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Influence of Injector Opening Pressure on Performance Parameters of Di Diesel Engine with Three Levels of Insulation with Diesel Operation

N. Janardhan[1], M.V.S. Murali Krishna[2*] and P.Ushasri[3]

[1, 2]Mechanical Engineering Department, Chaitanya Bharathi Institute of Technology,
Gandipet, Hyderabad 500 075, Andhra Pradesh, India,
[1]E-mail: narambhatlu.datta@gmail.com, [2]E-mail:maddalivs@gmail.com,
[3]Mechanical Engineering Department, College of Engineering, Osmania University, Hyderabad 500 007, Andhra Pradesh, India.
E-mail: emailushasri@gmail.com

Abstract: *Experiments were conducted to evaluate the performance of direct injection (DI) diesel engine with different levels of low heat rejection (LHR) combustion chamber such as i) ceramic coated cylinder head, ii) air gap insulated piston and air gap insulated liner and iii) ceramic coated cylinder head along with air gap insulation with pure diesel operation with varied injector opening pressure. 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 engine. Engine with ceramic coated cylinder head showed marginal improvement in performance at all loads, engine with air gap insulated piston and air gap insulated liner showed improved performance at 80% of the full load operation while engine with ceramic coated cylinder head, air gap insulated piston and air gap insulated liner showed deteriorated performance at all loads at manufacturer's recommended injector opening pressure of 190 bar when compared with conventional engine with diesel operation. However, performance improved further with an increase of injector opening pressure with different versions of the combustion chamber.*

Keywords: *Conventional engine, LHR combustion chamber, Performance, Exhaust emissions.*

1. INTRODUCTION

Conservation of diesel fuel is necessary due to i) depletion of fossil fuel, as diesel fuel is being consumed in not only in transport sector but also in agriculture sector, ii) increase of fuel prices in International market leading to burden on economic sector of Govt. of India. [1]. In this context, the Nation should pay high tributes and gratitude towards Dr. Diesel, for his remarkable invention of diesel engine, which is being used for mass transport and heavy duty engines and civilization of any country is linked with number of vehicles being used by its public. [2]

In the last one or two decades, the concept of adiabatic engine has gained importance. Various concepts of low heat rejection (LHR) combustion chambers are being developed employing the techniques like ceramic coating in the components, air gap in the piston and the other components. It is also found that the coatings provided on cylinder head is simple technique with advanced coating techniques. The technique of providing air gap in the piston is less effective in achieving lower brake specific fuel consumption and reduction of pollutants. It also provides lower degree of insulation causing combustion chamber of diesel engine less hot. Hence the technique of air gap insulated piston, air gap insulated liner and insulated cylinder head is finding favor from the various researchers from the point of view of effectiveness ease of manufacturer and operation. LHR combustion chambers were classified as low degree (LHR–1) insulation such as ceramic coated combustion chambers, medium grade (LHR–2) insulation such as air gap insulated combustion chamber and high grade (LHR–3) insulation such as the combination of low grade and medium grade LHR combustion chambers.

Investigations were carried out by various researchers on LHR–1 combustion chambers- ceramic coated engines with pure diesel operation.[3–5]. It was reported from their investigations that brake specific fuel consumption (BSFC) improved in the range 5–9% and smoke levels decreased with ceramic coated combustion chamber. Investigations were extended to

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vegetable oil operation and plastic oil operation with ceramic coated LHR combustion chamber and reported from their investigations that performance improved marginally with ceramic coated LHR combustion chamber when compared with pure diesel operation on conventional engine. [6-7]

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 [8]. Investigations were carried out with LHR–2 combustion chamber with air gap insulated piston with nimonic crown threaded with the body of the piston fuelled with pure diesel with varied injection timing and reported brake specific fuel consumption improved by 5% [9].

By controlling the injector opening pressure and the injection rate, the spray cone angle is found to depend on injector opening pressure. Few investigators reported that injector opening pressure has a significance effect on the performance and formation of pollutants inside the direct injection diesel engine.[10-11]

Experiments were conducted on high grade LHR combustion chamber with pure diesel operation.[9]. It was reported that performance deteriorated with pure diesel operation at recommended injector opening pressure of 190 bar.

