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A Cycle Simulation Model for Bio-Fueled Conventional Engine

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Abstract

In recent years, much research has been carried out to find suitable alternative fuel to petroleum products. The use of renewable fuels like ethanol, biogas and biodiesel in diesel engines is significant in this context. Diesel engines are mainly used in industrial, transport and agricultural applications due to their high efficiency and reliability. However, they suffer from high smoke and nitrogen oxide (NOx) emissions. The more stringent government regulations on exhaust emission and the fast depletion of world-wide petroleum reserves provide a strong encouragement to the search for alternative fuels. Biodiesel is oxygenated, biodegradable, non-toxic and environmentally friendly. Experimental analysis of the engine with various biodiesel and its blends requires much effort and time. Hence, a theoretical model is developed to analyze the performance characteristics of the compression ignition engine fueled by biodiesel and its blends.

1. Objectives

In this work an attempt is made to study the following:

- Mathematical model for predicting the variation of fuel injection angle by using jatropha straight vegetable oil, diesel and their blends.
- Mathematical model for observing the effect of injection pressure on engine performance.
- Modeling for analyzing the effect of engine speed.
- Simulation study of the developed model for engine performance by F-Chart Software.

2. Proposed Methodology

Ignition, combustion and pollutant formation in diesel engines is strongly influenced by the fuel spray, while fuel spray is significantly affected by such properties as vapor pressure, viscosity etc.

The present study describes a cycle simulation model. This thermodynamic based model follows the changing thermodynamic state of the working fluid through the engine intake, compression, combustion, expansion and exhaust processes for predicting the performance of a diesel engine fuelled by biodiesel. The model will predicts the performance of a CI engine in terms of brake power and brake thermal efficiency for fuel considered for the present study. These parameters will be calculated for jatropha straight vegetable oil for different angle of injection, injection pressure and inlet temperature of injected fuel. Fuel properties at varying temperature, injection pressure and operating parameters are specified as inputs to the model.

In this work, F-Chart software will be used for calculating the performance of a compression ignition engine when it is powered by bio- fuels.

3. Introduction

Internal combustion engines, particularly diesel engines, are widely used as prime movers in the transport industry, power generation and also used in agricultural applications. Diesel engines operate at a relatively higher efficiency compared to their counterparts. Current and future legislation on emissions require engine developers to produce cleaner and more efficient power plant systems. The rapid development of computer technology has encouraged the use of complex simulation techniques to quantify the effect of the fundamental processes in the engine systems. The advances achieved by current automotive engines would have been impossible without the simulation models providing these insights [1, 2].

The theoretical models used in case of internal combustion engines can be classified into two main groups viz., thermodynamic models and fluid dynamic models. These models are further classified into two groups namely single-zone models and multi-zone models. Multi-zone models are also called computational fluid dynamics models. These are also applied for the simulation of combustion process in the internal combustion engines. They are based on the numerical calculation of mass, momentum, energy and species conservation equations in either one, two or three dimensions to follow the propagation of flame or combustion front within the engine combustion chamber. Lyn et al. [3] analyzed the effects of injection timing, injection velocity and fueling rate on the delay period. An increase in speed at constant load increases the peak pressure and temperature, due to the decrease in heat transfer, resulting in a slight decrease in delay period as analysed by Wong et al. [4]. Hadenberg et al. [5] developed the empirical relation to calculate the delay period. They reported that, the predicted delay period had been matched with the experimental results over a wide range of operating conditions. Computational fluid dynamics (CFD) models are extensively used for flow visualization, fuel-air mixing combustion analysis and turbulent studies by different researchers [6–10]. Research has been carried out in the field of fuel injection, injection rate, injection duration, injection pressure, l/d ratio of the nozzle on engine performance and emissions during the 1990s [11–13].

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As stated above, researchers have experimentally evaluated the performance characteristics of conventional diesel engines fuelled by biodiesel and its blends. However, experiments require enormous effort, money and time. A realistic numerical simulation model could reduce such effort. Numerical simulation based on mathematical modeling of diesel engine processes have long been used as an aid by design engineers to develop new design concepts.

