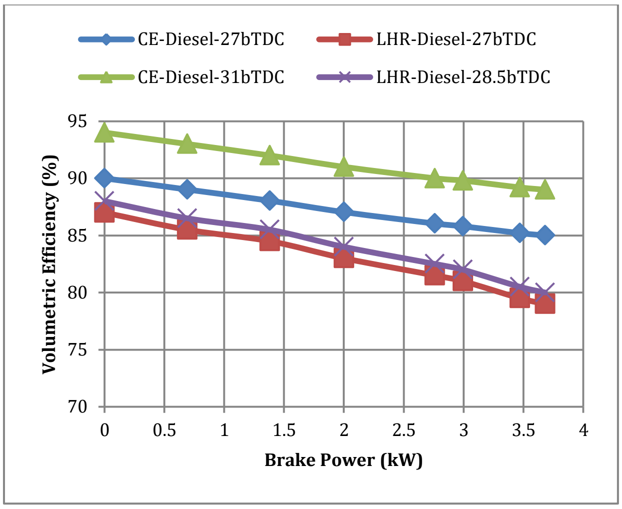


Fig.10 Variation of volumetric efficiency with brake power with conventional engine (CE) and LHR engine at recommended injection timing and optimum injection timing

This was due to increase of gas temperature with the load. LHR-3 engine gave lower volumetric efficiency at all loads when compared with CE. This was because of 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 at all loads increased marginally with advanced injection timing with both versions of the combustion chamber. This was due to reduction of combustion chamber wall temperature, which in turn depends on EGT. 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.

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

Conclusions

1) LHR-3 engine showed deteriorate performance at the full load operation in terms of brake thermal efficiency, exhaust gas temperature, volumetric efficiency and sound levels at 27 ° bTDC in comparison with conventional engine at 27 ° bTDC.

2) LHR-3 engine at 28° 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 ° 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%. .

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.

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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, et al. (2013), Effect of fuel injection timing and pressure on combustion, emissions and performance characteristics of a single cylinder diesel engine., Fuel, 111, pp 374–83.

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 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), pp 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,pp 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, pp* 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),pp 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.

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),pp 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), pp 213–221.*

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

Karthikeyan, S., Arunachalam, M., Srinivasan Rao, P. and Gopala Krishanan, 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..

Alkidas, A.C.,The influence of partial suppression of hear rejection on the performance and emissions of a divided-chamber diesel engine, SAE Paper No. 860309, 1986.

Baluswamy, N. and Selva Kumar, P., Experimental investigation of the performance of adiabatic diesel engines, Proceedings of XI National Conference on I.C. Engines and Combustion, I.I.T., Madras, 1989

Jabez Dhinagar, S., Siva Ram Prasada Rao, N., Nagalingam, B and Gopala Krishnan, K.V., (1989), Compensating the volumetric efficiency drop of an adiabatic diesel engine for improved the performance, Proceedings of XI National Conference on I.C Engines and Combustion, I.I.T., Madras.

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),pp *530-540.*