Potential of a Low Heat Rejection Diesel Engine with Renewable Fuels

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Abstract: Investigations are carried out to evaluate the performance of a low heat rejection (LHR) diesel engine consisting of air gap insulated piston with 3-mm air gap, with superni (an alloy of nickel) crown and air gap insulated liner with superni insert with different operating conditions of crude jatropha oil with varied injection timing and injection pressure. Performance parameters of brake thermal efficiency (BTE), exhaust gas temperature (EGT) and volumetric efficiency (VE) are determined at various magnitudes of brake mean effective pressure. Pollution levels of smoke and oxides of nitrogen (NOx) are recorded at the peak load operation of the engine. Combustion characteristics of the engine of peak pressure (PP), time of occurrence of peak pressure(TOPP), maximum rate of pressure rise (MRPR) and time of occurrence of maximum rate of pressure (TOMRPR) are measured with TDC (top dead centre) encoder, pressure transducer, console and special pressure-crank angle software package. Zero dimensional, multi-zone combustion model is assumed to predict combustion characteristics and validated with experimental results. Conventional engine (CE) showed deteriorated performance, while LHR engine showed improved performance with crude jatropha oil operation at recommended injection timing and pressure and the performance of both version of the engine is improved with advanced injection timing and higher injection pressure when compared with CE with operating on pure diesel. Peak brake thermal efficiency increased by 4%, smoke levels decreased by 4% and NOx levels increased by 37% with vegetable oil operation on LHR engine at its optimum injection timing, when compared with pure diesel operation on CE at manufacturer's recommended injection timing.

Keywords: Crude Jatropha Oil, Combustion Characteristics, LHR Engine, Multi-Zone Combustion Model, Performance, Pollution Levels, Zero-Dimensional

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1 INTRODUCTION

The increase of vehicle population at an alarming rate due to advancement of civilization and the use of diesel fuel not only in transport sector but also in agricultural sector, has lead to fast depletion of diesel fuels and increase of pollution levels originated from these fuels, thus the search for alternate fuels has become pertinent for engine manufacturers, users and researchers involved in the field of combustion research.

It is a well known fact that about 30% of the energy supplied is lost through the coolant and 30% is wasted through friction and other losses, thus leaving only 30% of energy utilization for useful purposes. In view of the above issue, the major thrust in engine research during the last one or two decades has been on development of LHR engines. Several adopted methods for achieving cool LHR engines are i. using ceramic coatings on piston, liner and cylinder head ii. creating air gap in the piston and other components with low-thermal conductivity materials like superni, cast iron and mild steel etc. Though ceramic coatings provided insulation and improved brake specific fuel consumption (BSFC), peeling of coating was reported by various researchers [1-4] after certain hours of trials. Creating an air gap in the piston involved the complications of joining two different metals.

Though Parker et al. [5] observed effective insulation provided by an air gap, the bolted design employed by them could not provide complete sealing of air in the air gap. Rama Mohan [6], [7] made a successful attempt of screwing the crown made of low thermal conductivity material, nimonic (an alloy of nickel) to the body of the piston, by keeping a gasket, made of nimonic, in between these two parts. The concept of LHR engine is to reduce heat loss to the coolant, specifically from the piston top to the body of the piston. It should be expected that the thickness of the air gap plays an important role on the insulation effect in LHR engines. Rama Mohan [6], [7] and Dhinagar et al. [8] followed this path in order to get higher insulation effect. Low degree of insulation provided by these researchers [6], [7] was not able to burn low cetane fuels of vegetable oils.

The complexities involved like heterogeneous combustion of diesel fuel and air in minute fraction of time, various sizes of fuel droplets and their penetration and evaporation etc., make diesel combustion usually not amenable for effective modelling. Miyairi [9] used a two-zone combustion model. A computer simulation model was used by Rafiqul Islam et al. [10] for predicting the performance of the ceramic-coated direct injection (DI) diesel engines. Ramamohan et al. [6], [7] employed zero dimensional multi-zone combustion models for predicting the performance of LHR engine and validated theoretical results with experimental data and found 5-7% deviation between these two.

The idea of using vegetable oil as fuel has been around since 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. Rehman et al. [11] carried out experimental investigations on diesel-karangi oil blends, esters with CE with varying injection pressure and injection timing and reported that the engine performance with esters was comparable with that of diesel fuel, while with blends, it was slightly inferior and performance was improved concurrent with increase of injection pressure and advancement of injection timing up to 3 degrees.

Srinivasa Rao et al. [12] conducted experiments on jatropha oil with CE. They varied injection pressures and evaluated performance parameters, like specific energy consumption and BTE. They reported that marginal improvement in the performance was observed concurrent with variation of injection pressure. Drastic smoke levels were observed with jatropha oil and these levels decreased marginally with variation of injection pressure. Sudhakar Babu et al. [13] carried out investigations with CE with the blends of rapeseed oil and diesel at recommended injection timing and pressure. They reported deterioration in the performance and increased pollution levels of smoke and carbon monoxide, when compared with pure diesel operation on CE. Kyle W.Scholl et al. [14] conducted experiments on four-cylinder, four-stroke, normally aspirated direct injection CE with soyabean oil methyl ester and reported combustion characteristics and exhaust emissions to be compatible with diesel fuel.

K. Pramanik [15] conducted experiments on CE with blends of 40-50% of jatropha oil and diesel by volume and reported improvement in specific fuel consumption. Shailendra Sinha et al. [16] conducted experimental investigations on four-stroke, four-cylinder CE with blends of 40-50% of rice bran oil based bio-diesel with diesel by volume and reported improvement in exhaust emissions. M. K. Gajendra Babu et al. [17] conducted experiments on CE with karanja oil based bio-diesel and reported improvement in BTE, exhaust emissions but increased NOx emissions and slightly increased brake specific fuel consumption (BSFC).

Jiwak et al. [18] conducted experiments on CE with blends of pongamia oil based on bio-diesel and diesel with varied injection timing and reported that improvements in engine performance and emission characteristics were observed at standard injection timing but slightly increased NOx emissions.

However NOx emissions decreased with negligible effect of fuel consumption with retarded injection timing. T. Ganapathy et al. [19] studied the viability of jatropha oil in CE and the results showed that BTE and power output were slightly lowered and fuel consumption was slightly higher when jatropha oil methyl esters were used in place of diesel fuel. Rajesh Kumar Pandey et al. [20] used karanja oil bio-diesel in CE and studied engine performance, and pollution levels by varying compression ratio with different blends of karanja based bio-diesel fuel. Experimental result showed that poor performance was obtained at lower compression ratio while running on karanja oil based bio-diesel and performance of the engine was improved at compression ratio of 18:1.

