MODELING OF COMPRESSION ENGINES USING BIODIESEL AS FUEL

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Abstract

A single zone C.I engine model has been developed in an in-house code that would be suitable for conventional diesel as well as for alternative fuel (biodiesels), to predict the pressure and temperature variation along crank angle, work done, mean effective pressure, power output and other engine parameters suitable for biodiesel. The model has been validated with the experimental results. Performance of the diesel engine depends on injection timing, delay period as well as the duration to get complete combustion are presented and discussed. The indicated and brake work done obtained are 13.20 kJ/cycle and 10.52 kJ/cycle respectively for B100 and the indicated and brake specific fuel consumption obtained are 0.289 kg/kWh and 0.362 kg/kWh.

***Key Words**: Biodiesel, Biodiesel engine, engine model.*

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1. INTRODUCTION

Most of the developments in automotive engines were to improve the power and fuel economy as well as to reduce the environmental pollutant for conventional fuel. But now there is a call for stringent environmental issue and fuel crisis in world wide. Composition and quality of conventional fuel is not widely varied since it is obtained from petroleum source after refinement. But it is frequently for alternative fuels which are obtained from different sources and processes. So there is challenge to adopt adequate development of the engine, which is suitable for alternative fuels which may be varied with its composition as well as heating value. The development of I.C. engine was not an easy task as its performance is depended on number of factors such as fuel compositions, heating value, combustion characteristics as well engine design and other features. The built and test method of the engine is not suitable due to its high cost and time consuming. So many scientists and engineers adopted another methodology which is the modeling approach based on the thermodynamics, fluid flow and heat transfer to predict the engine performance. Actual combustion phenomena are very much complex but it can be simplified by taking some assumptions The use of alternative fuels considerably decreases harmful exhaust emissions as well as ozone-producing emissions. Alternative fuels can be less expensive to use not just in terms of the fuel itself but also in terms of a longer service life. Lot of research have been done on the performance on the engine using alternative fuel [1-9]. Ramadhas et al. [1] developed a thermodynamic model to analyze the performance characteristics of the CI engine fueled by biodiesel and its blends and concluded that with the increase in compression ratio peak pressure, peak temperature and brake thermal efficiency are increased. Gogoi and Baruah [2] developed a single zone combustion model to predict the performance of diesel engine. The effect of engine speed and compression ratio on brake power and brake thermal efficiency is analyzed through the model. Lesnik et al. [3] carried out an experiment and numerical analysis on a heavy-duty bus diesel engine using mineral diesel fuel, neat biodiesel fuel and concluded that a reduction in engine power and torque when increasing the percentage of biodiesel fuel in the fuel blends due to lower calorific value of biodiesel fuel. Samanta [4] developed a single zone, thermodynamic model of internal combustion engine using biogas as chemical energy source and the effects of different parameters were investigated and optimization points were identified for the computational engine model. Abou Al-Sood et al. [5] developed a thermodynamic model for the evaluation of the performance of the four stroke DI diesel engine. The model was validated with experimental and predicted results for different engines operating at different conditions. Awad et al.[6] proposed a unique single zone combustion model for diesel fuel and biodiesel to predict the cylinder pressure for the better understanding of combustion characteristics. Lata and Misra [7] developed a mathematical model to predict pressure, net heat release rate, mean gas temperature, and brake thermal efficiency for dual fuel diesel engine operated on hydrogen, LPG and mixture of LPG and hydrogen as secondary fuels. Pyari et al. [8] developed a single zone thermodynamic model that takes into account the heat transfer to the chamber walls, the blow-by leakage, the fuel injection and engine deformations, along with the instantaneous change in gas properties. The model was validated in different operation points and showed a good capability for accurate predictions of engine performance. Ganapathy et al.[9] developed a two zone thermodynamic model for the analysis of Jatropha biodiesel engine to determine the optimum engine design and operating parameters. To maximize the performance of Jatropha biodiesel engine the signal to noise ratio (SNR) related to higher-the-better (HTB) quality characteristics has been used.

In the present work a single zone thermodynamic model has been developed which is built coupled with number of sub-models for conventional fuel and applied for biodiesel based on law of conservation of energy, energy release rate and equation of state. Among the sub models, the ignition delay model is same as that of Stone [10] and heat transfer model is not considered in the present work. In the present work, effect of suction and exhaust stroke have been incorporated in the intake and exhaust model. Also effects of friction losses at the different parts of the engines have been taken care of in the frictional loss model. Model predicted results have been validated with the experimental data as well as by the numerical data reported by Ramadhas et al.[1].

