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Computer Modeling of CI Engine Performance for Parametric Study of Injection Timing Fuelled with Palm Oil Methyl Ester (POME) and its Blends With Diesel.

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Abstract

Parametric study of fuel injection timing has been conducted for the analysis of CI engine performance using developed computer simulation model. A zero dimensional single zone thermodynamic model is developed for compression ignition engine cycle simulation. The model is also used for predicting the engine performance at different loads with various test fuels. 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 fuel injection timing on peak pressure, net heat release rate, brake specific energy consumption 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.

Key words: Biodiesel, compression ignition engine, double vibe function, simulation**.**

1. 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. Parametric study has been conducted for the analysis of CI engine performance fuelled with POME and its blends with diesel.

2. Description Of Mathematical Modeling

2.1 List of symbols

r = compression ratio.

L = length of connecting rod (mm).

B = bore diameter (mm).

V_{disp} = displacement volume (m3).

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 (m3).

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/m2.K).

A = interior surface area of cylinder (m2).

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 (m3/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.

2.2 Energy balance equation

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

$$m\frac{du}{d\theta} = \frac{dQ_r}{d\theta} - \frac{dw}{d\theta} \qquad (1)$$

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

$$m\frac{du}{d\theta} = \frac{dQ_r}{d\theta} - \frac{dQ_h}{d\theta} - \frac{dw}{d\theta} \qquad (2)$$

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

$$\frac{dT}{d\theta} = \frac{1}{mC_V}\frac{dQ_r}{d\theta} - \frac{h_c A(T - Tw)}{mC_V} - \frac{RT}{C_V V}\frac{dV}{d\theta} \qquad (3)$$

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

2.3 Cylinder volume at any crank angle

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

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

2.4 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 [16].

2.5 Combustion Process

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

$$6.908\frac{Q_d}{\theta_d}m_d\left(\frac{\theta}{\theta_d}\right)^{m_d-1}\exp\left[-6.908\left(\frac{\theta}{\theta_d}\right)^{m_d}\right] \qquad (5)$$

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 [17]. The amount of heat released in premixed mode is 40% of the total heat released per cycle is assumed.

2.6 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

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

Where Heat transfer coefficient due to convection (h_c) is given by Hohenberg equation [18].

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

2.7 Ignition delay

An empirical formula, developed by Hardenberg and Hase [19] is used for predicting Ignition delay in crank angle degrees.

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

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

2.8 Gas properties calculation

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

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

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

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

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.

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

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

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

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

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

2.9. Friction losses

Total friction loss calculated by the equation [20].

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

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.

3. Methodology

3.1. 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.

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

Table1. 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	5.2KW @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(at25°C)	0.830	0.880

4. Results and Discussion

4.1 Effect of fuel injection timing on

4.1a) Peak pressure.

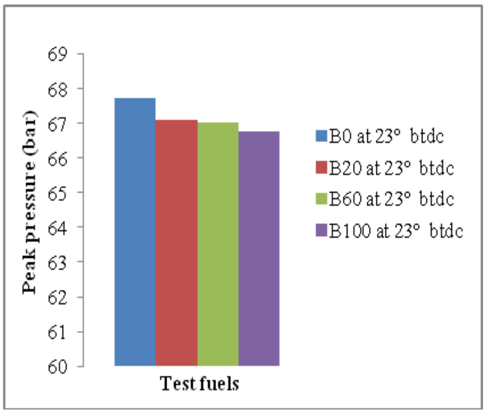

Figure 1(i). Variation of peak pressure of various test fuels with fuel injection at 23° btdc