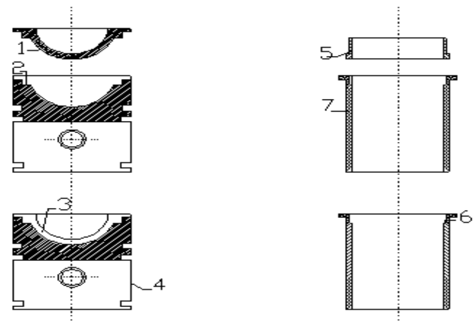
S. Choudhury et al. [21] studied karanja oil and jatropha oil as alternate fuel in CE and reported that all emission parameters were within maximum limits concluding a safer use as an alternate fuel. A. Siva Kumar et al. [22] used vegetable oils like jatropha and fish oil and compared performance parameters, emissions and combustion parameters in CE and reported that bio-diesel was proved to be a better fuel than diesel on the aspects of air-fuel ratio, volumetric efficiency, mechanical efficiency, BTE and indicated thermal efficiency. M. Pugazhvadirn et al. [23] experimented on waste frying oil a non-edible vegetable oil as an alternate fuel for diesel engine and reported CO and smoke emissions were reduced using preheated waste frying oil at 135°C.

Bhaskar et al. [24] conducted experimental investigations on jatropha oil with LHR engine, which consisted of ceramic-coated cylinder head and air gap cylinder liner. They reported that the improvement in the performance and reduction in pollution levels of hydrocarbon and smoke with LHR version of the engine with jatropha oil when compared with CE. Ignition improvers to jatropha oil further improved the performance and reduced the pollution levels. Pandu Rangadu et al. [25] conducted experiments on LHR engine with the monoesters of thumba oil and reported improvement in the performance when compared with CE with the same fuel.

S. Jabez Dhinagar et al. [26] tested three vegetable oils, neem oil, rice bran oil and karanja oil in LHR engine and reported that performance of vegetable oils was improved with pre-heating. S. Naga Sarada et al. [27] conducted experiments on LHR engine, which contains air gap insulated piston with superni crown and air gap insulated liner with superni insert with carbureted methanol and crude jatropha oil and reported that maximum induction of methanol is 35% on mass basis with best possible efficiency at all loads in CE while it is 60% in LHR engine. Performance was improved by 60% alcohol induction in LHR engine.

However, aldehyde levels increased with increase of methanol induction in both versions of the engines when compared with pure jatropha oil operation on CE. Krishna Murthy [28] conducted experiments on LHR engine which contained air gap insulated piston with superni crown, air gap insulated liner with superni insert and ceramic coated cylinder head with jatropha oil based bio-diesel and reported that performance was deteriorated with bio-diesel in CE and improved with LHR engine.

The present paper attempts to evaluate the performance of LHR engine, which contains air gap piston with superni crown and air gap insulated liner with superni insert with different operating conditions of crude jatropha oil with varying engine parameters including change of injection pressure and timing and compared with CE at recommended injection timing and injection pressure. An attempt is made to develop a zero dimensional, multi-zone combustion model and correlate theoretical analysis with experimental results.



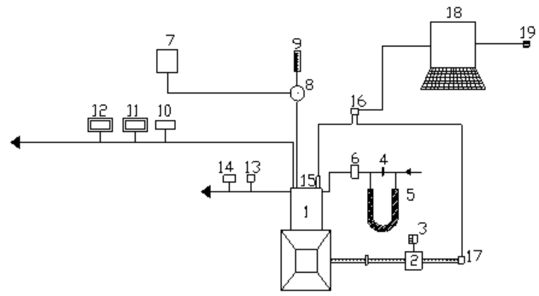
1. Crown, 2.Gasket, 3.Air gap, 4.Piston body 5. Superni inert, 6. Air gap, 7 Liner

Fig. 1 Assembly details of air gap insulated piston and air gap insulated liner

2 EXPERIMANTAL PROCEDURE

Fig. 1 gives the details of air gap insulated piston with superni crown and air gap insulated liner with superni insert employed in the experimentation. LHR diesel engine contains a two-part piston; the top crown made of low thermal conductivity material, superni-90 screwed to aluminum body of the piston, providing a 3-mm air gap in between the crown and the body of the piston. The optimum thickness of air gap in the air gap piston is found to be 3-mm [6], for better performance of the engine with diesel as fuel.

A superni-90 insert is screwed to the top portion of the liner in such a manner that an air gap of 3-mm is 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. The properties of vegetable oil are taken from reference No [28]. Experimental setup used for the investigations of LHR diesel engine with crude jatropha oil (CJO) operation is shown in Fig. 2.



1.Engine, 2.Electical Dynamo meter, 3.Load Box, 4.Orifice 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 meter, 12.Netel Chromatograph 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.Pentium Personal Computer and 19. Printer

Fig. 2 Experimental Set-up

CE has an aluminum alloy piston with a bore of 80 mm and a stroke of 110mm. The rated output of the engine is 3.68 kW at a rate speed of 1500 rpm. The compression ratio is 16:1 and manufacturer's recommended injection timing and injection pressures are 27^{o}bTDC and 190 bar respectively. The fuel injector has 3-holes of size 0.25-mm. The combustion chamber consists of a direct injection type with no special arrangement for swirling motion of air. The engine is connected to electric dynamometer for measuring brake power of the engine. Burette method is used for finding fuel consumption of the engine. Air-consumption of the engine is measured by air-box method.
The naturally aspirated engine is provided with water-cooling system in which inlet temperature of water is maintained at 60^{o}C by adjusting water flow rate. Engine oil is provided with a pressure feed system and no temperature control is incorporated, for measuring the lube oil temperature. Copper shims of suitable size are provided in between the pump body and the engine frame, to vary the injection timing and its effect on the performance of the engine is studied, along with the change of injection pressures from 190 bar to 270 bar (in steps of 40 bar) using nozzle testing device. The maximum injection pressure is restricted to 270 bar due to practical difficulties involved.
Exhaust gas temperature (EGT) is measured with thermocouples made of iron and iron-constantan. Pollution levels of smoke and NO_x are recorded by AVL smoke meter and Netel Chromatograph NOx analyzer respectively at the peak load operation of the engine. Piezo-electric transducer, fitted on the cylinder head to measure pressure in the combustion chamber is connected to a console, which in turn is connected to Pentium personal computer. TDC encoder provided at the extended shaft of the dynamometer is connected to the console to measure the crank angle of the engine. A special P-θ software package evaluates combustion characteristics such as PP, TOPP, MRPR and TOMRPR from the signals of pressure and crank angle at the peak load operation of the engine. Pressure-crank angle diagram is obtained on the screen of the personal computer.