The present paper attempted to evaluate the performance of different configurations of LHR combustion chamber, with pure diesel operation with varied injector opening pressure. Comparative performance studies were made with different versions of LHR combustion chamber with conventional engine (CE) with diesel operation.

2. MATERIAL AND METHOD

The physical-chemical properties of the diesel fuel are presented in Table-1. Fuels with flash point above 52^{o} C are considered as safe. Thus diesel is extremely safe fuel to handle. Diesel has high cetnane number, which shows efficient combustion in compression ignition engine. Diesel fuel has moderate viscosity. Hence there are no injection problems.

Table.1. Properties of diesel fuel

Fuel	Specific gravity	Kinematic viscosity @ 40^{o}C (cSt)	Flash point (Open cup) (^{o} C)	Low Calorific value (kJ/kg)
Diesel	0.84	2.25	66	42000

Engine with high grade LHR combustion chamber contained a two–part piston; the top crown made of low thermal conductivity material, superni–90 screwed to aluminum body of the piston, providing a 3mm air gap in between the crown and the body of the piston. The optimum thickness of air gap in the air gap piston was found to be 3-mm for improved performance of the engine with diesel as fuel [5]. The height of the piston was maintained such that compression ratio was not altered. A superni-90 insert was screwed to the top portion of the liner in such a manner that an air gap of 3-mm was maintained between the insert and the liner body. At 500^{o}C the thermal conductivity of superni-90 and air are 20.92 and 0.057 W/m–K respectively. Partially stabilized zirconium (PSZ) of thickness 500 microns was coated on inside portion of cylinder head.
Schematic diagram of experimental setup used for the investigations on compression ignition diesel engine with diesel operation is shown in Fig 1. The test fuel used in the experimentation was pure diesel. The specifications of the experimental engine are shown in Table 2. The combustion chamber consisted of a direct injection type with no special arrangement for swirling motion of air. The engine was connected to an electric dynamometer for measuring its brake power. Burette method was used for finding fuel consumption of the engine. Air consumption of the engine was measured by an air–box method (Air box was provided with an orifice flow meter and U–tube water manometer). Air box was provided with damper to minimize the pressure pulsations of the engine.

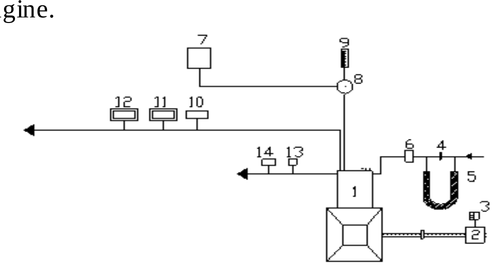

Figure 1: Schematic diagram of experimental set–up

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,

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13.Outlet jacket water temperature indicator and
14. Outlet-jacket water flow meter.

The naturally aspirated engine was provided with water–cooling system in which outlet temperature of water was maintained at 80^o C by adjusting the water flow rate. Engine oil was provided with a pressure feed system. No temperature control was incorporated, for measuring the lube oil temperature. Injector opening pressure was changed from 190 bar to 270 bar (in steps of 40 bar) using nozzle testing device. The maximum injector opening pressure was restricted to 270 bar due to practical difficulties involved. Exhaust gas temperature and coolant water outlet temperatures were measured with thermocouples made of iron and iron–constantan connected to temperature indicators. Sound levels were measured with sound analyzer at various values of brake mean effective pressure of the engine. The accuracy of the analyzer is ±1%.

Operating Conditions

Test fuel used in the experimentation was diesel. Different injector opening pressures attempted in this experiment were 190, 230 and 270 bar. The various combustion chambers used in experiment were conventional combustion chamber, low grade insulated combustion chamber with ceramic coated cylinder head (LHR–1), medium grade insulated combustion chamber (LHR–2) with air gap insulated piston and air gap insulated liner and high grade insulated combustion chamber (LHR–3) with ceramic coated cylinder head, air gap insulated piston and air gap insulated liner. The engine was started with diesel fuel and allowed to have a warm up for about 15 minutes. Each test was repeated twelve times to ensure the reproducibility of data according to the procedure adopted in error analysis. (Minimum number of trials must be not less than ten). The results were tabulated and comparative studies of performance parameters, exhaust emissions were reported at different operating conditions of the compression ignition engine.

