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Thermodynamic Modelling for Performance Analysis of Compression Ignition Engine Fuelled With Biodiesel and its Blends With Diesel

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Abstract: ***In this work, computer simulation framework for compression ignition engine cycle simulation is developed and engine performance is predicted. Double wiebe's function is used to model the rate of heat release due to combustion to predict heat released during premixed as well as diffusive phase of combustion. Effect of convective heat transfer and variation in specific heat of test fuels are also considered during development of model. Suitable correlations are established between adjustable parameters of wiebe's function, relative air-fuel ratio and engine operating conditions, such that the simulated heat release profile matches closely with experimental results. The simulation model is used to analyze the performance, combustion and emission characteristics of single cylinder 3.5 kW rated power diesel engine fuelled with Diesel (D0), Palm Oil Methyl Ester (POME) and POME-diesel blends. The model validation is done by comparing the predicted parameters like brake thermal efficiency and in-cylinder pressure with experimental results and are found in closer approximation. The model is also used to predict net heat release rate, exhaust gas temperature, NO_x and soot.***

Index Terms: ***biodiesel, C. I. engine, POME, simulation.***

I. Introduction

The demand for energy around the world is increasing due to rapid industrialization and increase in vehicular population for transportation, specifically the demand for petroleum fuels. Use of petroleum at present rate may deplete it in near future. Among the petroleum products demand for diesel is very high due to wide use of diesel engines for industrial prime movers and transportation. The high price of the diesel and high pollution levels from diesel engines have caused for the search of renewable and alternative fuels to diesel. Use of bio fuels such as vegetable oil, ethanol, etc. could reduce the dependency on petroleum products and the pollution problem. Researchers [1-3] have studied the performance of compression ignition (CI) engine fuelled with straight vegetable oil (SVO). Higher specific fuel consumption, more carbon monoxide emissions were observed, also engine coke up has been observed for long run of engine due to higher viscosity of straight vegetable oil.

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Reduction in viscosity of straight vegetable oil may improve the engine performance. Viscosity of SVO can be reduced by converting it into biodiesel, blending/emulsifying with ethanol, pre-heating SVO etc. Vegetable oil esters have physical properties very close to diesel which may result in engine performance very close to diesel. [4] Have compared the performance of diesel engine using methyl esters of honge, jatropha and sesame oil with diesel. Slight reduction in thermal efficiency, increase in smoke emissions and increased ignition delay & combustion duration with esters compared to diesel were observed. [5] Used the biodiesel of frying palm oil and its blends with diesel as fuel in a four cylinder, naturally aspirated indirect injection diesel engine. The results show slight increase in brake specific fuel consumption with increase in 5% of bio diesel in the blend and slight drop in the engine power. [6] Has investigated the CI engine of 5.2 kW rated power fuelled with palm oil methyl ester (POME) and diesel and found significant improvement in engine performance with POME. [7] Developed a theoretical model for analysis of performance characteristics of CI engine fueled by biodiesel and its blends. The effects of relative air-fuel ratio and compression ratio on the engine performance for different fuels are also analyzed using this model. Ignition delay was modeled with wolfer's relation. A rate of heat release was modeled using single wiebe's function. Rubber seed oil biodiesel was used to run the engine. This model showed increase in peak pressure, peak temperature and brake thermal efficiency with increase in compression ratio and decrease with increase in relative air-fuel ratio. [8] Developed simulation model to estimate the cylinder pressure, heat release rate, brake thermal efficiency, brake specific fuel consumption and engine out emissions. To model combustion process single wiebe's function is used. Pungam methyl ester and its blends with diesel are used as test fuel. Above literature survey shows that efforts are being made on use of biodiesel and its blends with diesel in CI engine to address the issues related to the performance, combustion and emission characteristics experimentally and theoretically. Theoretical model developed using single Wiebe function which does not predict the heat release rate during pre-mixed and diffusive phase of combustion separately. Experimental analysis requires costly logistic support; hence a simulation model could be developed to predict the engine performance.