3 COMBUSTION MODELLING

A zero-dimensional, multi-zone model is attempted to predict the performance of LHR diesel engine, with air gap insulated piston and liner. However, there are certain assumptions such as i) There is no interaction between two elements, ii) Pressure is uniform over the entire combustion chamber, iii) Fuel jet breaks into droplets right at the exit plane of the nozzle and iv) Injection pressure and injection rate are constant over a cycle. The concept of dividing spray is similar to that of Hiroyasu [29].
The model is a closed cycle simulation and has been divided into five sub-models, (1) fuel injection (2) spray penetration and air entrainment (3) spray evaporation (4) combustion model (5) heat transfer model. The spray emerging from the injection nozzle in the form of a cone is divided into a number of small elements which are specified by two variables i and j in axial and radial directions. The air package contains fresh air and residual gas for diesel operation is identified by a single index i=1. The number of droplets and their Sauter mean diameter in each element are computed in the following manner [29] in fuel injection sub model.

$$\left(D_{SM}\right)_{i,j} = (23.9)(P_{inj} - P)^{-0.135} \rho_a^{0.12} \left(Q_{inj}^{1}\right) \tag{1}$$

The location of each element in space is determined from the following relation [29]

$$N' = \int_{t_i}^{t_{i+1}} \frac{Q_{inj}^{1}\ dt}{J\frac{\pi}{6}\left(D_{SM}\right)_{i,j}^{3}} \tag{2}$$

The brake up time for the element can be found from the following relation.

$$t_{bu} = \frac{29\,\rho_l\, d}{\left(\rho_g \Delta P\right)^{0.5}} \tag{3}$$

where $t_{\theta I} < t_{bu}$, the spray tip penetration in axial direction ($X_{i,j}$) is given by [29].

$$X_{i,j} = 0.39\left(\frac{2\Delta P}{\rho_l}\right)^{0.5} t_{\theta i}\ \text{for}\ (j=1) \tag{4}$$

where $t_{\theta i} > t_{bu}$, the spraying penetration in axial direction is given by

$$X_{i,j} = 2.95\left(\frac{\Delta P}{\rho_g}\right)^{1/4}\left(d_n t_{\theta i}\right)^{1/2} \text{ for (j=1)} \quad (5)$$

The radial location of the other element is given by the following relation [29],

$$X_{i,j} = i^{\mathrm{Exp}\left(-8.557x10^{-3}(j-1)^2\right)} \quad \text{for (j > 1)} \quad (6)$$

Taking equation (4), and differentiate w.r.t. time,

$$\frac{dX_{i,j}}{dt} = \frac{X_{i,j}}{t_{\theta i}} \quad (7)$$

Taking equation (5), and differentiate with respect to the time, for $t_{\theta i} > t_{bu}$

$$\frac{dX_{i,j}}{dt} = \frac{X_{i,j}}{2t_{\theta i}} \quad (8)$$

The air entrainment into each package is obtained by momentum balance. The conservation of momentum for each package can be expressed as

$$M_{f_{i,j}} V_{inj}^1 = \left[M_{f_{i,j}} + M_{ae_{i,j}}\right]\frac{dX_{i,j}}{dt} \quad (9)$$

Simplifying the above equation,

$$M_{f_{i,j}} V_{inj}^1 = M_{f_{i,j}}\frac{dX_{i,j}}{dt} + M_{ae_{i,j}}\frac{dX_{i,j}}{dt}$$

Hence entrainment of air in an element is given by

$$M_{ae_{i,j}} = M_{f_{i,j}}\left[\frac{V_{inj}^1}{\frac{dX_{i,j}}{dt}} - 1\right] \quad (10)$$

The rate of air entrainment $\dot{M}_{ae_{i,j}}$ is obtained by differentiating the above equation with respect to time, assuming constant injection velocity and injection rate, as the rate of fuel injected/ degree of crank rotation is a function of injection cam velocity, diameter of injector plunger and flow area of tip orifices,

Therefore,

$$\dot{V}_{inj}^1 = 0; \qquad \dot{M}_{f_{i,j}} = 0$$

$$\dot{M}_{ae_{i,j}} = -\frac{\dot{V}_{i,j}^1}{V_{i,j}^1}\left[M_{ae_{i,j}} + M_{f_{i,j}}\right] \quad (11)$$

The entrainment of individual species into each package can be found from the mass fraction of that species in the air package (i=1)

$$\frac{dM_{NS_{i,j}}}{dt} = Y_{NS_{i,j}}\left[\frac{dM_{ae_{i,j}}}{dt}\right] \quad (12)$$

The evaporation rates of droplet in an element are computed using Spalding's droplet evaporation relation [30].

$$\frac{dM_f}{dt} = 4\pi\, r_s \rho_s D_s^l\left[\log(1+\beta')\right] \quad (13)$$

The rate of decrease droplet radius is given by the following expression [31]

$$\frac{dr_s}{dt} = \frac{-\rho_s}{\rho_l}\frac{D_s^l}{r_s}\left[\log(1+\beta')\right] \quad (14)$$

Total evaporation rate in the element is given by

$$\frac{dM_{v_{i,j}}}{dt} = N'_{i,j}\frac{dM_{f_{i,j}}}{dt} \quad (15)$$

The rate of heat removal from gas phase due to latent heat of evaporation is given by

$$\frac{dQ_{v_{i,j}}}{dt} = L\frac{dM_{v_{i,j}}}{dt} \quad (16)$$

Where L is obtained from the following expression [31].

$$L = 9764.2\ [1799.4 - 1.8T_l]^{0.5} \quad (17)$$

The mass of air and mass of gaseous fuel determines the equivalence ratio (Phi) in the gaseous phase in the element as

$$\mathrm{Phi} = \frac{M_{fg}/M_a}{(M_{fg}/M_a)st}$$

In an element after "s" short period of time from fuel injection, ignition occurs in the gaseous mixture. The ignition delay of gaseous mixture at varying temperature, pressure and equivalence ratio is obtained from [30]

