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Investigation on Effect of Variation in Compression Ratio on Performance and Combustion Characteristics of C.I Engine Fuelled With Palm Oil Methyl Ester (POME) and Its Blends By Simulation

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Abstract - The paper describes the development of zero dimensional single zone thermodynamic model for compression ignition engine cycle simulation. Rate of heat release due to combustion is modeled with double wiebe function, takes care of premixed as well as diffusive phase of combustion. Adjustable parameters of wiebe function are obtained by fitting it to experimental mass fraction burned profile by least square method. Empirical correlations are established between adjustable parameters of wiebe function, relative air-fuel ratio and engine operating conditions. The simulation is used to analyze the engine performance fuelled with diesel, Palm Oil Methyl Ester (POME) and its blends. Effect of change in compression ratio on peak pressure, net heat release rate and brake thermal efficiency is analyzed and discussed. The model is validated by comparing predicted peak pressure and brake thermal efficiency with diesel and POME –diesel blends at 17.5:1 compression ratio with that of experimental results.

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I. Introduction

Energy is prominent requirement of present society. Internal combustion engines have been the prime movers for generating power for various applications for more than a century [1]. The increasing demand, depletion and price of the petroleum prompted extensive research worldwide on alternative energy sources for internal combustion engines. Use of straight vegetable oils in compression ignition engine for long term deteriorates the engine performance and is mainly because of higher viscosity [2-6]. The best way to use vegetable oils as fuel in compression ignition engines is to convert it into biodiesel [7]. Biodiesels such as rape seed, soybean, sunflower and Jatropha, etc. are popular substitutes for diesel [8]. In the present energy scenario efforts are being focused on use of bio diesel in compression ignition engine, but there are many issues related to performance and emission [8]. The optimum operating parameters can be determined using experimental techniques but experimental procedure will be time consuming and expensive [9]. Computer simulation [10] serves as a tool for a better understanding of the variables involved and also helps in optimizing the engine design for a particular application thereby reducing cost and time. The simulation approach allows examining the effects of various parameters and reduces the need for complex experimental analysis of the engine [11]. A validated simulation model could be a very useful tool to study engines running with new type of fuels.

A zero-dimensional single-zone model as compared with multi-zone models is much simpler, quicker and easier to run. [12, 13] and it is capable of predicting engine performance and fuel economy accurately with a high computational efficiency [14]. Hence a zero-dimensional single-zone model is developed similar to the one developed previously by the authors [15] where single Wiebe function is used. In this paper double Wiebe function is used to model heat release rate.

II. Description Of Mathematical Modeling

a) List of symbols

r = compression ratio.

L = length of connecting rod (mm).

B = bore diameter (mm).

V_{disp} = displacement volume (m^3).

θ = angular displacement in degrees with respect to bottom dead center (BDC).

θ_s = crank angle at the start of combustion.

γ = specific heat ratio.

P = pressure (bar).

V = volume (m^3).

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m_c = number of moles of carbon in one mole of fuel.

m_h = number of moles of hydrogen in one mole of fuel.

m_o = number of moles of oxygen in one mole of fuel.

m = mass of the charge (kg).

h_c = coefficient of heat transfer due to convection (W/m^2.K).

A = interior surface area of cylinder (m^2).

T = instantaneous gas temperature (Kelvin).

T_w = cylinder wall temperature (Kelvin).

R = universal gas constant (kJ/kmole.kelvin).

C_m = piston mean speed (m/s).

U = internal energy.

H = enthalpy.

C_P = specific heat at constant pressure (kJ/kg.kelvin).

C_V = specific heat at constant volume (kJ/kg.kelvin).

$\Delta\theta$ = combustion duration in crank angle (degrees).

Q_r = heat released per cycle (kJ).

$\frac{dQ_r}{d\theta}$ = rate of heat released during combustion (kJ/degree CA).

$\frac{dQ_h}{d\theta}$ = rate of heat transfer (kJ/degree CA).

$\frac{dw}{d\theta}$ = rate of work done.

$\frac{du}{d\theta}$ = rate of change of internal energy.

$\frac{dV}{d\theta}$ = incremental change in cylinder volume (m^3/degree CA).

$\frac{dT}{d\theta}$ = rate of temperature change (Kelvin / degree CA).

Q_p = heat released during premixed phase (kJ).

Q_d = heat released during diffusive phase (kJ).

m_p = shape factor of premixed phase.

m_d = shape factor of diffusive phase.

θ_p =burning duration of premixed phase.