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Computer simulation [9] serves as a tool for a better understanding of effect of the engine operating variables involved on engine performance, combustion and emission characteristics thereby reducing cost and time. Hence, a zero dimensional, single zone thermodynamic model for compression ignition engine cycle simulation is developed. Double wiebe's function is used to model the rate of heat release due to combustion to predict rate of heat release during premixed as well as diffusive phase of combustion. The model is used to predict and analyze the engine performance, combustion and emission characteristics fuelled with diesel, biodiesel and its blends with diesel.

II. MATHEMATICAL MODELING

A. Energy Balance Equation

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

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

Where $\frac{du}{d\theta}$ is rate of change of internal energy, $\frac{dQ_r}{d\theta}$ is rate of heat released and $\frac{dw}{d\theta}$ is rate of work done.

Upon simplification by considering ideal gas law and rate of heat transfer we get equation (1) as

$$\frac{dT}{d\theta} = \frac{1}{mc_v}\frac{dQ_r}{d\theta} - \frac{h_c A(T - T_w)}{mc_v} - \frac{RT}{c_v V}\frac{dV}{d\theta} \qquad (2)$$

B. Cylinder Volume at Any Crank Angle

The slider crank angle formula is used to find the cylinder volume at any crank angle [9].

$$V(\theta) = V_{disp}\left[\frac{r}{r-1} - \frac{1-\cos\theta}{2} + \frac{1}{2}\sqrt{\left(2\frac{L}{S}\right)^2 - \sin^2\theta}\right] \qquad (3)$$

Where r = compression ratio, L=length of connecting rod and S= stroke length.

C. Combustion Process

$$\frac{dQ_r}{d\theta} = 6.908\frac{Q_p}{\theta_p}m_p\left(\frac{\theta}{\theta_p}\right)^{m_p-1} exp\left[-6.908\left(\frac{\theta}{\theta_p}\right)^{m_p}\right] + 6.908\frac{Q_d}{\theta_d}m_d\left(\frac{\theta}{\theta_d}\right)^{m_d-1} exp\left[-6.908\left(\frac{\theta}{\theta_d}\right)^{m_d}\right] \qquad (4)$$

The heat release rate is computed with equation (4). The parameters θ_p & θ_d represent the duration, m_p & m_d are shape factors and Q_p and Q_d represent the integrated energy release for premixed and diffusion combustion phases respectively. The amount of heat released in premixed phase is 50% of heat release due to the amount of fuel injected during ignition delay period is assumed.

D. Ignition Delay

An empirical formula, developed by Hardenberg and Hase [11] is used for predicting Ignition delay in crank angle degrees.

$$ID = (0.36 + 0.22C_m)exp\left[E_A\left(\frac{1}{RT} - \frac{1}{17190}\right)\left(\frac{21.2}{P-12.4}\right)^{0.63}\right] \qquad (5)$$

Where ID = ignition delay period.

E_A is apparent activation energy

E. Gas Properties Calculation

The gaseous mixture properties like internal energy, enthalpy, specific heats at constant pressure and constant volume are obtained based on the chemical composition of the reactant mixture, pressure, and temperature [9].

F. Friction Losses

Total friction loss calculated by the equation (6) given below [12].

$$FP = C + 1.44\frac{C_m * 1000}{B} + 0.4(C_m)^2 \qquad (6)$$

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.

