# INFLUENCE OF INJECTION TIMING ON PERFORMANCE PARAMETERS OF DI DIESEL ENGINE WITH AIR GAP INSULATION WITH LINSEED BIODIESEL

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## ABSTRACT

Investigations were carried out to study evaluate the performance of a medium grade low heat rejection (LHR) diesel engine consisting of air gap insulated piston with superni (an alloy of nickel) crown and air gap insulated liner with superni insert with different operating conditions [normal temperature and pre–heated temperature] of linseed biodiesel with varied injection timing. Performance parameters of brake thermal efficiency, exhaust gas temperature, volumetric efficiency and coolant load were evaluated at different values of brake mean effective pressure (BMEP) of the engine. Comparative studies were made with conventional engine (CE) with biodiesel and also with mineral diesel operation with similar working condition. The optimum injection timing was $31^{o}$ bTDC (before top dead centre) with conventional engine while it was $29^{o}$ bTDC for engine with LHR combustion chamber with biodiesel operation. Performance improved with engine with LHR combustion chamber with biodiesel in comparison with CE.

**Keywords**: Crude vegetable oil; biodiesel; exhaust emissions; LHR combustion chamber.

## 1. Introduction

Vegetable oils are promising substitutes for diesel fuel, as they are renewable in nature and properties are comparable to diesel fuel in scenario of depletion of fossil fuels and ever increase of fuel prices in International Market and increase of pollution levels with fossil fuels. The idea of using vegetable oil as fuel has been around from the birth of diesel engine. Rudolph diesel, the inventor of the engine that bears his name, experimented with fuels ranging from powdered coal to peanut oil and hinted that vegetable oil would be the future fuel [1]. Several researchers experimented the use of vegetable oils as fuel on conventional engines. They reported that the performance was poor, citing the problems of high viscosity, low volatility and their polyunsaturated character. [1–3].These problems can be solved to some extent, if neat vegetable oils are chemically modified (esterified) to bio-diesel. Experiments were conducted on conventional diesel engine with biodiesel operation. They reported that biodiesel increased efficiency marginally and decreased particulate emissions and increased oxides of nitrogen. [4–6]. The drawbacks (high viscosity and low volatility) of biodiesel call for LHR engine which provide hot combustion chamber for burning these fuels which got high duration of combustion.

The concept of engine with LHR combustion chamber is to minimize heat loss to the coolant by providing thermal insulation in the path of the coolant thereby increases the thermal efficiency of the engine. Several methods adopted for achieving LHR to the coolant are i) using ceramic coatings on piston, liner and cylinder head (low grade LHR combustion chamber) ii) 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. (medium grade LHR combustion chamber) and iii) combination of low grade and medium grade LHR combustion chamber resulted in high grade LHR combustion chamber.

Studies were made on medium grade LHR engine with biodiesel with varied injection timing and injection timing. They reported that performance was improved, decreased particulate emissions, increased $NO_x$ levels in comparison with neat diesel operation on CE. [7–11]

Experiments were conducted on preheated biodiesel in order to equalize their viscosity to that of mineral diesel may ease the problems of injection process [12–14]. Investigations were carried out on engine with preheated vegetable oils. It was reported that preheated biodiesel marginally increased thermal efficiency, decreased particulate matter emissions and $NO_x$ levels, when compared with normal biodiesel.

The present paper attempted to evaluate performance of engine with LHR combustion chamber which contained air gap insulated piston and air gap insulated liner with different operating conditions of linseed biodiesel with varied injection timing and compared with CE with biodiesel operation and also with mineral diesel operation working on similar working conditions.

## 2. MATERIAL AND METHOD

**2.1 Preparation of biodiesel**

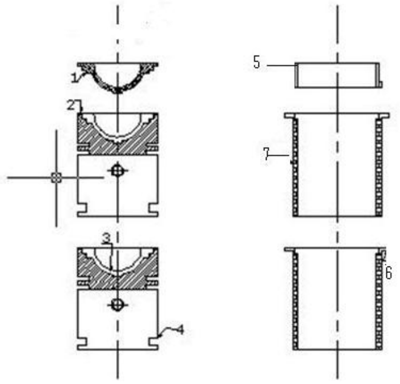
The chemical conversion of esterification reduced viscosity four fold. Linseed oil contains up to 70 % (wt.) free fatty acids. The methyl ester was produced by chemically reacting crude linseed oil with methanol in the presence of a catalyst (KOH). A two–stage process was used for the esterification of the crude linseed oil [5, 15]. The first stage (acid-catalyzed) of the process is to reduce the free fatty acids (FFA) content in linseed oil by esterification with methanol (99% pure) and acid catalyst (sulfuric acid-98% pure) in one hour time of reaction at 55°C. Molar ratio of linseed oil to methanol was 9:1 and 0.5% catalyst (w/w). In the second stage (alkali-catalyzed), the triglyceride portion of the linseed oil reacts with methanol and base catalyst (sodium hydroxide–99% pure), in one hour time of reaction at 65°C, to form methyl ester (biodiesel) and glycerol. To remove un–reacted methoxide present in raw methyl ester, it is purified by the process of water washing with air–bubbling. The properties of the Test Fuels used in the experiment were presented in Table-1.