Table.2. Specifications of the test engine

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

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
HSU=Hartridge smoke unit
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-1= Insulated combustion chamber with ceramic coated cylinder head
LHR-2= Insulated combustion chamber with air gap insulated piston and air gap insulated liner
LHR-3= Combination of LHR-1 & LHR-2 combustion chambers
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, ^{o} C
T_I=inlet temperature of water, ^{o} C
T_o=outlet temperature of water, ^{o}C
V=voltmeter reading, volt

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V_s=stroke volume, m^3
VE=Volumetric efficiency, %

Definitions of used values:

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

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

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

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

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

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

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

$\eta_v = \frac{m_a \times 1}{60 \times \rho_a \times N \times V_s}$ equation (8)

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

3. RESULTS AND DISCUSSION

Brake thermal efficiency was calculated by using equations (1-3). Brake mean effective pressure was calculated from equation (4). Curves in Fig.2 indicates that BTE increased up to 80% of the full load in the engine with different versions of the combustion chamber and beyond this load it decreased with diesel operation. This was due to increase of fuel conversion efficiency because of improved oxygen–fuel ratios and beyond 80% of the full load, performance deteriorated with decrease of oxygen fuel ratios, volumetric efficiency and increase of friction power. Engine with LHR–1 combustion chamber increased BTE marginally at all loads when compared with conventional engine. This was due to improved oxygen–fuel ratios. Engine with LHR–2 combustion chamber increased BTE up to 80% of the full load operation and beyond this load it decreased, when compared with conventional engine with pure diesel operation. The increase of BTE was due to improved air fuel ratios and improved combustion. As the combustion chamber was insulated to greater extent, it was expected that high combustion temperatures would be prevalent in engine with LHR–2 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. Engine with LHR–3 combustion chamber showed deteriorated performance at all loads when compared with conventional engine. This was due to increase of ignition delay.

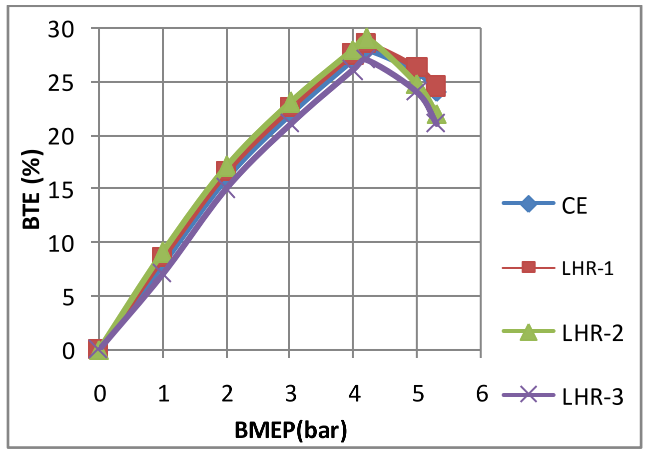

Figure 2: Variation of brake thermal efficiency (BMEP) with brake mean effective pressure (BMEP) with pure diesel operation in conventional engine and engine with different versions of combustion chambers at an injection timing of 27^o bTDC and injector opening pressure of 190 bar.

Fig.3 indicates that exhaust gas temperature (EGT) increased with increase of BMEP for different versions of the combustion chamber. This was because of increased fuel consumption with which gas temperatures increased and hence EGT. Engine with LHR–1 combustion chamber and LHR–3 combustion chamber increased EGT when compared with conventional engine at all loads of operation. This was due to restriction of heat rejection through presence of insulated components in combustion chamber.

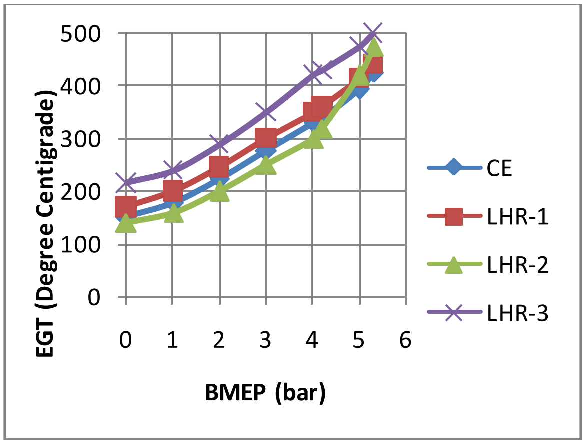

Figure 3: Variation of exhaust gas temperature (EGT) with brake mean effective pressure (BMEP)with pure diesel operation in conventional engine and engine with different versions of the combustion chamber at an injection timing of 27^{o}bTDC and injector opening pressure of 190 bar.

