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# Performance Characteristics of CI Engine Fuelled with Biodiesel and its Blends by Simulation

Sanjay Patil, Dr. M.M.Akarte

**Abstract—**In order to overcome the scarcity of hydrocarbon and its depletion, bio-diesel is used as a substitute for diesel or with its blend. In this paper an effort is made to develop a zero dimensional single zone thermodynamic model for compression ignition (CI) engine cycle simulation. Single Wiebe function is used to fit the experimental Mass Fraction Burned (MFB) profile to model heat released due to combustion. Heat transfer and variable specific heats are also considered. The authors of this paper have used a model to compute the performance of the CI engine powered by diesel, palm oil methyl ester (POME) and it blends with diesel. The affect of compression ratio and relative air-fuel ratio on the engine performance is analyzed. The predicted performance parameters like cumulative heat release for diesel and brake thermal efficiency with diesel, POME and various blends at 17.5:1 compression ratio are validated by comparing them with experimental results.

**Index Terms—** thermodynamic modeling, simulation, biodiesel, diesel engine.

## 1 INTRODUCTION

ENERGY is prominent requirement of present society. Internal combustion engines have been the prime movers for generating power for various applications for more than a century [1].The increasing demand, depletion and price of the petroleum prompted extensive research worldwide on alternative energy sources for internal combustion engines. Use of straight vegetable oils in compression ignition engine for long term deteriorates the engine performance and is mainly because of higher viscosity [2-6]. The best way to use vegetable oils as fuel in compression ignition engines is to convert it into biodiesel [7].Biodiesels such as rape seed, soybean, sunflower and Jatropha, etc. are popular substitutes for diesel [8]. In the present energy scenario efforts are being focused on use of bio diesel in compression ignition engine, but there are many issues related to performance and emission [8]. The optimum operating parameters can be determined using experimental techniques but experimental procedure will be time consuming and expensive [9]. Computer simulation [10] serves as a tool for a better understanding of the variables involved and also helps in optimizing the engine design for a particular application thereby reducing cost and time. The simulation approach allows examining the effects of various parameters and reduces the need for complex experimental analysis of the engine [11]. A validated simulation model could be a very useful tool to study engines running with new type of fuels.

The theoretical models used in the case of internal combustion engines can be classified into two main groups viz., thermodynamic models and fluid dynamic models. Thermodynamic models are mainly based on the first law of thermodynamics and are used to analyze the performance characteristics of engines. The pressure, temperature and other required properties are evaluated with respect to crank angle or with respect to time. On the other hand fluid dynamics models are also applied for the simulation of combustion process. These governing equations in these models are partial differential equations which deal with spatial distribution of the gas velocity, temperature and composition within the engine cylinder. The solutions for those equations are tedious. Large computer memory and more computational time is needed to solve the partial differential equations. A zero-dimensional single-zone model as compared with multi-zone models is much simpler, quicker and easier to run. [12, 13] and it is capable of predicting engine performance and fuel economy accurately with a high computational efficiency [14]. Hence a zero-dimensional single-zone model is developed based on first law of thermodynamics and programmed in MATLAB to predict the engine performance for varying blend, compression ratio and relative air-fuel ratio.


- *Sanjay patil, Guru Nanak Dev Engineering College, Bidar- India sanjaypatil1021@yahoo.co.in*
- *MMAkarte, National Institute of Industrial Engineering Mumbai-India. mmakarte@rediffmail.com*


## 2 DESCRIPTION OF MATHEMATICAL MODELING

### 2.1 List of symbols

$r$ = compression ratio.

$L$ = length of connecting rod (mm).

$B$ = bore diameter (mm).

$V_{disp}$ = displacement volume ($m^3$).

$\theta$ = angular displacement in degrees with respect to bottom dead center (BDC).

$\theta_s$ = crank angle at the start of combustion.

$\gamma$ = specific heat ratio.

$P$ = pressure (bar).