$t = 4 \times 10^{-3} (P)^{-2.5} (Phi)^{-1.04} Exp(4000)/T$ [30]

Once the delay period is over, depending upon the availability of oxidant and vaporized fuel, the chemical reaction is assumed to take place at stoichiometric proportion. Heat release rate due to combustion of fuel is shown by

$$\frac{dQ_c}{dt} = \frac{M_f\, Cv}{t} \qquad \text{for } Phi < 1$$

$$\frac{dQ_c}{dt} = \frac{M_{02}.Cv}{t\,(O/F)} \qquad \text{for } Phi > 1$$

Heat transfer to cylinder walls is obtained using Annand's relation [32]. This relation considers the net heat transfer as the summation of both convective and radiative heat transfer.

$$\frac{dQ_w}{dt} = \frac{aA.K}{D_l}\left[(Re)^b\right]\left[T - T_w\right] + Ac\left(T^4 - T_w^4\right) \tag{19}$$

A = Total surface area exposed to heat transfer = $A_p + A_{li} + A_h$

The wall temperature is obtained based on FEM analysis carried out for LHR engine.

Global Equations

The net heat release in the element is given by

$$\frac{dQ_n}{dt} = \frac{dQ_{C_{i,j}}}{dt} - \frac{dQ_{v_{i,j}}}{dt} - \frac{dQ_{w_{i,j}}}{dt} \tag{20}$$

The temperature of an element at any instant is obtained by applying energy equation to each element.

$$\dot{T} = \frac{\dot{Q}_n + v\dot{p} + \dot{M}_{ae}(h_a - h) + \dot{M}_v(h_f - h) - MT\dot{R}}{M(C_v + R)} \tag{21}$$

For air package

$$\dot{M} = -\dot{M}_{ae};\quad \dot{M}_v = 0;\quad h = h_a;\quad \dot{R} = 0;\quad \dot{Q}_n = \dot{Q}_w \tag{22}$$

By substituting the conditions for air package given in equation (21)

$$\dot{T} = \frac{\dot{Q}_w + V\dot{P}}{M(C_v + R)} \tag{23}$$

Pressure is obtained from the conservation of momentum

$$\dot{p} = \left[\frac{1}{V - V_l}\right]\left[\left\{\sum \dot{M}_{i,j} R_{i,j} T_{i,j} + \sum M_{i,j} \dot{R}_{i,j} T_{i,j}\right\} + \frac{\sum MR\left\{\dot{Q}_n + V\dot{p} + \dot{M}_{ae}(h_a - h_{i,j}) + \dot{M}_V(h_f - h_{i,j}) - MT\dot{R}_{i,j}\right\}}{M.C_{p_{i,j}}} - P(\dot{V} - \dot{V}_l)\right] \tag{24}$$

Rate of change of gas constant is computed by the following relation

$$R = \overline{R}\sum \frac{Y_{NS}}{W_{NS}} \tag{25}$$

By differentiating equation (25),

$$\frac{dR}{dt} = \overline{R}\sum \frac{1}{W_{ns}}\left[\frac{\dot{M}_{ns}}{M} - \frac{Y_{ns}\dot{M}}{M}\right] \tag{26}$$

Knowing the pressure and temperature of a particular element, its volume can be found by applying the equation of state to that individual element.

$$V_{i,j} = \frac{M_{i,j}\, R_{i,j}\, T_{i,j}}{p} \tag{27}$$

After finding the volume of all elements including the air element, the volume constraint should be checked. The volume constraint (where the liquid volume is neglected) is given by

$$\sum V_{i,j} = V_{cyel} \tag{28}$$

Equations (12), (13), (14), (23) and (24) formed the set of governing equations for pure diesel or non-edible version oil operations formed the set of governing equations which are solved using Runge-Kutta fourth order scheme. Computerization started at IVC and closed at EVO. The process is pure compression until the point of injection, and the main computations described in the model are computed after this instant (point of injection).

4 RESULTS AND DISCUSSION

A. Performance Parameters

The variation of brake thermal efficiency (BTE) with brake mean effective pressure (BMEP) in the conventional engine (CE) with normal temperature (NT) of CJO, at various injection timings at an injection pressure of 190 bar, is shown in Fig. 3. The variation of BTE with BMEP with pure diesel operation on CE is also shown for comparison purpose.

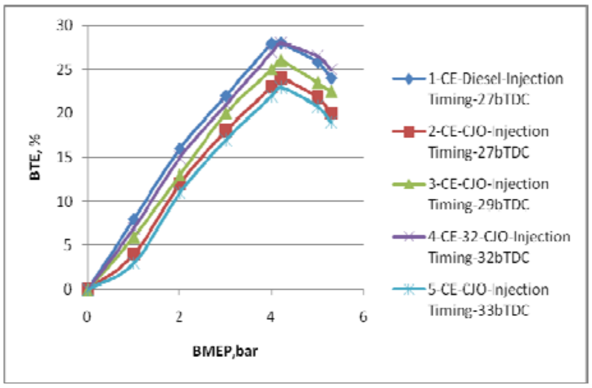


Fig. 3 Variation of brake thermal efficiency (BTE) with brake mean effective pressure (BMEP) in conventional engine (CE) at different injection timings

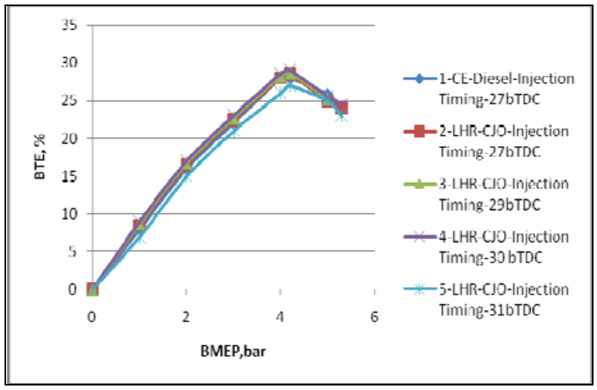


Fig. 4 Variation of brake thermal efficiency (BTE) with brake mean effective pressure (BMEP) in low heat rejection (LHR) engine at different injection timings

CE with vegetable oil showed deterioration in the performance for the entire load range when compared with pure diesel operation on CE at recommended injection timing. Although carbon accumulations on the nozzle tip might play a partial role for the general trends observed, the difference of viscosity between the diesel and vegetable oil provided a possible explanation for the deterioration in the performance of the engine operating on vegetable oil. The result of lower jet exit Reynolds numbers with vegetable oil adversely affected the atomization. The amount of air entrained by the fuel spray is reduced, since the fuel spray plume angle is reduced, resulting in slower fuel- air mixing. In addition, less air entrainment by the fuel spay suggested that the fuel spray penetration might increase and resulted in more fuel reaching the combustion chamber walls.

