

# Experimental Investigations on Performance Parameters of High Grade Low Heat Rejection Diesel Engine with Preheated Cotton Seed Biodiesel

D. Srikanth, M.V.S. Murali Krishna, P. Usha Sri

Department of Mechanical Engineering, Sagar Group of Educational Institutions, Chevella, Hyderabad, Telangana State, I, India

Department of Mechanical Engineering, Chaitanya Bharathi Institute of Technology, Hyderabad, Telangana India

Department of Mechanical Engineering, College of Engineering, Osmania University, Hyderabad, Telangana State, India

**ABSTRACT:** Biodiesels derived from vegetable oils present a very promising alternative for diesel fuel, since they have numerous advantages compared to fossil fuels. They are renewable, biodegradable, provide energy security and foreign exchange savings besides addressing environmental concerns and socio-economic issues. However drawbacks associated with biodiesel of high viscosity and low volatility which cause combustion problems in CI engines, call for engine with hot combustion chamber. They have 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 with low heat rejection combustion chamber with crude cotton seed biodiesel. It consisted of an air gap insulated piston, an air gap insulated liner and ceramic coated cylinder head with different operating conditions of cotton seed biodiesel with varied injection timing. The optimum injection timing with conventional engine (CE) was  $31^\circ$  bTDC (before top dead centre), while it was  $28^\circ$  bTDC for engine with LHR combustion chamber with biodiesel. Comparative studies were made for engine with LHR combustion chamber and CE at manufacturer's recommended injection timing ( $27^\circ$  bTDC) and optimum injection timing with biodiesel operation. Engine with LHR combustion chamber with biodiesel showed improved performance at  $27^\circ$  bTDC and at optimum injection timing over CE.

**KEYWORDS:** Vegetable oil, Biodiesel; LHR combustion chamber; Fuel performance;

## I. INTRODUCTION

Fossil fuels are limited resources; hence, search for renewable fuels is becoming more and more prominent for ensuring energy security and environmental protection. It has been found that the vegetable oils are promising substitute for diesel fuel, because of their properties are comparable to those of diesel fuel. They are renewable and can be easily produced. When Rudolph Diesel, first invented the diesel engine, about a century ago, he demonstrated the principle by employing peanut oil. He hinted that vegetable oil would be the future fuel in diesel engine [1]. Several researchers experimented the use of vegetable oils as fuel on conventional engines (CE) and reported that the performance was poor, citing the problems of high viscosity, low volatility and their polyunsaturated character. It caused the problems of piston ring sticking, injector and combustion chamber deposits, fuel system deposits, reduced power, reduced fuel economy and increased exhaust emissions [1-5].

The problems of crude vegetable oils can be solved to some extent, if these oils are chemically modified (esterified) to biodiesel. Studies were made with biodiesel on CE [6-10]. They reported from their investigations that biodiesel

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operation showed comparable thermal efficiency, decreased particulate emissions and increased nitrogen oxide (NO<sub>x</sub>) levels, when compared with mineral diesel operation.

Experiments were conducted on preheated vegetable oils in order to equalize their viscosity to that of mineral diesel may ease the problems of injection process [11–13]. Investigations were carried out on engine with preheated vegetable oils. They reported that preheated vegetable oils marginally increased thermal efficiency, decreased particulate matter emissions and NO<sub>x</sub> levels, when compared with normal biodiesel.

The drawbacks associated with biodiesel (high viscosity and low volatility) call for hot combustion chamber, provided by low heat rejection (LHR) combustion chamber. The concept of the engine with LHR combustion chamber is reduce heat loss to the coolant with provision of thermal resistance in the path of heat flow to the coolant. Three approaches that are being pursued to decrease heat rejection are (1) Coating with low thermal conductivity materials on crown of the piston, inner portion of the liner and cylinder head (low grade LHR combustion chamber); (2) air gap insulation where air gap is provided in the piston and other components with low-thermal conductivity materials like superni (an alloy of nickel), cast iron and mild steel (medium grade LHR combustion chamber); and (3) high grade LHR engine contains air gap insulation and ceramic coated components.

Experiments were conducted on engine with high grade LHR combustion chamber with biodiesel. It consisted of an air gap (3 mm) insulation in piston as well as in liner and ceramic coated cylinder head. The engine was fuelled with biodiesel with varied injector opening pressure and injection timing [14–20]. They reported from their investigations, that engine with LHR combustion chamber at an optimum injection timing of 28° bTDC with biodiesel increased brake thermal efficiency by 10–12%, at full load operation—decreased particulate emissions by 45–50% and increased NO<sub>x</sub> levels, by 45–50% when compared with mineral diesel operation on CE at 27° bTDC.

The present paper attempted to determine the performance of the engine with high grade LHR combustion chamber. It contained an air gap (3.2 mm) insulated piston, an air gap (3.2 mm) insulated liner and ceramic coated cylinder head with cotton seed biodiesel with different operating conditions with varied injection timing and injector opening pressure. Results were compared with CE with biodiesel and also with diesel at similar operating conditions.

