A Simulation Model for Predicting the Performance of Bio Fuels Conventional Engine

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Abstract -A theoretical method based on various existing process models has been developed for the performance evaluation of a compression ignition engine by using diesel and Jatropha straight vegetable oil as an input fuel. A few researches have been carried out performance testing of conventional diesel engines fuelled by different biofuel and its blends. In the present work, a simulation model based on two different zones of combustion has been developed and discussed to predict the performance of compression ignition engine. The effects of fuel injection pressure and advance angle of fuel injection point are analyzed on the basis of brake thermal efficiency, brake specific fuel consumption, as well as for nitric oxide emissions. A good synchronization has been observed between the model's prediction and the available theoretical and experimental results. Engineering equation solver has been used as the programming input in the model.

Keywords: Diesel engine; Thermodynamic modeling; Engine performance and emissions; Jatropha straight vegetable oil; Enginerring equation solver

1. Introduction

Today transport sector is almost dependent on fossil fuels, particularly, petroleum-based fuels such as gasoline, diesel fuel, liquefied petroleum gas and natural gas. In the last 25 years, the world wide consumption of petroleum has steadily increased. In 1985, total world wide petroleum consumption was 2807 million tons but in 2008 figure reached up to 3928 million tons .However, the petroleum fuel is becoming scarce and its use is associated with the increase of environmental problems. According to review of world energy, current oil would suffice to last only 42 years [1]. The consumption of oil is approximately one-third of all the energy used in the world. The total world oil consumption was about 63 million barrels per day in 1980. Since then, world consumption of petroleum products has increased to about 84 million barrels per day in 2005. Rapid economic growth in China and India could significantly increased world demand for oil. India imported nearly 70% of its crude oil requirement (90 million tonnes) during 2003–04. With the same trend of rise in consumption this figure will rise to 95% by 2030 [2]. To overcome the problem of rising energy demand and reducing petroleum reserves, fuels such as biofuels can be considered as one of the solutions. Typical biofuels are biodiesel and alcohol. The importance of biodiesel has been pointed out by many previous researchers. The biodiesel can be obtained from renewable sources such as vegetable oils or animal fats through a transesterification process [3].

Biodiesel are monoalkyl esters of fatty acids which derived from vegetable oils or animal fats and it is a clean and renewable fuel. Biodiesel is usually produced by the transesterification of vegetable oils or animal fats with methanol or ethanol [4]. The vegetable oil is transesterified using methanol in the presence of NaOH as a catalyst. The parameter involved in the processing such as catalyst amount, molar ratio of alcohol to oil, reaction temperature and reaction time are optimized [5].Biodiesel has various advantages such as its renewable,biodegradable,non-toxic, is safe for use in all conventional diesel engines and offers the same performance and engine durability as petroleum diesel fuel. The use of biodiesel has grown dramatically during the last few years. But feedstock costs account for a large percent of the direct biodiesel production costs including capital cost and return.

A technique to reduce the costing of biodiesel production is to use the less expensive feedstock containing fatty acids such as inedible oils, animal fats, and waste food oil and by products of the refining vegetables oils [6]. As an option the inedible vegetable oils, mostly produced by seed-bearing trees and shrubs can provide an alternative. With no competing food uses, the characteristics turn attention to jatropha curcas that grows in tropical and subtropical climates across the developing world. During the literature survey an outcome brings in to front which shows that oil content of jatropha kernel consisting by 63.16%. Oil content of jatropha kernel was found higher than linseed, soybean, and palm kernel which were 33.33%, 18.35% and 44.6% respectively [7]. High oil contents of jatropha curcas indicated that jatropha curcas are suitable as non-edible vegetable oil.

Existing vegetable oils mainly contain triglycerides (90% to 98%) and small amounts of mono and di-glycerides. Commonly found fatty acids in vegetable oils are stearic, palmitic, oleic, linoleic and linolenic acids. Vegetable oils can also be produced even on a small scale for on-farm utilization to run tractors, pumps and small engines for power generation/irrigation. Suitability of vegetable oils as fuels for diesel engines mainly depends on the combustion characteristics and the type of engine used along with operating conditions. Vegetable oils can be used directly or blended with diesel to operate compression ignition engines. The different blends of vegetable oils with diesel has been tested to run engine successfully world wide. Brazil used pre-combustion chamber engines with a blend of 10% vegetable oil while maintaining same power output without any engine modifications. Short-term engine performance tests indicated good potential for most vegetable oils as input fuel for engine. Some studies reported lower exhaust emissions including PAHs and PM [8] [9].

The high viscosity and low volatility of raw vegetable oils are generally noticed as major drawbacks for their direct utilization as fuels in diesel engines. The usage of raw vegetable oils in operation of diesel engines leads to injector coking, severe engine deposits, filter gumming problems, piston ring sticking and thickening of the lubricating oil. Different methods such as preheating, blending, ultrasonically assisted methanol transesterification and supercritical methanol transesterification are being used to reduce the viscosity and make them suitable for an input fuel supply in C.I engines [10][11].

