International Journal of Science and Research (IJSR)
ISSN (Online): 2319-7064
Index Copernicus Value (2013): 6.14 | Impact Factor (2013): 4.438

Combustion Modeling of Biodiesel Fuelled Direct Injection CI Engine

Tushar Naitam[1], M. M. Deshmukh[2]

[1]PG Student, Department of Mechanical Engineering, Govt. College of Engineering, Amravati

[2]Associate Professor, Department of Mechanical Engineering, Govt. College of Engineering, Amravati

Abstract: ***Increasing of costly and depleting fossil fuels are prompting researchers to use edible as well as non edible vegetable oils as a promising alternative to petro-diesel fuels. Biodiesel and its blends are considered as suitable and most promising fuel for diesel engine. The properties of biodiesel found similar to that of diesel. A comprehensive computer code using "C" language was developed for CI engine. Experimental investigation is carried out on four stroke, single cylinder direct injection CI engine. In this analysis 10% to 40% raw oil (derived from Karanja seeds) blended with diesel and used as working fuel. Combustion characteristics such as cylinder pressure, temperature and heat release rate were analyzed. On the basis of the first law of thermodynamics the properties at each degree crank angle was calculated. Suitable correlations and models are used to find out the combustion parameters. Wiebe model is used to predict the instantaneous heat release rate. The predicted results are found satisfactory with the experimental result.***

Keywords: Transesterification, CI engine, Numerical modeling, Combustion parameters, Simulation**.**

1. Introduction

The petroleum fuels fulfil our energy needs in transportation, agriculture, industrial sector and many other basic requirements. There are very limited resources of these petroleum fuels because of their excessive use. Also researchers are dealing with the problems of pollutant emission like CO2, HC, NOx, etc. from these petroleum fuels. Now they are using alternative fuel like biodiesel in CI engine. Biodiesel are having higher viscosity and density but having low calorific value as compared to diesel fuel. It can be produced from various edible and non edible vegetables oil or animal fats. Biodiesel is non-toxic, biodegradable and renewable alternative fuel which can be used in diesel engine without any modification [3].

Combustion modelling of CI engine depends on the characteristics of fuels. Modeling is the process of designing the model of real system and carry out the experiment with it to understand the behaviour of the system. Theoretical models are used in case of CI engine to understand the combustion phenomena. This can be divided into two groups; thermodynamics models and fluid dynamics models. Thermodynamics models are based on first law of thermodynamics and are used to analyze the performance characteristics of engines. Cylinder pressure, temperature, brake power, brake specific fuel consumption etc. can be found out by using suitable correlations and empirical equations. This model is further classified into single-zone models and multi-zone models. Multi-zone models are also called as fluid dynamics models. They are based on numerical calculations of mass, momentum, energy and species conversion equations in either two or three dimensions to understand the combustion phenomena. Multi-zone models need detail information and large computational time [16].

2. Transesterification of Oil

Several processes have been developed for the production of biodiesel, among which trasesterification process is widely used.

The formation of methyl esters by transesterification of vegetable oil requires raw oil (Karanja seed oil), 15% of methanol and 5% of KOH on mass basis. However, transesterification is an equilibrium reaction in which excess alcohol is required to derive the reaction with an alcohol in presence of a catalyst to produce methyl esters. Glycerol was produced as a by-product of transesterification reaction. The mixture was heated at temperature of 55ºC to 60ºC and stirred continuously and then allowed to settle under gravity in a separating funnel. Two distinct layers form after gravity settling for 24 hrs. The upper layer was of karanja methyl esters and lower was of glycerol. The lower layer of glycerol was separated out. The separated karanja methyl ester was mixed with some warm water (around 10% volume of ester) to remove the catalyst present in karanja methyl ester and allowed to settle under gravity for another 24 hrs. The catalyst got dissolved in water, which was separated and removed the moisture. The karanja methyl ester was then blended with mineral diesel in various concentrations for preparing biodiesel blends to be used in CI engine for conducting various engine tests [4].

In this study biodiesel is used as an alternate fuel in diesel engine. Prepared biodiesel is mixed with neat diesel in various concentrations (10%, 20%, 30% and 40%) by volume and is termed as BD10, BD20, BD30 and BD40 respectively.