## II. MATERIAL AND METHOD

Cottonseeds have approximately 18% (w/w) oil content. India's cottonseed production is estimated to be around 35% of its cotton output (approximately 4.5 million metric tons). Approximately 0.30 million metric ton cottonseed oil is produced in India and it is an attractive biodiesel feedstock [5]

### 2.1 Preparation of biodiesel

The chemical conversion of esterification reduced viscosity four fold. Crude cotton seed oil contains up to 70 % (wt.) free fatty acids. The methyl ester was produced by chemically reacting crude cotton seed oil with methanol in the presence of a catalyst (KOH). A two-stage process was used for the esterification of the crude cotton seed oil [5]. The first stage (acid-catalyzed) of the process is to reduce the free fatty acids (FFA) content in cotton seed 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 cotton seed oil to methanol was 9:1 and 0.75% catalyst (w/w). In the second stage (alkali-catalyzed), the triglyceride portion of the cotton seed 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. [5].

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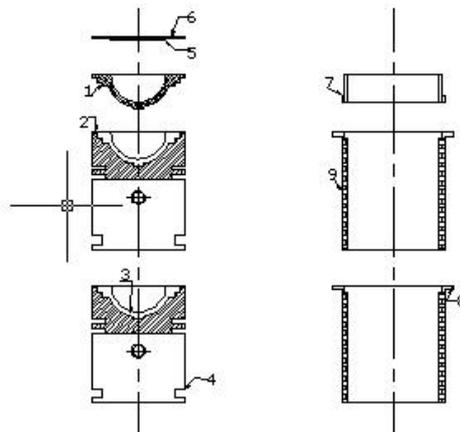
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Table.1 Properties of test fuels [5]

| Property                                 | Units        | Diesel (DF)                     | Biodiesel(BD)                    | ASTM Standard |
|--|--------------|---------------------------------|----------------------------------|---------------|
| Carbon Chain                             | --           | C <sub>8</sub> -C <sub>28</sub> | C <sub>16</sub> -C <sub>24</sub> | ---           |
| Cetane Number                            | -            | 51                              | 56                               | ASTM D 613    |
| Specific Gravity at 15°C                 | -            | 0.8275                          | 0.8673                           | ASTM D 4809   |
| Bulk Modulus at 15°C                     | MPa          | 1408.3                          | 1564                             | ASTM D 6793   |
| Kinematic Viscosity @ 40°C               | cSt          | 2.5                             | 5.44                             | ASTM D 445    |
| Air Fuel Ratio (Stoichiometric)          | --           | 14.86                           | 13.8                             | --            |
| Flash Point (Pensky Marten's Closed Cup) | °C           | 120                             | 144                              | ASTM D93      |
| Cold Filter Plugging Point               | °C           | Winter 6° C<br>Summer 18°C      | 3° C                             | ASTM D 6371   |
| Pour Point                               | °C           | Winter 3°C<br>Summer 15°C       | 0°C                              | ASTM D 97     |
| Sulfur                                   | (mg/kg, max) | 50                              | 42                               | ASTM D5453    |
| Low Calorific Value                      | MJ/kg        | 42.0                            | 39.9                             | ASTM D 7314   |
| Oxygen Content                           | %            | 0.3                             | 11                               | --            |

### 2.3 Engine with LHR combustion chamber

Fig.1 shows assembly details of insulated piston, insulated liner and ceramic coated cylinder head. Engine with LHR combustion chamber contained a two-part piston; the top crown made of superni was screwed to aluminium body of the piston, providing an air gap (3.2 mm) in between the crown and the body of the piston by placing a superni gasket in between the body and crown of the piston. A superni insert was screwed to the top portion of the liner in such a manner that an air gap of 3.2 mm was maintained between the insert and the liner body.



1.S1. 1.Piston crown with threads, 2. Superni gasket, 3. Air gap in piston, 4. Body of piston, 5. Ceramic coating on inside portion of cylinder head, 6. Cylinder head, 7.Superni insert with threads, 8.Air gap in liner, 9.Liner

Fig.1 Assembly details of air gap insulated piston, air gap insulated liner and ceramic coated cylinder head

At 500 °C the thermal conductivity of superni and air are 20.92 and 0.057 W/m-K. Partially stabilized zirconium (PSZ) of thickness 500 microns was coated by means of plasma coating technique. The combination of low thermal conductivity materials of air, superni and PSZ provide sufficient insulation for heat flow to the coolant, thus resulting in LHR combustion chamber.