The present model describes a cycle simulation for engine testing. This 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 diesel and jatropha straight vegetable oil. The model predicts the performance of a C.I engine in terms of brake thermal efficiency and brake specific fuel consumption for diesel and jatropha straight vegetable oil fuels considered for the present study. An engineering equation solver program, developed by F-Chart Software is used for conventional engine. It is a general equation-solving program that can numerically solve thousands of coupled non-linear algebraic and differential equations. The program can also be used to solve differential and integral equations, do optimization, provide uncertainty analyses, perform linear and non-linear regression, convert units, check unit consistency, and generate publication-quality plots. A major feature of EES is the high accuracy thermodynamic and transport property database that is provided for hundreds of substances in a manner that allows it to be used with the equation solving capability [12].

2. Jatropha oil

It is non-edible oil being singled out for large-scale plantation on wastelands. Jatropha curcas plant can grow under adverse conditions. It is a drought-resistant, perennial plant, living up to fifty years and has capability to grow on marginal soils such as in India. It requires very little irrigation and grows in all types of soils (from coastline to hill slopes). The oil content of Jatropha seed ranges from 30% to 40% by weight and the kernel itself ranges from 45% to 60% [10]. Fresh Jatropha oil is slow-drying, odorless and colorless, but it turned yellow after aging .The only limitation of this crop is that the seeds are toxic and the press cake can not be used as animal fodder. The press cake can only be used as organic manure. The fact that Jatropha oil can not be used for nutritional purposes without detoxification makes its use as energy or fuel source for better performance as bio diesel [13]. In Madagascar, Cape Verde and Benin, Jatropha oil was used as mineral diesel substitute during the Second World War. However, when low concentration of Jatropha oil used with diesel in different blends in C.I engines, the performance and emissions characteristics are found similar to that of diesel [14].Some researchers tried to reduce viscosity of Jatropha oil by heating it and also blending it with

diesel and study the performance &emission characteristics [10].

3. The model

Due to the complexity of the diesel engine combustion and the turbulent fuel air- mixing it is difficult to develop a effective model that does not have too long computational time. There are different approaches to implement a diesel combustion model i.e. single-zone or multi-zone[15][16].To develop model that is accurate enough, has a acceptable simulation time and has a complexity level that reflects the timeframe of this paper, a two-zone model has been selected. The model includes only those processes occurring during closed cycle, when all valves are closed. The compression phase begins at the point of closing of the inlet valve (IVC) and continues up to crank angle at which combustion begins. The period from the end of combustion to the exhaust valve opening (EVO) is the expansion phase. The compression and expansion phases are considered as polytrophic.

3.1. Energy balance equation

The energy balance equations for the present model can be expressed in differential form. Then the first law of thermodynamics for compression phase, considering only one zone (pure air) is given as:

$$\frac{dQ}{d\theta} = \frac{dU}{d\theta} + p\frac{dV}{d\theta} \qquad (1)$$

Where $\left(\frac{dQ}{d\theta}\right)$, is the rate of heat loss to the combustion chamber walls, $\left(\frac{dU}{d\theta}\right)$ is the rate of change of internal energy, and $\left(p\frac{dV}{d\theta}\right)$ is the rate of work done on the system. During the combustion and expansion phases, at the start of combustion the entire volume is assumed to be the unburned region and at the end of combustion the entire volume is assumed to be the burned region. Therefore, the energy balance equation, using the two zone model in the combustion chamber is given [17] [18] and can be expressed by,

$$m\frac{du}{d\theta} + u\frac{dm}{d\theta} = \left(\frac{dQ}{d\theta}\right) - \left(p\frac{dV}{d\theta}\right) \qquad (2)$$

The various term of the equation are elaborated below as:

$\left(\frac{du}{d\theta}\right)$, is the rate of change of internal energy of mixture of mass m.

$\left(\frac{dm}{d\theta}\right)$, is the rate of change of mixture mass m.

$\left(\frac{dQ}{d\theta}\right)$, is the net heat release rate which is the difference between the rate of heat release during the combustion period and rate of heat transfer from in combustion chamber gases to the combustion chamber walls .

$\left(p\frac{dV}{d\theta}\right)$, is the rate of work done by the system.

Where m is the mass of the mixture (air + fuel) contained in the combustion chamber, θ is the crank angle, V is the instantaneous cylinder volume, P is the cylinder pressure and Q is the heat energy.

3.2. Volume at any crank angle

The instantaneous cylinder volume V_θ is given by:

$$V_\theta = V_c + \left\{\pi\frac{d^2}{4}\frac{l}{2}\right\}\left\{1 + m - \left(m^2 - \sin^2\theta\right)^{0.5} - \cos\theta\right\} \qquad (3)$$

Where $m = (2.L).l^{-1}$, d is Cylinder bore, L is connecting rod length l is stroke length, V_θ is the volume at any crank angle, V_c is the clearance volume and θ is the crank angle.