The engine with LHR–2 combustion chamber at the recommended injection timing recorded lower EGT up to 80% of the load and beyond that load it increased compared to conventional engine at the recommended injection timing. This was because of improved oxygen fuel ratios up to 80% of the full load operation. Beyond this load, heat rejection was restricted through the piston and liner, thus maintaining the hot combustion chamber as result of which the exhaust gas temperature increased.

Coolant load was calculated by using equation (5). Curves in Fig.4 indicate that coolant load increased with increase of BMEP for different versions of the combustion chamber This was due to increase of combustion temperatures due to consumption of fuel. Engine with LHR–1 combustion chamber and LHR–3 combustion chamber showed reduction in coolant load at all loads when compared with conventional engine. This was due to provision of insulation in the path of heat flow to the coolant. The engine with LHR–2 combustion chamber gave lesser 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. 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. This was due to improved oxygen- fuel ratios with which gas temperatures decreased. Beyond 80% of the full load, coolant load in the engine with LHR–2 combustion chamber increased over and above that of the convectional engine, with which efficiency was deteriorated at full load of the engine with 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 full load; a greater amount of heat was being transferred through the un-insulated cylinder head, where higher temperatures exist.

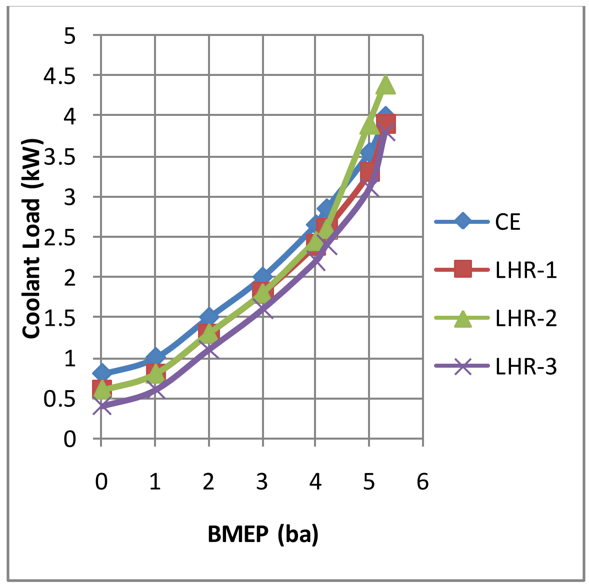

Figure 4: Variation of coolant load with brake mean effective pressure (BMEP) with pure diesel operation in conventional engine and engine with different versions of the combustion chamber at an injection timing of 27^{o}bTDC and injector opening pressure of 190 bar.

Volumetric efficiency was calculated by using equations (6-9). Curves in Fig.5 indicate that volumetric efficiency decreased with the increase of brake mean effective pressure (BMEP) in different versions of the combustion chamber. This was due to increase of gas temperature with the load. At the recommended injection timing, volumetric efficiency in the engine with different versions of insulated combustion chambers decreased at all loads when compared to the conventional engine. This was due

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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.