1.1 Model Descriptions

A thermodynamic model of the compression ignition engine (C.I. Engine) based on single zone [11] approach considering different sub-models and the variations of specific heats with local temperature [Abu Nada et al.[12]. Heat transfer loss from the engine has been considered as percentage heat leakage from the engine [13]. The following assumptions have been taken to develop for the actual cyclic operation of the C.I. Engines.

- The air inside the cylinder and the combustion product mixture are assumed to be homogenous and ideal gas.
- Inside the combustion chamber chemical equilibrium exist among the gaseous species.
- The specific heats of the gaseous species vary with the local temperature.
- The variations of the specific heats have been considered with the change in the crank rotation due to change in temperature and condition.
- The suction, compression, expansion and exhaust stroke of engine cycle are considered.
- Combustion chamber design has no effects in the engine model.
- At the compression stroke the initial pressure is assumed to be atmospheric pressure.
- The heat release model has been developed using ignition delay, pressure and temperature at ignition delay and equivalence ratio.
- Inlet valve and the exhaust valve open and close at BDC only.
- Heat loss from the considered as a percentage of fuel loss from the cycle.

The relationship between the cylinder volume and the crank angle is the function of compression ratio, bore, stroke and connecting rod length. Combustion model has been developed based on the conservation principle applied to a combustion chamber as the control volume. The relation from ideal gas law and its differential form,

$$PV = mR_gT \quad \text{and} \quad \frac{dV}{V} + \frac{dP}{P} = \frac{dT}{T} \tag{1}$$

The relation from conservation of energy can be expressed in differential form and applied to a control volume. The equation becomes for net heat release rate is

$$\frac{dQ_n}{d\theta} = mc_v\frac{dT}{d\theta} + P\frac{dV}{d\theta} \tag{2}$$

Rearranging equations (1 and 2)

$$\frac{dP}{d\theta} = \frac{k-1}{V}\frac{dQ_n}{d\theta} - \frac{kP}{V}\frac{dV}{d\theta} \tag{3}$$

The net heat term has both heat added and heat loss in case of combustion

$Q_n = Q_{in}x_b - Q_l$ Where, Q_{in} indicates total heat input and x_b as fraction of heat release,The heat loss can be obtained as,

$$\frac{dQ_l}{d\theta} = \frac{h_g(\theta)A_w(\theta)}{\omega}(T_w - T_g(\theta)) \tag{4}$$

According to Rakapoulos [14] the temperature of wall can be taken as constant. In the present work the ratio of temperature is taken as $T_w / T_1 = 1.2$

The mass fraction of fuel burned in an internal combustion engine can be expressed as a function of crank angle using the Wiebe function,

$$x_b(\theta) = 1 - \exp\{-a(\frac{\theta-\theta_s}{\Delta\theta})^n\}$$

(5)

Now, according to Stone [10]

$$x_b = \alpha f_1(\lambda) + (1-\alpha) f_2(\lambda) \tag{6}$$

α =pre-mixed diffusion combustion parameter for compression ignition engine,

$f_1(\lambda)$ =pre-mixed phase, $f_2(\lambda)$ =diffusion burning phase

Now, $\alpha = 1 - 0.875\frac{\phi^{0.350}}{\tau^{0.375}}$, ϕ =equivalence ratio.

Ignition delay is the time taken to start the combustion after the injection of the fuel. For the calculation of ignition delay many expressions were found in literature as a function of pressure, temperature, cetane number and equivalence ratio. The following empirical relation is used for the calculation of ignition delay [14].

$$\tau = \frac{3.52\exp(2100/T_{id})}{\left(p_{id}/100\right)^{1.022}} \tag{7}$$

τ =ignition delay, T_{id} =temperature at ignition delay= $T_{id} = T_1 r_c^{k-1}$, p_{id} =pressure at ignition delay $p_{id} = p_1 r_c^{k}$.

Here, k =adiabatic index= $\frac{c_p}{c_v}$

So, $$\frac{dx_b}{d\theta} = \frac{\alpha}{\Delta\theta}\frac{df_1}{d\lambda} + \frac{1-\alpha}{\Delta\theta}\frac{df_2}{d\lambda} \tag{8}$$

So, the heat release and mass burn fraction are related as

$$\frac{dQ_{net}}{d\theta} = Q_{in}\frac{dx_b}{d\theta} - loss \tag{9}$$

Net heat input (Q_{in}) can be expressed as

$$Q_{in} = V_f \eta_v Q_{LHV} \tag{10}$$

Where, V_f is the volume of fuel inside the combustion chamber, η_v is the volumetric efficiency and Q_{LHV} is the lower heating value of fuel.