3.3. Gas properties calculation

The gas properties are functions of temperature and its composition. The higher temperature can be achieved for that system where the combustion is completed and the system became adiabatic. It depends on the chemical composition of the reactant mixture, pressure and temperature of the mixture and combustion process. A hydrocarbon fuel can be represented by $C_xH_yO_z$. The chemically suitable amount of oxygen (O_{cc}) required for combustion per mole of fuel can be written as:

$$O_{cc} = m_c + 0.25m_h - 0.5m_O \quad (4)$$

The minimum amount of oxygen required (O_{min}) for combustion in the reactants per mole of fuel (to convert H_2 to H_2O and C to CO) can be expressed as:

$$O_{\min} = O_{cc} - 0.5m_c \quad (5)$$

Here m_c, m_h and m_o are respectively the number of moles of carbon, hydrogen and oxygen atoms in one mole of fuel. From these mole fraction calculations, mixture properties such as enthalpy, internal energy, specific volume and specific heat at constant pressure are calculated.

3.4. Internal energy

The internal energy (u) can be written in terms of the internal energy of unburned zone Uu and burned zone U_b with (*x*) as a mass fraction burned as:

$$u = \frac{U}{m} = xU_b + (1-x)U_u \quad (6)$$

Similarly the specific volume can be written in terms of the specific volume of the unburned zone (u) and the burned zone (b) as:

$$v = \frac{V}{m} = xv_b + (1-x)v_u \quad (7)$$

Differentiating Eq. (6) and (7) with respect to crank angle and replacing the partial derivative terms with the logarithmic forms gives the internal energy and specific volumes for burnt and unburnt zone. The mass fraction burned can be calculated by Weibe function by using following expression [17]:

$$x = 1 - \exp\left[-a\left(\frac{\theta - \theta_0}{\Delta\theta}\right)^{c+1}\right] \quad (8)$$

Where θ_0 is the start of combustion and $\Delta\theta$ is the combustion duration and a is an adjustable parameter that characterizes the completeness of combustion process. The parameter c represents the rate of combustion. The value of c for all fuel is taken as 2.0 and a as 5.0[17].

3.5. Heat loss from combustion chamber

The heat loss from the combustion chamber can also be expressed by using both burned and unburned zones as follows:

$$\frac{dQ}{d\theta} = \frac{Q_l}{\omega} = \frac{Q_b + Q_u}{\omega} \quad (9)$$

Where

$$Q_b = hA_b(T_b - T_w) \quad (10)$$

$$Q_u = hA_u(T_u - T_w) \quad (11)$$

The surface areas of the two different zones are given by A_u and A_b. These areas can be related to a mass fraction burned *x* by using an empirical formula [19]:

$$A_b = \left(\frac{\pi d^2}{2} + \frac{4V}{d}\right)x^{0.5} \quad (12)$$

$$A_u = \left[\left(\frac{\pi d^2}{2} + \frac{4V}{d}\right)\left(1 - x^{0.5}\right)\right] \quad (13)$$

Where d is the bore of the cylinder. The convective heat transfer coefficient h for Eq. (9) and (10) is given by [20].

3.6. Heat transfer coefficient

Heat transfer coefficient of gas for each degree crank angle is calculated from the following equation as:

$$h = 0.26(k/d)\,\mathrm{Re}^{0.6} \quad (14)$$

Where k is the thermal conductivity for each change in viscosity and Re is the Reynolds number for each time step is calculated as follows:

$$\mathrm{Re} = \frac{\rho d V_p}{\mu} \quad (15)$$

Where ρ, is the density of gas mixture μ is the dynamic viscosity and V_p is the mean piston speed.

3.7. Ignition delay

The ignition delay is the time duration between start of fuel injection and start of combustion. Many expressions for ignition delay are found in literature as a function of mixture pressure, temperature, fuel cetane number [21] [22]. The following empirical correlation is used to obtain the value of ignition delay [23]:

$$t_d = \frac{2.64}{P^{0.8}\phi^{0.2}} \exp\left(\frac{1650 - 20CN}{RT}\right) \quad (16)$$

Where R is universal gas constant, CN is fuel cetane number and ¢ is fuel air equivalence ratio. However, the constants and exponents in the above correlation are to be better calibrated against experimental results and conclusion.

3.8 Mass of fuel injected

Considering that nozzle open area is constant during the injection period, total mass of the fuel injected for each crank angle is calculated as follows:

$$m_f = C_d A_n \sqrt{2\rho_f \Delta P}\left(\frac{\Delta\theta_f}{360N}\right) n \quad (17)$$

Where n is the number of injector nozzle holes, C_d is the coefficient of discharge of injector nozzle, A_n is the cross sectional area of nozzle, ΔP is the pressure drop in the nozzle, N is the engine speed and $\Delta\theta_f$ is the fuel injection period.