### 2.4 Experimental set-up

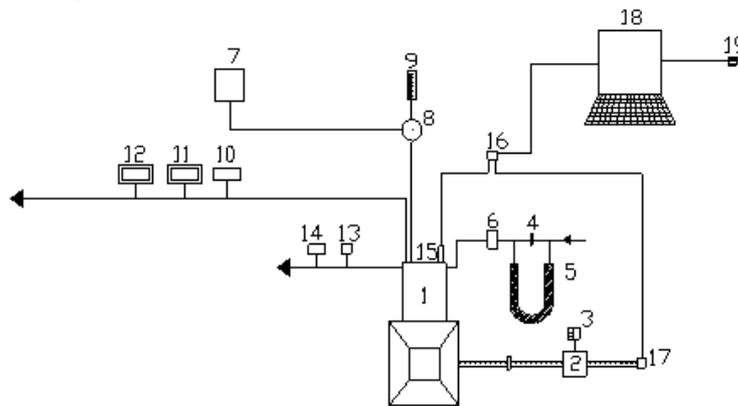
The schematic diagram of the experimental setup used for the investigations on the engine with LHR combustion chamber with cotton seed biodiesel is shown in Fig.2. Specifications of Test engine are given in Table 2. The engine

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was coupled with an electric dynamometer (Kirloskar), which was loaded by a loading rheostat. The fuel rate was measured by Burette. The accuracy of brake thermal efficiency obtained is  $\pm 2\%$ . Provision was made for preheating of biodiesel to the required levels ( $90^{\circ}\text{C}$ ) so that its viscosity was equalized to that of diesel fuel at room temperature. Air-consumption of the engine was obtained with an aid of air box, orifice flow meter and U-tube water manometer assembly. The naturally aspirated engine was provided with water-cooling system in which outlet temperature of water was maintained at  $80^{\circ}\text{C}$  by adjusting the water flow rate. The water flow rate was measured by means of analogue water flow meter, with accuracy of measurement of  $\pm 1\%$ .



1.Four Stroke Kirloskar Diesel Engine, 2.Kirloskar Electrical Dynamometer, 3.Load Box, 4.Orifice flow meter, 5.U-tube water manometer, 6.Air box, 7.Fuel tank, 8, Pre-heater 9.Burette, 10. Exhaust gas temperature indicator, 11.AVL Smoke opacity meter,12. Netel Chromatograph  $\text{NO}_x$  Analyzer, 13.Outlet jacket water temperature indicator, 14. Outlet-jacket water flow meter, 15.AVL Austria Piezo-electric pressure transducer, 16.Console, 17.AVL Austria TDC encoder, 18.Personal Computer and 19. Printer.

Fig.2 Schematic diagram of experimental set-up

Engine oil was provided with a pressure feed system. No temperature control was incorporated, for measuring the lube oil temperature. Copper shims of suitable size were provided in between the pump body and the engine frame, to vary the injection timing.

Table.2 Specifications of 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 $\times$ cylinder position $\times$ stroke            | One $\times$ Vertical position $\times$ four-stroke |
| Bore $\times$ stroke  | 80 mm $\times$ 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 | $27^{\circ}\text{bTDC} \times 190$ bar              |
| Number of holes of injector and size                                      | Three $\times$ 0.25 mm                              |
| Type of combustion chamber  | Direct injection type                               |

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Coolant water jacket inlet temperature, outlet water jacket temperature and exhaust gas temperature were measured by employing iron and iron-constantan thermocouples connected to analogue temperature indicators. The accuracies of analogue temperature indicators are  $\pm 1\%$ .

Exhaust emissions of particulate matter and nitrogen oxides ( $\text{NO}_x$ ) were recorded by smoke opacity meter (AVL India, 437) and  $\text{NO}_x$  Analyzer (Netel India; 4000 VM) at full load operation of the engine. Analyzers were allowed to adjust their zero point before each measurement. To ensure that accuracy of measured values was high, the gas analyzers were calibrated before each measurement using reference gases.

## III. RESULTS AND DISCUSSION

### 3.1 Performance parameters

The optimum injection timing with CE was  $31^\circ$  bTDC, while it was  $28^\circ$  bTDC for engine with LHR combustion chamber with diesel operation [21–22]. Fig.3 shows variation of brake thermal efficiency with brake mean effective pressure (BMEP) in conventional engine with biodiesel at various injection timings. BTE increased up to 80% of the full load and beyond that load, it decreased with biodiesel operation at various injection timings.

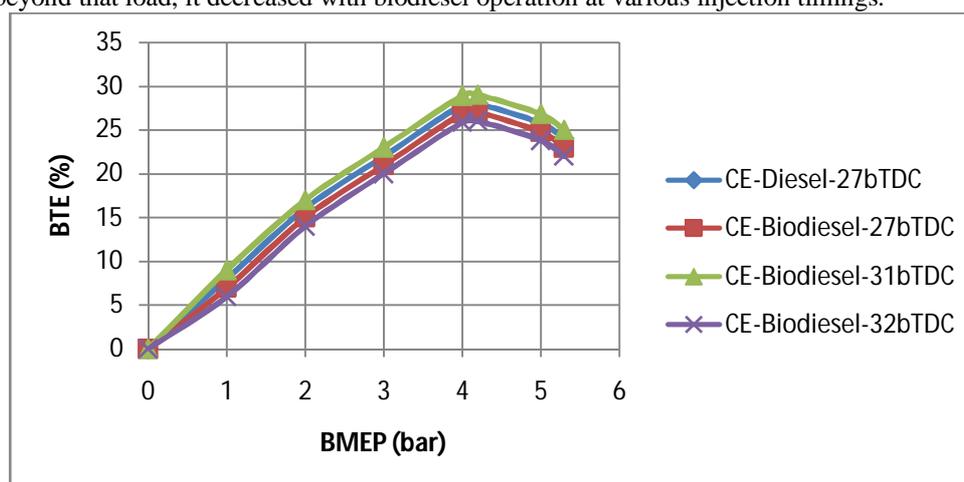


Fig. 3 Variation of brake thermal efficiency (BTE) with brake mean effective pressure (BMEP) in conventional engine (CE) with biodiesel at various injection timings at an injector opening pressure of 190 bar.