θ_d =combustion duration.

b) Energy balance equation

According to the first law of thermodynamics, the energy balance equation for the closed cycle is

$$m\frac{du}{d\theta}=\frac{dQ_r}{d\theta}-\frac{dw}{d\theta} \tag{1}$$

The heat term (rate of heat release) can be split into the heat released due to combustion of the fuel and the heat transfer that occurs to the cylinder walls or from the cylinder walls to gases. The equation (1) can be written as

$$m\frac{du}{d\theta}=\frac{dQ_r}{d\theta}-\frac{dQ_h}{d\theta}-\frac{dw}{d\theta} \tag{2}$$

Replacing the work transfer by $p\frac{dV}{d\theta}$ or by the ideal gas law $PV=mRT\frac{dV}{d\theta}$, rate of heat transfer by $h_c=A(T-T_w)$ and the internal energy can be related to specific heat through the relationship $\frac{du}{d\theta}=C_V\frac{dT}{d\theta}$

Upon simplification we get equation (2) as

$$\frac{dT}{d\theta}=\frac{1}{mC_V}\frac{dQ_r}{d\theta}-\frac{h_cA(T-Tw)}{mC_V}-\frac{RT}{C_VV}\frac{dV}{d\theta} \tag{3}$$

Solving above equation by Range-kutta fourth order algorithm, the temperature at various crank angles during combustion can be calculated.

c) Cylinder volume at any crank angle

The slider crank angle formula is used to find the cylinder volume at any crank angle [10]

$$V(\theta)=V_{disp}\left[\frac{r}{r-1}-\frac{1-\cos\theta}{2}+\frac{1}{2}\sqrt{\left(2\frac{L}{S}\right)^2-\sin^2\theta}\right] \tag{4}$$

d) Compression and Expansion strokes

The compression stroke starts from the moment the inlet valve closes (IVC) to the moment the fuel injection starts. The expansion stroke starts from the moment combustion ends to the moment the exhaust valve opens (EVO).During these processes the temperature and pressure at each step are calculated using ideal gas equation and an isentropic process [15].

e) Combustion Process

$$\frac{dQ_r}{d\theta}=6.908\frac{Q_p}{\theta_p}m_p\left(\frac{\theta}{\theta_p}\right)^{m_p-1}\exp\left[-6.908\left(\frac{\theta}{\theta_p}\right)^{m_p}\right]+6.908\frac{Q_d}{\theta_d}m_d\left(\frac{\theta}{\theta_d}\right)^{m_d-1}\exp\left[-6.908\left(\frac{\theta}{\theta_d}\right)^{m_d}\right] \tag{5}$$

The parameters θ_p and θ_d represent the duration of the premixed and diffusion combustion phases. Also, Q_p and Q_d represent the integrated energy release for premixed and diffusion phases respectively. Shape factors m_p and m_d for premixed and diffuse phase of combustion have to be such that the simulated heat release profile matches closely with experimental data. These shape factors are obtained by fitting wiebe function to experimental mass fraction burned profile using least square method. Prior knowledge of actual overall equivalence ratio is necessary because the fuel/air equivalence ratio depends on the amount of fuel injected inside the cylinder, from which the mass of fuel admitted can be calculated [18]. The amount of heat released in premixed mode is 40% of the total heat released per cycle is assumed.

f) Heat transfer

The convective heat transfer between gases and cylinder wall is considerable and hence it directly affects the engine performance. The convection heat transfer in kJ/degree crank angle is given by

$$\frac{dQ_h}{d\theta} = h_c A(T - T_w) \qquad (11)$$

Where Heat transfer coefficient due to convection (h_c) is given by Hohenberg equation [19].

$$h_c = \frac{130P^{0.8}(C_m + 1.48)^{0.8}}{V^{0.06}T^{0.4}} \qquad (12)$$

g) Ignition delay

An empirical formula, developed by Hardenberg and Hase [20] is used for predicting Ignition delay in crank angle degrees.

$$ID = (0.36 + 0.22C_m)\exp\left[E_A\left(\frac{1}{RT} - \frac{1}{17{,}190}\right)\left(\frac{21.2}{P - 12.4}\right)^{0.63}\right] \qquad (13)$$