Furthermore droplet mean diameters (expressed as Sauter mean) are larger for vegetable oil leading to reduce the rate of heat release as compared with diesel fuel. This also, contributed the higher ignition (chemical) delay of vegetable oil due to lower cetane number. According to the qualitative image of combustion under the crude vegetable oil operation with CE, the lower BTE is attributed to the relatively retarded and lower heat release rates. BTE increased with advancement of injection timing in CE with vegetable oil at all loads, when compared with CE at the recommended injection timing and pressure. This is due to initiation of combustion at an earlier period and efficient combustion with increase of air entrainment in fuel spray giving higher BTE. BTE increased at all loads when the injection timing is advanced to 32^{o}b TDC in the CE at normal temperature of vegetable oil. The increase of BTE at optimum injection timing over the recommended injection timing with vegetable oil with CE could be attributed to its longer ignition delay and combustion duration.

BTE increased at all loads when the injection timing is advanced to 32^{o}bTDC in CE, at the preheated temperature (PT) of CJO. The performance is improved further in CE with the preheated vegetable oil for the entire load range when compared with normal vegetable oil. Preheating of vegetable oil reduced the viscosity, which improved the spray characteristics of the oil. Variation of BTE with BMEP in the LHR engine with CJO, at various injection timings at an injection pressure of 190 bar, is shown in Fig. 4.

LHR version of the engine showed improved performance for the entire load range compared with CE with pure diesel operation. High cylinder temperatures helped in better evaporation and faster combustion of the fuel injected into the combustion chamber. Reduction of ignition delay of vegetable oil in the hot environment of the LHR engine improved heat release rates and efficient energy utilization. Preheating of vegetable oil improves performance further in LHR version of the engine. The optimum injection timing is found to be 30^{o}bTDC with LHR engine with normal CJO operation. Since the hot combustion chamber of LHR engine reduced ignition delay and combustion duration and hence the optimum injection timing is obtained earlier with LHR engine when compared with CE operating on vegetable oil. Injection pressure is varied from 190 bars to 270 bars to improve the spray characteristics and atomization of vegetable oils and injection timing is advanced from 27 to 34^{o}bTDC for CE and LHR engine.

The improvement in BTE at higher injection pressure is due to improved fuel spray characteristics. However, unlike the CE , the optimum injection timing with the LHR engine does not vary even at higher injection pressure. Hence it is concluded that the optimum injection timing is 32^{o}bTDC at 190 bar, 31^{o}bTDC at 230 bar and 30^{o}bTDC at 270 bar for CE. The optimum injection timing for LHR engine is 30^{o}bTDC irrespective of injection pressure. Improvement in the peak BTE is observed concurrent with increase of injection pressure and with advancement of injection timing with vegetable oil in both versions of the engine.

Peak BTE is higher in LHR engine when compared with CE with different operating conditions of vegetable oils. Preheating of vegetable oil improved the performance in both versions of the engine compared with vegetable oil at normal temperature. Preheating reduced the viscosity of vegetable oils, which in turn reduced the impingement of the fuel spray on combustion chamber walls, causing efficient combustion thus improving BTE. Fig. 5 shows variation of the exhaust gas temperature (EGT) with BMEP in CE and LHR engine with CJO at NT at recommended and optimized injection timings at an injection pressure of 190 bar.

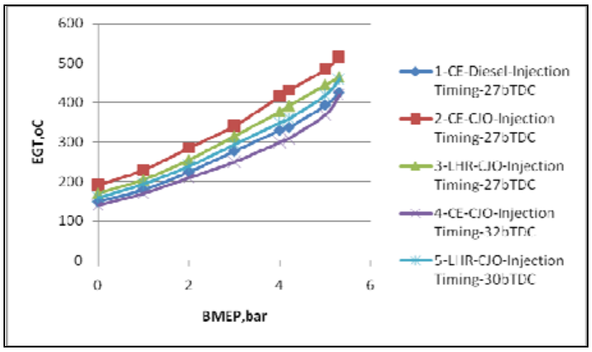


Fig. 5 Variation of exhaust gas temperature (EGT) with brake mean effective pressure (BMEP) in conventional engine (CE) and low heat rejection (LHR) engine at recommended injection timing and optimized injection timings

CE operating on vegetable oil at recommended injection timing recorded higher EGT at all loads when compared with CE operating on pure diesel. Lower heat release rates and retarded heat release associated with specifically high energy consumption lead to EGT increase in CEs. Ignition delay in the CE with different operating conditions of vegetable oil increased the duration of the burning phase. LHR engine recorded lower value of EGT when compared with CE operating on vegetable oil. This is due to reduction of ignition delay in the hot environment as a result of insulation in the LHR engine, which caused expansion of gases in the cylinder producing higher output power and lower heat rejection.

This showed that the performance is improved with LHR engine over CE operating on vegetable oil. The magnitude of EGT at peak load decreased concurrent with advancement of injection timing and with increase of injection pressure in both versions of the engine with vegetable oil. Preheating of vegetable oil further reduced the magnitude of EGT, compared with normal vegetable oil in both versions of the engine. Fig. 6 shows the variation of the volumetric efficiency (VE) with BMEP in CE and LHR engine with CJO operation at the recommended and optimized injection timings at an injection pressure of 190 bar.

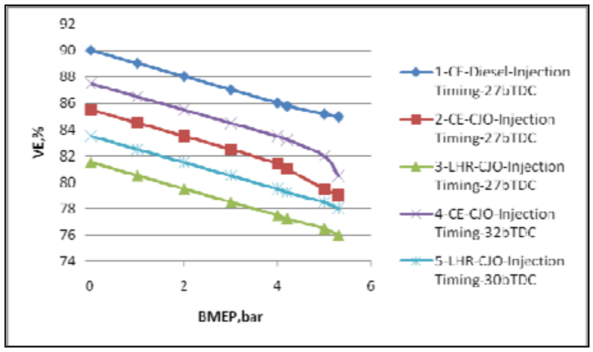


Fig. 6 Variation of volumetric efficiency (VE) with brake mean effective pressure (BMEP) in conventional engine (CE) and low heat rejection (LHR) engine at recommended injection timing and optimized injection timings

VE decreased along with an increase of BMEP in both versions of the engine. This is due to increase of gas temperature with increasing load. At the recommended injection timing, VE in both versions of the engine with CJO operation decreased at all loads when compared with CE, operating on pure diesel. This is due to increase of temperature of incoming charge in the hot environment created with the simultaneous provision of insulation, causing reduction in the density and hence the quantity of air with LHR engine.VE increased marginally in CE and LHR engine at optimized injection timings when compared with recommended injection timings operating on vegetable oil. This is due to decrease of un-burnt fuel fraction in the cylinder leading to increase in VE in CE and reduction of gas temperatures with LHR engine.

