INTERNATIONAL JOURNAL OF
ENERGY AND ENVIRONMENT

Volume 1, Issue 6, 2010 pp.1039-1056

Journal homepage: www.IJEE.IEEFoundation.org

Two-zone modeling of diesel / biodiesel blended fuel operated ceramic coated direct injection diesel engine

B. Rajendra Prasath[1], P. Tamil Porai[2], Mohd. F. Shabir[1]

[1] Department of Mechanical Engineering, Tagore Engineering College, Anna University Chennai, India.
[2] Department of Mechanical Engineering, Anna University Chennai, India.

Abstract

A comprehensive computer code using ”C” language was developed for compression ignition (C.I) engine cycle and modified in to low heat rejection (LHR) engine through wall heat transfer model. Combustion characteristics such as cylinder pressure, heat release, heat transfer and performance characteristics such as work done, specific fuel consumption (SFC) and brake thermal efficiency (BTE) were analysed. On the basis of first law of thermodynamics the properties at each degree crank angle was calculated. Preparation and reaction rate model was used to calculate the instantaneous heat release rate. The effect of coating on engine heat transfer was analysed using a gas-wall heat transfer calculations and total heat transfer was based on ANNAND’s combined heat transfer model. The predicted results are validated through the experiments on the test engine under identical operating conditions on a turbocharged D.I diesel engine. In this analysis 20% of biodiesel (derived from Jatropha seed oil) blended with diesel was used in both conventional and LHR engine. The simulated combustion and performance characteristics are found satisfactory with the experimental results.

Keywords: Biodiesel, Compression ignition, Direct injection, Low heat rejection, Turbocharged.

1. Introduction

Diesel engines are playing a major role in medium and heavy-duty applications due to superior characteristics such as lower fuel consumption, higher engine power output and lower emissions (such as CO and HC) compared with gasoline operated engines. Depletion of petroleum based conventional fuel increases the research interest to find the suitable alternate, especially diesel fuel. Among the various alternate fuels, vegetable oil was identified as a suitable oil to replace the diesel fuel. The usage of vegetable oil is not new, the inventor of diesel engine Rudolf Diesel, reportedly used groundnut oil (peanut) as a fuel for demonstration purpose in 1900. However, higher viscosity and low volatility of straight vegetable oils (SVO) are generally considered to be the major drawbacks for their utilization as fuels in diesel engines. The usage of raw vegetable oils in diesel engines leads to injector coking, severe engine deposits, filter gumming problems, piston ring sticking and thickening of the lubricating oil [1-8]. However, these problems could be reduced through the transesterification of vegetable oil called as biodiesel [9, 10]. Biodiesel produced [11-15] from the non-edible Jatropha seed oil (JSO) decreases the viscosity and improves the cetane number and heating value.

Ceramic coatings in I.C engine are acting like a thermal insulation in order to improve the engine performance and efficiency [16-21] and leaves higher energy in the engine exhaut. The engine with

thermal insulation is called low heat rejection (LHR) engine, suppressing the heat rejection to the engine coolant side. The engines parts such as piston, cylinder head, cylinder liners and valves are coated with partially stabilized zirconia (PSZ) of 0.5 mm thickness. The superior advantages of LHR engines like improved fuel economy, higher energy in exhaust gases and capability of handling higher viscous fuel encouraged the usage of biodiesel.
The rapid development of computer technology narrow down the time consumption for engine test through the simulation techniques. The insight of the combustion process is analysed thoroughly, which enhance the engine power out put and consider as heart of the engine process [22-25]. Thermodynamic models are mainly based on the first law of thermodynamics and used to analyse the combustion and performance characteristics of engines. A computer code was developed using "C" language for conventional engine and converted in to LHR engine application through the wall heat transfer model [26].

2. Transesterification of vegetable oil

The vegetable oil was transesterified using methanol in the presence of sodium hydroxide (NaOH) as a catalyst (Figure 1 and 2). The parameter involved in the processing such as catalyst amount, molar ratio of alcohol to oil, reaction temperature and reaction time are optimized [27, 28].

$$\begin{array}{lclclcl}
CH_2OOCR & & CH_3OH & & CH_3OOCR & & CH_2OH \\
CHOOCR & + & CH_3OH & \rightarrow & CH_3OOCR & + & CHOH \\
CH_2OOCR & & CH_3OH & & CH_3OOCR & & CH_2OH \\
\text{Triglycerides} & & \text{Methanol} & & \text{Methyl Ester} & & \text{Glycerol}
\end{array}$$

Figure1. Transesterification chemistry of vegetable oil

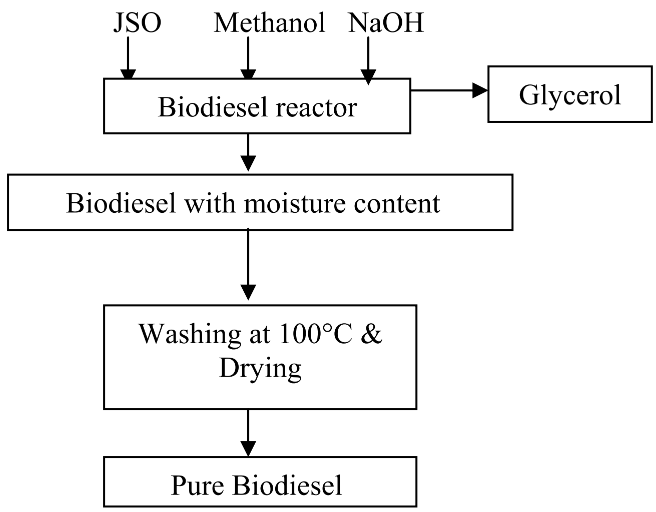

Figure 2. Transesterification process of Jatropha seed oil

Known quantity of vegetable oil was taken in a biodiesel reactor. A water-cooled condenser and a thermometer with cork were connected to the side openings. The required amount of catalyst (NaOH) was weighed and dissolved completely in the required amount of methanol by using a stirrer to form sodium methoxide solution. Meanwhile, the oil was warmed by placing the reactor in water bath maintained at the selected temperature. The sodium methoxide solution was added into the oil and stirred vigorously by means of a mechanical stirrer. The required temperature was maintained throughout the reaction time and the reacted mixture was kept in the separating drum. The mixture was allowed to separate and settled down by gravity settling into a clear, golden liquid biodiesel on the top with the light brown glycerol at the bottom. The glycerol was drained off from the separating drum leaving the biodiesel at the top. This pure biodiesel was measured on weight basis and the important fuel and chemical properties were determined (Table 1).

In this experimental study diesel and biodiesel was used as a fuel for conventional engine (Turbocharged) and LHR engine. Biodiesel and diesel with 20% and 80% by volume was mixed thoroughly and thus a stable mixture (hereafter referred as biodiesel) was prepared to use in the test engine.