Increase of fuel conversion of efficiency up to 80% of full load and decrease of mechanical efficiency and volumetric efficiency beyond 80% of the full load and were the responsible factors for variation of BTE with respect to BMEP. Curves in Fig.3 indicate that CE with biodiesel at  $27^\circ$  bTDC showed comparable performance at all loads. The presence of oxygen in fuel composition might have improved performance with biodiesel operation, when compared with mineral diesel operation on CE at  $27^\circ$  bTDC. CE with biodiesel operation at  $27^\circ$  bTDC decreased peak BTE by 3%, when compared with diesel operation on CE. Low calorific value and high viscosity of biodiesel might have showed comparable performance with biodiesel operation in comparison with neat diesel. CE with biodiesel operation increased BTE at all loads with advanced injection timing, when compared with CE with diesel operation at  $27^\circ$  bTDC. Initiation of combustion at early period and increase of contact period of fuel with air improved performance with biodiesel when compared with diesel operation at  $27^\circ$  bTDC. CE with biodiesel operation increased peak BTE by 3% at an optimum injection timing of  $31^\circ$  bTDC, when compared with diesel operation at  $27^\circ$  bTDC.

Fig.4 shows variation of brake thermal efficiency with brake mean effective pressure (BMEP) in engine with LHR combustion chamber with biodiesel at various injection timings. This curve followed similar trends with Fig.3. From Fig.4, it is observed that at  $27^\circ$  bTDC, engine with LHR combustion chamber with biodiesel showed the improved performance at all loads when compared with diesel operation on CE. High cylinder temperatures helped in improved evaporation and faster combustion of the fuel injected into the combustion chamber. Reduction of ignition delay of the biodiesel in the hot environment of the engine with LHR combustion chamber might have improved heat release rates.

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Engine with LHR combustion chamber with biodiesel operation increased peak BTE by 14% at an optimum injection timing of 28° bTDC in comparison with mineral diesel operation on CE at 27° bTDC.

Hot combustion chamber of LHR engine reduced ignition delay and combustion duration and hence the optimum injection timing (28° bTDC) was obtained earlier with engine with LHR combustion chamber when compared with CE (31° bTDC) with biodiesel operation. Fig.5 presents bar charts showing the variation of peak BTE with test fuels. From Fig.5, it is observed that engine with LHR combustion chamber with biodiesel operation increased peak BTE by 3% at 27° bTDC and 10% at 31° bTDC in comparison with CE with biodiesel operation at 27° bTDC and at 31° bTDC. Improved evaporation of biodiesel in hot environment provided by the engine with LHR combustion chamber might have improved peak thermal efficiency of the engine.

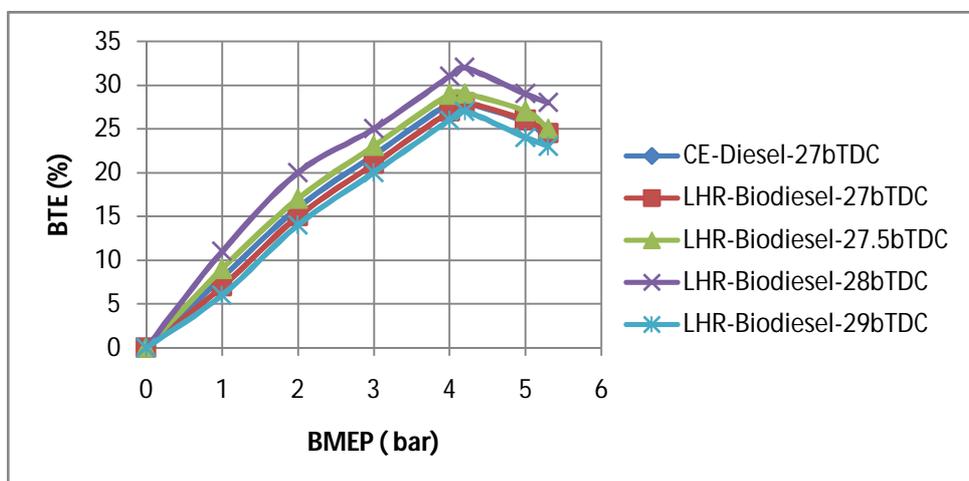


Fig.4. Variation of brake thermal efficiency (BTE) with brake mean effective pressure (BMEP) in engine with LHR combustion chamber with biodiesel at various injection timings at an injector opening pressure of 190 bar.