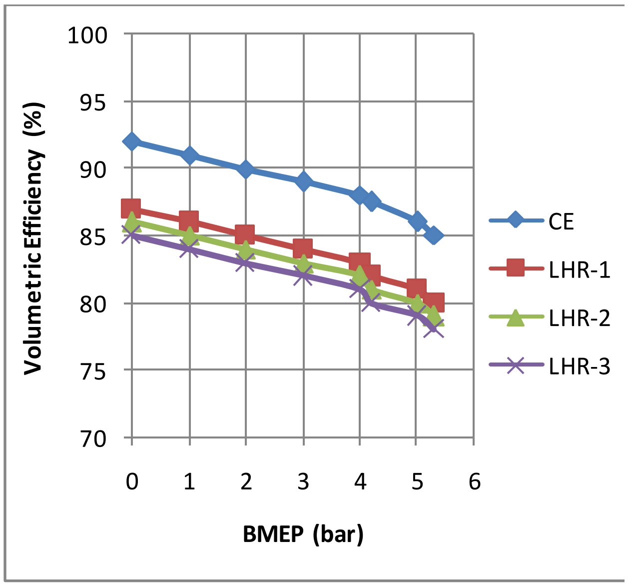

Figure 5: Variation of volumetric efficiency with brake mean effective pressure(BMEP) with pure diesel operation in conventional engine and engine with different versions of the combustion chamber at an injection timing of 27^{o}bTDC and injector opening pressure of 190 bar.

Fig.6 shows that sound intensity increased up to 80% of the full load and decreased at 80% of the full load and beyond it increased again for different versions of the combustion chamber. This was due to increase of thermal efficiency at 80% of the full load. Beyond that load, thermal efficiency decreased due to reduction of volumetric efficiency. Sound levels decreased with engine with LHR–1 combustion chamber, while it increased for engine with LHR–3. combustion chamber at all loads, when compared with conventional engine. This was due to improved combustion for engine with LHR–1 combustion chamber, while combustion deteriorated with engine with LHR–3 combustion chamber when compared with conventional engine. Sound levels decreased in engine with LHR–2 combustion chamber up to 80% of the full load when compared with conventional engine. This showed that combustion improved with engine with LHR–2 combustion chamber up to 80% of the load. However, sound intensity increased over and above conventional engine. This was due to deterioration of the performance of the engine with LHR–2 combustion chamber beyond 80% of the full load.

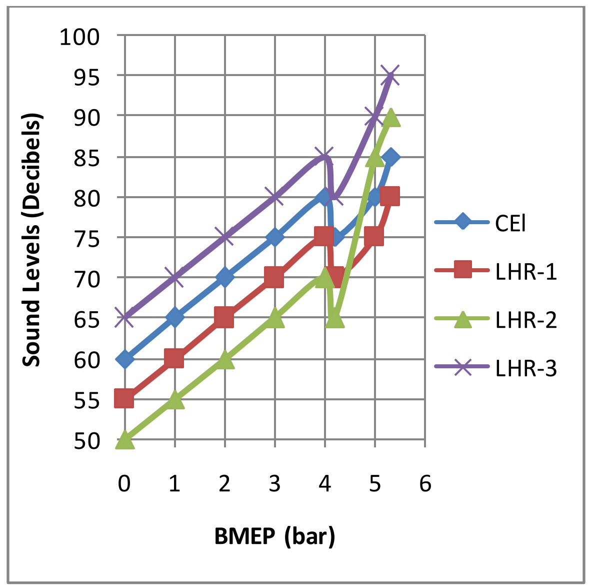

Figure 6: Variation of sound levels with brake mean effective pressure (BMEP) with pure diesel operation in conventional engine and engine with different versions of the LHR combustion chamber at an injection timing of 27^{o} bTDC and injector opening pressure of 190 bar.

Effect of Injector Opening Pressure

Table.3 shows variation of performance parameters with injector opening pressure for conventional engine and engine with different versions of combustion chamber with pure diesel operation. Peak BTE increased with increase of injector opening pressure marginally in different versions of the engine, at the recommended injection timing. This was due to improved fuel spay characteristics of the fuel at the increased injection pressure. Poor performance at lower injection pressures indicated slow mixing probably because of insufficient spray penetration with consequent slow mixing during

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diffusion burning. Peak BTE was higher with LHR–2 combustion chamber at an injector opening pressure of 190 bar, when compared with other versions of the combustion chamber. This was due to improved oxygen fuel ratios.

Brake specific fuel consumption (BSFC) decreased with increase of injector opening pressure with both versions of the combustion chamber. This was due to increase of fuel spray characteristics. Poor performance at lower injection pressures indicated slow mixing probably because of insufficient spray penetration with consequent slow mixing during diffusion burning. BSFC of the conventional engine was improved at a higher injector opening pressure of 270 bar, when compared to the same version of the engine at the recommended injection pressure of 190 bar. This was due to improved atomization of the fuel at higher injector opening pressure. BSFC was lower with LHR–1 combustion chamber at full load operation when compared with other configurations of insulated combustion chamber. This was due to improved volumetric efficiency and utilization of air.