Table1. Properties of diesel fuel, B100 and B20

Properties	Diesel Fuel (DF)	Biodiesel (B100)	20%DF / 80%B100 (B20)
Density @ 15°C(kg/m^3)	830	880	840
Viscosity @ 40°C(cSt)	2.8	4.6	3.15
Flash point (°C)	55	170	80
Cetane number	45	50	46
Lower Heating Value (MJ/kg)	42	36	40.5

3. Test engine and experimental procedure

The experiment was conducted on a four-cylinder four-stroke turbocharged water-cooled D.I diesel engine. The engine specifications and valve timing diagram is shown in Table 2 and Figure 3.

Table 2. Specification of the test engine

Bore, mm	111
Stoke length, mm	127
Connecting rod length, mm	251
Compression ratio	16: 1
Displacement volume, liter	4.9134
Maximum power, HP	75
Injection pressure, bar	200
Inlet valve open (IVO)	8° bTDC
Inlet valve closing (IVC)	45° aBDC
Exhaust valve Open (EVO)	45° bBDC
Exhaust valve close (EVC)	12° aTDC
Fuel injection timing, degrees	26° bTDC

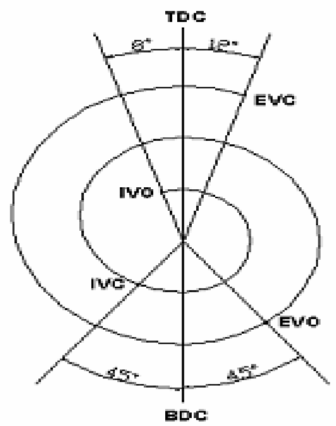

Figure 3. Valve timing diagram

The tests were performed at full load condition for the engine speeds from 1000 to 1500 rpm with 100 rpm step (Figure 4). The ambient, cooling water and exhaust gas temperature before the turbine inlet was measured by K type thermocouples. The thermocouples were connected to a multi-channel temperature indicator. Fuel consumption was measured with a constant volume burette with stopwatch. Airflow rate was measured through air box connected with water manometer. First, diesel fuel was used as fuel and then biodiesel was used as fuel. After the completion of test on conventional engine, the engine parts were coated with plasma sprayed Partially Stabilized Zirconia (PSZ) of 0.5 mm thickness, as the test engine was converted in to LHR engine. The same test procedure was repeated for the LHR engine and the experimental results were compared with predicted results.

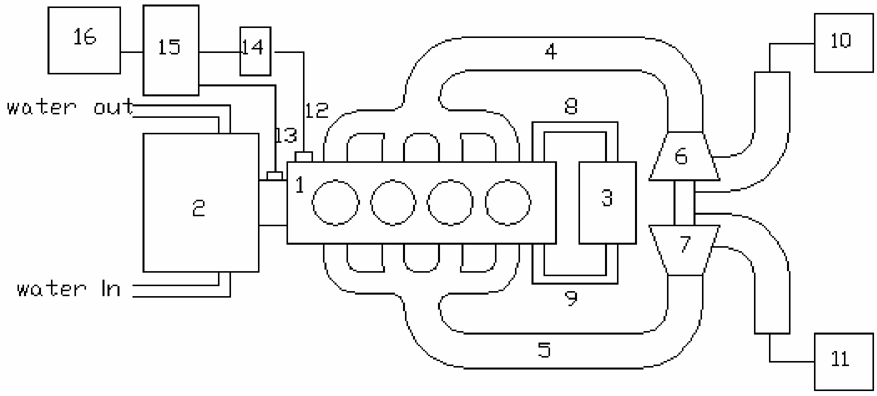

Figure 4. Schematic of experimental set up

Both the diesel and biodiesel are used in conventional engine and LHR engine with different speeds such as 1000, 1100, 1200, 1300, 1400, 1500 rpm. The simulation and experimental studies were made for four combinations such as Turbocharged engine operated with diesel (denoted by TC Diesel), Turbocharged engine operated with biodiesel (denoted by TC Biodiesel), Turbocharged LHR engine operated with diesel (denoted by LHR Diesel) and Turbocharged LHR engine operated with biodiesel (denoted by LHR Biodiesel)

4. Conversion of conventional turbocharged engine to LHR engine

Increasing the coating thickness decreases the volumetric efficiency and hence work done. In this experimental study ceramic coating thickness was optimized as 0.5 mm [29,30]. The engine combustion chamber was coated with partially stabilized zirconia (PSZ) of 0.5 mm thickness, which includes the piston crown, cylinder head, valves, and outside of the cylinder liner. The equal amount of material has been removed from the various parts of the combustion chamber and PSZ was coated uniformly. After PSZ coating, the engine was allowed to run about 10 hours, then test were conducted on it. The present work involves to usage of biodiesel in LHR engine to analyse the combustion and performance characteristics. To validate the theoretical results, experiments were conducted on a turbocharged direct injection (DI) diesel engine and LHR engine using diesel and biodiesel under identical operating conditions.

5. Theoretical considerations

In this analysis the molecular formula for diesel and biodiesel are approximated, as $C_{10}H_{22}$ and $C_{19}H_{34}O_2$ [31]. The combustion model is developed for the C.I engine and suitable for any hydrocarbon fuel and their blends.

5.1 Calculation of number of moles of reactants and products

In this simulation during the start of combustion, the moles of different species are considered includes O_2, N_2 from intake air and CO_2, H_2O, N_2 and O_2 from the residual gases. The overall combustion equation considered for the fuel with C-H-O-N is

$$C_xH_yO_z + \lambda * y_{cc} * (O_2 + 3.773N_2) \longrightarrow xCO_2 + (y/2)H_2O + (\lambda - 1)y_{cc}O_2 + \lambda y_{cc} * 3.773N_2 \quad (1)$$

Stochiometric AFR $y_{cc} = x + (y/4) - (z/2)$

Total number of reactants and products during the start of combustion as well every degree crank angle was calculated from the equations.

$$tmr = 1 + \lambda * y_{cc} * 4.773 \tag{2}$$

$$tmp = x + (y/4) + 3.773 * \lambda * y_{cc} + (\lambda - 1) * y_{cc} \tag{3}$$

5.2 Volume at any crank angle

Volume at any crank angle is calculated from this equation

$$V_\theta = V_C + \left(\pi * \frac{d^2}{4} * \frac{l}{2}\right) * \left(1 + Z - \left(Z^2 - \sin^2\theta\right)^{\frac{1}{2}} - \cos\theta\right) \tag{4}$$

where $Z = 2*(L/l)$

5.3 Calculation of specific heat

Specific heat at constant volume and constant pressure for each species is calculated using the expression given below

$$Cv(T) = (B - \bar{R}) + \frac{C}{T} \tag{5}$$

$$Cp(T) = B + \frac{C}{T} \tag{6}$$

where A, B and C are the coefficients of the polynomial equation

5.4 Initial pressure and temperature during start of compression

Initial pressure and temperature at the beginning of the compression process is calculated as follows

$$P_2 = \left(\frac{V_1}{V_2}\right) * \left(\frac{T_2}{T_1}\right) * P_1 \tag{7}$$

and

$$T_2 = T_1 * \left(\frac{V_1}{V_2}\right)^{\frac{R}{Cv(T_1)}} \tag{8}$$

5.5 Calculation of enthalpy and internal energy

Enthalpy of each species is calculated from the expression given below which is used to calculate the peak flame temperature of the cyclic process.

$$H(T) = A + B*T + C*\ln(T) \tag{9}$$

The internal energy for each species and overall internal energy are calculated from the expressions given below

$$U(T) = A + (B - \bar{R})*T + C*\ln(T) \tag{10}$$

$$U(T) = \sum (x_i U_i(T)) \tag{11}$$

where A, B and C are the coefficients of the polynomial equation.