Where ID = ignition delay period.
E_A is apparent activation energy

h) Gas properties calculation

A hydrocarbon fuel can be represented by $C_xH_yO_z$. The required amount of oxygen Y_{cc} for combustion per mole of fuel is given by:

$$Y_{cc} = m_c + 0.25m_h - 0.5m_o \qquad (14)$$

The minimum amount of oxygen required $(Y_{\min})$ for combustion per mole of fuel is

$$Y_{\min} = Y_{cc} - 0.5m_c$$

The gaseous mixture properties like internal energy (U), enthalpy (H) specific heats at constant pressure (C_P) and constant volume (c_v) depend on the chemical composition of the reactant mixture, pressure, temperature and combustion process and can be calculated using following equations.

$$U(T) = A + (B - R)*T + C*\ln(T) \qquad (15)$$

$$H(T) = A + B*T + C*\ln(T) \qquad (16)$$

$$C_p(T) = B + \frac{C}{T} \qquad (17)$$

$$C_V(T) = (B - R) + \frac{C}{T} \qquad (18)$$

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

i) Friction losses

Total friction loss calculated by the equation [21].

$$FP = C + 1.44\frac{C_m * 1000}{B} + 0.4(C_m)^2 \qquad (19)$$

Where *FP* is total friction power loss and *C* is a constant, which depends on the engine type, C = 75 kPa for direct injection engine.

III. Methodology

a) Simulation

A thermodynamic model based on the First law of thermodynamics has been developed. The molecular formula of diesel fuel is taken as $C_{10}H_{22}$ and biodiesel is approximated as $C_{19}H_{34}O_2$. A computer program has been developed using MATLAB software for numerical solution of the equations used in the thermodynamic model described in Section 2. This computes pressure, temperature, brake thermal efficiency, brake specific fuel consumption and net heat release rate etc, for the fuels considered for analysis. Fuels considered for analysis are namely B20, B60, and B100, 20%, 60%, and 100% POME with petroleum diesel respectively.

b) Experimental

A stationary single cylinder, 4 stroke, water cooled diesel engine developing 5.2 KW at 1500 rpm is used for investigation. The technical specifications of the engine are given in Table 1. The fuel properties are determined using standard procedure and tabulated in table 2. The cylinder pressure data is recorded by using piezoelectric transducer for 80 cycles. The average of data for 80 cycles is computed to evaluate mass fraction

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burned profile and combustion duration within the framework of first law of thermodynamics.

Table 1 : Specifications of Engine

Sl.No	Parameter	Specification
1	Type	Four stroke direct injection single cylinder diesel engine
2	Software used	Engine soft
3	Injector opening pressure	200 bar
4	Rated power	5.2 KW @1500 rpm
5	Cylinder diameter	87.5 mm
6	Stroke	110 mm
7	Compression ratio	17.5:1
8	Injection timing	23 degree before TDC

Table 2 : Properties of Diesel and POME

Properties	Diesel(B0)	POME(B100)
Viscosity in cst(at 30°C)	4.25	4.7
Flash point(°C)	79	190
Fire point(°C)	85	210
Carbon residue (%)	0.1	0.64
Calorific value(kj/kg)	42700	36000
Specific gravity(at 25°C)	0.830	0.880

IV. Results and Discussion

a) Effect of compression ratio on

i. Peak pressure

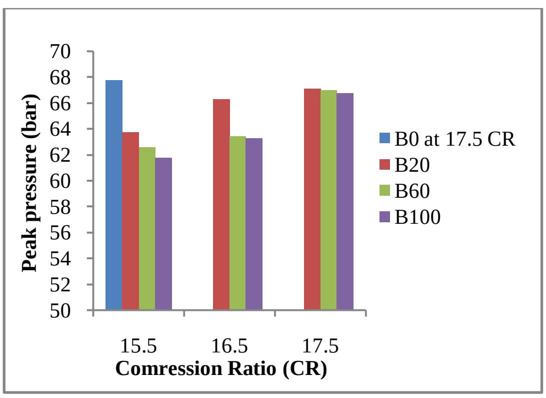

Figure 1 : Variation of Peak pressure with test fuels

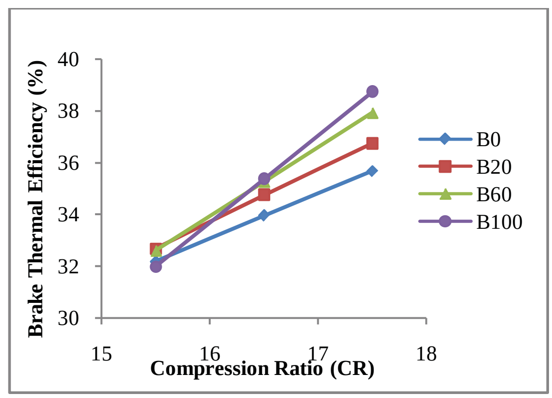

Figure 2 : Variation of Brake thermal efficiency at different Compression Ratio with test fuels at different Compression Ratio

Figure 1. shows the variation of peak pressure with various test fuels at different compression ratios. With increase in compression ratio, the peak pressure is increased for all test fuels. At every compression ratio, the peak pressure decreases with increase in proportion of biodiesel in the blend and also found that the peak pressures of all test fuels are less in comparison with that of diesel.