This study will describes a model for estimating the physical properties of biodiesels based on their chemical composition and structure over a wide temperature range up to the critical temperature, which can be used in combustion modeling. Biodiesel is relatively simple as a chemical since it contains no more than six or seven fatty acid esters. Therefore, it is possible to estimate the properties of each pure component and then compute the mixture properties based on some mixing rules. While this study will be focuses on the properties of jsvo.

The present study will also describe a simple simulation model. This thermodynamic based model follows the changing thermodynamic state of the working fluid through the engine intake, compression, combustion, expansion and exhaust processes for predicting the performance of a diesel engine fuelled by biodiesel.

4. Literature Review

[Gogoi T.K. and Baruah D.C., 2010] developed simulation model for predicting the performance of a single cylinder four stroke diesel engine fuelled by diesel and various blends of diesel and biodiesel. The brake power and brake thermal efficiency predicted by the model under varying speed and CR conditions for different fuels are analyzed that gives characteristic of diesel engine. The model also predicts a higher rate of pressure and temperature rise for the blends during combustion as compared to diesel. A mathematical modeling for analyzing the combustion and performance characteristics of the compression ignition engine and LHR engine was presented by [Prasath B. R. et.al., 2010]. The modeling results shows that with increase in speed the peak pressure, peak temperature and brake thermal efficiency increases and decreases the specific fuel consumption. [Hountalas D. A., 2000] developed the method, based on a general simulation model initially for direct injection diesel. The model manages to predict the behavior of the various engine subsystems as well as exhaust gas temperature. The simulation model after validation is used to predict engine behavior under fault condition.An experimental result on an engine test bench were used to build analytical empirical model of engine working cycle, based on fuzzy sets theory presented by [Kekez M and Radziszewski L., 2010]

5. Model for Prediction the Performance

Volume at any crank angle is calculated from this equation,

$$V_\theta = V_c + \left\{\frac{\pi d^2}{4} \times \frac{l}{2}\right\} \times \{1 + m - (m^2 - \sin^2\theta)^{0.5} \cos\theta\}$$

Where, $m = 2 \times (L/l)$

d = Cylinder diameter, m

L = Connecting rod length, m

l = Stroke length, m

V_θ = Volume at any crank angle, m^3

V_c = Clearance volume, m^3

θ = Crank angle, degree

5.1 Calculation of specific heat

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

$$C_v(T) = (B - R) + C/T$$

Total specific heat of all species is, $C_p(T) = B + C/T$

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

5.2 Initial Temperature and Pressure During Start of Combustion

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

$$P_2 = P_1 \times (V_1/V_2) \times (T_2/T_1) \text{ and}$$

$$T_2 = T_1 \times (V_1/V_2)^{(R \times T/C_v)}$$

T_1 = Initial temperature, K

T_2 = Final temperature, K

P_1= Initial pressure, bar

P_2= Final pressure, bar

R= Characteristic gas constant, kJ/kgK

C_v = Specific heat at constant volume, kJ/kgK

5.3 Work Done

Work done in each crank angle is calculated from

$dW = \{(P_1 + P_2)/2\} \times (V_2 - V_1)V_1$ = Initial volume, m^3

V_2 = final volume, m^3

6. Viscosity of Air at each Temperature is Calculated from

$\mu_{air} = 3.3 \times 10^{-7}(T)^{0.7}$ **Viscosity of combustion gases are calculated from:**

$\mu_{prod} = \mu_{air}/(1 + 0.027\Phi)$

Φ=Equivalence ratio Reynolds number for each time step is calculated as the viscosities varies

$R_e = \rho d V_p/\mu_{prod}$

ρ= Density of gas mixture, kg/m^3

V_p= Mean piston speed, m/min

Thermal conductivity is calculated for each change in viscosity

$k = C_p\mu_{prod}/0.7$

6.1 Heat Transfer

The heat transfer between the trapped gas and the surrounding wall is calculated by following equation.