5.6 Work done
Work done in each crank angle is calculated from

$$dW = \left(\frac{P_1 + P_2}{2}\right)(V_2 - V_1) \tag{12}$$

5.7 Total heat transfer
The gas-wall heat transfer is found out using ANNAND'S combined heat transfer model. Radiation heat transfer was also estimated using this model. A wall heat transfer model is used to find out the instantaneous wall temperature. First term of this equation shows that Prandtl number for the gases forming the cylinder contents will be approximately constant at a value 0.7, claims that Reynolds number is the major parameter affecting convection. The second is a straightforward radiation term assuming gray body radiation.

$$\frac{dQ}{dt} = ak\frac{\mathrm{Re}^b}{d}(T_g - T_W) + c(T_g^4 - T_W^4) \tag{13}$$

a, b and c are the constants of the equation

5.8 Heat transfer coefficient
Heat transfer coefficient of gases for each degree crank angle is calculated from the following equation.

$$h_g = 0.26 * (k/d) * R_e^{0.6} \tag{14}$$

5.9 Wall heat transfer model
Thermal network model was developed (for wall heat transfer analysis) are used to analyze the effect of coating on engine heat transfer. The model used to analyze the heat transfer through cylinder to the coolant and thereby to find instantaneous wall temperature (Figure 5). Initial temperature is found out using the following expression

$$T_W = T_g - \left(Q_w / 2\pi h_g r l\right) \tag{15}$$

The total resistance (Rt) offered by the cylinder liner, piston rings, cylinder head, ceramic coating and piston for the heat transfer from cylinder gases to coolant is given by the following equation.

$$R_t = \frac{1}{(h_g * 2 * \pi * r_1 * l)} + \frac{1}{h_{co} * 2 * \pi * r_3 * l)} + \frac{\log(r_2 / r_1)}{2 * \pi * k_1 * l} + \frac{2 * t_p}{k_p * 2 * \pi * r^2} + \frac{\log(r_5 / r_9)}{k_r * 2 * \pi * l_r} + \frac{\log(r_3 / r_4)}{k_c * 2 * \pi * l_c} + \frac{2 * t_c}{k_c * 2 * \pi * r^2} \tag{16}$$

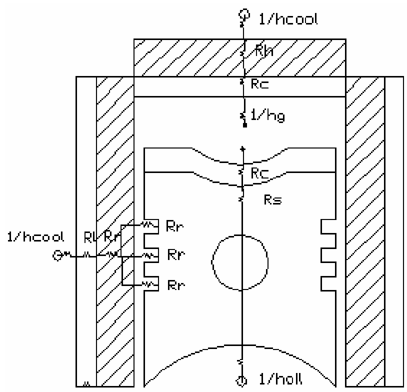

Figure 5. Thermal network model for wall heat transfer

5.10 Energy equation

According to first law of thermodynamics the energy balance equation is given by

$$U(T_2) = U(T_1) - dW - dQ + dM_f Q_S \tag{17}$$

To find the correct value of T_2, both sides of the above equation should be balanced. So the above equation is rearranged as shown below

$$ER = U(T_2) - U(T_1) - dW - dQ + dM_f Q_S \tag{18}$$

If the numerical value of ER is less than the accuracy required, then the correct value of T_2 has been established, otherwise a new value of T is calculated for new internal energy and C_V values.

$$ER' = C_V(T_2) * tmp \tag{19}$$

Newton-Rapshon technique

$$(T_2)_n = (T_2)_{n-1} - \frac{ER}{ER'} \tag{20}$$

5.11 Mass of fuel injected

Considering that nozzle open area is constant during the injection period, mass of the fuel injected for each crank angle is calculated using the following expression

$$M_f = C_d A_n \sqrt{2\rho_f \Delta P}\left(\frac{\Delta\theta_f}{360N}\right) \tag{21}$$

5.12 Unburned zone temperature

$$T_u = T_{SOC}\left(\frac{p}{p_{SOC}}\right)^{\frac{\gamma-1}{\gamma}} \tag{22}$$

5.13 Preparation rate

The preparation rate is calculated using the following equation given by

$$P_r = KM_f^{(1-x)} M_u^{x} P_{O2}^{L} \tag{23}$$

$$K = 0.085 N^{0.414} M_f^{1.414} \Delta\theta_f^{-1.414} n^{-1.414} d_n^{-3.644} \tag{24}$$

5.14 Reaction rate

The reaction rate is calculated using the following equation

$$R_r = \frac{K' P_{O2}}{N\sqrt{T}} e^{\frac{-act}{T}} \int (P_r - R_r) d\theta \tag{25}$$

5.15 Initial temperature during the start of combustion

Initial temperature during the start of combustion after the ignition delay is calculated using the equation given below

$$T_2 = T_1 + \frac{x_f Q_s}{C_v(T_1)} \tag{26}$$

6. Results and discussions

In this study combustion parameters like cylinder pressure, peak cylinder pressure and heat release are discussed. Performance parameters like brake thermal efficiency and specific fuel consumption are discussed. The results are compared with respect to LHR Biodiesel operation.

6.1 Cylinder pressure

In a CI engine the cylinder pressure is depends on the fuel-burning rate during the premixed burning phase. The high cylinder pressure ensures the better combustion and heat release. The Figures (6 –9) shows the typical pressure variation with respect to crank angle. The advantages of higher temperature of LHR engine increase the cylinder pressure. The cylinder pressure for LHR biodiesel is lower than LHR diesel about 1% and higher about 4% and 2.5% for TC biodiesel and TC diesel.