**Table 1:** Properties Test Fuels

| Test Fuel | Viscosity at 25°C (Centi-Stroke ) | Specific gravity at 25°C | Cetane number | Lower Calorific value (kJ/kg) |
|---|---|---|---|---|
| Diesel | 2.5 | 0.82 | 51 | 42000 |
| Biodiesel (BD) | 3.7 | 0.90 | 55 | 41000 |
| ASTM Standard | ASTM D 445 | ASTM D 4809 | ASTM D 613 | ASTM D 7314 |

**2.2 Fabrication of LHR engine**

The low heat rejection diesel engine contains a two part piston (Fig.1) – the top crown made of low thermal conductivity material, superni was screwed to aluminum body of the piston, providing a 3mm air gap in between the crown and the body of the piston by placing superni gasket in between piston crown and body of the piston. A superni insert was screwed to the top portion of the liner in such a manner that an air gap of 3mm is maintained between the insert and the liner body.

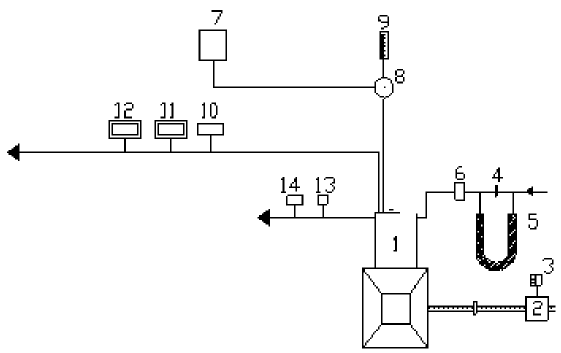



1.Superni crown with threads, 2. Superni gasket, 3. Air gap in piston, 4. Body of the piston, 5.Superni insert with threads 6. Air gap in liner, 7. Body of the liner

**Figure 1:** Assembly of the air gap insulated piston and air gap insulated liner

**2.3. Experimental Set-up**

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 linseed biodiesel in Fig.2 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 Dynamometer, 3.Load Box, 4.Orifice flow meter, 5.U-tube water manometer, 6.Air box, 7.Fuel tank, 8, Preheater 9.Burette, 10. Exhaust gas temperature indicator, 11.Smoke opacity meter, 12.NOx Analyzer, 13.Outlet–jacket water temperature indicator, 14. Outlet-jacket water flow meter, 15. Piezo-electric pressure transducer, 16.Console, 17.TDC encoder, 18.Personal Computer and 19. Printer.

**Figure 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 |
| Engine Displacement | 553 cc |
| 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 at full load | 5.31 bar |
| Manufacturer's recommended injection timing and injector opening 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 |

The naturally aspirated engine was provided with water-cooling system in which outlet temperature of water is maintained at 80$^{o}$C by adjusting the water flow rate. Engine oil was provided with a pressure feed system. No temperature control was incorporated, for measuring the lube oil temperature. The naturally aspirated engine was provided with water-cooling system in which outlet temperature of water is maintained at 80$^{o}$C by adjusting the water flow rate, which was measured by water flow meter (Part No.14). Exhaust gas temperature (EGT) and coolant water outlet temperatures were measured with thermocouples made of iron and iron-constantan attached to the exhaust gas temperature indicator (Part No.10) and outlet jacket temperature indicator (Part No.13). The accuracies of analogue temperature indicators are ±1%. 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.

**2.4 Definitions of Parameters**

$m_f = \frac{10 \times \rho \times 3600}{t \times 1000}$--(1)

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

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

$BSEC = \frac{1}{BTE}$ ----(4)

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

$$A = \frac{\pi \times D^2}{4}$$

CL= $m_w \times c_p \times (T_o - T_i)$ (6)

$m_a = C_d \times a \times \sqrt{2 \times 10 \times g \times h \times \rho_a} \times 3600$ (7)