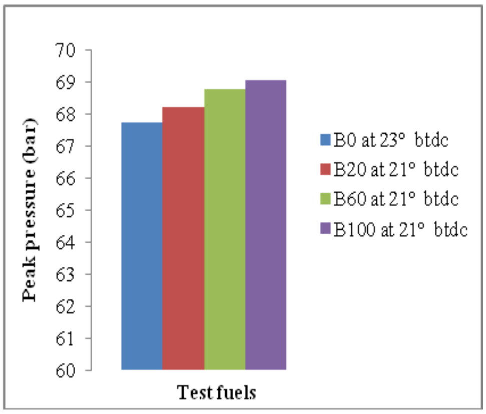

Figure 1(ii). Variation of peak pressure of various test fuels with fuel injection at 21° btdc

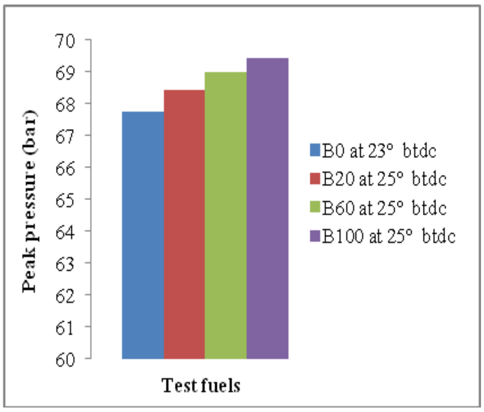

Figure 1 (iii). Variation of peak pressure of Various test fuels with fuel injection at 25° btdc

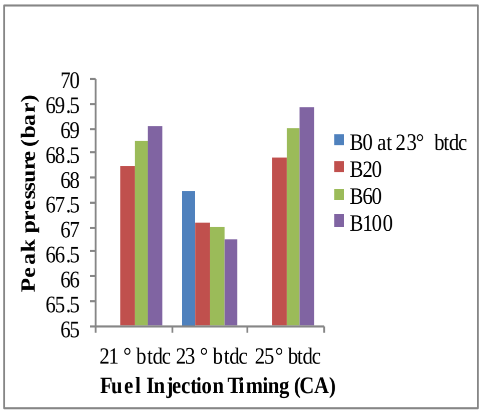

Figure 1 (iv). Variation of Peak pressure of various test fuels at different fuel injection timings

Figures 1(i,ii,iii & iv). Shows the variation of peak pressure with various test fuels at different fuel injection timing. Fuel injection timing is retarded by 2 degrees and advanced by 2 degrees.

The results are predicted with fuel injection at 21 degree before TDC, 23 degree before TDC and 25 degree before TDC for all test fuels. With retarded injection timing peak pressure is lowered for all test fuels. This is due to the fact that at retarded injection timing ignition delay gets shortened and less fuel available for premixed combustion. With advanced injection timing peak pressure is increased for all test fuels. This is due to the fact that at advanced injection timing ignition delay period increases and more fuel available for premixed combustion.

4.1b) Brake thermal efficiency.

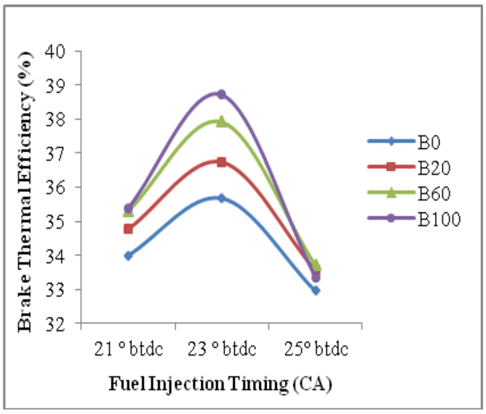

Figure 2. Variation of brake thermal efficiency with test fuels at different fuel injection timing

Figures.2. Shows the variation of brake thermal efficiency with various test fuels at different fuel injection timing. Advancement and retarding of fuel injection timing reduces the brake thermal efficiency, however more reduction in brake thermal efficiency is observed with advancement of fuel injection timing than retarding for all the test fuels.

4.1c) Brake specific fuel consumption.