Engine with LHR combustion chamber with biodiesel operation showed higher peak BTE than diesel operation on same configuration of the engine. This showed that engine with LHR combustion chamber was more suitable for biodiesel.

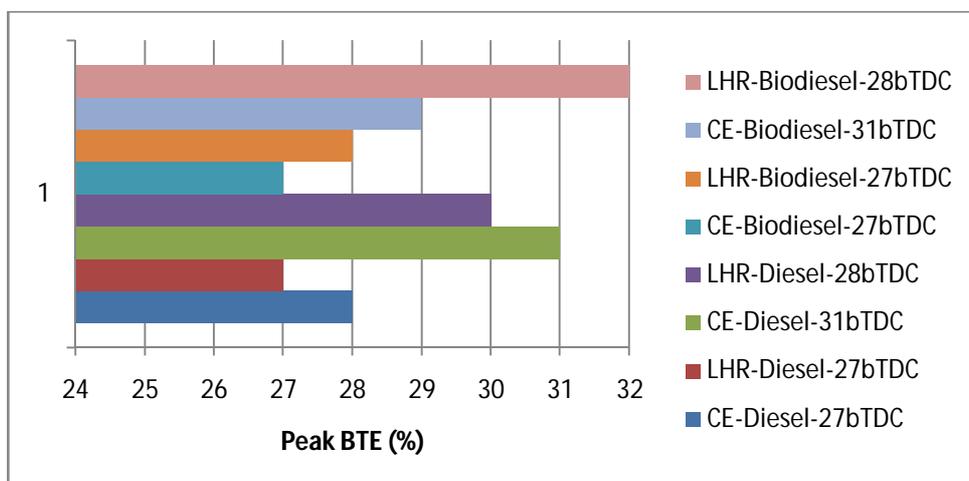


Fig.5 Bar charts showing the variation of peak brake thermal efficiency (BTE) with test fuels with both versions of the engine at recommended and optimized injection timings at an injector opening pressure of 190 bar.

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Fig.6 presents bar charts showing the variation of brake specific energy consumption (BSEC) at full load with test fuels. From Fig.6, it is shown that BSEC at full load decreased with advanced injection timing with test fuels. This was because of increase of resident time of fuel with air thus improving atomization and thus combustion. BSEC was comparable with biodiesel with CE at 27° bTDC and 31° bTDC, when compared with CE with diesel operation at 27° bTDC and at 31° bTDC. Improved combustion with higher cetane number and presence of oxygen in fuel composition with higher heat release rate with biodiesel may lead to produce comparable BSEC at full load. Engine with LHR combustion chamber with biodiesel decreased BSEC at full load operation by 6% at 27° bTDC and 3% at 28° bTDC, when compared diesel operation with engine with LHR combustion chamber at 27° bTDC and at 28° bTDC. This once again confirmed that engine with LHR combustion chamber was more suitable for biodiesel operation than neat diesel operation. Engine with LHR combustion chamber with biodiesel decreased BSEC at full load operation by 3% at 27° bTDC and 2% at 28° bTDC, in comparison with CE at 27° bTDC and at 31° bTDC with biodiesel. Improved evaporation rate and higher heat release rate of fuel with LHR combustion chamber might have improved the performance with LHR engine.

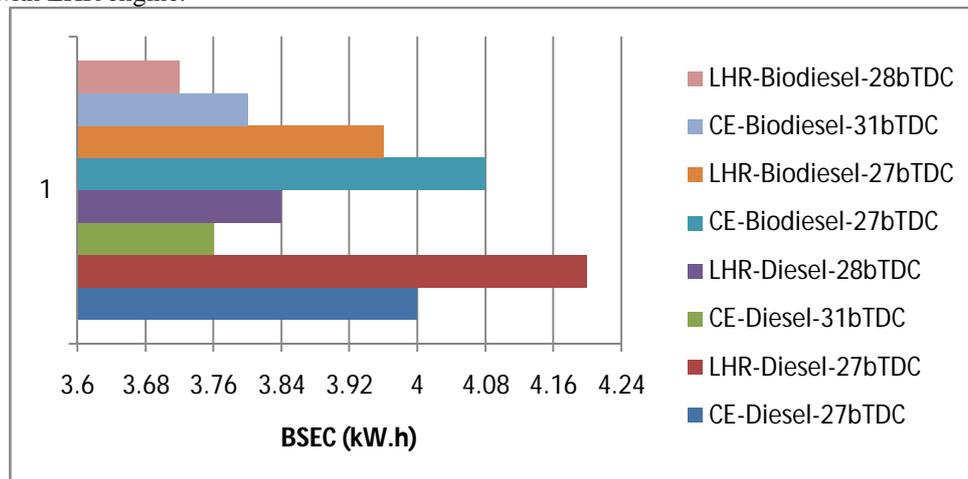


Fig.6.Bar charts showing the variation of brake specific energy consumption (BSEC) at full load operation with test fuels with both versions of the engine at recommended and optimized injection timings at an injector opening pressure of 190 bar.