$V$ = volume (m3).

$m_c$ = number of moles of carbon in one mole of fuel.

$m_h$ = number of moles of hydrogen in one mole of fuel.

$m_o$ = number of moles of oxygen in one mole of fuel.

$m$ = mass of the charge (kg).

$h_c$ = coefficient of heat transfer due to convection (W/m2.K).

$A$ = interior surface area of cylinder (m2).

$T$ = instantaneous gas temperature (Kelvin).

$T_w$ = cylinder wall temperature (Kelvin).

$R$ = universal gas constant (kJ/kmole.kelvin).

$C_m$ = piston mean speed (m/s).

$U$ = internal energy.

$H$ = enthalpy.

$C_P$ = specific heat at constant pressure (kJ/kg.kelvin).

$C_V$ = specific heat at constant volume (kJ/kg.kelvin).

$n$ = wiebe's shape factor.

$\Delta\theta$ = combustion duration in crank angle (degrees).

$Q_r$ = heat released per cycle (kJ).

$\frac{dQ_r}{d\theta}$ = rate of heat released during combustion (kJ/°CA).

$\frac{dQ_h}{d\theta}$ = rate of heat transfer (kJ/°CA).

$\frac{dw}{d\theta}$ = rate of work done.

$\frac{du}{d\theta}$ = rate of change of internal energy.

$\frac{dV}{d\theta}$ = incremental change in cylinder volume (m3/°CA).

$\frac{dT}{d\theta}$ = rate of temperature change (Kelvin /°CA).

### 2.2 Energy balance equation

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

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

The heat term (rate of heat release) can be split into the heat released due to combustion of the fuel and the heat transfer that occurs to the cylinder walls or from the cylinder walls to gases. The equation (1) can be written as

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

Replacing the work transfer by $p\frac{dV}{d\theta}$ or by the ideal gas law $PV = mRT$, $\frac{dV}{d\theta}$, rate of heat transfer by $h_c = A(T - T_w)$ and the internal energy can be related to specific heat through the relationship

$$\frac{du}{d\theta} = C_V\frac{dT}{d\theta}$$

Upon simplification we get equation (2) as

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

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

### 2.3 Cylinder volume at any crank angle

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

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

### 2.4 Compression and Expansion strokes

The compression stroke starts from the moment the inlet valve closes (IVC) to the moment the fuel injection starts. The expansion stroke starts from the moment combustion ends to the moment the exhaust valve opens (EVO).During these processes the temperature and pressure at each step are calculated using ideal gas equation and an isentropic process [15].

$$P_{i+1} = P_i\left(\frac{V_i}{V_{i+1}}\right)^{\gamma} \quad (5)$$

$$T_{i+1} = T_i\left(\frac{V_i}{V_{i+1}}\right)^{\gamma-1} \quad (6)$$

$$\gamma = \frac{C_p}{C_V} \quad (7)$$

Equations 5 and 6 are used to calculate the pressure and temperature at each cylinder volume. The $C_p$ and $C_v$ are specific heat at constant pressure and specific heat at constant volume of the charge are calculated based on charge composition and temperature using polynomial equation.

$$C_p = \frac{\sum N_i C_{p_i}}{\sum N_i} \quad (8)$$

Where $N_i$ is the number of moles of any gases CO (carbon monoxide), $CO_2$ (carbon dioxide), $H_2O$ (water), $N_2$ (nitrogen) and $O_2$ (oxygen).

$$C_V = C_p - R \quad (9)$$

### 2.5 Combustion Process

Heat release due to combustion is calculated by using single Weibe's heat release correlation [16, 17].

$$\frac{dQ_r}{d\theta} = a(n+1)\left(\frac{Q_r}{\Delta\theta}\right)^n \left(\frac{\theta-\theta_S}{\Delta\theta}\right)^n \exp\left[-a\left(\frac{\theta-\theta_S}{\Delta\theta}\right)^{n+1}\right] \quad (10)$$

Where a and n are wiebe constants. These constants are different at every operating condition and they have to be such that the simulated Mass Fraction Burned profile (MFB) matches closely with experimental MFB profile. These constants are obtained by fitting the Weibe's heat release correlation to the experimental MFB profile by least square method. Prior knowledge of actual overall equivalence ratio is necessary because the fuel/air equivalence ratio depends on the amount of fuel injected inside the cylinder, from which the mass of fuel admitted can be calculated [18].