$\rho_a = \frac{P_a \times 10^5}{750 \times R \times T_a}$ (8)

$$a = \frac{\pi \times d^2}{4}$$

$\boldsymbol{\eta_v} = \frac{\boldsymbol{m_a \times 2}}{\boldsymbol{60 \times \rho_a \times N \times V_s}}$ (9)

$$V_s = A \times L$$

**2.5. Methodology**

the fuel ($m_f$) consumed in kg/h was calculated by knowing density of fuel (ρ) in gm/cc measured with hydrometer and time taken for 10 cc of fuel measured with stop watch by using the equation–1. Brake power (BP) in kW at different percentages of load was calculated by knowing the voltmeter signal (V) and ammeter signal (I) and efficiency of dynamometer [($\eta_d$) generally assumed as 0.85] by using the equation–2. Brake thermal efficiency (BTE) was determined by knowing mass of fuel consumed, brake power and calorific value of the fuel (CV) by using the equation–3. Brake specific energy consumption (defined as energy consumed by the engine in producing unit brake power, a parameter to compare two fuels with different properties on same configuration of the engine) was determined by using the equation–4, knowing the value of BTE. Brake mean effective pressure (BMEP) of the engine in bar was determined by knowing area of cylinder (A) in square meter, bore of the cylinder (D=0.080 m), stroke of piston (L= 0.110 m), number of power cycles per minute (n), which is equal to N/2, where N is the speed of the engine (1500 rpm) and number of cylinders (k=1) by using the equation–5. Coolant load (CL) in kW was calculated by knowing mass flow rate of coolant; ($m_w$) measured known quantity of coolant in unit time, specific heat of coolant (4.18 k J/kg–K), inlet temperature of coolant ($T_i$) and outlet temperature of coolant ($T_o$) by using the equation–6. Mass flow rate of air (ma) inducted in engine in kg/h was calculated by knowing coefficient of discharge ($c_d$=0.65), area of orifice meter (a) in square meter (diameter of orifice meter, d=0.020 m ), difference of water column in U–tube water manometer (h in cm) and density of air ($\rho_a$) in kg/m$^3$ using the equation–7. Density of air was calculated from equation–8, by knowing pressure of air ($P_a$) in mm of mercury measured by barometer and temperature of ambient air ($T_a$) in Kelvin. Volumetric efficiency of engine (ηv) was calculated by equation–9, by knowing speed of the engine (N=1500 RPM), mass flow rate of air and stroke volume of cylinder ($V_s$), in m$^3$ which is equal to area of cylinder and stroke length of piston.

**2.6 Operating Conditions:** The different configurations used in the experimentation were conventional engine and engine with LHR combustion chamber. The various operating conditions of the vegetable oil used in the experiment were normal temperature (NT) and preheated temperature (PT–It is the temperature at which viscosity of the vegetable oil is matched to that of diesel fuel, 80$^o$C). Various test fuels used in the experiment were biodiesel and diesel.

# 3. RESULTS AND DISCUSSION

### 3.1 Fuel Performance

The optimum injection timing was 31$^o$ bTDC with CE, while it was 29$^o$ bTDC for engine with low grade LHR combustion chamber with mineral diesel operation [16].

From Fig.3, it is observed CE with biodiesel at 27$^o$ bTDC showed comparable performance at all loads due to improved combustion with the presence of oxygen, when compared with mineral diesel operation on CE at 27$^o$ bTDC. CE with biodiesel operation at 27$^o$ bTDC decreased peak BTE by 3%, when compared with diesel operation on CE. This was due to low calorific value and high viscosity of biodiesel. CE with biodiesel operation increased BTE at all loads with advanced injection timing, when compared with CE with biodiesel operation at 27$^{o}$ bTDC. This was due to initiation of combustion at early period and increase of resident time of fuel with air leading to increase of peak pressures. CE with biodiesel operation increased peak BTE by 7% at an optimum injection timing of 31$^{o}$ bTDC, when compared with diesel operation at 27$^{o}$ bTDC.

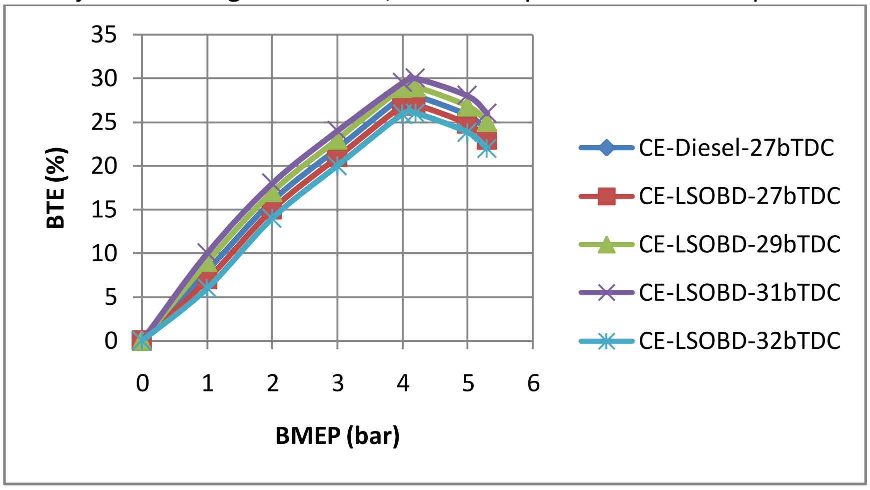



**Figure 3:** Variation of brake thermal efficiency (BTE) with brake mean effective pressure (BMEP) in conventional engine (CE) and with various injection timings at an injector opening pressure of 190 bar with biodiesel

Curves in Fig.4 indicate that LHR version of the engine at recommended injection timing showed the improved performance at all loads compared with CE with neat diesel operation. High cylinder temperatures helped in improved evaporation and faster combustion of the fuel injected into the combustion chamber. Reduction of ignition delay of the vegetable oil in the hot environment of the LHR combustion chamber improved heat release rates and efficient energy utilization. The optimum injection timing was found to be 29$^{o}$ bTDC with LHR combustion chamber with different operating conditions of biodiesel operation.