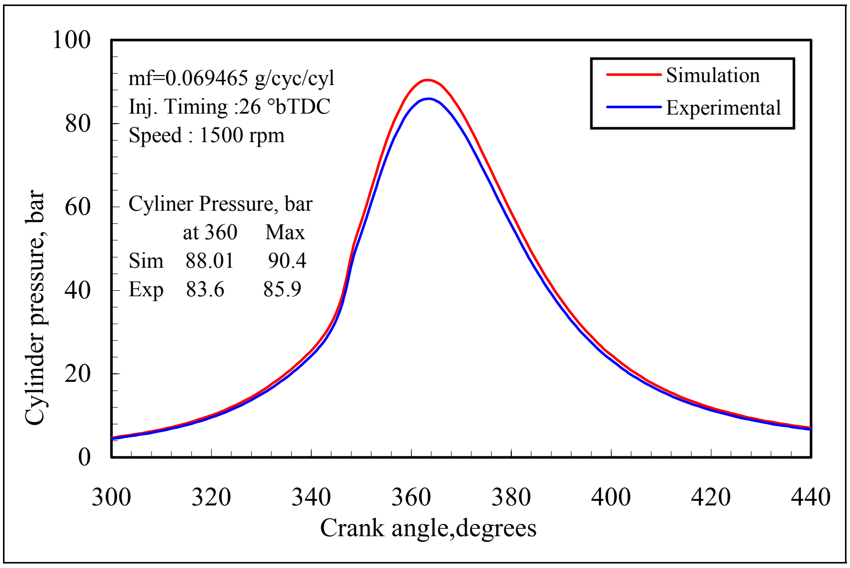

Figure 6. Cylinder pressure with crank angle for TC diesel at full load

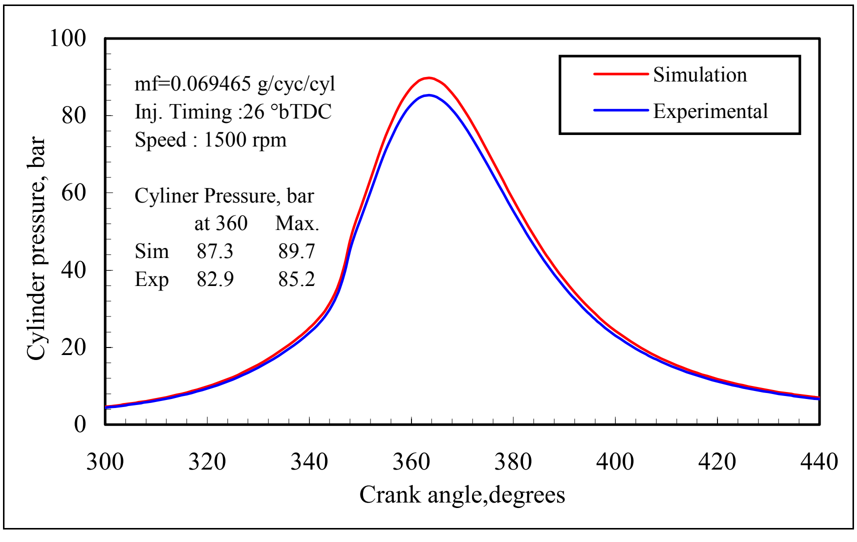

Figure 7. Cylinder pressure with crank angle for TC biodiesel at full load

The lower calorific value of biodiesel and higher initial temperature due to the domination of residual gases decreases the volumetric efficiency of LHR engine, thus reduces the peak pressure and work done. The cylinder pressure for TC biodiesel is lower than TC diesel due to the reduction in heat supply for blended fuel. It is noted that the maximum pressure obtained for biodiesel is closer with TDC than diesel fuel. The fuel-burning rate of biodiesel in early stage of combustion is more than the diesel fuel, which bring the peak pressure close to TDC. During the expansion process, the torque produced by the engine is kind enough to go for the next cycle with the consumption of more fuel.

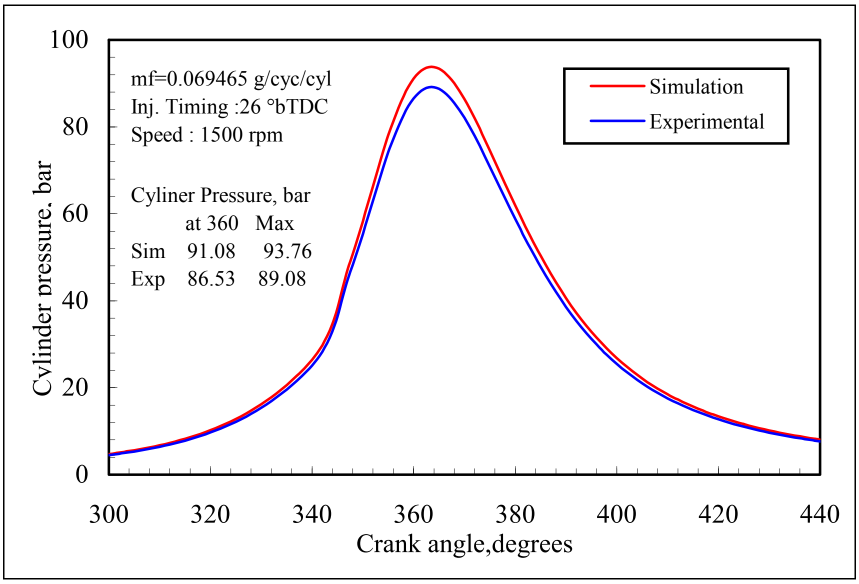

Figure 8. Cylinder pressure with crank angle for LHR diesel at full load

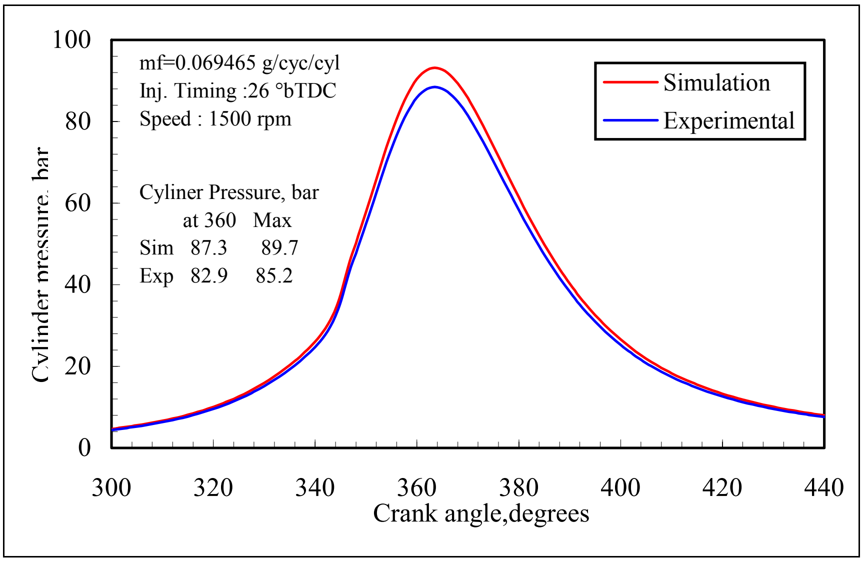

Figure 9. Cylinder pressure with crank angle for LHR Biodiesel at full load

6.2 Cylinder temperature

High pressure of compressed mixture increases its burning rate. This increases the peak pressure inside the combustion chamber. The comparisons of peak temperatures inside the cylinder for various combinations are shown in Figure 10. The presence of oxygen in the biodiesel makes complete combustion of fuel thereby producing more CO_2 and hence more heat is released from the gases. Thus, the peak temperature of biodiesel-fueled engine is higher than that of diesel fueled engine. The peak cylinder temperature for LHR biodiesel is higher than LHR diesel TC biodiesel and TC diesel about 1%, 4.4%-6.1% and 5.8%-7.4% respectively from higher speed to lower speed.