G. Oxides of Nitrogen (NO_x)

An oxide of nitrogen formation has been predicted using procedure explained by Turns [13]. The following equation is used for computation of nitric oxide

$$\frac{d[NO]}{dt} = 2k_{1f}\left(\frac{k_p P^o}{R_u T}\right)^{0.5}[N_2][O_2]^{0.5} \qquad (7)$$

Where

$$k_{1f} = 1.82 * 10^{14} e^{\left[\frac{-38370}{T(k)}\right]}$$

$$k_p P^o = e^{\left(\frac{-\Delta G^o T}{R_u T}\right)}$$

$[N_2]$ and $[O_2]$ are equilibrium nitrogen and oxygen concentrations in moles.

$$[N]_2 = 0.21 * \frac{P}{R_u T}$$

$$[O]_2 = 0.79 * \frac{P}{R_u T}$$

H. Soot

The following equation has been used for prediction of soot [14].

$$\frac{dm_{soot}}{dt} = C_{BS} * \emptyset * m_f * P^{0.5} * e^{\left(\frac{-E_{sf}}{R_a T}\right)} \qquad (8)$$

Where C_{BS} is constant

E_{sf} is the activation energy of the soot formation reaction.

III. METHODOLOGY

A. Simulation

A thermodynamic model has been developed using First law of thermodynamics. The molecular formula of diesel fuel is taken as $C_{10}H_{22}$ and for POME (biodiesel) is approximated as $C_{19}H_{34}O_2$.

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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 exhaust gas temperature etc, for the fuels considered for analysis. During investigation, theoretical results are predicted for various blends of POME with diesel namely D0, P20, P40, P60, P80 and P100 with 0%, 20%, 40%, 60%, 80% and 100% POME with petroleum diesel respectively.

B. Experimental

A TV-1, stationary, single cylinder, water cooled variable compression ratio diesel engine developing 3.5 kW at 1500 rpm is used for this investigation. It is coupled to a water cooled eddy current dynamometer for loading. It is equipped with thermocouples to measure temperature of coolant, exhaust gas at inlet and outlet of calorimeter and water temperature. A manometer is used to measure air flow rate and burette is used to measure fuel flow. The cylinder pressure data is recorded by using piezoelectric transducer. The technical specifications of the engine and the fuel properties are given in Table 1 and 2 respectively.

Table1. Specifications of Engine

Sl.No	Parameter	Specification
1	Type	Four stroke direct injection single cylinder VCR diesel engine
2	Software used	Engine soft
3	Injector opening pressure	200 bar
4	Rated power	3.5 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(D0)	POME(P100)
Viscosity in cst(at 30°C)	4.25	4.7
Flash point(°C)	79	170
Fire point(°C)	85	200
Carbon residue (%)	0.1	0.62
Calorific value(kj/kg)	42000	36000
Specific gravity(at25°C)	0.830	0.870

IV. RESULTS AND DISCUSSION

A. Performance parameters

Figure 1 shows the variation of brake thermal efficiency (BTE) at various loads for all test fuels. Improvement in brake thermal efficiency is observed with increase in proportion of POME in the blend. This is due to the presence of oxygen molecule in the biodiesel which enhances combustion phenomenon. At full load, the BTE for P100 is 26.97 % and for D0 is 24.98%

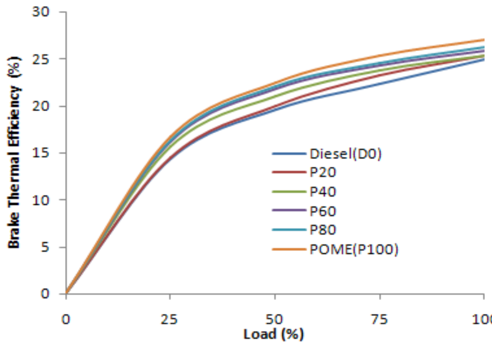

Figure.1 variation of BTE with test fuels at different loads.

Figure 2 & 3 show the variation of brake specific fuel consumption (BSFC) and brake specific energy consumption (BSEC) for all test fuels at various loads respectively. With increase in load, the BSFC decreased for all the test fuels. Increase of percentage of POME in the blend reduces the calorific value of charge which resulted in more fuel consumption for developing same power. Hence increase in proportion of POME in the blend increased the brake specific fuel consumption. Decrease in BSEC is observed with increase in proportion of POME in the blend. This is due to the presence of inherent oxygen in biodiesel, better combustion, etc.