Fig.7 presents bar charts showing variation of exhaust gas temperature (EGT) at full load with test fuels. From Fig.7, it is noticed that, exhaust gas temperature (EGT) at full load operation decreased with advanced injection timing with test fuels. This was 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 EGT. CE with biodiesel operation increased EGT at full load operation by 6% at 27° bTDC and 7% at 31° bTDC in comparison with CE with neat diesel operation at 27° bTDC and at 31° bTDC. Though calorific value (or heat of combustion) of biodiesel is lower than that of diesel, density of biodiesel is higher, therefore greater amount of heat was released in the combustion chamber leading to produce higher EGT at full load operation with biodiesel operation than neat diesel operation. This was also because of higher duration of combustion of biodiesel causing retarded heat release rate. Similar findings were obtained by other researchers [6–8]. From Fig.7, it is noticed that engine with LHR combustion chamber with biodiesel operation increased EGT at full load operation by 5% at 27° bTDC and 5% at 28° bTDC, when compared with diesel operation on same configuration of the engine at 27° bTDC and at 28° bTDC. High duration of combustion due to high viscosity of biodiesel in comparison with diesel might have increased EGT at full load. Engine with LHR combustion chamber with biodiesel increased EGT at full load operation by 17% at 27° bTDC and 13% at 28° bTDC, in comparison with CE at 27° bTDC and at 31° bTDC. This indicated that heat rejection was restricted through the piston, liner and cylinder head, thus maintaining the hot combustion chamber as result of which EGT at full load operation increased with reduction of ignition delay.

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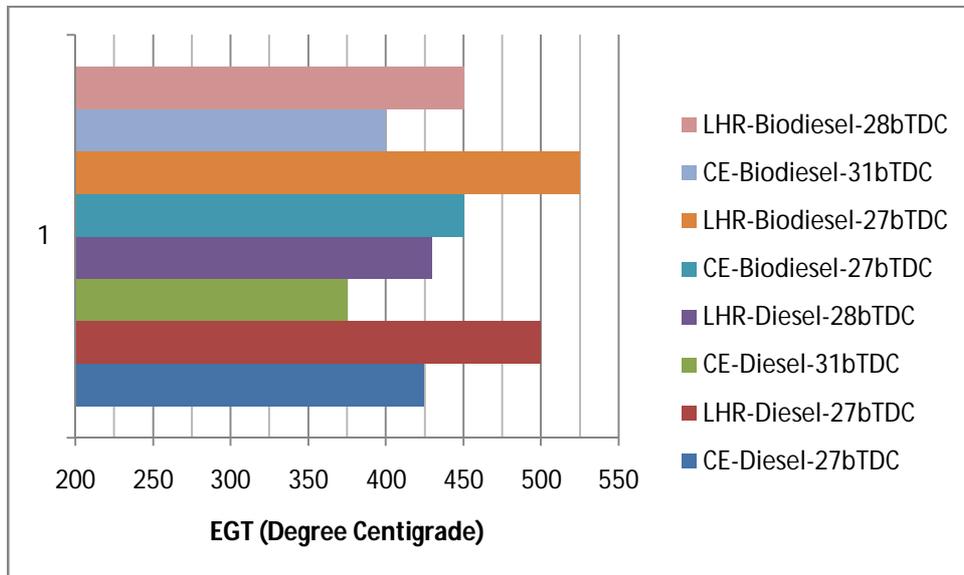


Fig.7 Bar charts showing the variation of exhaust gas temperature (EGT) at full load operation with test fuels with both versions of the engine at recommended and optimized injection timings at an injector opening pressure of 190 bar.

Table.3 shows performance parameters of peak BTE, BSEC at full load and EGT at full load,  
Table.3

Comparative data on Peak Brake Thermal Efficiency, Brake Specific Energy Consumption and Exhaust Gas Temperature at full load

| Injection timing/<br>Combustion Chamber Version | Test fuel | Peak Brake Thermal Efficiency (%) |      | Brake Specific Energy consumption at full load operation (kW.h) |      | Exhaust Gas Temperature (°C) at full load operation |     |
|---|-----------|-----------------------------------|------|---|------|---|-----|
|   |           | Fuel Operating Condition          |      | Fuel Operating Condition  |      | Fuel Operating Condition                            |     |
|   |           | NT                                | PT   | NT  | PT   | NT  | PT  |
| 27(CE)  | DF        | 28                                | --   | 4.0   | --   | 425   | --- |
|   | BD        | 27                                | 28   | 4.08  | 4.04 | 450   | 500 |
| 27(LHR)   | DF        | 27                                | --   | 4.2   | --   | 500   | --  |
|   | BD        | 28                                | 29   | 3.96  | 3.92 | 525   | 500 |
| 28(LHR)   | DF        | 30                                | --   | 3.84  | --   | 430   | --- |
|   | BD        | 32                                | 32.5 | 3.72  | 3.68 | 450   | 425 |
| 31(CE)  | DF        | 31                                | --   | 3.76  | --   | 375   | --- |
|   | BD        | 29                                | 29.5 | 3.80  | 3.76 | 400   | 425 |

From Table 3, it is noticed that preheating of the biodiesel improved the performance in both versions of the combustion chamber when compared with the biodiesel at normal temperature. Preheating (PT) reduced the viscosity of the biodiesel, causing efficient combustion thus improving BTE.