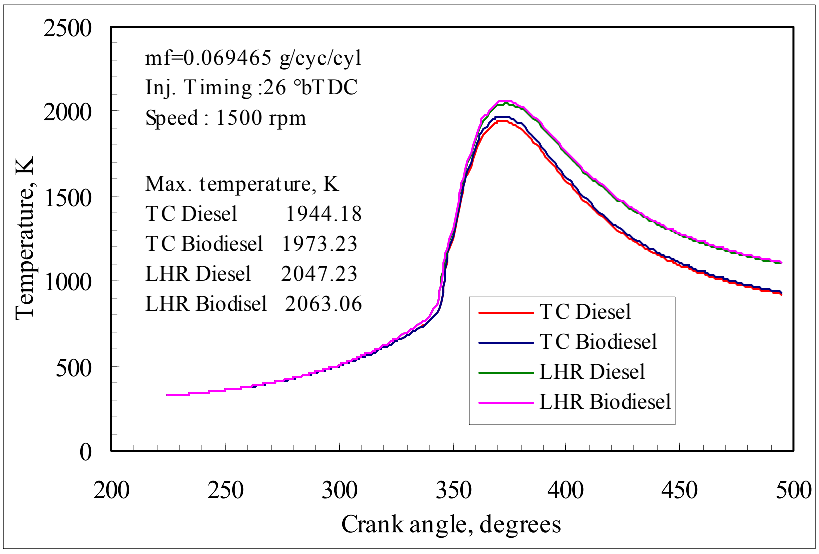

Figure10. Cylinder temperature with crank angle at full load

6.3 Heat release rate

Figures 11 and 12 show the variation of rate of heat release with crank angle. Both fuel experiences that the rapid premixed combustion followed by diffusion combustion. The premixed fuel burns rapidly and releases the enormous amount heat followed by the controlled heat release. The heat release rate during the premixed combustion is responsible for the high peak pressure. C.I engine combustion is divided in to four stages such as ignition delay, uncontrolled combustion, controlled combustion and after burning.

Ignition delay: The higher operating temperature reduces the ignition delay for both the fuel. However oxygenated nature of biodiesel further reduces the ignition delay. Combustion starts well in advance for LHR operation due to higher surrounding temperature.

Premixed combustion: Fuel droplets accumulated during the ignition delay period is responsible for the sudden heat release during this period is called premixed or uncontrolled combustion. Due to higher ignition delay of TC operation lower the heat release and higher the heat release for LHR operation. The peak heat release for LHR was advanced few degree crank angle than TC operation due to shorter ignition delay and higher combustion temperature.

Diffusion combustion: After the evaporation of fuel droplets during the delay period, the fuel jets coming from the nozzle are converted in to fuel vapor is called as diffusion combustion or controlled combustion. During this period both fuel in TC and LHR operation experience the smooth combustion and follow the similar pattern. However a marginal improvement was noticed for biodiesel.

After burning: The higher boiling point fuel molecules release the heat during this period called after burning.

Findings from the analysis are summarized as follows

- Decreases the ignition delay period for LHR engine operation.
- Premixed burning period decreases and diffusion-burning period increases for LHR operation.
- Total combustion duration increases for LHR operation.

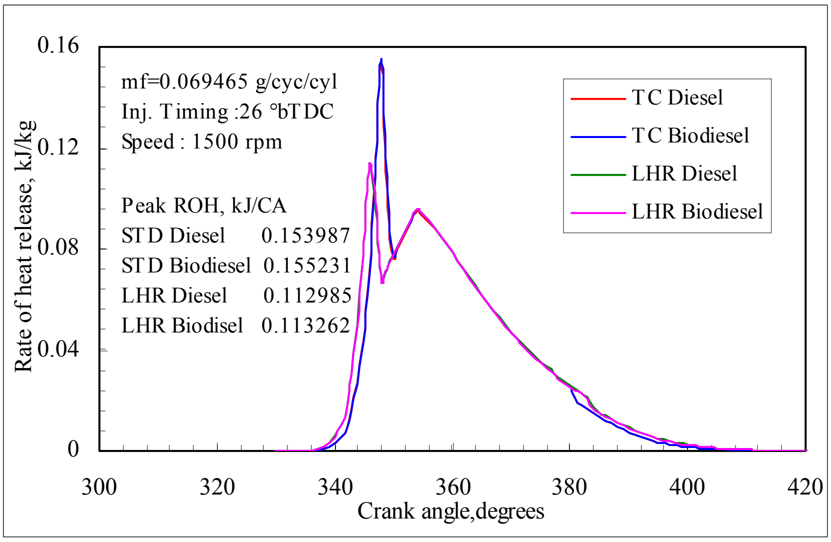

Figure 11. Rate of heat release with crank angle at full load

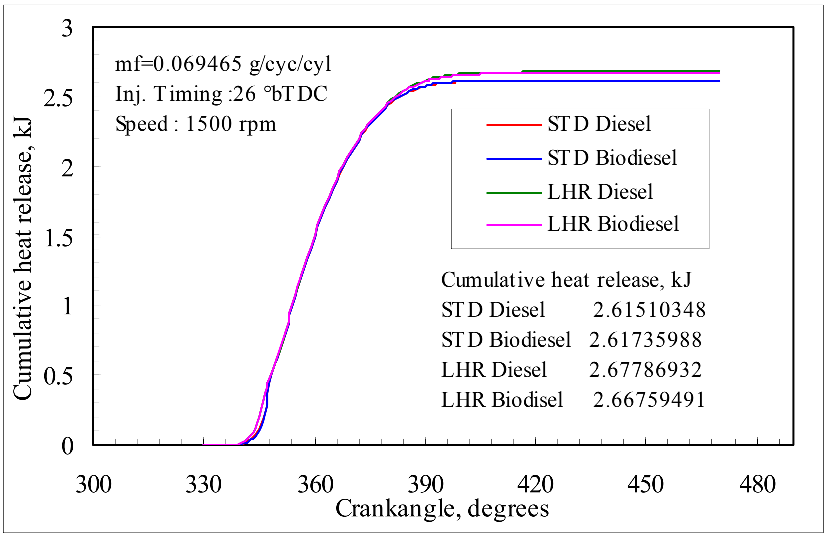

Figure 12. Cumulative heat release with crank angle at full load

6.4 Total heat transfer

ANNAD'S combined (convection and radiation) heat transfer model was used to predict the engine heat transfer. In general LHR engine suppress the heat transfer to the coolant side, and increases the exhaust energy. The variation of engine heat transfer with crank angle is shown in Figure 13. The heat rejected to the coolant through cylinder walls are reduced for LHR engine due to ceramic coatings. Higher amount of heat can be trapped inside the cylinder, however all the trapped heat cannot be converted in to the useful work output, hence higher the exhaust temperature. A marginal improvement on the brake thermal efficiency was noticed due to low heat loss.