### 2.6 Heat transfer

The convective heat transfer between gases and cylinder wall is considerable and hence it directly affects the engine performance. The convection heat transfer in kJ/degree crank angle is given by

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

Where heat transfer coefficient due to convection $(h_c)$ is given by Hohenberg equation [19].

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

### 2.7 Ignition delay

An empirical formula, developed by Hardenberg and Hase [20] is used for predicting Ignition delay in crank angle degrees.

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

Where $ID$ = ignition delay period.

$E_A$ is apparent activation energy

### 2.8 Gas properties calculation

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

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

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

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

The gaseous mixture properties like internal energy $(U)$, enthalpy $(H)$ specific heats at constant pressure $(C_P)$ and constant volume ($C_v$) depend on the chemical composition of the reactant mixture, pressure, temperature and combustion process and can be calculated using following equations.

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

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

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

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

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

### 2.9 Friction losses

Total friction loss calculated by the equation [21].

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

Where FP is total friction power loss and C is a constant, which depends on the engine type,

$C$ = 75 kPa for direct injection engine.

## 3 METHODOLOGY

### 3.1 Simulation

A thermodynamic model based on the First law of thermodynamics has been developed. Simulation program is made to model the compression, combustion and expansion processes as closed cycle period and the exhaust and intake processes by the gas exchange processes using the control volume approach. Single Wiebe function is used to fit the experimental mass fraction burned profile.

The molecular formula of diesel fuel is taken as $C_{10}$ $H_{22}$ and biodiesel is approximated as $C_{19}H_{34}O_2$. A computer program in matlab has been developed for the numerical solution of the

equations used in the thermodynamic model described in Section 2. This computes pressure, temperature, mass fraction burned and the performance parameters like brake thermal efficiency, peak pressure and MFB profile for the fuels considered for analysis. Fuels considered for analysis are diesel, blend of 40% biodiesel & 60% diesel and biodiesel (B0, B40 and B100).

### 3.2 Experimental

A stationary single cylinder, 4 stroke, water cooled diesel engine developing 5.2 KW at 1500 rpm is used for investigation. The technical specifications of the engine are given in Table 1. The fuel properties are determined using standard procedure and tabulated in table 2. The author [22] has used the experimental setup to collect data over the various blends

TABLE1. SPECIFICATIONS OF ENGINE

| Sl. No | Parameter | Specification |
|---|---|---|
| 1 | Type | Four stroke DI single cylinder diesel |
| 2 | Software used | Engine soft |
| 3 | Nozzle opening | 200 bar |
| 4 | Rated power | 5.2 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 |

TABLE.2. PROPERTIES OF DIESEL AND POME

| Sl. No. | Properties | Diesel | POME(B100) |
|---|---|---|---|
| 1 | Viscosity in cst at(30°C) | 4.25 | 4.7 |
| 2 | Flash point(°C) | 79 | 190 |
| 3 | Fire point(°C) | 85 | 210 |
| 4 | Carbon residue (%) | 0.1 | 0.64 |
| 5 | Calorific value(kj/ kg) | 42000 | 36000 |
| 6 | Specific gravity(at25°C) | 0.830 | 0.880 |

as detailed in table 3: below.