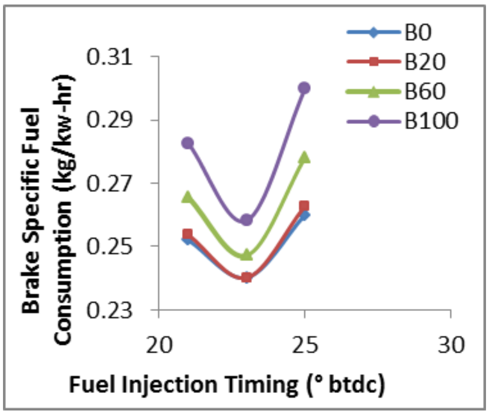

Figure 3. Variation of brake specific fuel Consumption with test fuels at different fuel injection timings

Figures.3. Shows the variation of brake specific fuel consumption with various test fuels at different fuel injection timing. With both advancing and retarding of fuel injection timing increase in brake specific fuel consumption for all test fuels is observed.

4.1 d) Net Heat Release Rate.

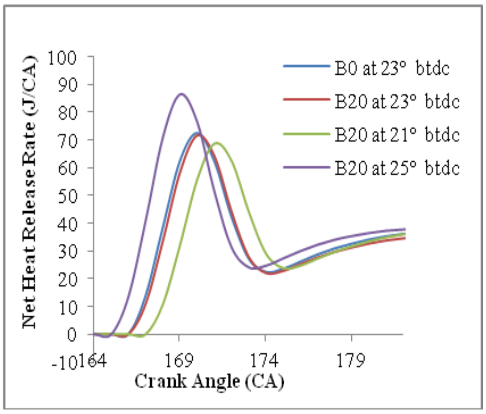

Figure 4(i). Variation of net heat release rate with B20 at different fuel injection timings

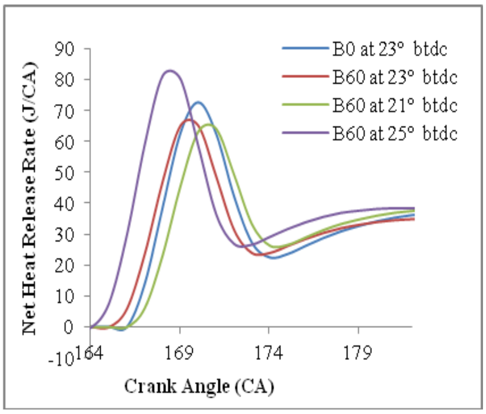

Figure 4(ii). Variation of net heat release rate with B60 at different fuel injection timings

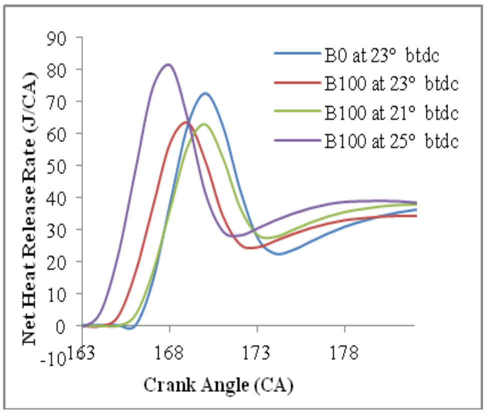

Figure 4(iii). Variation of net heat release rate with B100 at different fuel injection timings

Figures 4(i,ii & iii). Shows the variation of net heat release rate with various test fuels at different fuel injection timing. Retarding the fuel injection timing has resulted in reduced ignition delay; less fuel gets burned in premixed phase. It is also observed that delay in start of combustion, maximum het release rate moved away from conventional injection timing i.e 23° btdc. Advancement of fuel injection timing has resulted in increased ignition delay more fuel burning in premixed phase and early start of combustion than conventional injection timing i.e 23° btdc.