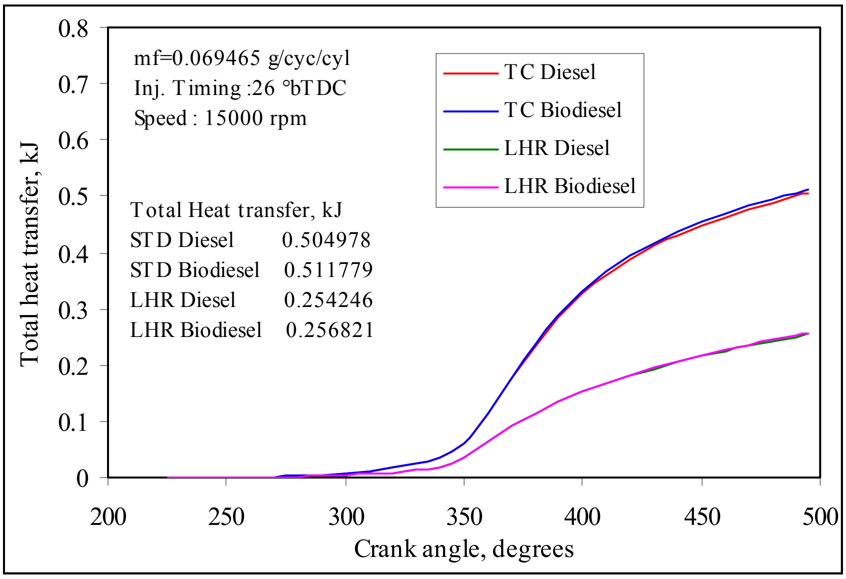

Figure 13. Total transfer with crank angle at full load

6.5 Work done

The variation of rate of work done is shown in Figure 14. Work done for LHR diesel operation is higher due to higher peak pressure, followed by LHR biodiesel, TC diesel and TC biodiesel operation. The advantages of higher operating temperature for LHR operation are responsible for the higher work done. The higher work done leads to high thermal efficiency. Higher initial temperature of LHR operation reduces the volumetric efficiency. The energy lost during the compression process is compensated from expansion process due to increased combustion efficiency

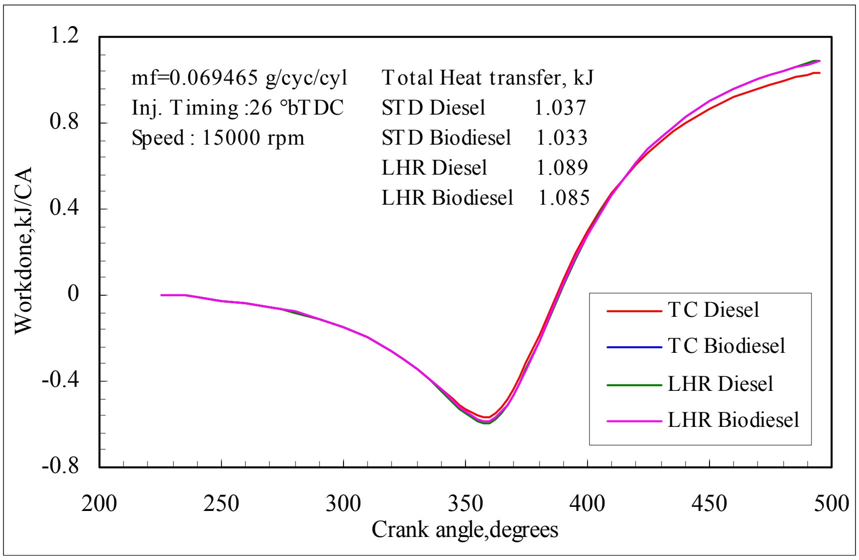

Figure 14. Work done with crank angle at full load

6.6 Brake thermal efficiency

The variation of brake thermal efficiency with engine speed is shown in Figure15. It can be seen that the brake thermal efficiency of LHR biodiesel engine is lower than that of LHR diesel and higher than TC biodiesel and TC diesel operation. The calorific value of biodiesel is lower than (about 12%) that of diesel because of the presence of oxygen in its molecule. However due to the presence of oxygen molecule in the biodiesel lowers the calorific value and hence the reduction in brake thermal efficiency was obtained. Hence, the brake thermal efficiency of LHR biodiesel is lower than that of LHR diesel.

The blends resulted in slightly higher efficiency than the diesel fuel, mainly due to more complete combustion [32-34]. Lower initial temperatures due to low calorific value and oxygenated nature of biodiesel in LHR engine increase the work done than TC operation.

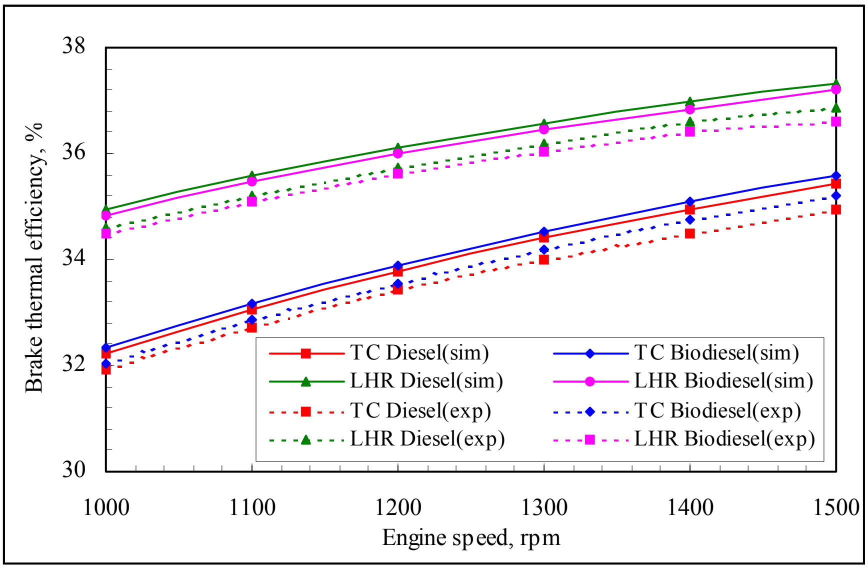

Figure 15. Variation of brake thermal efficiency with engine speed at full load

6.7 Specific fuel consumption

Figure 16. shows the variations of specific fuel consumption (SFC) with respect to the engine speed. The SFC values of the biodiesel were slightly higher than diesel fuel in both TC and LHR operation. Lower calorific value of biodiesel leads to increase the fuel consumption.