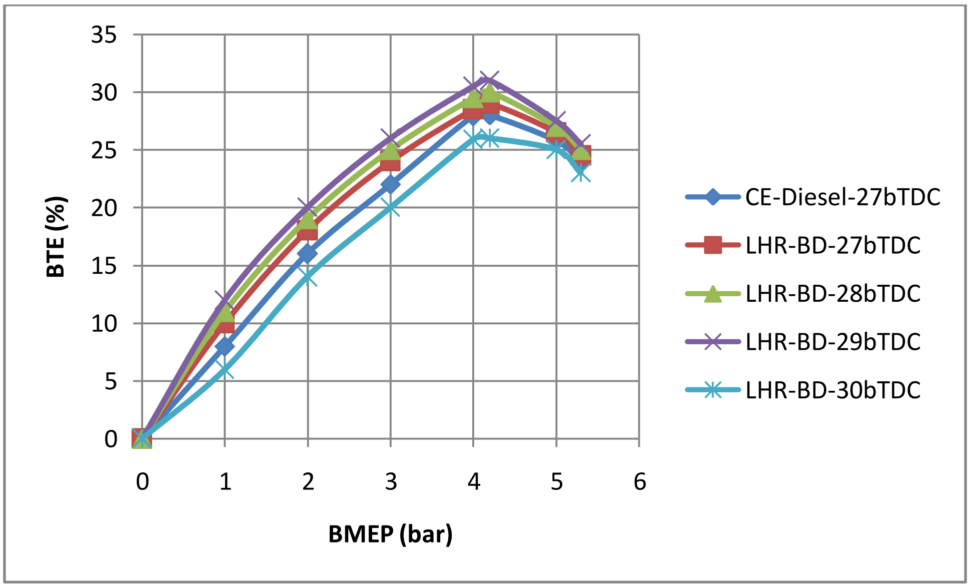



**Figure 4:** Variation of brake thermal efficiency (BTE) with brake mean effective pressure (BMEP) in LHR combustion chamber at different injection timings with biodiesel (LSOBD) operation.

Since the hot combustion chamber of LHR combustion chamber reduced ignition delay and combustion duration and hence the optimum injection timing was obtained earlier with LHR combustion chamber when compared to conventional engine with the biodiesel operation.

Part load variations are very small for the performance parameters with both versions of the combustion chamber with test fuels, and hence bar charts were drawn at full load operation in order to compare the performance.

From Fig.5, it is noticed that engine with LHR combustion chamber increased peak BTE by 7% at 27° bTDC and 3% at 29° bTDC when compared with biodiesel operation on CE at 27° bTDC and 31° bTDC . This was due to improved combustion in hot environment provided by LHR combustion chamber.

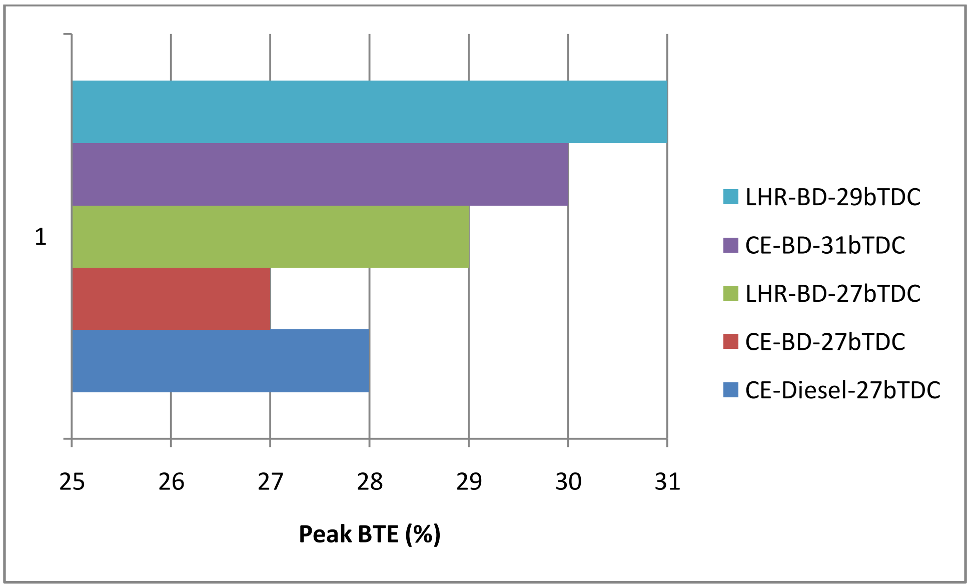



**Figure 5:** Bar charts showing the variation of peak brake thermal efficiency (BTE) with both versions of the combustion chamber at recommended and optimum injection timing with biodiesel operation.