Fuel injection timing is retarded by 2 degrees and advanced by 2 degrees. The results are predicted with fuel injection at 21 degree before TDC, 23 degree before TDC and 25 degree before TDC for all test fuels. With retarded injection timing peak pressure is lowered for all test fuels. This is due to the fact that at retarded injection timing ignition delay gets shortened and less fuel available for premixed combustion.
With advanced injection timing peak pressure is increased for all test fuels. This is due to the fact that at advanced injection timing ignition delay period increases and more fuel available for premixed combustion.

4.2 Effect of load on

Peak pressure and Brake thermal efficiency.

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.

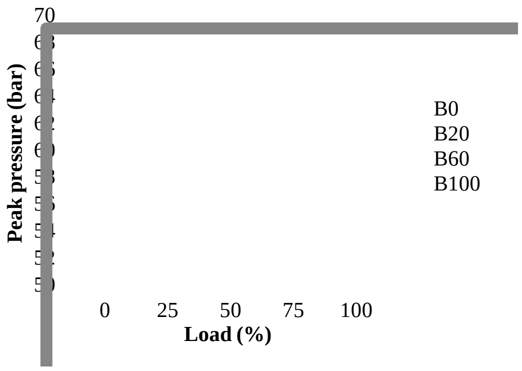

Figure.4. Variation of Peak pressure with test fuels

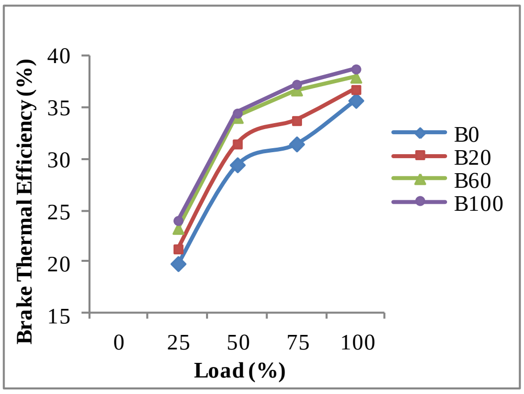

Figure.5. Variation of Brake thermal efficiency with test fuels at different load

5. Model Validation

With the help of developed model theoretical results are predicted for peak pressure and brake thermal efficiency 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 that of experimental results and are found in closer approximation.

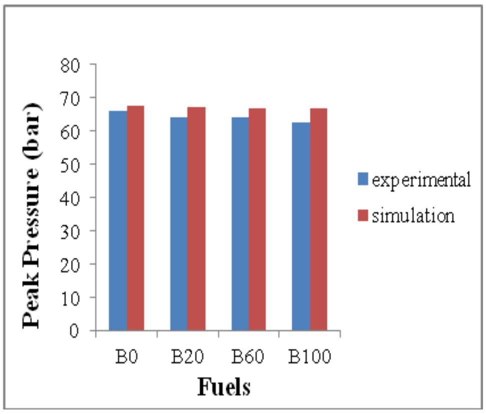

Figure.6. Peak Pressure at full load

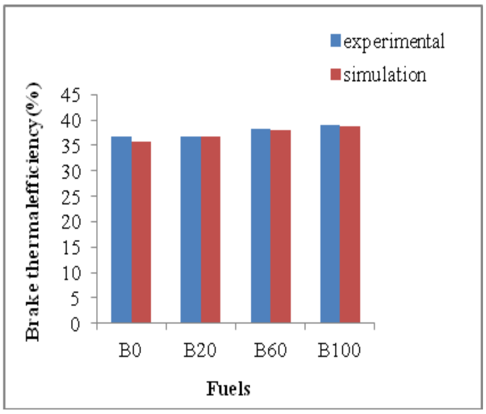

Figure.7. Brake thermal efficiency at full load

6. Conclusions

The thermodynamic model developed is used for analyzing the performance characteristics of the compression ignition engine. The modeling results showed that, both advancement and retarding of fuel injection timing reduces the 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 engine performance and as a tool for parametric study of fuel injection timing.

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