Increase in compression ratio enhances the pressure and temperature of air–fuel mixture in compression stroke results in increased peak pressure. Increase in proportion of biodiesel in blend burns more fuel during diffusion phase of combustion and lower calorific value of blend causes in decrease of peak pressure.

ii. Brake thermal efficiency

Figure 2. Shows the variation of brake thermal efficiency for various test fuels at different compression ratios. It is observed that brake thermal efficiency for all the test fuel is increased with increase in compression ratio. From the results it is also observed that the brake thermal efficiency at every compression ratio is increased with increase in proportion of biodiesel in the blend. This is due to the presence of oxygen molecule in the biodiesel which enhances combustion phenomenon. The brake thermal efficiency of test fuels is lower at compression ratio of 15.5:1 and 16.5:1 and higher at compression ratio of 17.5:1 in comparison with

iii. *Net Heat Release Rate*

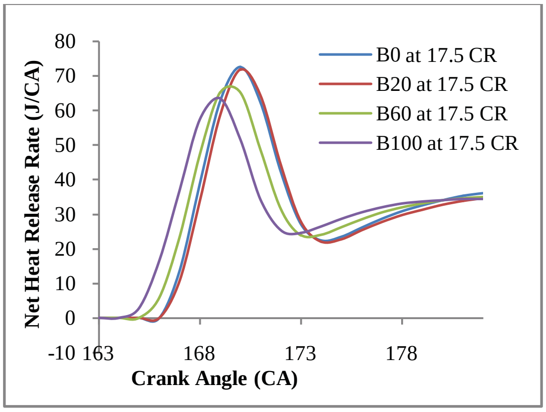

Figure 3 (i) : Variation of Net heat release rate with test fuels at 17.5 Compression Ratio

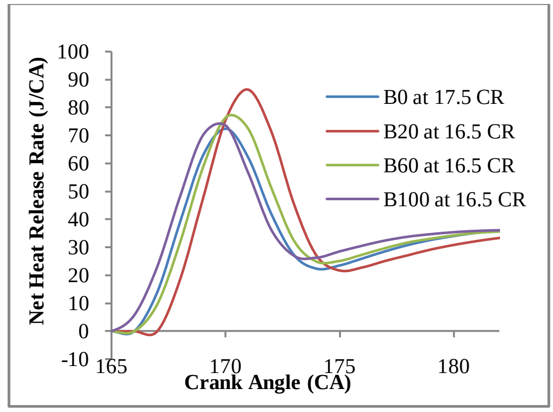

Figure 3 (ii) : Variation of Net heat release rate with test fuels at 16.5 Compression Ratio

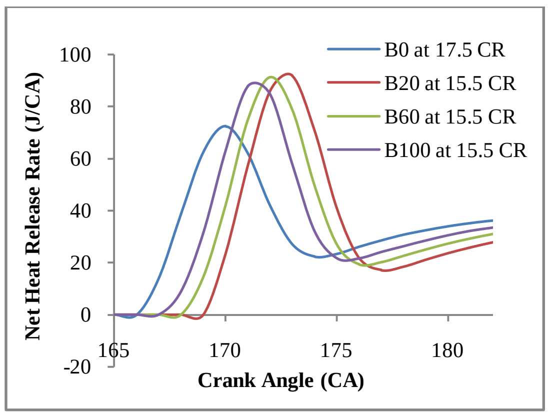

Figure 3 (iii) : Variation of Net heat release rate with test fuels at 15.5 Compression Ratio

Figures 3(i. ii & iii). Shows the variation of net heat release rate for various test fuels at different compression ratios. From the results it is observed that decrease in compression ratio increases heat release in premixed phase; however occurrence of maximum heat release moved away from TDC. This is because decrease in compression ratio increases the ignition delay period, which causes more fuel to burn late in the expansion stroke. Same trend is observed for all the test fuels. Increase in proportion of biodiesel increases the cetane number of blend, decreasing the delay period. Decrease in delay period burns less amount of fuel in premixed phase, hence decrease in net heat release rate is observed at every compression ratio.