TABLE 3. DETAILS OF BLEND

| | |
|---|---|
| B0 | 100% diesel |
| B20 | Blend of 20% POME and 80% diesel |
| B40 | Blend of 40% POME and 60% diesel |
| B60 | Blend of 60% POME and 40% diesel |
| B80 | Blend of 80% POME and 20% diesel |
| B100 | 100% POME |

The cylinder pressure data is recorded by using piezoelectric transducer for 80 cycles. The average of data for 80 cycles is computed to evaluate mass fraction burned profile and combustion duration within the framework of first law of thermodynamics.

## 4 RESULTS AND DISCUSSION

### 4.1 Effect of compression ratio on

#### 4.1.1 Peak pressure

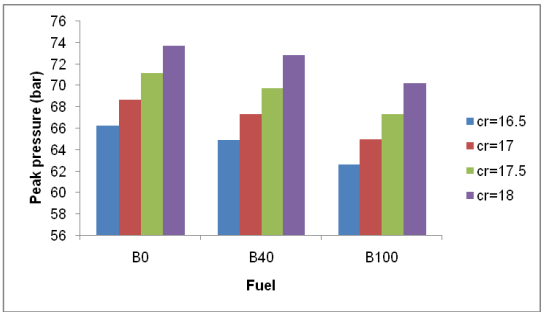


Fig.1. Variation in Brake Peak Pressure

Figure.1 shows variation in peak pressure with respect to compression ratio for different test fuels. With increase in compression ratio the air-fuel mixture compressed to high pressure and temperature which results in better combustion and hence high peak pressure. The above figure shows the same trend for all test fuels. The peak pressure is reduced as the percentage of biodiesel in the blend is increased; this is due to diffusion phase of combustion is predominant as compared to diesel and lower calorific value of biodiesel.

#### 4.1.2 Brake thermal efficiency

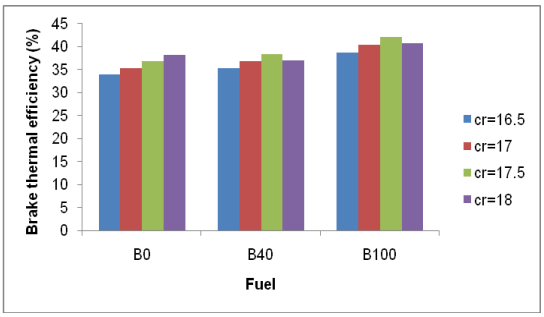


Fig.2. comparison of brake thermal efficiency

Fig. 2 shows the comparison of brake thermal efficiency with respect to compression ratios for different test fuels. With increasing compression ratio the brake thermal efficiency is increased for all the test fuels. It is observed that the brake thermal efficiency is higher with B100 than that of diesel fueled engine due to the presence of oxygen molecule in the biodiesel enhances combustion.

### 4.2 Effect of Relative air-fuel ratio on

#### 4.2.1 Peak Pressure

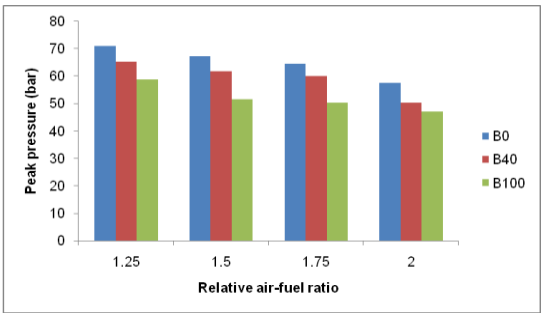


Fig.3. comparison of peak pressure

Fig. 3 shows the comparison of peak pressure with respect to relative air –fuel ratio for different test fuels. Lean mixture results in less heat release and less peak pressure whereas rich mixture results in higher peak pressure. The peak pressure is decreasing as relative air-fuel ratio is increasing. Lean mixture dilutes the charge and hence lower temperature and pressure are achieved at higher relative air-fuel ratios.