$$dQ = a \times \frac{k}{d} \times R_e{}^b (T_w - T) + (T_w{}^4 - T^4)$$

Where T_w is the temperature of the cylinder wall, R_e is the Reynolds number with d is the cylinder bore. The value of a varies with speed and engine design. With normal combustion, 0.35<a<0.8 with b=0.7, c=0 for combustion period otherwise c=3.3$\times 10^{-8}$ W/$m^2 k^4$

6.2 Heat transfer coefficient

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

$$h_g = 0.26 \times k/d \times R_e{}^{0.6}$$

6.3 Mass of fuel Injected

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

$$M_f = C_d\, A_n \left(2\rho_f \Delta P\right)^{0.5} \left(\frac{\Delta\theta_f}{360N}\right)$$

C_d = Coefficient of discharge of injector nozzle

A_n= Cross sectional area of nozzle, m^2

ΔP= Pressure drop across the nozzle, bar

N= Engine speed, rpm

$\Delta\theta_f$= Fuel injection period, degree

6.4 Nitric Oxide Formation

As the consideration of chemical equilibrium cannot predict correctly the NO concentration, the generally accepted kinetics formation scheme proposed by Lavoie et al is used. The equations, which describe the model, together with their forward reaction rate constants k_{if}(in m^3/kmol/s), are

$$N + NO \rightarrow N_2 + O \quad , \quad k_{if} = 3.1 \times 10^{10} exp^{\left(-\frac{160}{T}\right)}$$

$$N + O_2 \rightarrow NO + O, \quad k_{2f} = 6.4 \times 10^{6} exp^{\left(-\frac{3125}{T}\right)}$$

$$N + OH \rightarrow NO + H, \quad k_{3f} = 4.2 \times 10^{10}$$

The change of (NO) concentration (in kmol/m^3) is expressed as follows:

$$\left(\frac{l}{V}\right) \times \frac{d(NO)V}{dt} = \frac{2(1-\alpha_2)R_1}{l} + \frac{AR_1}{(R_2 + R_3)}$$

Where R_i is the one way equilibrium rate for reaction *i*, defined as

$$R_1 = k_{if}(N)_e(NO)_e \ , \quad R_2 = k_{2f}(N)_e(O_2)_e \ , \quad R_3 = k_{3f}(N)_e(OH)_e$$

With index **e** denoting equilibrium concentration and term $\alpha = (NO)/(NO)_e$

6.5 Soot Formation

Exhaust gas soot concentration related to normal conditions are as follows,

$$[C] = \lim_{\theta B} 480 \ \frac{d[c]}{dt} \times \frac{d\theta}{6n} \times \frac{\left(\frac{0.1}{P}\right)}{\gamma}$$

Where γ=1.33 is an exhaust gas adiabatic exponent, [C] is the current soot concentration in the cylinder, θ is the crank angle and P is the cylinder pressure.

7. General Description

The equations of the model exposed in the above section are solved numerically using with a time step size of 0.5° crank angle for the independent variable θ (Crank angle, degree). The steps used are described in detail in the following subsections. Before start of calculations, the design characteristics of the engine in hand are provided, as well as the operating data at the start of the cycle, i.e., at the IVC event. The calculations stop at the EVO event. The corresponding program is written in F-Chart software. The computer program structure follows very closely the step by step procedure described below.

7.1 Computational of Compression Phase

1. Introduce the data at the IVC event, i.e. P_1, T_1 , trapped composition (air with no fuel) and compute V_1 from engine geometry . Select the crank angle step size delta theta ($\Delta\theta$), here, to 0.5°.
2. Calculate the initial internal energy E_1 using its T_1 relation and similarly for the heat capacities C_p and C_v.
3. For the new crank angle $\theta_2 = \theta_1 + \Delta\theta$, compute V_2 from the engine geometry.
4. Estimate temperature T_2 assuming an isentropic change:
   $$T_2 = T_1 \times (V_1/V_2)^{(R \times T/C_v)}$$
   then, find pressure P_2 from perfect gas state equation :
   $$P_2 = P_1 \times (V_1/V_2) \times (T_2/T_1)$$
5. Calculate the internal energy E_2 using its T_2 relation.
6. Calculate the work in step :
   $$dW = \{(P_1 + P_2)/2\} \times (V_2 - V_1)$$
7. Calculate dQ from heat loss model.
8. Apply the first law of thermodynamics for a closed system:
   $$f(E) = E_2 - E_1 + dW - dQ = 0$$
   Solve the equation with respect to T_2 using the Newton-Rapson numerical method so that a better estimate of T_2 is found.
9. Calculate P_2 for the revised value of T_2 using the gas state equation at the time moments 1 and 2.
10. Repeat steps 5-9 until the error *f*(E) in the first law equation is negligible .
11. Continue this way, until θ_2 equals the value at the start of fuel injection.