The final relation of combustion chamber pressure with the crank angle is obtained as

$$\frac{dP}{d\theta} = \frac{k-1}{V}[Q_{in}\frac{dx_b}{d\theta} - \frac{h_g(\theta)A_w(\theta)}{\omega}(T_w - T_g(\theta))] - \frac{kP}{V}\frac{dV}{d\theta} \tag{11}$$

The above equation is solved by using explicit finite difference technique with second – order accuracy as [12].

The pressure variation during the intake and exhaust stroke can be obtained as [15]:

$$\frac{dP}{dt} = kP(\frac{1}{M}\frac{dM}{dt} - \frac{1}{V}\frac{dV}{dt}) \tag{12}$$

The equation (9) requires mass flow rate dM/dt, which is to be found from the equations of fluid mechanics. For the intake stroke, mass flow rate can be obtained as :

$$\frac{dM_{in}}{dt} = AP\sqrt{\frac{2k}{RT(k-1)}(\frac{P_{in}}{P})^{\frac{k-1}{k}}\left[(\frac{P_{in}}{P})^{\frac{k-1}{k}} - 1\right]}$$

[When (Pin/P)<PRcric] (13)

$$\frac{dM_{in}}{dt} = AP\sqrt{\frac{k}{RT}(\frac{2}{k+1})^{(k+1)/(k-1)}}$$

[(Pin/P)>PRcric] (14)

The critical pressure ratio that defines the two regimes is called PR_{crit} and depends only on the value of k of the gas.

$$PR_{crit} = (\frac{k+1}{2})^{\frac{k}{k-1}} \tag{15}$$

Where P_{in} the intake manifold pressure and P is is the cylinder pressure.

Similarly for the exhaust stroke there is a little change in the mass flow rate equation which is mentioned below.

$$\frac{dM_{ex}}{dt} = AP\sqrt{\frac{2k}{RT(k-1)}(\frac{P}{P_{ex}})^{\frac{k-1}{k}}\left[(\frac{P}{P_{ex}})^{\frac{k-1}{k}} - 1\right]}$$

[When (P/Pex)<PRcrit] (16)

$$\frac{dM_{ex}}{dt} = AP\sqrt{\frac{k}{RT}(\frac{2}{k+1})^{(k+1)/(k-1)}}$$

[When (P/Pex)>PRcrit] (17)

Where P_{ex} is the exhaust manifold pressure and P is the cylinder pressure.

Where valve area can be obtained as:

$$A = A_{in}(|\sin\theta|)^{1/3} \tag{18}$$

Where A_{in} is the wide-open valve area. In this model the value of A_{in} has been taken as 3.5 cm2 [15].

The mean effective losses of power due to friction in different moving parts are calculated by using the following empirical relations by Gogoi[12].

i) Mean effective pressure (MEP) lost due to friction in the piston and rings

$$FMEP1 = 12.85 * \frac{P_{sl}}{B*S} * \frac{100*Upe}{1000} \tag{19}$$

Where, P_{sl} is the piston skirt length (0.65 to 0.85 times of Bore) [13], B is the cylinder bore (m), S is the stroke length (m) and Upe is the mean piston speed (m/s).

ii) MEP lost in bearing friction

$$FMEP2 = 0.0564 * \frac{B}{S} * \frac{N}{1000} \tag{20}$$

Where, N is the engine speed in rpm.

iii) Mean effective pressure (MEP) absorbed in overcome friction due to the valve gear

$$FMEP3 = 0.226 * \left(30 - \frac{4N}{1000}\right) * \frac{G * Dvi^{1.75}}{B^2 * S} \tag{21}$$

Where G is the number of intake valves per cylinder and D_{vi} is the diameter of the intake valve.

iv) MEP lost in pumping

$$FMEP4 = 0.0275 * \left(\frac{N}{1000}\right)^{1.5} \tag{22}$$

v) MEP lost in friction due to the wall tension of rings

$$FMEP5 = \frac{10*0.377*S*n_{pr}}{B^2} \tag{23}$$

Where, n_{pr} is the number of rings.

In this model, two rings have been considered.

vi) MEP lost in overcoming combustion chamber and wall pumping losses

$$FMEP6 = \sqrt{\frac{p_{imep}}{11.45}} * 0.0915 * \left(\frac{N}{1000}\right)^{1.5} \tag{24}$$

Where p_{imep} is the indicated mean effective pressure.