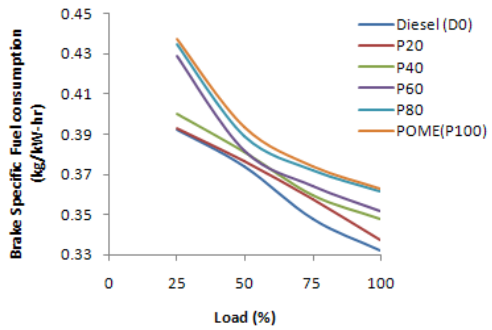

Figure.2 variation of BSFC with test fuels at different loads.

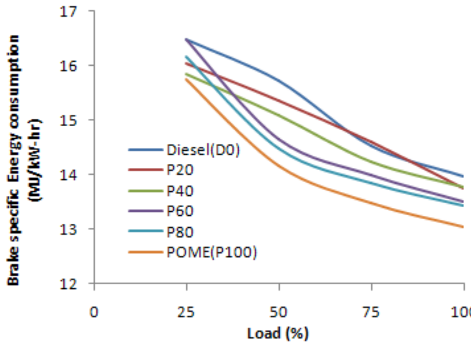

Figure.3 variation of BSEC with test fuels at different loads.

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Figure 4 shows the variation of exhaust gas temperature at different loads. From the results, it is observed that increase in load increased the exhaust gas temperature for all the test fuels. It is also observed that with increase in percentage of POME in blend exhaust gas temperature increased. The reason may be better combustion during diffusion phase and overall increase in operating temperature.

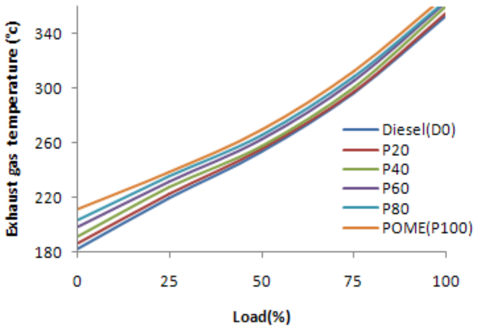

Figure.4 variation of exhaust gas temperature with test fuels at different loads.

B. Combustion Characteristics

Figure 5 & 6 shows the variation of peak pressure and net heat release rate at various loads for all test fuels. With increase in load, the peak pressure is increased for all the test fuels. Increase in proportion of biodiesel in the blend resulted in slight reduction of peak pressure. This may be due to poor atomization and lower calorific value of biodiesel compared to diesel. From the figure 6 it is observed that net heat release rate is lower for POME due to its lower calorific valve and higher cetane number which resulted in reduced delay period. Reduced delay period causes less amount heat release during premixed phase of combustion.

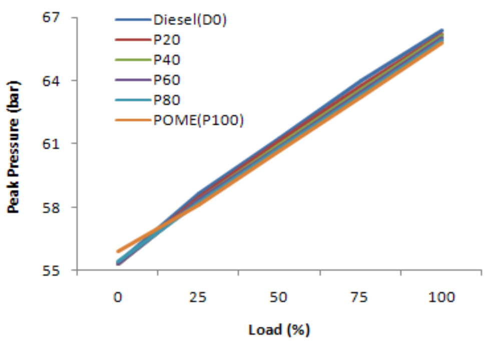

Figure.5 variation of peak pressure with test fuels at different loads.

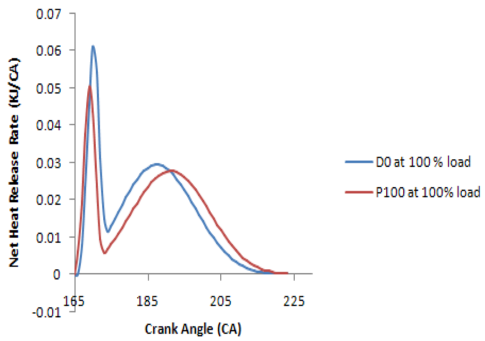

Figure.6 variation of net heat release rate with test fuels.