Exhaust gas temperature (EGT) decreased marginally with increase of injector opening pressure with diesel operation. This was due to improved air fuel ratios with improved spray characteristics. EGT was lower with LHR–1 insulated combustion chamber when compared with other insulated combustion chambers. This was due to lower insulation provided with LHR–1 combustion chamber.

Coolant load was lower with LHR–3 combustion chamber when compared with other versions of the insulated combustion chamber. This was due to high degree of insulation provided with LHR–3 combustion chamber.

Volumetric efficiency increased marginally with increase of injector opening pressure in conventional engine and engine with different versions of the LHR combustion chamber. This was due to improved spray characteristics leading to reduce gas temperatures. This was also reduction of exhaust gas temperatures with increase of injector opening pressure as volumetric efficiency depends on combustion chamber wall temperature. Volumetric efficiency was lower with LHR–3 combustion chamber due to provision of high degree of insulation, with which gas temperatures increased leading to reduce density of air and hence volumetric efficiency.

Table.3 Variation of performance parameters with injector opening pressure with diesel operation

Parameter /Unit	Conventional Engine			Engine with LHR combustion chamber								
				LHR–1			LHR–2			LHR–3		
				Injector Opening Pressure (bar))			Injector Opening Pressure (bar)			Injector Opening Pressure (bar)		
	190	230	270	190	230	270	190	230	270	190	230	270
Peak BTE(%)	28	29	30	28.5	29	29.5	29	30	30.5	27	28	29
BSFC (kg/kW.h)	0.34	0.33	0.32	0.33	0.32	0.31	0.35	0.34	0.33	0.36	0.35	0.34
Exhaust Gas Temperature (^{o}C)	425	410	395	440	420	400	475	450	425	500	475	450
Coolant Load (kW)	4.0	4.2	4.4	3.9	3.7	3.5	4.5	4.2	3.8	3.8	3.6	3.4
Volumetric Efficiency (%)	85	86	87	80	81	82	79	80	81	78	79	80
Sound Levels (Decibels)	85	80	75	80	75	70	90	85	80	95	90	85

Sound levels decreased marginally with increase of injector opening pressure in both versions of the combustion chamber at full load operation. This showed that combustion improved with increase of injector opening pressure. This was because of increase of fuel atomization characteristics. Sound levels decreased with LHR–1 combustion chamber, as gas temperatures were lower and sound velocity depends on temperature of the gas. However, sound levels were intermittent in engine with LHR–3

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combustion chamber, as ceramic material absorbs sound levels, though temperatures were higher due to the provision of the insulation.

4. CONCLUSIONS

Comparative studies were made on performance parameters and pollution levels with different configurations of the combustion chamber with pure diesel operation.

In comparison with conventional engine, engine with LHR–1 combustion chamber with ceramic coated cylinder head with diesel operation increased peak brake thermal efficiency by 2%, decreased brake specific fuel consumption by 3%, increased exhaust gas temperature by 4%, decreased coolant load by 3%, decreased volumetric efficiency by 6%, decreased sound levels by 6% at an injector opening pressure of 190 bar.

In comparison with conventional engine, engine with LHR–2 combustion chamber with air gap insulated piston and air gap insulated liner with diesel operation increased peak brake thermal efficiency by 3%, brake specific fuel consumption by 3%, exhaust gas temperature by 12%, coolant load by 12%, decreased volumetric efficiency by 7%, and increased sound levels by 6%, at an injector opening pressure of 190 bar.

In comparison with conventional engine, engine with LHR–3 combustion chamber with air gap insulated piston, air gap insulated liner and ceramic coated cylinder head with diesel operation decreased peak brake thermal efficiency by 3%, increased brake specific fuel consumption by 6%, exhaust gas temperature by 18%, decreased coolant load by 5%, decreased volumetric efficiency by 8% and increased sound levels by12%.

However, performance parameters improved with engine with various configurations of LHR combustion chamber at an injector opening pressure of 270 bar in comparison with 190 bar.

4.1 Future Scope of Work

Injection timing can also be varied along with change of injector opening pressure. Investigations can be extended to alternate fuels for diesel fuel like vegetable oils, biodiesel and alcohols.

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