### 4.2.2 Brake thermal efficiency

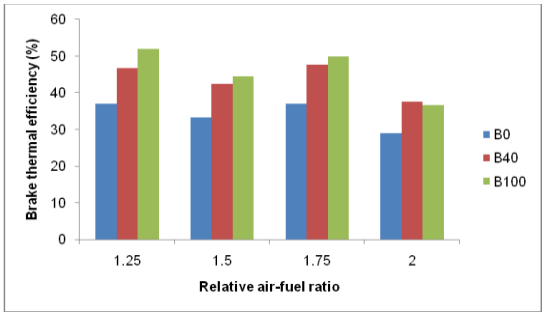


Fig.4. comparison of brake thermal efficiency

Fig. 4 shows the comparison of brake thermal efficiency with respect to relative air –fuel ratio for different test fuels. It can be observed from the fig.6 that the increase in relative air-fuel ratio dilutes the concentration of the charge and in complete combustion hence reduces the brake thermal efficiency. The same trend is observed for all the test fuels.

## 5 MODEL VALIDATION

With the help of developed model theoretical results are predicted for brake thermal efficiency for all test fuels and cumulative heat released for diesel at full load. The same are compared with that of experimental results on the basis of [22] for validation. The figures below highlight the features.

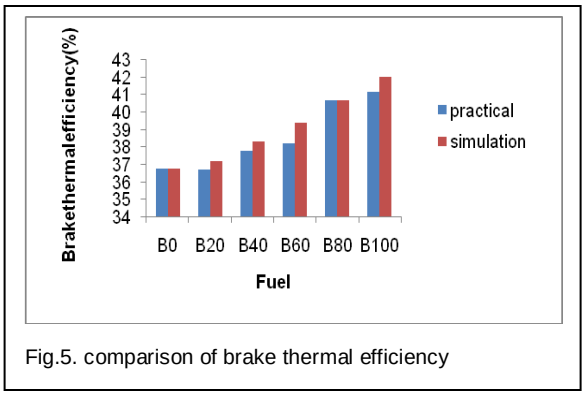


Fig.5. comparison of brake thermal efficiency

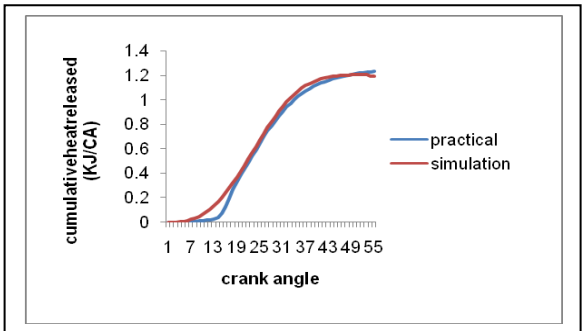


Fig.6. comparison of mass fraction burned

Predicted full load brake thermal efficiency when engine is fuelled with B0, B20, B40, B60, B80 and B100 and mass fraction burned profile for diesel are compared with experimental results are found in closer approximation and hence the developed simulation model has been proved to be reliable and adequate for the proposed objectives.

## 6 CONCLUSIONS

A thermodynamic model is developed for analyzing the performance characteristics of the compression ignition engine. The model is developed in such way that it can be used for characterizing any hydrocarbon-fueled engines, viz., diesel or biodiesel or their blends. The modeling results showed that, with increase in compression ratio peak pressure and brake thermal efficiency are increased. With increase in relative air-fuel ratio, the above parameters are decreased. The performance characteristics of the engine follow the same trend for all test fuels. The predicted results are compared with the experimental results of the engine fueled by B0, B40 and B100. This model predicted the engine performance characteristics in closer approximation to that of experimental results. Hence, it is concluded that this model can be used for the prediction of the performance characteristics of the compression ignition engine fueled by any type of hydrocarbon fuel.

## REFRENCES

[1] Jamil Ghojel, Damon Honnery. Heat release model for the combustion of diesel oilemulsions in DI diesel engines. Applied Thermal Engineering 25 (2005)

2072–2085.