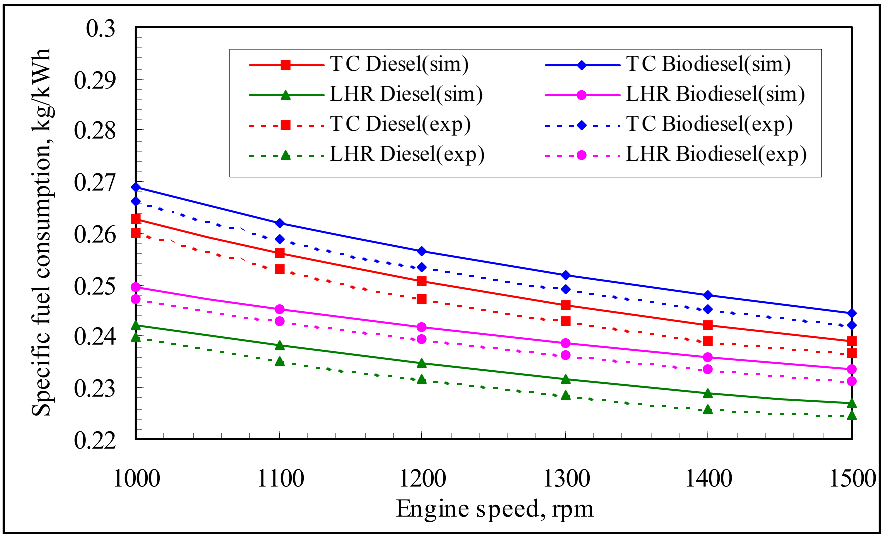

Figure 16. Iation of specific fuel consumption at full load

7. Conclusion

A mathematical model was developed using a set of codes for analyzing the combustion and performance characteristics of turbocharged and LHR engine. The model has been developed in such way that it can be used for characterizing any hydrocarbon fuels and their blends. The result shows that, increasing the brake thermal efficiency and decreasing the specific fuel consumption for LHR engine operated with diesel, shows better performance than LHR biodiesel operation but not up to the extent of lower level. However biodiesel in LHR engine shows better performance than conventional engine operation. This model predicted the engine performance characteristics in closer approximation to that of experimental results. Hence, the developed mathematical model is suitable for the prediction of the combustion and performance characteristics of the C.I engine and LHR engine.

Nomenclature

act	-	Activation energy, kJ/kmol K
An	-	Cross sectional area of nozzle, m^2
Cd	-	Coefficient of discharge of injector nozzle
Cp	-	Specific heat at constant pressure, kJ/kg
Cv	-	Specific heat at constant volume, kJ/kgK
d	-	Cylinder diameter, m
dn	-	Diameter of nozzle, m
Q	-	Heat transfer, kJ/CA
H	-	Enthalpy at any degree crank angle, kJ/kg
hco	-	Heat transfer coefficient of coolant, w/m^2K
hg	-	Heat transfer coefficient of gas, w/m^2K
K’	-	Constant of equation
kc	-	Thermal conductivity of coating material, W/m k
kp	-	Thermal conductivity of piston, W/m k
kr	-	Thermal conductivity of piston ring, W/m k
kr	-	Thermal conductivity of piston skirt, W/m k
L	-	Connecting rod length, m
l	-	Stroke length, m
ll	-	Length of liner, m
lc	-	Coating thickness, m
lc	-	Thickness of piston ring, m
Mf	-	Mass of fuel supplied, kg/h
Mu	-	Mass of fuel unprepared, g/CA
N	-	Engine speed, rpm
n	-	No. of nozzle holes
P_1	-	Initial pressure, bar
P_2	-	Final pressure, bar
Pr	-	Preparation rate, kg/h
Rr	-	Reaction rate, kg/h
P_{O2}	-	Partial pressure of oxygen, bar
Psoc	-	Pressure at the start of combustion, bar
Qs	-	Total heat supply, kJ/kg
Qw	-	Heat transfer through wall, kJ
R	-	Charecteristic gas constant, kJ/kgK
Re	-	Reynolds number
Rt	-	Total resistance, m2 k/W
T	-	Temperature at any crank angle, K
tp	-	Thickness of piston, m
T1	-	Initial temperature, K
T2	-	Final temperature, K
Tg	-	Gas temperature, K
Tw	-	Wall temperature, k
tmr	-	Total moles of reactants
tmp	-	Total moles of products
Tsoc	-	Temperature at the start of combustion, K
T	-	Temperature at any degree crank angle, K
Tu	-	Unburned zone temperature, K
Tw	-	Wall temperature, K
U	-	Internal energy at any degree crank angle, kJ/kgK
V1	-	Initial volume, m^3
V2	-	Final volume, m^3
Vc	-	Clearance volume, m^3

Vp	-	Mean piston speed, m/min
Vθ	-	Volume at any crank angle, m^3
W	-	Work done, kJ/kg
x	-	No. of moles of carbon
xf	-	Mass fraction of fuel burnt
y	-	No. of moles of hydrogen
ycc	-	Stochiometric air fuel ratio
z	-	No. of moles of oxygen

Greek

ΔP	-	Pressure drop across the nozzle, bar
Δθf	-	Fuel injection period, degrees
θ	-	Crank angle, degrees
λ	-	Excess air factor
ρ	-	Density of gas mixture, kg/m^3
ρf	-	Density of fuel, kg/m^3
μ	-	Kinematic viscosity, cSt

Abbreviations

BTE	-	Brake thermal efficiency
C.I	-	Compression ignition
CO	-	Carbon monoxide
D.I	-	Direct injection
HC	-	Hydrocarbon
JSO	-	Jatropha seed oil
LHR	-	Low heat rejection
PSZ	-	Partially stabilized zirconia
SFC	-	Specific fuel consumption
SVO	-	Straight vegetable oil
TC	-	Turbocharged engine

References

[1] Ryan TW, Bagby MO, “Identification of chemical changes occurring during the transient injection of selected vegetable oils”, SAE Paper No. 930933, 1993.

[2] Dorado MP, Ballesteros E, Arnal JM, Gomez J, Lofez FJ, “Exhaust emissions from a diesel engine fueled with transesterified waste olive oil”, Fuel 82:1311–5, 2003.

[3] Ziejewski M, Kaufman KR, ”Vegetable oils as a potential alternate fuel in direct injection diesel engines”, SAE Paper No. 831357, 1983.

[4] Schlick ML, Hanna MA, Schinstock JL, “Soybean and sunflower oil performance in a diesel engine’, Trans ASAE 31(5): 1345–9, 1988.

[5] Hemmerlein N, Korte V, Richter H, Schroder G, “Performance, exhaust emissions and durability of modern diesel engines running on rapeseed oil”, SAE Paper No. 910848, 1991.

[6] Lang X, Dalai AK, Bakhshi NN, Reaney MJ, Hertz PB, “Preparation and characterization of bio-diesels from various bio-oils”, Bioresource Technol 80:53–62, 2001..

[7] Monyem A, Van Gerpen JH, “The effect of biodiesel oxidation on engine performance and emissions”, Biomass Bioenergy 20:317–25, 2001.

[8] Al-Widyan M, Tashtoush G, Abu-Qudais M, “Utilization of ethyl ester of waste vegetable oils as fuel in diesel engines”, Fuel Process Technol 76:91–10, 2002.

[9] Zhang Q, FeldmanM, Peterson CL, ”Diesel engine durability when fueled with methyl ester of winter rapeseed oil”, ASAE paper No. 881562, 1988.

[10] Perkins LA, Peterson CL, Auld DL, “Durability testing of transesterified winter rape oil as fuel in small bore, multi-cylinder, DI, CI engines”, SAE Paper No. 911764, 1991.

[11] Sanjib Kumar Karmee, Anju Chadha, “Preparation of biodiesel from crude oil of Pongamia pinnata”, Bioresource Technology 96, 1425–1429, 2005.