The properties of different biodiesel blends were found out which are as follows;

Table 1: Properties of diesel and biodiesel

Properties	Diesel	B20	KME (B100)
Density [kg/m3]	831	852	875
Viscosity at 40° C [cSt]	2.6	3.1	5.2
Flash point [° C]	55	105	174
Calorific value [MJ/Kg]	42.5	41.8	38.3

3. Test Engine and Experimental Procedure

The experiment was conducted on 4 stroke, single cylinder, direct injection diesel engine. It is integrated with speed sensors, pressure transducers, thermocouples, air flow meters, fuel flow meters and in-line torque meter. The specification of test engine is given is table 2.

Table 2: Specifications of test engine

Make	Comet
Type	4 Stroke, direct injection
Bore	80 mm
Stroke	110 mm
No. of Cylinder	1
Injection Pressure	200 bar
Compression ratio	16:1
Rated power	3.7 KW @ 1500 rpm
Cooling type	Air cooled
Loading type	Eddy current dynamometer

The control panel is attached with the test engine for monitoring the engine operations. This control panel is connected with the computer on which we can visualize the performance and combustion characteristics. The test rig is installed with ICE software for obtaining various curves and results during operation. Calorimeter is provided to find out the heat carried away by exhaust gas. Test engine is equipped with eddy current dynamometer as shown in figure 1.

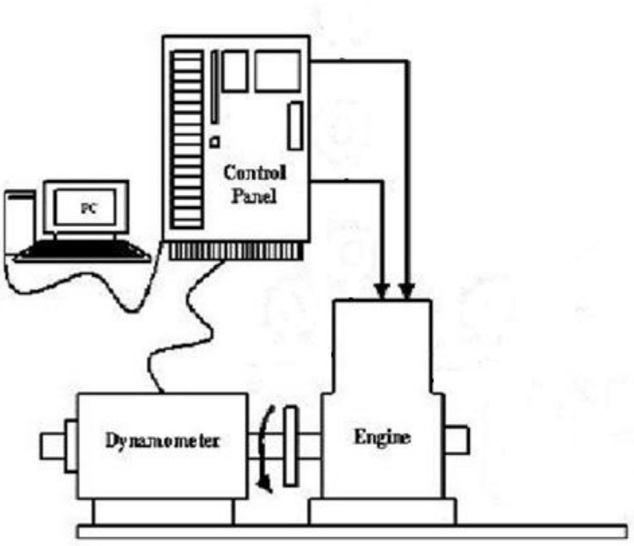


Figure 1: Schematic diagram of Experimental Setup

Initially, the experiment was conducted on pure diesel at constant speed 1500 rpm. The engine was operated for 10 minutes at each load without taking data to stabilize the engine under new condition. This is done to ensure that fuel from previous measurements remaining in flow meter, fuel filter and fuel pipes have been removed. The basic input values of engine parameters and fuels were provided and the experiment is carried out. The results were recorded for variable loads using eddy current dynamometer. The above procedure was repeated at same operating conditions for all the biodiesel blends. The four types of blends (B10, B20, B30 and B40) were used in this experiment.

4. Theoretical Analysis

The present work is deals with the combustion modeling of CI engine. This combustion modeling is carried out on the basis of first law of thermodynamics. Suitable correlations and models are considered to find out the combustion parameters.

4.1 Calculation of number of moles of reactants and products

In this simulation during the start of combustion, the moles of different species are considered includes O2, N2 from intake air and CO2, N2 and O2 from the residual gases. The overall combustion equation considered for the fuel with C-H-O-N is [3];

$$C_xH_yO_z + \lambda * y_{cc} * (O_2 + 3.773N_2) \rightarrow xCO_2 + (y/2)H_2O + (\lambda - 1)y_{cc}O_2 + \lambda y_{cc} * 3.773N_2 \quad (1)$$

Where,
x - No. of moles of carbon.
y - No. of moles of hydrogen.
z - No. of moles of oxygen.
Stoichiometric air fuel ratio $y_{cc} = x + (y/4) - (z/2)$.
λ - Excess air factor.
Total number of reactants and products during the start of combustion as well every degree crank angle was calculated from the equations;

$$tmr = 1 + \lambda * y_{cc} * 4.773 \quad (2)$$

$$tmp = x + (y/4) + 3.773 * \lambda * y_{cc} + (\lambda - 1) * y_{cc} \quad (3)$$