From Table.3, it is noticed that EGT at full load operation increased marginally with preheated biodiesel with CE, 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 decreased marginally with engine with LHR combustion with preheated (PT)biodiesel due to improved combustion.

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Fig.8 presents bar charts showing variation of coolant load with test fuels. CE with biodiesel increased coolant load by 3% at 27° bTDC and 10% at 31° bTDC when compared with neat diesel operation on CE at 27° bTDC and 31° bTDC as observed from Fig.8.

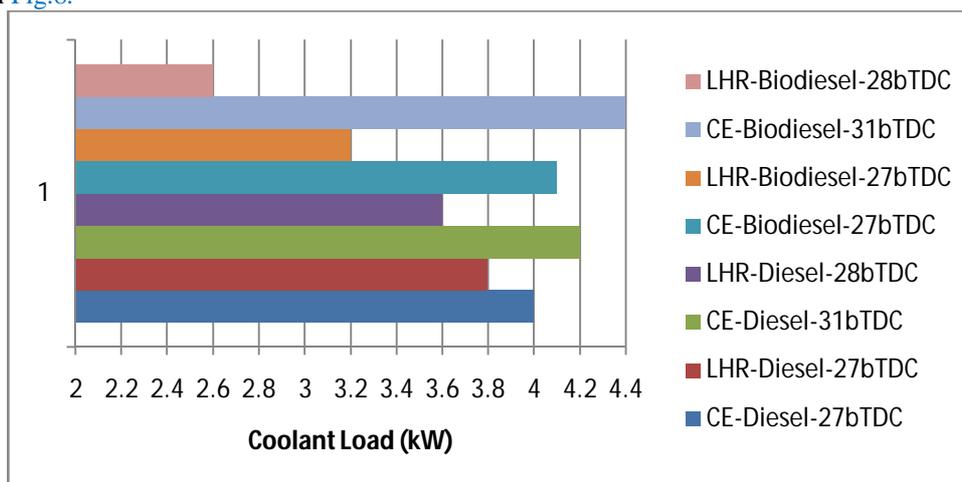


Fig.8 Bar charts showing the variation of coolant load at full load operation with test fuels with both versions of the engine at recommended and optimized injection timings at an injector opening pressure of 190 bar.

Increase of un-burnt fuel concentration at the combustion chamber walls may lead to increase of gas temperatures with biodiesel produced higher coolant load than diesel operation. Coolant load at full load operation increased in CE, while decreasing the same in engine with LHR combustion chamber with advanced injection timing with biodiesel. 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, with an increase of gas temperatures, when the injection timing was advanced to the optimum value.. The reduction of coolant load in engine with LHR combustion chamber might be due to the reduction of gas temperatures with improved combustion. Hence, 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 because of recovery from coolant load at their optimum injection timings with test fuels. Engine with LHR combustion chamber with biodiesel operation decreased coolant load operation by 16% at 27° bTDC and 28% at 28° bTDC, when compared diesel operation with same configuration of the engine at 27° bTDC and at 28° bTDC. More conversion of heat into useful work with biodiesel than diesel might have reduced coolant load with biodiesel. Fig.11 indicates that engine with LHR combustion chamber with biodiesel decreased coolant load at full load operation by 7% at 27° bTDC and 41% at 28° bTDC, in comparison with CE at 27° bTDC and at 31° bTDC. Provision of thermal insulation and improved combustion with engine with LHR combustion chamber might have reduced coolant load with LHR engine in comparison with CE with biodiesel operation.

Fig.9 shows bar charts showing variation of volumetric efficiency at full load with test fuels.

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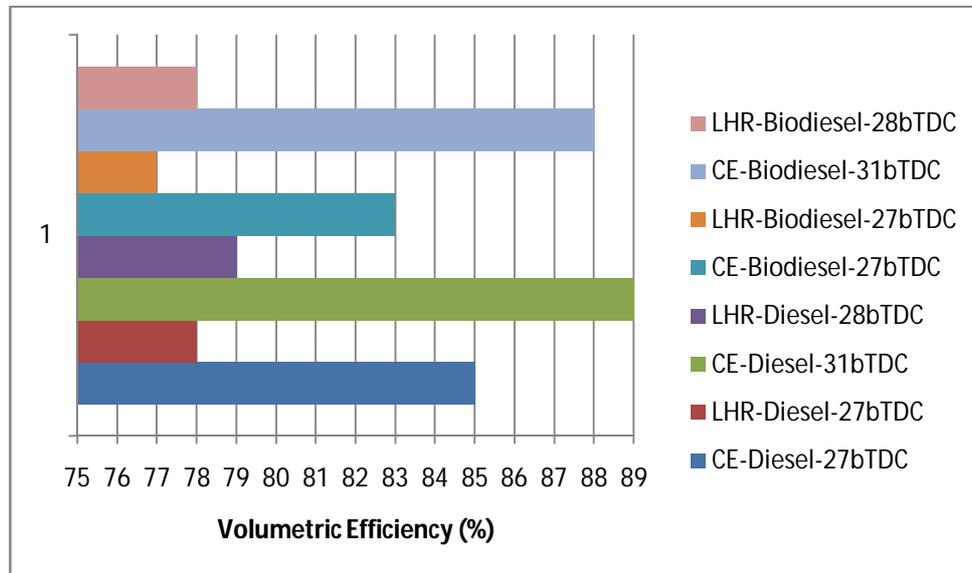