Figure 7 & 8 show the variation of NO_x and soot for all test fuels at various load conditions respectively. From the figure 7, it is observed that increase in percentage of POME in blend resulted in increase in NO_x emissions and decrease in soot emissions. Higher proportion of biodiesel in the blend enhances the combustion quality and results in an increased combustion temperature with higher NO_x emissions and decrease in soot due to presence of additional oxygen which makes mixture leaner.

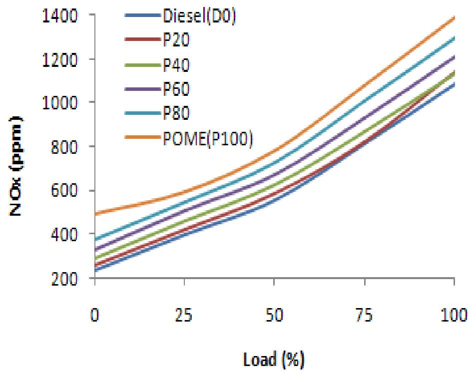

Figure.7 variation of NO_x with test fuels at different loads.

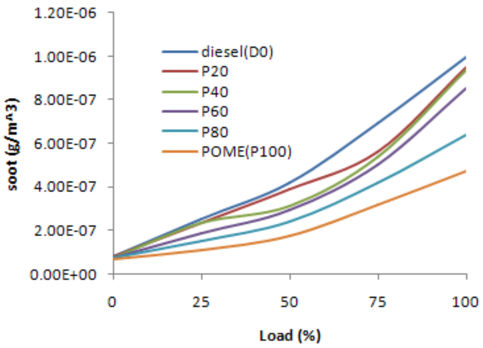

Figure.8 variation of soot with test fuels at different loads.

V. MODEL VALIDATION

Figs.9 & 10 shows brake thermal efficiency and in-cylinder pressure obtained using simulation model and experimentally for test fuels D0 and P100. It is observed that the theoretical results are in closer approximation with experimental results.

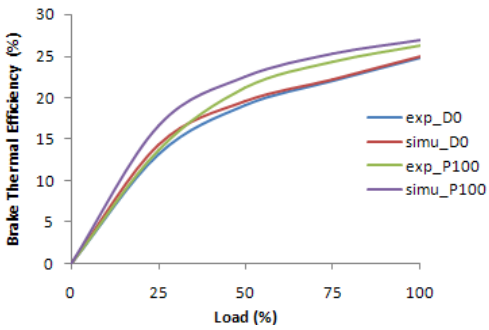

Figure.9 comparison of experimental and simulated brake thermal efficiency for D0 and P100.

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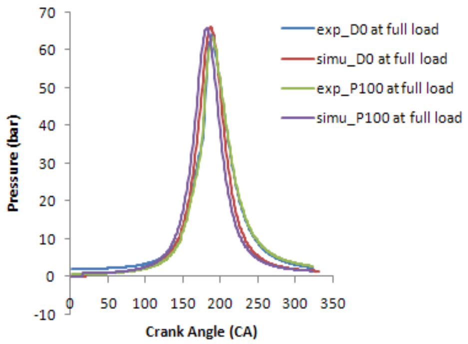

Figure.10 comparison of experimental and simulated in-cylinder pressure for D0 and P100.

VI. CONCLUSION

From above investigation following conclusions are drawn.

- Thermal efficiency with POME and its blends are slightly improved compared to diesel.
- The brake specific energy consumption with P100 is lower than diesel at all loads.
- Exhaust gas temperature is slightly increased with increase in POME proportion in the blend.
- With increase in percentage of POME in blend NO_x is increased and soot emissions are decreased as compared to diesel.
- The predicted BTE and peak pressure are in good agreement with that of experimental results.

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