VE increased marginally with the advancement of injection timing and with the increase of injection pressure in both versions of the engine. This is due to better fuel spray characteristics and evaporation at higher injection pressures leading to marginal increase of VE. This is also due to the reduction of residual fraction of the fuel, with the increase of injection pressure. Preheating of vegetable oil marginally improved VE in both versions of the engine, because it reduced un-burnt fuel causing efficient combustion rather than normally tempered oil.

B. Pollution Levels

Barsic et al. [33] reported that fuel physical properties such as density and viscosity could have a greater influence on smoke emission than fuel chemical properties. Fig. 7 shows variation of smoke levels with

BMEP in CE and LHR engine operating on vegetable oil at the recommended and optimized injection timings at an injection pressure of 190 bars.

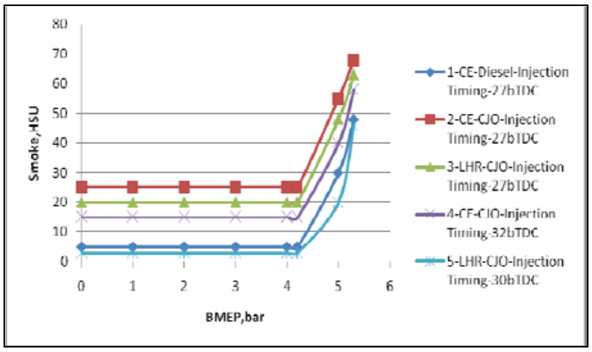


Fig. 7 Variation of smoke intensity in Hartridge Smoke Unit (HSU) with brake mean effective pressure (BMEP) in conventional engine (CE) and low heat rejection (LHR) engine at recommend injection timing and optimized injection timings

Drastic increase of smoke levels is observed at the peak load operation in CE at different operating conditions of vegetable oil, compared with pure diesel operation on CE. This is due to the higher magnitude of the ratio of C/H of CJO (0.83) when compared with pure diesel (0.45). The increase of smoke levels is also due to decrease of air-fuel ratios and VE with vegetable oil compared with pure diesel operation. Smoke levels are related to the density of the fuel. Since vegetable oil has a higher density compared to diesel fuels, smoke levels are higher with vegetable oil. However, LHR engine marginally reduced smoke levels due to efficient combustion and less amount of fuel accumulation on the hot combustion chamber walls of the LHR engine at different operating conditions of vegetable oil compared with the CE. Density influences the fuel injection system.

Decreasing the fuel density tends to increase spray dispersion and spray penetration. Preheating of vegetable oils reduced smoke levels in both versions of the engine, when compared with normal temperature of vegetable oil. This is due to i) the reduction of density of vegetable oils, as density is directly proportional to smoke levels, ii) the reduction of the diffusion combustion proportion in CE with preheated vegetable oil, iii) the reduction of viscosity of vegetable oil, by which the fuel spray does not impinge on combustion chamber walls at lower temperatures and is smoothly directed towards combustion chamber.

Smoke levels decreased at optimized injection timings and with increase of injection pressure, in both versions of the engine, at different operating conditions of vegetable oil. This is due to improvement in the fuel spray characteristics at higher injection pressures and increase of air entrainment, at advanced injection timings, causing lower smoke levels. Fig. 8 shows variation of NOx levels with BMEP in CE and LHR engine with CJO at recommended and optimized injection timings at an injection pressure of 190 bar.

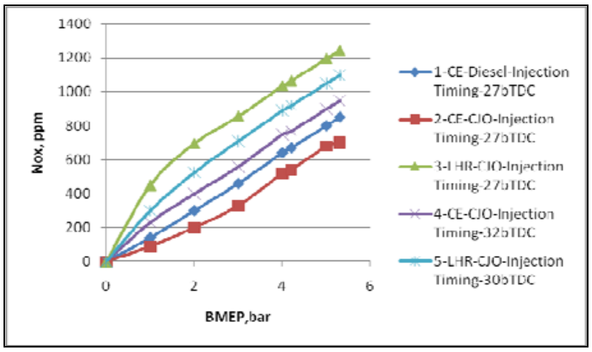


Fig. 8 Variation of NOx levels with brake mean effective pressure (BMEP) in conventional engine (CE) and low heat rejection (LHR) engine at recommended injection timing and optimized injection timings

NOx levels are lower in CE while they are higher in LHR engine at different operating conditions of vegetable oil at the peak load when compared with diesel operation. This is due to lower heat release rate because of high duration of combustion causing lower gas temperatures operating on vegetable oil on CE, which reduced NOx levels. Increase of combustion temperatures along with faster combustion and improved heat release rates in LHR engine causes higher NOx levels. As expected preheating of vegetable oil further increased NOx levels in the CE whereas in the LHR engine it had an adverse effect; decreasing NOx level in the LHR engine the same amount it increased NOx level in the CE.

This is due to improved heat release rates and increased mass burning rate of the fuel by which combustion temperatures increase leading to increase in NOx emissions in the CE and decrease of combustion temperatures in the LHR engine with improvement in air-fuel ratios leading to decreasing NOx levels in LHR engine. NOx levels increased with advancement of injection timing and with increase of injection pressure in CE with different operating conditions of vegetable oil. Residence time and combustion temperatures had increased, when injection timing is advanced operating on vegetable oil, in turn causing higher NOx levels. With the increase of injection pressure, fuel droplets penetrate and find oxygen counterpart easily. Turbulence of the fuel spray increased the spread of droplets thus leading to increase in NOx levels. However, decrease of NOx levels is observed in LHR engine, due to decrease of combustion temperatures as seen Fig. 11, when injection timing is advanced and along with increase of injection pressure.