Brake specific fuel consumption, is not used to compare the two different fuels, because their calorific value, density, chemical and physical parameters are different. Performance parameter, brake specific energy consumption (BSEC), is used to compare two different fuels by normalizing brake specific energy consumption, in terms of the amount of energy released with the given amount of fuel.

From Fig.6, it is noticed that engine with LHR combustion chamber decreased BSEC at full load operation by 19% at 27° bTDC and 27% at 29° bTDC when compared with biodiesel operation on CE at 27° bTDC and 31° bTDC. This was due to improved injection process and mass flow rate of fuel in hot environment provided by LHR combustion chamber. BSEC decreased with advanced injection timing with test fuels. This was due to initiation of combustion and increase of atomization of fuel with more contact of fuel with air. BSEC of biodiesel is almost the same as that of neat diesel fuel as shown in Fig.6. Even though viscosity of biodiesel is slightly higher than that of neat diesel, inherent oxygen of the fuel molecules improves the combustion characteristics. This is an indication of relatively more complete combustion.

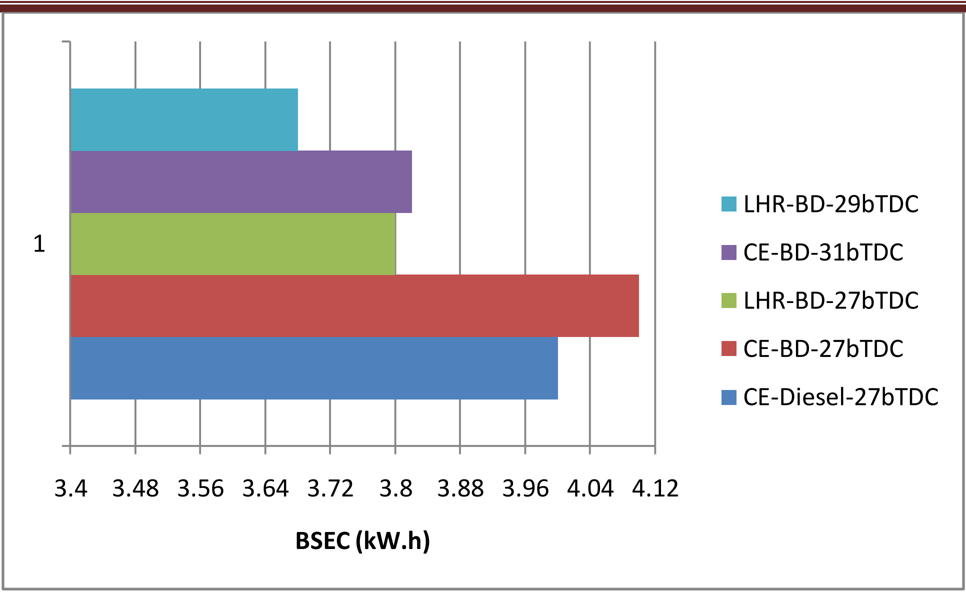



**Figure 6:** Bar charts showing the brake specific energy consumption (BSEC) with both versions of the combustion chamber at recommended and optimum injection timing with biodiesel operation at full load operation.

From Fig.7, it is noticed that engine with LHR combustion chamber increased exhaust gas temperature by 11% at full load operation at 27$^o$ bTDC and 10% at 29$^o$ bTDC when compared with biodiesel operation on CE at 27$^o$ bTDC and 31$^o$ bTDC. This indicated that heat rejection was restricted through insulated piston and insulated liner, thus maintaining the hot combustion chamber as result of which the exhaust gas temperature increased.