3.9. Pressure drop in the nozzle

The pressure drop in the nozzle is calculated as follows [24]:

$$\Delta P = 0.5\rho_f \left(\frac{u_{inj}}{C_d}\right)^2 \quad (18)$$

Where u_{inj} the spray velocity from nozzle hole is given as:

$$u_{inj} = \left(\frac{dm_f}{d\theta}\right)\left(\frac{6N}{\rho_f A_n}\right) \quad (19)$$

Where $\left(\frac{dm_f}{d\theta}\right)$ is the fuel injection rate (kg/^{0}CA)

$$\left(\frac{dm_f}{d\theta}\right) = \left(\frac{m_f}{n\Delta\theta_f}\right) \quad (20)$$

3.10. Sauter mean diameter

It is the ratio of mean volume to the mean surface area of the fuel droplets and has an important role in defining the fuel atomization characteristics. Smaller SMD results better fuel atomization and ultimately the fuel combustion efficiency.

$$SMD = 3.08\nu^{0.335}(\sigma\rho_f)^{0.737}\rho_a^{0.06}(\Delta P)^{-0.54} \quad (21)$$

Where ν is the kinematic viscosity, σ is surface tension of fuel, ρ_f is the density of fuel, ρ_a is the density of air and ΔP is the pressure drop across the nozzle.

3.11. Instantaneous pressure at any crank angle

$$\left(\frac{dP}{d\theta}\right) = \left(\frac{A + B + C}{D + E}\right) \quad (22)$$

Where A,B,C,D and E are the variables which depend on the internal energy ,specific volume, temperature of the burnt and unburnt zone, specific heat at constant pressure and crank angle.

3.12. Net work done

Work done in each crank angle is calculated from (Heywood, 1998):

$$dW = \left(\frac{P_1 + P_2}{2} \right)(V_2 - V_1) \quad (23)$$

Where P_1 and P_2 are the change in pressure for each crank angle inside the combustion chamber, and V_1 and V_2 are the changes in volume inside the combustion chamber for the same crank angle.

3.13. Nitric oxide formation

As the consideration of chemical equilibrium cannot predict the actual (NO) concentration. The general accepted kinetics formation scheme proposed by [25] is used. The equations which describe the model together with their forward reaction rate constants k_{if} (m^3/kmol/s) are as follows:

$$N + NO \leftrightarrow N_2 + O \quad k_{if} = 3.1 \times 10^{10} \exp^{(-160/T)} \quad (a)$$

$$N + O_2 \leftrightarrow NO + O \quad k_{2f} = 6.4 \times 10^{6} \exp^{(-3125/T)} \quad (b)$$

$$N + OH \leftrightarrow NO + H \quad k_{3f} = 4.2 \times 10^{10} \quad (c)$$

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

$$\frac{1}{V}\left(\frac{d((NO)V}{dt} \right) = 2(1-\alpha^2)\left(\frac{R_1}{1 + \alpha R_1 / (R_2 + R_3)} \right) \quad (24)$$

Where R_i is the one way equilibrium rate for reaction i, defined as: $R_1 = k_{1f} (N)_e (NO)_e$, $R_2 = k_{2f} (N)_e (O_2)_e$, $R_3 = k_{3f} (N)_e (OH)_e$

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

4. Procedure for theoretical solution

The equations of the model adopted in the previous section are suitable for any hydrocarbon fuel as diesel, vegetable oil and biodiesel etc. These equations are solved numerically using with time step size of 2^0 crank angles. The engine geometrical parameters, molecular weight of gaseous products and various constants used in the modeling are defined. The input parameters used in modeling are injection pressure, crank angle and the molecular formula of the diesel and jatropha oil. The several physical, chemical and thermal properties are also used as input parameters as shown in Table.1 [11].The properties of gaseous constituents such as enthalpy, internal energy and specific heats are calculated as a function of temperature . The pressure and temperature of the gases in the combustion chamber are calculated for every two degree crank angle. From the typical design of the engine, the combustion chamber volume at every degree crank angle is calculated with the help of (Eq.3). Ignitation delay of gaseous mixtures is determined by (Eq.16). The outputs of the modeling program are instantaneous pressure, temperature, volume and the performance parameters that include BTE, BSFC and NO_x emission.

Property	Diesel	Jatropha oil
Density (kg/m^3)	840	917
Kinematic viscosity at 40 ^{0}C (cSt)	2.44	35.98
carbon residue (%, w/w)	0.1	0.8
Ash content (%, w/w)	0.01	0.03
Calorific value (MJ/kg)	45.343	30.071
Carbon (%, w/w)	80.33	76.11
Hydrogen (%, w/w)	12.36	10.52
Nitrogen (%, w/w)	1.76	0
Oxygen (%, w/w)	1.19	11.06
Sulfur (%, w/w)	0.25	0
Cloud point (^{0}C)	3	9
Pour point (^{0}C)	-6	4
Flash point (^{0}C)	71	229

Table 1: Properties of diesel and jatropha oil

In brief, the numerical solution stages consists of calculation of compression phase(one zone), calculations of combustion and expansion phases for both zones, calculations in the unburned zone, calculations in the burning zone and calculation of the mean state of zones.

5. Results and discussions

In the combustion modeling the molecular formula of diesel fuel is taken as C_{10} H_{22} .From the fatty acid composition, Jatropha classifies it as a linoleic or oleic acid type, which are unsaturated fatty acids [10].So based on these properties, the molecular formula of Jatropha oil is approximated as $C_{18}H_{32}O_2$

[3][26]. In general, this combustion model, developed for C.I engine analysis is suitable for any hydrocarbon fuel. This includes diesel, biodiesel or their blends as well as vegetables oil. The engine model is analyzed for the variation in fuel injection pressure and point of fuel injection for the diesel as well as jatropha oil is presented as follows.