7.2 Same Manner Computational Analysis of the following Phases can be Carried Out

(a) Computation analysis of combustion and expansion phases for the zones.
(b) Calculations for air (unburned) zone.
(c) Calculations for the burning zones, etc.

7.3 Part Program For Compression Phase

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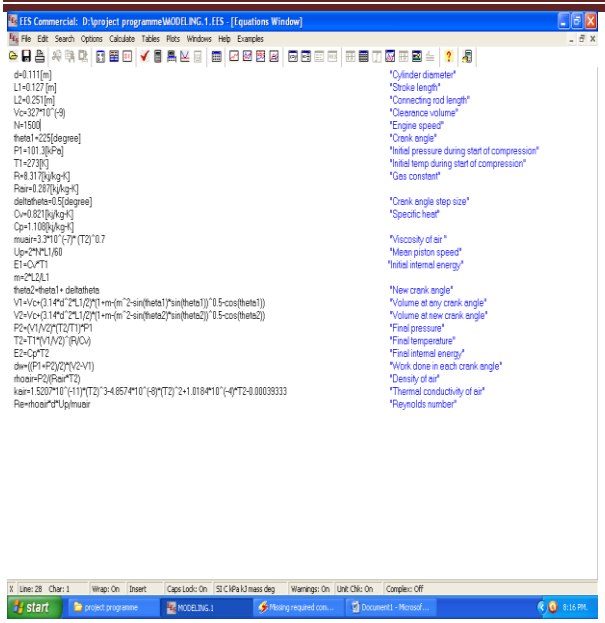
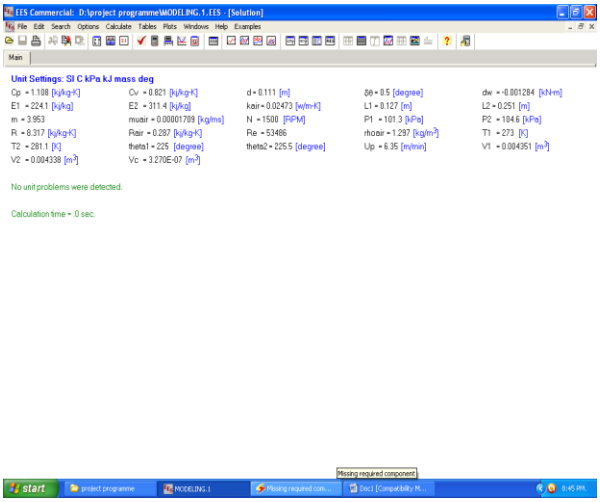


7.4 Result of the Above Part Program



8. Simulation

The liquid density of biodiesel decreases linearly at the lower temperatures until the initial boiling point (IBP) is reached. Therefore, it decreases at a higher rate, which could be attributed to an increasing liquid thermal expansion rate. However, the trends will be regarded as representative of the fuel density in the combustion chamber. The same predicted density behavior of biodiesel from $150^{\circ}C$ to $240^{\circ}C$ in steps of $10^{\circ}C$ is shown below with the use F- Chart software.

9. Expected Outcomes

a) Model can predict the physical properties of Jatropha Straight Vegetable Oil with variation in the inlet temperature.
b) Developed model will be suitable for single cylinder four stroke diesel engines fuelled with JSVO.
c) Also suitable for predicting the performance under varying crank angle, injection pressure condition and speed.
d) Possibility of all calculated parameter should be within the limit.

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