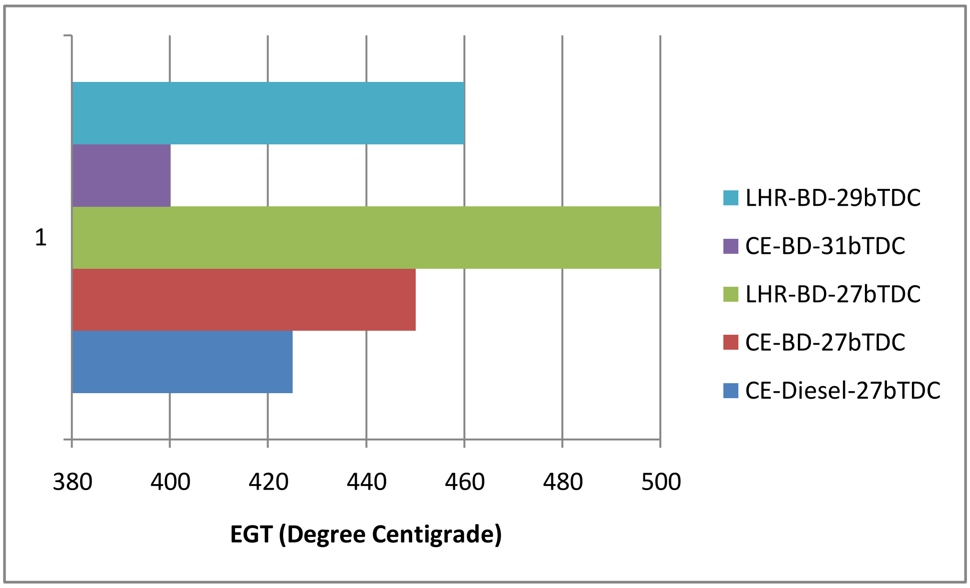



**Figure 7:** Bar charts showing the exhaust gas temperature (EGT) with both versions of the combustion chamber at recommended and optimum injection timing with biodiesel operation at full load operation

EGT at full load operation decreased with advanced injection timing with both versions of the combustion chamber. This was due to increase of contact of fuel with air and thus improving atomization characteristics of fuel leading to reduce EGT. This was also due to increase of expansion of gases with advanced injection timing.

From Fig.8, it is noticed that engine with LHR combustion chamber decreased coolant load by 29% at full load operation at 27$^o$ bTDC and 27% at 29$^o$ bTDC when compared with biodiesel operation on CE at 27$^o$ bTDC and 31$^o$ bTDC. This was due to provision of insulation with which heat rejection was reduced. Coolant load at full load operation increased with CE while decreasing with engine with LHR combustion chamber with advanced injection timing with test fuels. In case of CE, un-burnt fuel concentration reduced with effective utilization of energy, released from the combustion, coolant load with test fuels increased marginally at full load operation, due to un-burnt fuel concentration reduced with effective utilization of energy, released from the combustion, with increase of gas temperatures, when the injection timing was advanced to the optimum value. However, the improvement in the performance of CE was due to heat addition at higher temperatures and rejection at lower temperatures, while the improvement in the efficiency of the engine with LHR combustion chamber was due to recovery from coolant load at their optimum injection timings with test fuels

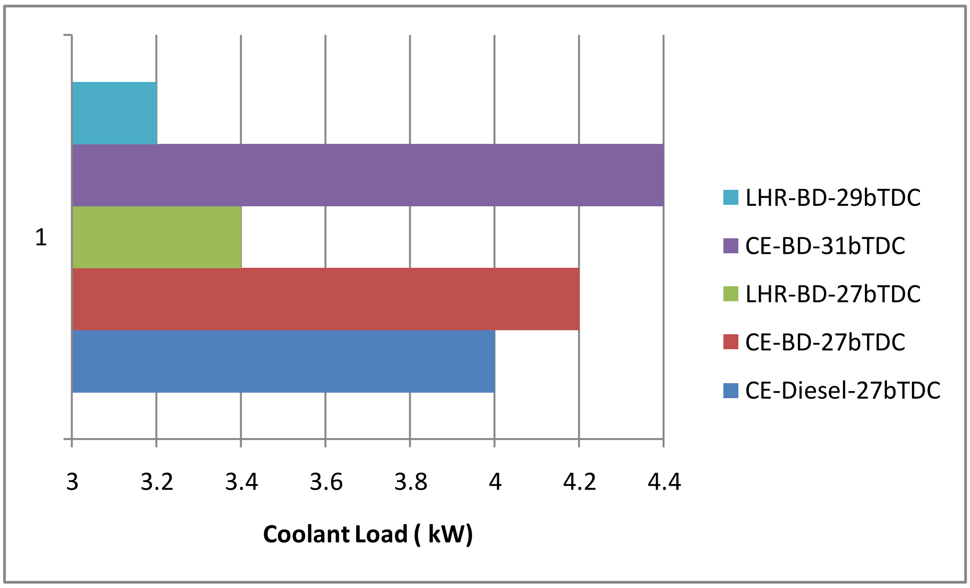



**Figure 8:** Bar charts showing the coolant load with both versions of the combustion chamber at recommended and optimum injection timing with biodiesel operation at full load operation

From Fig.9, it is noticed that engine with LHR combustion chamber decreased volumetric efficiency by 6% at full load operation at 27$^o$ bTDC and 9% at 29$^o$ bTDC when compared with biodiesel operation on CE at 27$^o$ bTDC and 31$^o$ bTDC. This was due 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. Volumetric efficiency increased marginally with both versions of the engine with test fuels with advanced injection timing. This was due to decrease of combustion chamber wall temperatures with improved air fuel ratios.