5.1. Effect of fuel injection point on brake thermal efficiency

The fuel injection point is taken for analysis because the time duration between the point of fuel injection and the start of combustion process has been felt be one factor which is not sufficient to allow the fuel particles to get atomized properly. The increase in time delay provides more time available for this precombustion. It is stated that the jatropha oil demands ignition timing to be more for better combustion and hence improves the engine performance and emission characteristics [27]. The advancing the fuel injection point by 1^0 results in increase in ignition delay of 0.07 sec .This results in better fuel atomization characteristics and enhanced the combustion efficiency. The increase in delay period is achieved by advancing the fuel injection point. In present model the reference or static point of fuel injection is taken 20^0 BTDC for diesel fuel. The fuel injection point used for the analysis of model is 20^0 BTDC to 26^0 BTDC in 2^0 steps. Advancing the fuel injection point is limited because increased delay period increases the possibility of engine detonation and effects efficiency. The possible reason for detonation and reduction in engine performance is due to the fact that advanced ignition timing results in an increased peak pressure and temperature before and around the TDC.

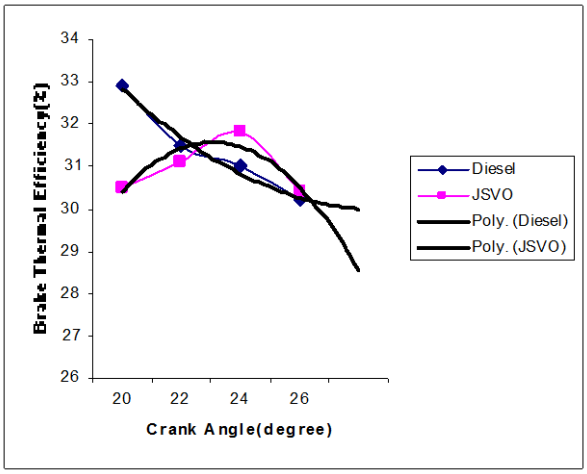

Figure 1: Brake thermal efficiency vs. crank angle (BTDC) at speed 1000 rpm, 80% of full load and static175 bar injection pressure

Figure.1 represents the predicted trend of BTE as a function of crank angle at speed 1000 rpm and 80% of full load (7.35 kW) .BTE decreases with advancing crank angle for diesel fuel. But in case of jatropha oil, it first increases up to 24^0 and then started decreasing. Due to increased brake power and less fuel energy input with jatropha oil, the BTE trend is increasing in case of jatropha oil. But it starts decreasing as injection timing is more advanced, as cylinder pressure and temperature during the delay period become lower. Therefore, the ignition delay period becomes longer [28]. This phenomena results in more fuel being burnt during premixed combustion phase following ignition delay period.

5.2. Effect of fuel injection pressure on brake thermal efficiency

The effect of varying fuel injection pressure at a constant fuel injection point has been investigated in the model at different pressures from175 bar to 205 bar in10 bar step variation. The investigation can be carried out with objective that the fuel atomization characteristics which indirectly depend upon fuel droplets size injected into the combustion chamber gets improved as the fuel droplets reduces at high fuel injection pressures. The sauter mean diameter has an important role in defining the fuel atomization characteristics. Smaller SMD results better fuel atomization and ultimately the fuel combustion efficiency. The SMD of diesel at room temperature and injection pressure 175 bar is found $3.82x10^{-5}$ m.The SMD of diesel can be reduced from 3.82×10^{-5} m to $3.5x10^{-5}$m as the pressure increases from 175 bar to 205 bar. The SMD of jatropha oil at room temperature and 175 bar is obtain $12.56x10^{-5}$ m.It reduces from $12.56x10^{-5}$ m to 11.53×10^{-5} m as increase in pressure from 175 bar to 205 bar. As jatropha oil heated up to 90^0C, the SMD at 175 bar can be reduced from 12.56×10^{-5} m to 4.33×10^{-5} m.This is substantial reduction in SMD by preheating. The preheating the fuel has lowered the viscosity from $48.7x10^{-6}m^2/s$ to $3.68x10^{-6}$ m^2/s.The comparison of SMD for jatropha oil and diesel at room temperature showed that the difference between them is very high $8.74x10^{-5}$m.This difference can be reduced to a good extent($0.5x10^{-5}$m) as the jatropha oil preheated. This appreciable improvement is due to the fact that the viscosity and surface tension have reduces to sufficiently high extent.

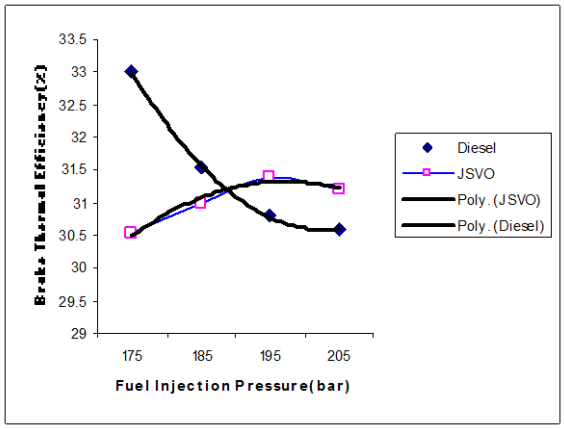

Figure 2: Brake thermal efficiency vs. injection pressure at speed 1000 rpm, 80% 0f full load and 20^0 static injection point (BTDC)

Figure.2 represents the predicted trend of BTE as a function of fuel injection pressure at constant speed, load and point of injection. BTE decreases as fuel injection pressure either decreases or increases from 195 bar.