b) *Effect of load on*

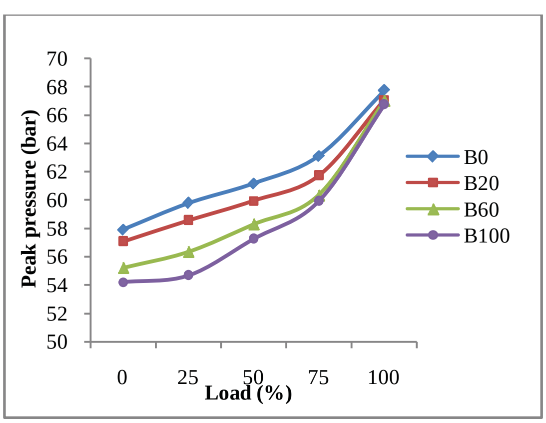

Figure 4 : Variation of Peak pressure with test fuels at different load

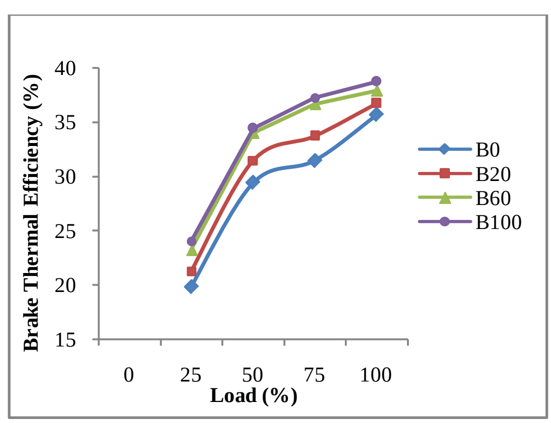

Figure 5 : Variation of Brake thermal efficiency with test fuels at different load

Figures 4 & 5.Shows the Variation of peak pressure and brake thermal efficiency with test fuels at different load. From the predicted results it is observed that increase in load increases the peak pressure and brake thermal efficiency. Same trend has been observed with all test fuels.

V. Model Validation

With the help of developed model theoretical results are predicted for brake thermal efficiency and peak pressure for all test fuels. The same are compared with that of experimental results. The figures below highlight the features. Predicted brake thermal efficiency and peak pressure at full load when engine is fuelled with B0, B20, B60 and B100 are compared with experimental results are found in closer approximation.

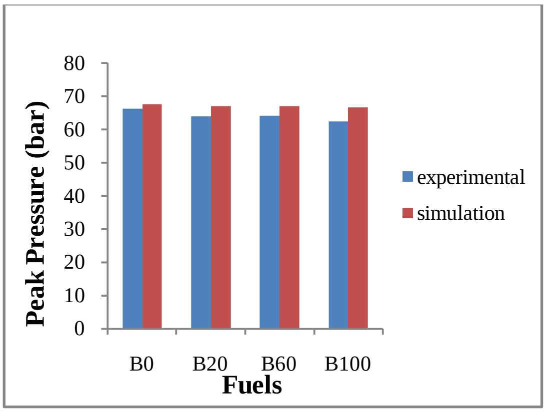

Figure 6 : Peak Pressure at full load

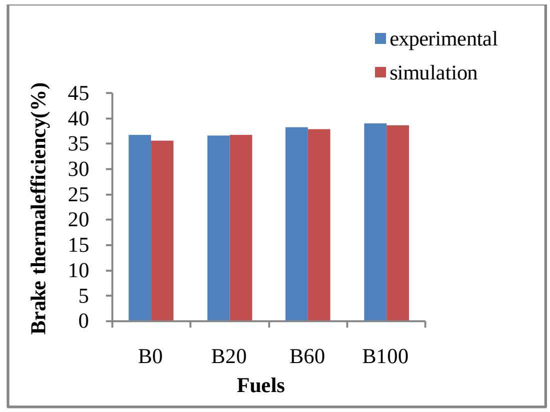

Figure 7 : Brake thermal efficiency at full load

VI. Conclusions

The thermodynamic model developed is used for analyzing the performance characteristics of the compression ignition engine. The modeling results showed that, with increase in compression ratio peak pressure and brake thermal efficiency are increased for all test fuels. At every compression ratio, increase in proportion of biodiesel in the blend decreased peak pressure and increased brake thermal efficiency. This model predicted the engine performance characteristics in closer approximation to that of experimental results. Hence, it is concluded that this model can be used for the prediction of the performance characteristics of the compression ignition engine fueled by any type of hydrocarbon fuel.

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