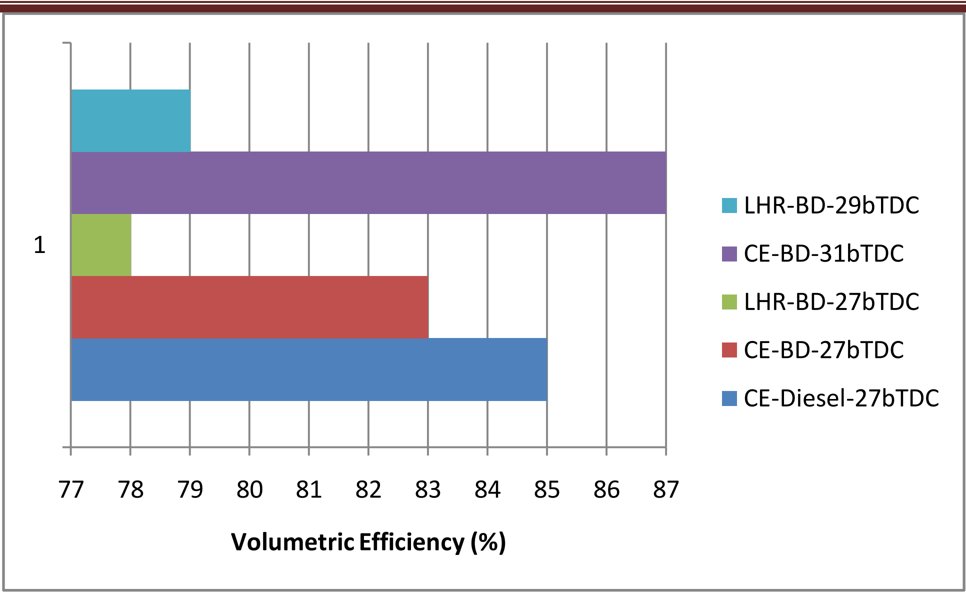



**Figure 9:** Bar charts showing the volumetric efficiency with both versions of the combustion chamber at recommended and optimum injection timing with biodiesel operation at full load operation

### 3.2 Preheated Biodiesel

Table.3 shows the data of peak BTE, BSEC at full load operation and EGT at full load operation. Peak BTE increased marginally with both versions of the combustion chamber with biodiesel operation. High cylinder temperatures helped in improved evaporation and faster combustion of the fuel injected into the combustion chamber. Reduction of ignition delay of biodiesel in the hot environment of the engine with LHR combustion chamber improved heat release rates and efficient energy utilization. Preheating of the biodiesel reduced the viscosity, which improved the spray characteristics of the oil causing efficient combustion thus improving brake thermal efficiency. The cumulative heat release was more for preheated biodiesel than that of biodiesel and this indicated that there was a significant increase of combustion in diffusion mode. This increase in heat release was mainly due to better mixing and evaporation of preheated biodiesel, which leads to complete burning.

From Table.3, it is noticed that BSEC at full load operation decreased with preheated condition of the test fuels. BSEC decreased with the preheated biodiesel at full load operation when compared with normal biodiesel. Preheating of the biodiesel reduced the viscosity, which improved the spray characteristics of the oil.

From same Table, it is noticed that EGT at full load operation of preheated biodiesel in CE was higher than that of normal biodiesel, which indicates the increase of diffused combustion due to high rate of evaporation and improved mixing between methyl ester and air. Therefore, as the fuel temperature increased, the ignition delay decreased and the main combustion phase (that is, diffusion controlled combustion) increased, which in turn raised the temperature of exhaust gases. However, EGT at full load operation marginally reduced with preheated biodiesel in engine with LHR combustion chamber due to improved air fuel ratios.

**Table 3: Data of Performance Parameters**

| Injection timing (deg. bTDC) | Combustion chamber version | Test Fuel | Peak BTE (%) | | BSEC at full load operation (kWh) | | EGT at full load operation ($^{o}$C) | |
|---|---|---|---|---|---|---|---|---|
| | | | NT | PT | NT | PT | NT | PT |
| 27 | CE | Diesel | 28 | ---- | 4.0 | -- | 425 | --- |
| | CE | BD | 27 | 28 | 4.1 | 4.0 | 450 | 425 |
| | LHR | Diesel | 29 | --- | 4.2 | -- | 475 | ---- |
| | LHR | BD | 29 | 30 | 3.8 | 3.76 | 500 | 475 |
| | LHR | Diesel | 29.5 | --- | 3.84 | --- | 450 | ---- |

| | | | | | | | | |
|---|---|---|---|---|---|---|---|---|
| 29 | LHR | BD | 31 | 31.5 | 3.68 | 3.64 | 460 | 440 |
| 31 | CE | Diesel | 31 | --- | 3.75 | --- | 375 | --- |
| | CE | BD | 30 | 30.5 | 3.82 | 3.78 | 400 | 375 |

Table.4 shows the data of performance parameters of coolant load and volumetric efficiency at full load operation. Coolant load decreased marginally with preheated biodiesel with both versions of the combustion chamber.