[12] Ayhan Demirba, “Comparison of transesterification methods for production of biodiesel from vegetable oils and fats”, Energy Conversion and Management 49 125–13, 2008.

[13] A.S. Ramadhas, S. Jayaraj, C. Muraleedhara, “Biodiesel production from high FFA rubber seed oil”, Fuel 84 335–340, 2005.

[14] L.C. Meher a, Vidya S.S. Dharmagadda b, S.N. Naik, “Optimization of alkali-catalyzed transesterification of Pongamia pinnata oil for production of biodiesel”, Bioresource Technology 1392–1397, 97, 2006.

[15] Abderrahim Bouaid, Mercedes Martinez, Jose Aracil, “A comparative study of the production of ethyl esters from vegetable oils as a biodiesel fuel optimization by factorial design”, Chemical Engineering Journal 134, 93–99, 2007.

[16] Abdullah Uzun a, I˙smet C¸ evik b, Mustafa Akc¸il, “Effects of thermal barrier coating on a turbocharged diesel engine performance”, Surface and Coatings Technology 116–119, 505–507, 1999

[17] James A. Leidel, “An Optimized Low Heat Rejection Engine for Automotive Use - An Inceptive Study”, SAE Paper No. 970068, 1997.

[18] Ekrem Bu¨yu¨kkaya , Tahsin Engin, Muhammet Cerit, “Effects of thermal barrier coating on gas emissions and performance of a LHR engine with different injection timings and valve adjustments”, Energy Conversion and Management 47, 1298–1310, 2006.

[19] I. TaymazT, K. C¸ akVr, A. Mimaroglu, “Experimental study of effective efficiency in a ceramic coated diesel engine”, Surface and Coatings Technology 200, 1182– 1185, 2005.

[20] S.Jaichandar and P.Tamilporai, “Low Heat Rejection Engines – An Overview”, SAE 2003-01-0405,2003.

[21] Imdat Taymaz, “The effect of thermal barrier coatings on diesel engine performance”, Surface and Coatings Technology 201, 5249–5252, 2007.

[22] B.Rajendra Prasath, P.Tamilporai, Mohd.F.Shabir, T.Senthilkumar, “Simulation and Analysis of Combustion Performance and Emission Characteristics of Biodiesel Fueled Low Heat Rejection Direct Injection Diesel Engine”, SAE Paper No. 2007-32-0094.

[23] Mohd.F.Shabir, Tamilporai P., and Rajendra Prasath B. (2010), ‘Analysis of Combustion, Performance and Emission Characteristics of Turbocharged LHR Extended Expansion DI Diesel Engine’, International Journal of Mechanical Systems Science and Engineering, Vol. 2, No. 2, pp. 126-137.

[24] Jonh. B. Heywood, “Internal combustion engines fundamentals”, Tata McGraw Hill Book Company; 1989.

[25] Ganesan V, “Computer simulation of compression ignition engines”, University Press (India) Ltd.; 2000.

[26] P.Tamilporai, N.Baluswamy, ” Simulation and analysis of heat transfer in low heat rejection direct injection diesel engines using a two zone model” 3 rd Asian – Pacific international symposium on Combustion and energy utilization.1998.

[27] Fangrui Maa, Milford A. Hannab,” Biodiesel production: a review” Bioresource Technology 70, 1-15, 1999.

[28] P. Chitra, P. Venkatachalam and A. Sampathrajan, ”Optimisation of experimental conditions for biodiesel production from alkali-catalysed transesterification of Jatropha curcus oil” Article from Department of Bio-Energy, AEC & RI, TamilNadu Agricultural University, Coimbatore-641 003, Tamil Nadu, India, 2005.

[29] P.Tamilporai,” Simulation and analysis of combustion and heat transfer in low heat rejection diesel engine using two zone combustion model and different heat transfer models” PhD thesis, Anna University, India.1998.

[30] B.Rajendra Prasath, P.Tamilporai, Mohd.F.Shabir, “ Theoretical Modeling and Experimental Study of Combustion and Performance Characteristics of Biodiesel in Turbocharged Low Heat Rejection D.I Diesel Engine”, World Academy of science, Engineering and Technology, 61,2010,pp.617-627.

[31] A.S. Ramadhas_, S. Jayaraj, C. Muraleedharan, “Theoretical modeling and experimental studies on biodiesel-fueled engine”, Renewable Energy 31, 1813–1826. 2006.

[32] Gonzalez Gomez ME, Howard-Hildige R, Leahy JJ, O. Reilly T, Supple B, Malone M, “Emission and performance characteristics of a 2 litre Toyota Diesel van operating on esterified waste cooking oil and mineral diesel fuel”, Environ Monit Assess 65,13–20, 2000.

[33] Pramanik K, “Properties and use of jatropha curcas oil and diesel fuel blends in compression ignition engine. Renew Energy 28(2): 239–48, 2003.

[34] Al-Widyan MI, Tashtoush G, Abu-Qudais M, " Utilization of ethyl ester of waste vegetable oils as fuel in diesel engines", Fuel Process Technol 76,91–103, 2000.

B. Rajendra Prasath, Graduated Bachelor of Engineering (Mechanical Engineering) in the year 2000 from Madurai Kamaraj University and Master of Engineering (Internal Combustion Engineering) in the year 2003 from, College of Engineering, Guindy, Anna university, Chennai, India. Currently pursuing his Ph.D degree in Anna university, Chennai, India. Working as a Assistant Professor in Tagore Engineering College, Anna University, Chennai, India. His area of interest is usage of bio fuels in internal combustion engines and emission reduction. Mr. B. Rajendra Prasath, life member in Indian society for technical education (ISTE).
E-mail address: br_prasath@rediffmail.com

P. Tamilporai, Graduated Bachelor of Engineering (Mechanical Engineering) form Govt. College of Engineering, Salem, Tamilnadu, India and Master of Engineering (Automobile Engineering) from College of Engineering, Guindy, Anna university, Chennai, India. He obtained his PhD degree from Anna University, Chennai, India. Currently working as Professor & Head, Internal Combustion Engineering Division, Department of Mechanical Engineering, Anna University, Chennai, India. He has vast experience in teaching and guiding research scholars. He has published so many research articles and papers in various national and international journals and conferences. His main interest of research is "Low Heat Rejection engines". Dr. P. Tamilporai Porai, life member in Indian society for technical education (ISTE).
E-mail address: tporai@annauniv.edu

Mohd. F. Shabir is currently working as Associate Professor, Department of Mechanical Engineering, Tagore Engineering College, Affiliated to Anna University, Chennai, India. He completed his B.E. (Mechanical Engineering) degree from University of Madras and M.E. (Internal Combustion Engineering) degree from Anna University, Chennai, India. Presently pursuing his PhD programme in Anna University, Chennai, India in the field of Simulation and Analysis of Low Heat Rejection Extended Expansion engine. Mr, Mohd.F.Shabir is the Member of Indian Society of Technical Education.
E-mail address: ffshabir2001@yahoo.co.in