4.2 Volume at Any Crank Angle

The cylinder volume at any crank angle is calculated from given equation [8]:

$$V = Vdisp\left[\frac{r}{r-1} - \frac{1-\cos\theta}{2} + \frac{L}{S} - 0.5\sqrt{\left(\frac{2L}{S}\right)^2 - \sin^2\theta}\right] \quad (4)$$

Where,
r – Compression ratio
L – Connecting rod length
S – Stroke
ϴ – Crank angle position

4.3 Pressure and temperature during compression

The initial pressure and temperature at the beginning of the compression process is calculated as follows [8];

$$P_2 = \left(\frac{V_1}{V_2}\right) * \left(\frac{T_2}{T_1}\right) * P_1 \quad (5)$$

and

$$T_2 = T_1 * \left(\frac{V_1}{V_2}\right)^{\frac{R}{C_v(T_1)}} \quad (6)$$

4.4 Heat transfer model

The gas-wall heat transfer is found out using following formula [3]:

$$\frac{dQ_{ht}}{dt} = h_c A_c \left(T_g - T_w\right) \quad (7)$$

Where,
hc - Heat transfer coefficient (w/m2K)
Ac - Convection heat transfer area (m2)
Tg & Tw are gas and wall temperature respectively (K)

Heat transfer coefficient correlation given by Hohenberg has been used to calculate convective heat transfer [16]:

$$h = 129.8V^{-0.06} P^{0.8} T^{-0.4} (Vp + 1.4)^{0.8} \quad (8)$$

Where,
P - Pressure
T - Temperature
V - Volume of the cylinder
V p - Mean piston speed, V p = 2LN/60

4.5 Combustion model

The combustion of fuel and air is a very complex process, and would require extensive modeling to fully capture. In this work Wiebe model is used which some time is spelled Wiebe function to simulate the combustion process. The Wiebe function is often used as a parameterization of the mass fraction burned and it has the following form [3];

$$x_b(\theta) = 1 - e^{-a\left(\frac{\theta - \theta_{ing}}{\Delta\theta}\right)^{m+1}} \quad (9)$$

and the burn rate is given by its differential form:

$$\frac{dx_{b(\theta)}}{d\theta} = \frac{a(m+1)}{\Delta\theta} * \left(\frac{\theta - \theta_{ing}}{\Delta\theta}\right)^{m} * e^{\left(\frac{\theta - \theta_{ing}}{\Delta\theta}\right)^{m+1}} \quad (10)$$

Where,
ϴ - Crank angle
a – Wiebe form factor
n – Wiebe efficiency factor
ϴi – Start of ignition
Δϴ – Combustion duration

4.6 Ignition delay

Ignition delay in direct injection diesel engines is of great interest to researchers and engineers because of its direct impact on the intensity of heat release rate. The delay period is composed of a physical and chemical delay. An empirical formula developed by Hardenberg and Hase for predicting the duration of the ignition delay in diesel engine. It is given in terms of crank angle [16].

$$\tau = \left(0.36 + 0.22V_p\right) Exp\left[E_a\left(\frac{1}{RT} - \frac{1}{17190}\right)\left(\frac{21.1}{P - 12.4}\right)^{0.63}\right] \quad (11)$$

Where,
V p - Mean piston speed (m/s)
R - Universal gas constant (8.3143 J/mol-K)
Ea - Apparent activation energy
Ea = 618840/ (CN+25)
CN - Cetane number.

5. Results and Discussions

In this study combustion parameters like cylinder pressure, heat release rate, ignition delay and combustion zone temperature are discussed. Simulated curve for ROHR has been tuned with experimental curve for various load conditions to find out the shape parameters of the functions. Correlation for these shape parameters are modified with adjusting coefficients using the least square curve fitting method.

5.1 Cylinder pressure

In CI engine the cylinder pressure depends on the fuel burning rate during the combustion. High cylinder pressure means better combustion and heat release rate. Figure 2 show the cylinder pressure of different blends at different crank angle position. It is observed that peak pressure for diesel is higher than biodiesel blends at all engine loads. This is due to the superior volatility, lower viscosity and carbon percentage of pure diesel. It has been found that the cylinder pressure increases with increasing load for both diesel and biodiesel blends.