**Table 4: Data of Performance Parameters**

| Injection timing (deg. bTDC) | Combustion chamber version | Test Fuel | Coolant Load at full load operation (kW) | | Volumetric Efficiency at full load operation (%) | |
|---|---|---|---|---|---|---|
| | | | NT | PT | NT | PT |
| 27 | CE | Diesel | 4.0 | --- | 85 | ---- |
| | CE | BD | 4.2 | 4.0 | 83 | 82 |
| | LHR | Diesel | 4.2 | --- | 79 | --- |
| | LHR | BD | 3.4 | 3.2 | 78 | 79 |
| 29 | LHR | Diesel | 3.7 | ---- | 80 | ---- |
| | LHR | BD | 3.2 | 3.0 | 79 | 80 |
| 31 | CE | Diesel | 4.2 | --- | 89 | --- |
| | CE | BD | 4.4 | 4.2 | 87 | 86 |

This was due to reduction of gas temperatures with preheated biodiesel. Volumetric efficiency decreased marginally with CE, while it increased slightly with engine with LHR combustion chamber with preheated biodiesel. This was due to reduction of combustion chamber wall temperatures with engine with LHR combustion chamber and increase of the same with CE.

# 4. SUMMARY

Advanced injection timing improved performance with biodiesel operation on engine with LHR combustion chamber. Preheated biodiesel improved performance further in both versions of the combustion chamber.

**Comparison with CE with biodiesel**

Engine with LHR combustion chamber increased peak BTE by 7% and 3%, at full load operation– it decreased BSEC by 19% and 27%; It increased EGT by 11% and 10%; It decreased coolant load by 29% and 27%; It decreased volumetric efficiency by 6% and 9% when compared with biodiesel operation on CE at 27$^{o}$ bTDC and 31$^{o}$ bTDC.

**Comparison with mineral diesel operation**

Conventional engine with biodiesel operation decreased peak BTE by 3% at 27$^{o}$ bTDC and 3% at 31$^{o}$ bTDC in comparison with CE at 27$^{o}$ bTDC and 31$^{o}$ bTDC with mineral diesel operation. Engine with LHR combustion chamber with biodiesel showed comparable peak BTE at 27$^{o}$ bTDC and 5% at 29$^{o}$ bTDC in comparison with same configuration of the combustion chamber with diesel operation at 27$^{o}$ bTDC and 29$^{o}$ bTDC.

Conventional engine with biodiesel operation increased BSEC at full load operation by 2% at 27$^{o}$ bTDC and 2% at 31$^{o}$ bTDC in comparison with CE at 27$^{o}$ bTDC and 31$^{o}$ bTDC with mineral diesel operation. Engine with LHR combustion chamber with decreased BSEC at full load by 6% at 27$^{o}$ bTDC and 4% at 29$^{o}$ bTDC in comparison with same configuration of the combustion chamber with diesel operation at 27$^{o}$ bTDC and 29$^{o}$ bTDC.

Conventional engine with biodiesel operation showed comparable coolant load at 27$^{o}$ bTDC and 5% at 31$^{o}$ bTDC in comparison with CE at 27$^{o}$ bTDC and 31$^{o}$ bTDC with mineral diesel operation. Engine with LHR combustion chamber with biodiesel reduced coolant load by 19% at 27$^{o}$ bTDC and 14% at 29$^{o}$ bTDC in comparison with same configuration of the combustion chamber with diesel operation at 27$^o$ bTDC and 29$^o$ bTDC.

Conventional engine with biodiesel operation reduced volumetric efficiency at full load by 2% at 27$^o$ bTDC and 4% at 31$^o$ bTDC in comparison with CE at 27$^o$ bTDC and 31$^o$ bTDC with mineral diesel operation. Engine with LHR combustion chamber with biodiesel showed comparable volumetric efficiency at full load at 27$^o$ bTDC and at 29$^o$ bTDC in comparison with same configuration of the combustion chamber with diesel operation at 27$^o$ bTDC and 29$^o$ bTDC.

**4.1. Research Findings**

Performance with engine with air gap insulation was evaluated with varied injection timing at different operating conditions of linseed biodiesel.

**4.2 Recommendations**

Engine with medium grade LHR combustion chamber showed improved performance in comparison with CE with biodiesel operation. Injection pressure can also be varied to improve the performance further with both versions of the combustion chamber with biodiesel operation.

**4.3 Scientific Significance**

Change of injection timing was attempted to improve the performance from the engine along with change of configuration of combustion chamber with different operating conditions of the biodiesel.

**4.4 Social Significance**

Use of renewable fuels will strengthen agricultural economy, which curbs crude petroleum imports, saves foreign exchange and provides energy security besides addressing the environmental concerns and socio-economic issues.

**4.5 Novelty**

Change of injection timing of the engine was accomplished by inserting copper shims between pump body and engine frame

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