[2] O.M. I. Nwafor & G. Rice. Performance of Rapeseed Oil Blends in a Diesel Engine Applied Energy. Vol. 54, No. 4, pp. 345-354, 1996.

[3] Vellguth G. Performance of vegetable oil and their monoesters as fuels for diesel engines. SAE 831358, 1983.

[4] Tadashi, Young. Low carbon build up, low smoke and efficient diesel operation with vegetable oil by conversion to monoesters and blending of diesel or alcohols. SAE 841161, 1984.

[5] Recep Altim, Selim C etinkaya , Huseyin Serdar , Yucesu, The potential of using vegetable oil fuels as fuel for diesel engines Energy Conversion and Management 42 (2001) 529-538 diesel engine , Applied Energy 86 (2009) 106–112.

[6] Murugesan, C.Umarani, R.Subramanian, N.Nedunchezhian. Bio-diesel as an alternative fuel for diesel engine- A review. Renewable and Sustainable Energy Reviews 13(2009)653-662.

[7] N.R. Banapurmath, P.G. Tewari, R.S.Hosmath. Performance and emission characteristics of a DI compression ignition engine operated on Honge, Jatropha and sesame oil methyl esters. Renewable Energy 33 (2008) 1982–1988.

[8] T. Ganapathy, K. Murugesan , R.P. Gakkhar, "Performance optimization of Jatropha biodiesel engine model using Taguchi approach" Applied Energy (2009).

[9] T. Ganapathy, K. Murugesan *, R.P. Gakkhar "Performance optimization of Jatropha biodiesel engine model using"Taguchi approach

[10] Ganesan, V., Computer simulation of Compression-Ignition engine processes, University Press(India) Ltd., Hyderabad, India, 2000.

[11] Udarapandian, "Performance and Emission Analysis of Bio Diesel Operated CI Engine" Journal of Engineering, Computing and Architecture Volume 1, Issue 2, 2007.

[12] G.H. Abd Alla, A.A. Soliman, O.A. Badar, M.F. Adb Rabbo, Combustion quasi-two zone predictive model for dual fuel engines, Energy Conversion and Management 42 (2001) 1477–1498.

[13] P.A. Lakshminarayanan, Y.V. Aghav, A.D. Dani, P.S. Mehta, Accurate prediction of the heat release in a modern direct injection diesel engine, Proceedings of the Institute of Mechanical Engineers 216 (2002) 663–675.

[14] (Krieger and Borman, 1966; Foster, 1985; Assanis and Heywood, 1986)

[15] Jamil Ghojel, Damon Honnery. Heat release model for the combustion of diesel oil emulsions in DI diesel engines. Applied Thermal Engineering 25 (2005) 2072–2085.

[16] V. Caika, J. Krammer, R. Tatshi, B. Weissbacher, An integral 1D/3Dworkflow for analysis and optimization of injection parameters of a diesel engine, SAE technical paper, 2001, (2001-28-0004).

[17] Lydia CDS. Simulation of the thermodynamic processes in diesel cycle internal combustion engines. SAE 931899, 1993.

[18] P. Arque`s, La combustion: Inflammation, combustion, pollution, applications, Ellipses, Paris, 2004, ISBN 2-7298-2037-X, p. 304.

[19] Hohenberg GF. Advanced approaches for heat transfer calculations. SAE 790825, 1979.

[20] J.B. Heywood, Internal Combustion Engines Fundamentals, Mc Graw Hill, 1988, ISBN 0-07-100499-8.

[21 Shroff, H. D., Hodgetts, D., Simulation and Optimization of Thermodynamic Processes of DieselEngine, SAE 740194, 1974.

[22] Sanjay Patil, Dr. M.M. Akarte and Nandkishore D. Rao., Experimental Investigation on CI Engine Performance Fueled with Biodiesel and its Blends, International Journal of Engineering Research and Technology Volume 4, Number 3 (2011), pp. 319-327.