C. Combustion Characteristics

Table 1 presents variation of PP, MRPR, TOPP and TOMRPR with Injection Y=Timing and Injection Pressure at the peak load operation of CE and LHR Engine operating on vegetable oil. Peak pressures are lower in CE while they were higher in LHR engine at the recommended injection timing and pressure, when compared with pure diesel operation on CE. This is due to increase of ignition delay, as vegetable oils require longer duration of combustion. Mean while the piston started making downward motion thus increasing volume when combustion takes place in CE. LHR engine increased the mass-burning rate of the fuel in the hot environment leading to producing higher peak pressures.

The advantage of using LHR engine for vegetable oil is obvious as it could burn low cetane and high viscous fuels. Peak pressures increased with the increase of injection pressure and with the advancing of the injection timing in both versions of the engine, operating on vegetable oil. Higher injection pressure produces smaller fuel particles with low surface to volume ratio; giving rise to higher PP. Rehman et al. [11] observed the similar trend with normal karanja oil in CE. With the advancement of injection timing to the optimum value in the CE, more amount of the fuel accumulated in the combustion chamber due to increase of ignition delay as the fuel spray found the air at lower pressure and temperature in the combustion chamber.

When the fuel- air mixture burns, it produces more combustion temperatures and pressures due to increase of fuel mass. With LHR engine, peak pressures increases due to effective utilization of the charge with the advancement of injection timing to the optimum value. The magnitude of TOPP decreased with the advancement of injection timing and with increase of injection pressure in both versions of the engine, at different operating conditions of vegetable oils. TOPP is more with different operating conditions of vegetable oils in CE, when compared with pure diesel operation on CE.

This is due to higher ignition delay with vegetable oil when compared with pure diesel fuel. This once again established the fact by observing lower peak pressures and higher TOPP, that CE operating on vegetable oil showed deterioration in performance when compared with pure diesel operation on CE. Preheating of vegetable oils showed lower TOPP, compared with vegetable oil at normal temperature. This once again was confirmed by observing the lower TOPP and higher PP, the performance of both versions of the engine is improved with preheated vegetable oil compared with normal vegetable oil. This trend of increase of MRPR and decrease of TOMRPR indicated better and faster energy substitution and utilization by vegetable oils, which could replace 100% diesel fuel. However, these combustion characters are within the limits hence vegetable oils could be effectively substituted for diesel fuel.

D. Combustion Modelling

The input data on the surface temperatures of the piston needed in the model are chosen from the temperature predicted from the finite element analysis from the reference [6], [7]. Fig. 9 shows comparison of typical pressure-crank angle data predicted from the multi-zone combustion model and the experimental values obtained from the special P-θ software package and TDC encoder for the case of CE and LHR engine at the recommended injection timing and at the recommended injection pressure. It can be seen from this figure that the LHR engine exhibited higher peak pressures in comparison with CE. The above trend indicated that there is larger amount of energy contained in combustion gases in case of LHR engine which is also indirectly confirmed by the higher surface temperatures estimated in the finite element analysis [6], [7] which formed the input data for the combustion model. It could be seen clearly that data predicted from the combustion model gave higher values of peak pressures in CE as well as LHR engine in comparison with the experimental results. This is because of the idealized assumptions assumed in the model, which may not exist in reality. A deviation of 8% is observed between the experimental results and theoretical values, while Rama Mohan [6], [7] reported a deviation of 6.5% with pure diesel operation.

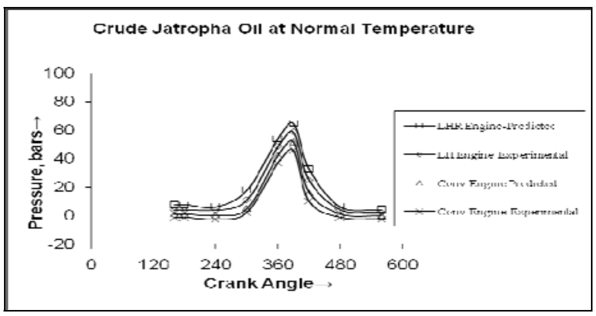


Fig. 9 Comparison of typical pressure-crank angle data predicted from the multi-zone combustion model and the experimental values obtained from the special P-θ software package and TDC encoder for the case of conventional and LHR engines at the recommended injection timing and injection pressure

Table 1 Variation of PP, MRPR, TOPP and TOMRPR with injection Y=Timing and injection pressure at the peak load operation of CE and LHR Engine operating on vegetable oil

Injection timing (°bTDC)/ Test fuel	Engine version	PP(bar)				MRPR (Bar/deg)				TOPP (Deg)				TOMRPR (Deg)			
		Injection pressure (Bar)				Injection pressure (Bar)				Injection pressure (Bar)				Injection pressure (Bar)			
		190		270		190		270		190		270		190		270	
		NT	PT	NT	PT	NT	PT	NT	PT	NT	PT	NT	PT	NT	PT	NT	PT
27/Diesel	CE	50.4	--	53.5	---	3.1	---	3.4	--	9	-	8	--	0	0	0	0
	LHR	48.1	--	53.0	--	2.9	--	3.1	--	10	--	9	--	0	0	0	0
27/ CJO	CE	46.9	47.7	49.9	50.3	2.4	2.5	2.9	3.0	11	10	11	10	1	1	1	1
	LHR	59.5	60.9	63.6	64.5	3.3	3.4	3.5	3.6	10	9	9	8	1	1	1	1
30/CJO	LHR	61.5	61.9	66.3	67.1	3.4	3.5	3.6	3.7	9	8	8	7	0	0	0	0
32/CJO	CE	51.5	52.8			3.3		3.4		8		8		0		0	

CJO- Crude jatropha oil, CE-Conventional engine, LHR-Low heat rejection, NT-Normal temperature, PT-Preheated temperature.

Fig. 10 depicts the typical diagram of variation of air zone temperature and gas zone temperature with respect to crank angle for both versions of the engine at the recommended and optimum injection timings at the recommended injection pressure operating on vegetable oil. Gas temperatures decreased in LHR engine while they increased in CE when injection timing advanced towards the optimum values.

Decrease of gas temperatures with LHR engine indicated saving of waste heat of the exhaust while converting the same into useful work. This is confirmed from the increased BTE observed with advanced injection timing in the LHR engine, as can be seen in Fig. 4. Increase of NOx emissions with the advanced injection timing in CE and NOx emission decrease of the same amount in the LHR engine proved validation of computer predicted gas temperatures in CE and LHR engine.