The total mean effective pressure (MEP) lost in pressure can be calculated as,

FMEP=(FMEP1+FMEP2+FMEP3+FMEP4+FMEP5+FMEP6)

The specific heat ratio is the most important thermodynamic factor in calculation of the heat release in the engine. The variation of specific heats of air is found in literature [12], it was calculated by the following equation.

$$C_p = 2.506\times10^{-11}T^2 + 1.454\times10^{-7}T^{1.5} - 4.246\times10^{-7}T + 3.162\times10^{-5}T^{0.5} + 1.3303 - 1.512\times10^4 T^{-1.5} + 3.063\times10^5 T^{-2} - 2.212\times10^7 T^{-3} \tag{25}$$

2. MODEL VALIDATION AND PERFORMANCE ANALYSIS

The results predicted by the developed model are compared with the experimental and numerical data reported by [1]. The parameters in Table 1 are given for the compression ignition engine model and have been compared with the experimental and numerical results published by [1], in his experimental report he carried out experiment on a four stroke, direct injection, and naturally aspirated single cylinder diesel engine. The specifications of the engine along with the operating conditions are given in the table 1.

Table -1: Engine specifications along with operating parameters [1]

Engine specifications	
Type	Four stroke, single cylinder, Compression ignition engine
Fuel	Biodiesel (B100) of rubber seed oil methyl ester
Bore × Stroke	0.088m×0.110m
R.P.M	1500
Equivalence Ratio	0.50
Compression ratio	16.5, 17.5, 18.5 (CI version)

The experimental test and the numerical analysis were carried out in compression ignition engine using B100 biodiesel as a fuel and the test were carried out on three compression ratio 16.5, 17.5 and 18.5. The calorific value of the B100 fuel is 36.50 MJ/kg. The pressure variation along the crank angle predicted by the developed model validated with the results obtained by [1] for different values of compression ratio (CR) at equivalence ratio 0.50 have been presented below.

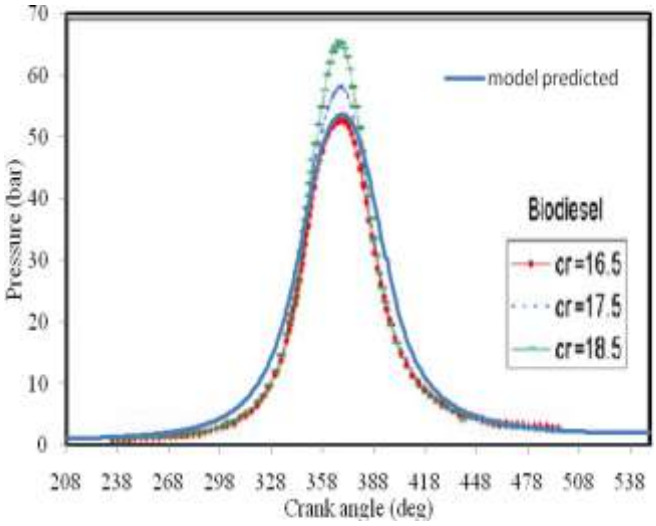

Fig -1: Pressure variation with crank angle when CR is 16.5 and the burning duration is 49^0

Figure 1 shows the variation of the pressure with respect to crank angle when the compression ratio is 16.5, the burning duration is 490 and the fuel injection timing is 120 BTDC (Before Top Dead Centre).

It is observed from the results shown in Figure that the variation pressure along with the crank angle are approximately same with the numerical data reported by [1] for different values of compression ratio. Therefore, the developed model can well interpret the performance parameters of the engine very well.

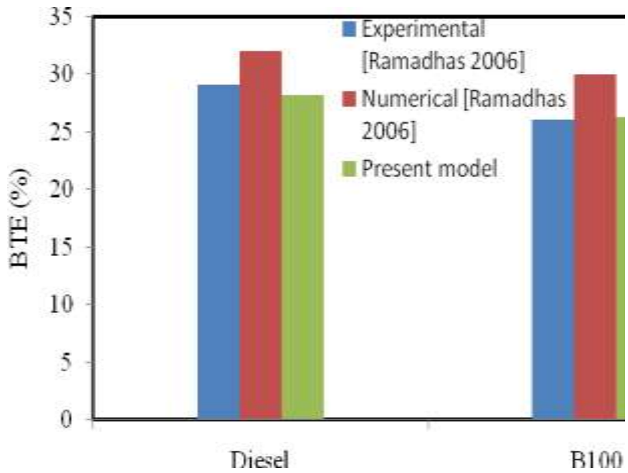

Fig -2: Comparison of brake thermal efficiency for different fuels

The present model has been validated with the experimental and numerical result for diesel and biodiesel (B100). The fig. 2 shows the brake thermal efficiency for diesel and biodiesel and here the present model result is successfully validated with the numerical and experimental results reported by [1]. Furthermore, validations of the other numerical data have been done in the table 2. It shows a reasonable agreement with the numerical data reported by [1].