Fig.9 Bar charts showing the variation of volumetric efficiency at full load operation with test fuels with both versions of the engine at recommended and optimized injection timings at an injector opening pressure of 190 bar.

It indicates that CE with biodiesel operation decreased volumetric efficiency at full load by 2% at 27° bTDC and comparable at 31° bTDC, when compared with diesel operation on CE at 27° bTDC and 31° bTDC. Increase of EGT might have reduced volumetric efficiency at full load, as volumetric efficiency depends on combustion wall temperature, which in turn depends on EGT. Volumetric efficiency at full load operation improved marginally with advanced injection timing with test fuels with both configurations of the combustion chamber. Reduction of EGT at full load might have improved volumetric efficiency with test fuels. From Fig.9, it is noticed that volumetric efficiencies at full load operation on engine with LHR combustion chamber at 27° bTDC and at 28° bTDC with biodiesel were marginally lower than diesel operation on same configuration of the engine at 27° bTDC and 28° bTDC. Increase of EGT was responsible factor for it. Fig.9 indicates that engine with LHR combustion chamber with biodiesel decreased volumetric efficiency at full load operation by 5% at 27° bTDC and 11% at 28° bTDC, in comparison with CE at 27° bTDC and at 31° bTDC. The reduction of volumetric efficiency with engine with LHR combustion chamber was because of increase of temperatures of insulated components of LHR combustion chamber, which heat the incoming charge to high temperatures and consequently the mass of air inducted in each cycle was lower. Similar observations were noticed by earlier researchers [21–22]. Table.4 shows coolant load and volumetric efficiency at full load. Coolant load at full load operation decreased with preheating of biodiesel, as noticed from Table.4. Improved spray characteristics might have reduced gas temperatures and hence coolant load. Volumetric efficiency at full load operation marginally reduced in CE, while increasing it in engine with LHR combustion chamber with preheated biodiesel as observed from Table.4. This was because of increase of EGT in CE, while decreasing of the same in engine with LHR combustion chamber.

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Table.4 Data of coolant load and volumetric efficiency at full load

| IT/<br>Combustion<br>Chamber<br>Version | Test<br>fuel | Coolant Load<br>(kW)        |     | Volumetric Efficiency<br>(%) at full load<br>operation (%) |    |
|---|--------------|-----------------------------|-----|--|----|
|   |              | Fuel Operating<br>Condition |     | Fuel Operating<br>Condition                                |    |
|   |              | NT                          | PT  | NT   | PT |
| 27(CE)                                  | DF           | 4.0                         | -   | 85   |    |
|   | BD           | 4.1                         | 3.9 | 83   | 82 |
| 27(LHR)                                 | DF           | 3.8                         | -   | 78   |    |
|   | BD           | 3.2                         | 3.2 | 77   | 78 |
| 28(LHR)                                 | DF           | 3.6                         | -   | 79   | -  |
|   | BD           | 2.6                         | 2.4 | 78   | 79 |
| 31(CE)                                  | DF           | 4.2                         | -   | 89   |    |
|   | BD           | 4.4                         | 4.2 | 88   | 87 |

## Summary

1. Engine with LHR combustion chamber is efficient for alternative fuel like biodiesel rather than neat diesel.
2. Engine with LHR combustion chamber with biodiesel improved its performance over CE at recommended injection timing and optimized timing.
3. The performance of the engine improved with advanced injection timing, and with preheating with both versions of the combustion chamber with biodiesel.
- 4.

## 4.2 Novelty

Engine parameters (injection timing and injection pressure) fuel operating conditions (normal temperature and preheated temperature) and different configurations of the engine (conventional engine and engine with LHR combustion chamber) were used simultaneously to improve performance, exhaust emissions and combustion characteristics of the engine. Change of injection timing was accomplished by inserting copper shims between pump frame and engine body.

## Highlights

- Fuel injection pressure & timings affect engine performance.
- Performance improved with preheating of biodiesel
- Change of combustion chamber design improved the performance of the engine

## Future Scope of Work

Engine with LHR combustion chamber gave higher NO<sub>x</sub> levels, which can be controlled by means of the selective catalytic reduction (SCR) technique using lanthanum ion exchanged zeolite (catalyst-A) and urea infused lanthanum ion exchanged zeolite (catalyst-B) with different versions of combustion chamber at full load operation of the engine [30].

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