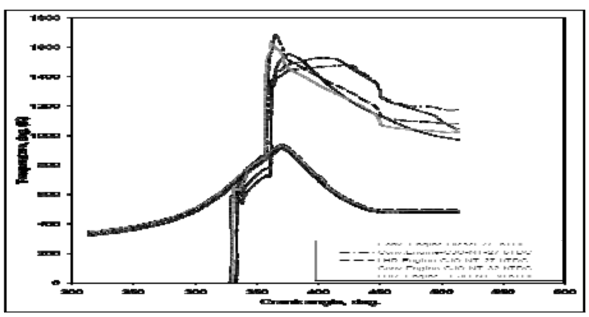


Fig. 10 Variation of air-zone temperature and gas-zone temperature with respect to crank angle in both versions of the engine with normal temperature (NT) of crude jatropha oil (CJO) operation at recommended and optimized injection timings

Fig. 11 shows variation of air zone temperature and gas zone temperature with respect to the crank angle for CE and LHR engine at the recommended injection timing and at an injection pressure of 270 bar operating on vegetable oil. It is noticed that CE at higher injection pressure showed higher magnitudes of gas temperatures while LHR engine produced lower gas temperatures when compared with the same versions of the engine at recommended injection pressure. The computer predictions of gas temperatures are validated with the trends of NOx emissions in CE and LHR engine.

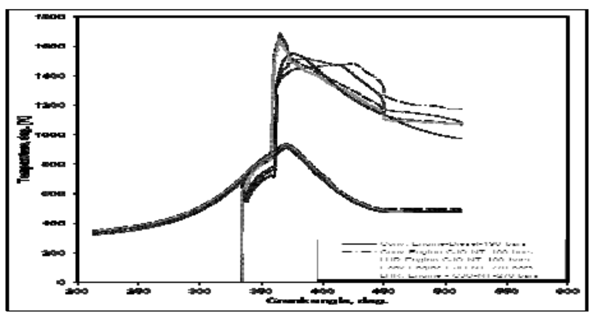


Fig. 11 Variation of air zone temperature and gas zone temperature with respect to the crank angle for CE and LHR engine at the recommended injection timing and at an injection pressure of 270 bars with normal temperature (NT) of crude jatropha oil (CJO) operation

5 CONCLUSION

Brake thermal efficiency increased by 4%, exhaust gas temperature increased by 35°C, volumetric efficiency decreased by 8%, smoke levels decreased by 4% and NOx levels increased by 37% with LHR engine at its optimum injection timing of 30^{o}bTDC in comparison with CE with pure diesel operation at its recommended injection timing of 27^{o}bTDC.

6 NOMENCLATURE

a	Constant used in Annand's equation
A	Total heat transfer surface area in m^2
b	Constant used in Annand's equation
BDC	Bottom Dead Centre
BMEP	Brake mean effective pressure in bar
BSFC	Brake specific fuel consumption in kg/h-kW
bTDC	Before top dead centre
BTE	Brake thermal efficiency in percentage
c	Constant used in Annand's equation
CE	Conventional engine
CV	Calorific value of the fuel in kJ/kg
D_l	Cylinder bore in meter
d_n	Diameter of nozzle in meter
D_{sm}	Sauter mean diameter of fuel particle in micrometers
EGT	Exhaust gas temperature in C^o
HSU	Hartridge smoke units
IVC	Inlet valve closing
J	Total number of radial divisions
K	Thermal conductivity in W/m-K
L	Latent heat of evaporation of fuel in J/kg,
LHR	Low heat rejection
M	Mass in kg
MRPR	Maximum rate of pressure rise in bar/degree
$N^{/}$	Number of droplets
NOx	Oxides of nitrogen
O/F	Oxygen to fuel ratio
O_2	Oxygen
P	Pressure in mega Pascal
P (θ)	Instantaneous gas pressure in bar
Phi	Equivalence fuel-air ratio
PP	Peak pressure in bar
Q	Heat transfer in W
Q^1_{inj}	Amount of fuel delivered per cycle per second in m^3 / sec
R	Gas constant in kJ/kg-K
$\overline{R}$	Universal gas constant kJ/ K
R	Radial co-ordinate of cylindrical co-ordinate system
R_e	Reynold's number
r_s	Radius of droplet in micrometers
SMD	Sauter mean diameter of droplet in microns
T	Surrounding temperature in Kelvin
t_{bu}	Brake up time for element in sec
TDC	Top Dead Centre
T_l	Temperature of liquid fuel in Kelvin
TOMRPR	Time of occurrence of maximum rate of pressure rise in degrees
TOPP	Time of occurrence of peak pressure in degrees
T_w	Wall temperature in Kelvin
t	Ignition delay of gaseous mixture
V	Volume or domain
V '	Velocity in m/s
V^1_{inj}	Velocity of injection
VE	Volumetric efficiency in percentage
V_{dis}	Displacement volume in m^3
W	Molecular weight
X_{ij}	Spray tip penetration in axial direction in meters
Y	Mass fraction of chemical species
27^o bTDC	Manufacturer's recommended injection timing
190 bars	Manufacturer's recommended injection pressure

Subscripts

a	Air
ae	Air entrainment
b	Burnt
bu	Brake up
c	Combustion
dis	Displacement
e	Entrainment
f	Fuel
g	Gas
fg	Gaseous fuel
h	Cylinder head
i	Index for an element in the axial direction
inj	Injection
j	Index for an element in the radial direction
l	Liquid
L	Latent heat of vaporization
li	Liner
NS	Index for chemical species vary from 1 to 5 referring to fuel (Referring to the fuel, O_2, N_2, CO_2 and H_2O respectively)
n	Net
o	Reference state
p	Piston
St	Stoichiometric ratio
s	Surface of droplet
u	Un-burnt
v	Vapor
w	Wall
298	Datum temperature in Kelvin

Greek symbols

θ	Crank angle in degrees
δ	Differential
μ	Dynamic viscosity in N-s/m ^{2}
Δ	Corresponding to change in a quantity
θ_i	Crank angle corresponding to the location of the package
θ_o	Crank angle corresponding to one complete cycle in degrees
β'	Spalding number
ρ_a	Density of air in kg/m^3
ρ_g	Gas density in kg/m^3
ρ_l	Liquid density in kg/m^3
ρ_s	Density of droplet in kg/m^3
τ	Ignition delay in milliseconds

Other symbols

$\cdot$	This is a symbol, which indicates the rate of change of that quantity

7 ACKNOWLEDGMENTS

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