Table -2: Engine other parameters compared with numerical data

Compres-sion ratio		Peak Pressure (bar)	BTE (%)	Peak Temperature (K)
16.5	Present Model	45	26.26	1353
	Ramadhas et al. Model	50	29	1420
17.5	Present Model	59	27	1360
	Ramadhas et al. Model	55	30.5	1425
18.5	Present Model	68	27.8	1386
	Ramadhas et al. Model	62	31.5	1434

The optimization of the single zone compression engine has been done at burning duration of 440 crank angle and for the combustion start timing at 200 BTDC with an equivalence ratio of 0.5 and compression ratio 16. The fig. 3, shows the variation of the temperature and pressure with respect to crank angle. The maximum temperature of 1390.99 K is obtained at 190 ATDC and the peak pressure of 60.60 bar is obtained at 100 ATDC.

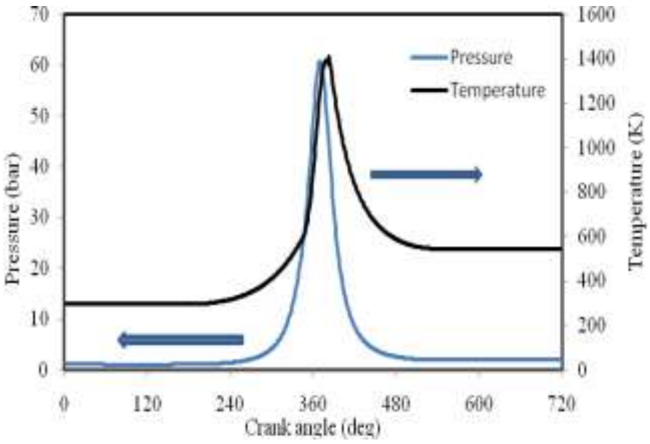

Fig -3: Temperature and Pressure variation along the crank angle

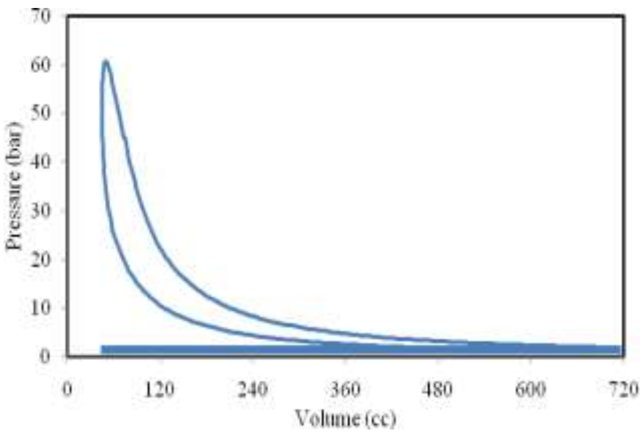

Fig -4: PV work done per cycle

Figure 4, shows the variation of pressure within the combustion chamber with respect to volume. The indicated work done is 13.20 kJ/cycle, the indicated thermal efficiency is 34.17 % with the indicated specific fuel consumption of 0.289 kg/kW-h. Also found that the indicated mean effective pressure as 5.26 bar and brake mean effective pressure as 4.18 bar. The brake work done is 10.52 kJ/cycle, the brake thermal efficiency is 27.21 % and the brake specific fuel consumption is 0.362 kg/kWh.

3. CONCLUSIONS

A single zone C.I engine model based on physical geometries has been developed that would be suitable for conventional diesel as well as for alternative fuel (biodiesels). The model has been successfully validated with the simulation results as well as the experimental results available in the literature. A close agreement has obtained for different performance parameters. Optimization of the different operating parameters such as injection timing, burning duration and compression ratio have been done based on the maximum thermal efficiency and minimum fuel consumption. For Biodiesel (B100) the optimum operating conditions are 44º burning duration, 20º BTDC injection timing and 16 compression ratio. The value of different parameters that have been obtained under the optimum conditions. The indicated and brake work done obtained are 13.20 kJ/cycle and 10.52 kJ/cycle respectively for B100. The indicated and brake thermal efficiency obtained are 34.17% and 27.21% respectively for B100. The indicated and brake specific fuel consumption obtained are 0.289 kg/kWh and 0.362 kg/kWh.

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