5.3. Effect of fuel injection point on brake specific fuel consumption

The predicted variation of BSFC keeping fuel injection pressure constant with varying fuel injection point is shown in Figure 3.

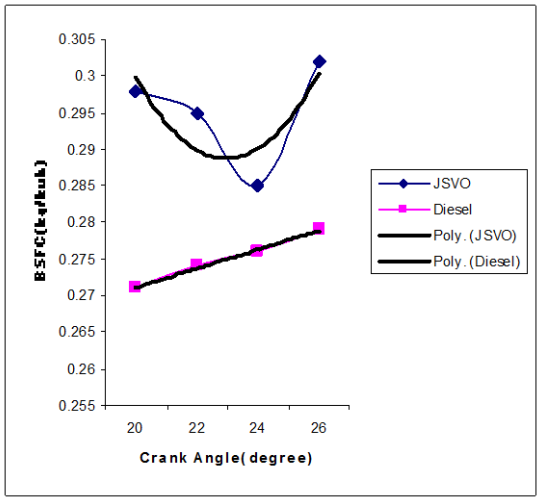

Figure 3: Brake specific fuel consumption vs. crank angle (BTDC) at speed 1000 rpm, 80% of full load and static 175 bar injection pressure

It has been noticed that the brake specific fuel consumption for diesel is lower than the Jatropha oil operation. However with the advantages of high operating temperature due to presence of oxygen molecule in the Jatropha oil lowers the calorific value, also having low viscosity, low cetane number, and higher surface tensions resulting in all together effect combustion characteristics. It can also be seen that the BSFC of engine with diesel fuel started increasing whenever operating point advances. But in the case of jatropha oil as the fuel injection angle advances from 20^0 BTDC to 24^0 BTDC it show decreasing pattern .When we further advance from 24^0 it shows rising . This is in correlation with BTE trends with change in fuel injection point [20].

5.4. Effect of fuel injection pressure on brake specific fuel consumption

The effect of fuel injection pressure keeping the angle constant has been plotted in Figure 4.

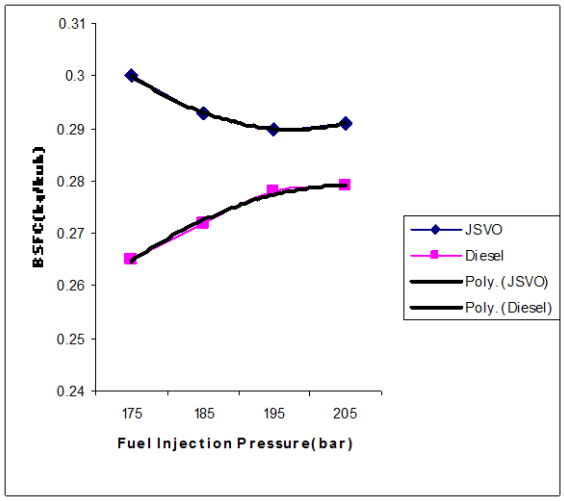

Figure 4: Brake specific fuel consumption vs. injection pressure at speed 1000 rpm, 80% of full load and 20^0 static injection point BTDC

The engine BSFC with diesel fuel get increases continuously when fuel injection pressure increases above 175 bar. But in case of jatropha oil which is higher than diesel at 175 bar initially started decreasing up to 195 bar but further increase in pressure its goes up. Lower viscous fuel breaks in to lighter fuel particle at the end of fuel injection which increases the atomization and better combustion. In contrast higher viscous fuels increase the mixture momentum due to heavier fuel particle size. This reduces the combustion efficiency for jatropha oil than diesel. But in contrast higher combustion

efficiency can be obtained for jatropha oil than diesel engine [10]. The increased mixture momentum and penetration depth is responsible for this improvement, which can be achieved by increasing the fuel injection pressure.

5.5. Effect of fuel injection point on NOx emission

It is important to consider NO_x as one of the major pollutant and emission concerns. When the combustion temperature is decreased the emission of NOx is reduced but the smoke and particulate emission increases. By retarding the injection timing the temperature inside the combustion chamber decreases and therefore a reduction in the NO_x emission can obtained.

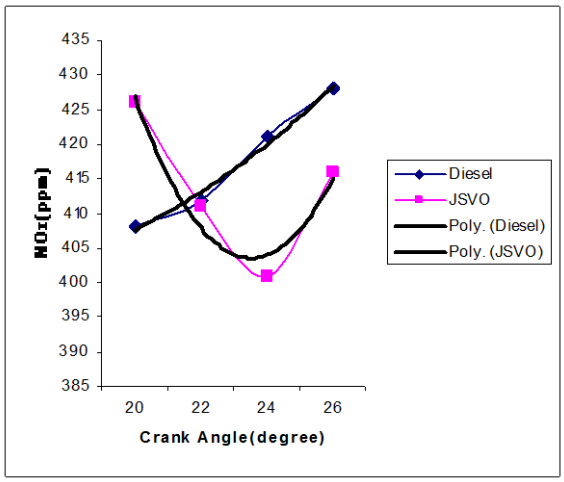

Figure 5: NOx emission vs. crank angle (BDC) at 1000 rpm, 80% of full load and static175 bar injection pressure

Figure 5 represents the predicted trend of fuel injection point variation on NO_x emission keeping fuel injection pressure constant. This trend show the continuous increasing the NOx with diesel fuel as the angle of fuel injection advances from 20^0 BTDC.This increased NO_x emission is due to the increase in peak cylinder pressure and temperature as the combustion occurs earlier in the cycle and more heat is released before and around the top dead centre. The charge elements which burns early in the cycle are subjected to higher temperature and pressure with advance in peak timing and remain at high temperatures for the longer period. These early burn elements contribute most to NO formation and hence higher NO formation rates result with advanced ignition timing. The NO_x emission for jatropha oil decreases with advance angle of fuel injection up to 24^0 BTDC but it started increasing for further increase. This reverse behavior of diesel and jatropha oil in NO_x emission on varying the point of fuel injection may be because the established combustion process in jatropha oil might be still later than diesel fuel. This consequence of late combustion could be having been due to low temperature even little beyond the TDC point. This indicates that combustion efficiency may improve in case of jatropha oil as fuel injection angle advances. The inverse result beyond 24^0 BTDC may be due to possibility of engine knocking and erratic behavior. Predicted trend by this model is very close to experimental analysis done by previous researcher [29].

5.6. Effect of fuel injection pressure on NOx emission

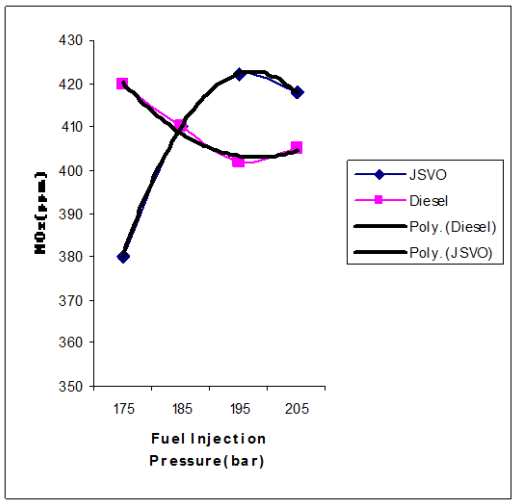

Figure 6: NOxemission vs. injection pressure at speed 1000 rpm, 80% of full load and 20^0 static injection point BTDC

From the Figure 6 it can be predicted that the NO emission in case of diesel fuel increases as the fuel injection pressure increases from 175 bar keeping constant point of fuel injection. But this trend was reversed when the pressure further increases from 195 bar. This may be due to the fact that peak pressure and temperature is very close to TDC or even in some cases after TDC which result in exposing the NO for very small time period to the excess air available for the formation of NOx .The reverse effect on engine beyond the injection pressure 195kgf/cm2 might be due to the possibility of engine knocking and erratic behavior of engine. In case of

jatropha oil it is much more than diesel fuel at 175 bar. But it reduces up to 195bar and then started increasing for further rise in pressure [25].

5.7. Model prediction

A few performance parameters like calorific value, density, cetane number and composition of the fuels were taken into consideration while defining the characteristics of fuel. However, viscosity is also an important property that plays a major role in injection and combustion process. The effect of viscosity and volatility are understood deeply if fuel spray combustion characteristics are considered in details. Further, it was assumed that the cylinder charge is a homogeneous gas mixture of fuel and air and calculation of fuel mass is based on complete combustion of fuel. The relative air fuel ratio was taken 1.5 for both the fuels, although it will vary in real engine situation. It is observed from the model results BTE values for diesel and Jatropha oil at 20^0 BTDC, 1000 rpm and 175 bar pressure are 32.9% and 30.4% respectively. BTE of a single cylinder direct injection diesel engine at 1500 rpm was found 27.4% with neat jatropha oil [28].The present model predicts slightly higher BTE and hence the error is marginal.

5.8. Validation of the model

To validate the model, a comparison is given between the results obtained from the simulation model and the ones obtained from an experimental investigation conducted at U.P.E.S.Dehradun. The comparison between the calculated and experimentally value determined for 1000 r.p.m engine speed and 7.35 kW load examined, is given in Fig. 7.

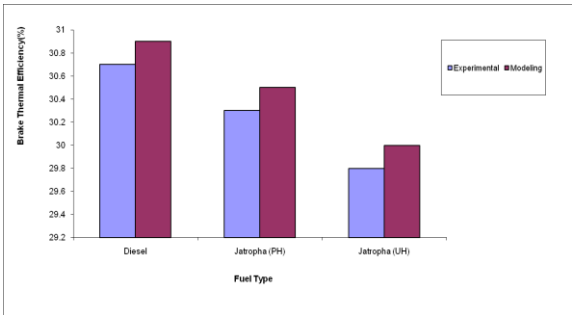

Fig.7 Comparison of brake thermal efficiency

The coincidence between calculated and experimental values is good, verifying the accuracy of the simulation model. The values obtained from the proposed model and the detailed one are practically the same for all test conditions examined. The basic data of the engine are given in Table 2.

Manufacturer	Field Marshal Engine Ltd.,India
Model	FM-4
Engine type	Vertical,4-stroke,singlecylinder,compression-ignition engine
Rated power	7.35 kW at 1000 rpm
Bore/stroke	120/139.7 (mm)
Compression ratio	17:1
Nozzle	DL30S12002MICO
Nozzle holder	9430031264MICO
Fuel pump plunger	9x03/323MICO
Nozzle opening pressure	145 bar
Sump capacity	4.5 litre

Furthermore, in order to validate the model against existing sophisticated models and to find the level of significance in case of diesel fuel BTE, a chi-square test is performed which as shown below.

5.9. Chi-square test

χ^2 test enables us to ascertain how well the theoretical distribution fit in to empirical distribution. If the calculated value of χ^2 is less than the tabular value at a specified level (generally 5%) of significance, the fit is considered to be good. If the calculated value of

χ^2 is greater than the tabular value, the fit is considered to be poor.

To calculate the value of χ^2

Observed value(O_i) %	30 .7	29.8	27.5	28.3	26.2
Expected value(E_i) %	31 .7	28.5	30.8	27.5	27.5
(O_i-E_i)	-1	1.3	-3.3	0.8	-1.3
$(O_i-E_i)^2$	1	1.69	10.89	0.64	1.69
$(Oi-Ei)^2/Ei$	0. 03	0.059	0.35	0.023	0.061

$\chi^2 = \Sigma [(O_i-E_i)^2 / E_i] = 0.523$

Tabular value of χ^2 at 5% level of significance for n-1 =4 is 7.815.

So $\chi^2_{0.05} = 7.815$ .Since the calculated value of χ^2 is less than that of the tabulated value. So theoretical results support the experimental results.

Manufacturer	Field Marshal Engine Ltd.,India
Model	FM-4
Engine type	Vertical,4-stroke,singlecylinder,compression-ignition engine
Rated power	7.35 kW at 1000 rpm
Bore/stroke	120/139.7 (mm)
Compression ratio	17:1
Nozzle	DL30S12002MICO
Nozzle holder	9430031264MICO
Fuel pump plunger	9x03/323MICO
Nozzle opening pressure	145 bar
Sump capacity	4.5 litre

Table 2. Engine specifications

6. Conclusion

A thermodynamic model was developed for analyzing the combustion and performance characteristics of C.I engine. The model has been developed in such a way that it can be used for characterizing any hydrocarbon fueled engines like diesel, vegetable oil or their blends. This predictive modeling results show that with advancing the crank angle and increasing the fuel injection pressure the break thermal efficiency decreases and break specific fuel consumption increases for diesel fuel and reverse

behavior was shown in case of jatropha oil up to a limit. The efficiency of engine with jatropha oil was improved from 31.0% to 34.8% which is 1.8 % higher than the diesel fuel. The BSFC was reduced from 0.30 kg/kWh to 0.26 kg/kWh.The predicted engine power output values showed a good agreement with the corresponding experimental data.

Nomenclature

A_n cross sectional area of nozzle, m^2
d cylinder bore, m
k thermal conductivity, W/m K
L connecting rod length, m
l stroke length, m
m mass of the mixture (air + fuel) contained in the combustion chamber, kgs
N engine speed, rpm
P cylinder pressure, pascal
Q heat energy, joule
R universal gas constant, kJ/molK
Re Reynolds's number
T temperature, K
V instantaneous cylinder volume, m^3
V_p mean piston speed, m/s
V_θ volume at any crank angle, m^3
V_c clearance volume, m3

Greek

θ crank angle ,degrees
$\Delta\theta$ computational step, ^oCA

ΔP pressure difference, pascal

ρ density, kg/m^3

¢ fuel air equivalence ratio, dimensionless
ν kinematic viscosity, m^2/sec
σ surface tension of fuel, N/m
γ specific heat ratio, dimensionless
μ dynamic viscosity, kg/m s

Subscripts

a air
b burned zone
c carbon
com combustion
f fuel
h hydrogen
inj fuel injection
min minimum
n species number, or nozzle hole
o oxygen
p piston
u unburned zone
w wall
0 initial value
1 state at the beginning of time step
2 state at the end of time step

Abbreviations

BTE brake thermal efficiency

BSFC brake specific fuel consumption (kg/kWh)

BTDC before top dead center

CN cetane number

EVO exhaust valve opening

EES engineering equation solver

IVC inlet valve closing

NO nitric oxide (ppm)

PM particulate matter

PAHs polycyclic aromatic hydrocarbons

rpm revolutions per minute

SMD sauter mean diameter (m)

TDC top dead center

JSVO jatropha straight vegetable oil

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