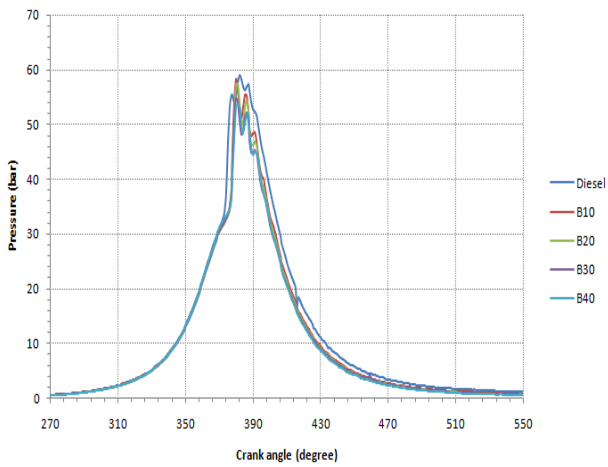


Figure 2: In-cylinder pressure vs. crank angle

Figure 3 show the comparison between experimental and predicted in-cylinder pressure. B20 blend is taken for simulation because it showed better results as compared to other blends. Some modification is required for B30 and B40 for getting similar value like B20. Cylinder pressure can be predicted from predicted heat release rate by adjusting coefficients using the least square curve fitting method. Predicted pressure is observed near about similar to the experimental pressure. The difference in the predicted and experimental values of pressure is because of limitations of the single Weibe model for the heat release prediction.

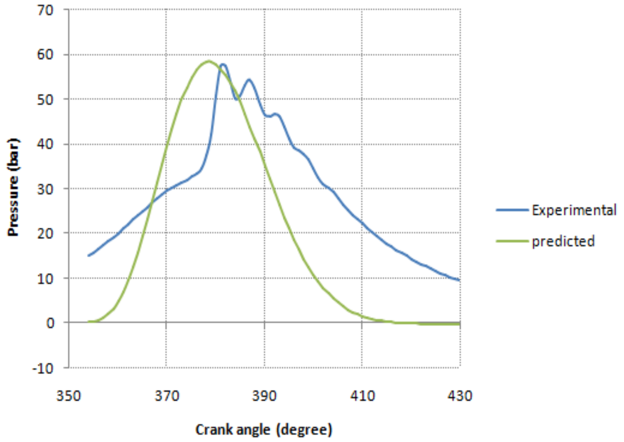


Figure 3: Comparison of experimental and predicted In-cylinder pressure for B20

5.2 Heat release rate

The heat release rate for different blends is shown in figure 4. Heat release rate during pre-mixed combustion is responsible for high peak pressure and high peak pressure means high rate of heat release. Superior volatility of diesel fuel ensures better air fuel mixing relatively higher heat release takes place when piston is at TDC. At lower engine loads, the heat release pattern is similar for diesel and biodiesel. However at higher loads, biodiesel blends showed quite well heat release rate during mixing controlled combustion this is due to the lower volatility and higher viscosity.

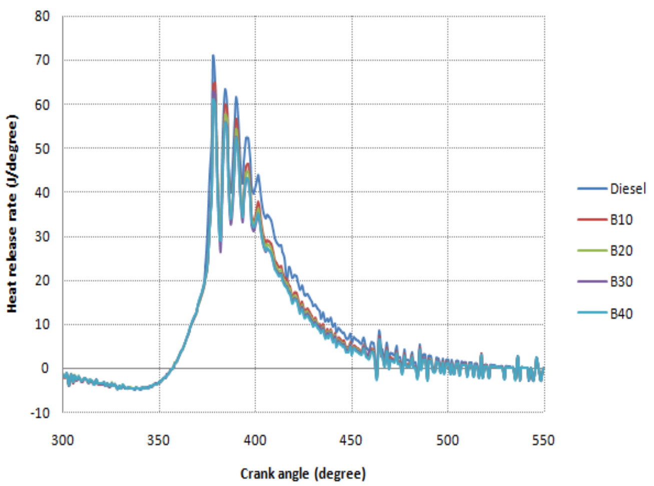


Figure 4: Heat release rate vs. crank angle

Wiebe heat release model is used to predict the heat release rate of engine cylinder. In this model, the Wiebe form factor is adjusted according to the experimental heat release rate so that predicted heat release rate is found near about similar to the experimental heat release rate. Difference in predicted and experimental heat release value is because of some limitations of single Wiebe model. It is observed from the figure 5 that predicted heat release is noted as 63 J/degree.

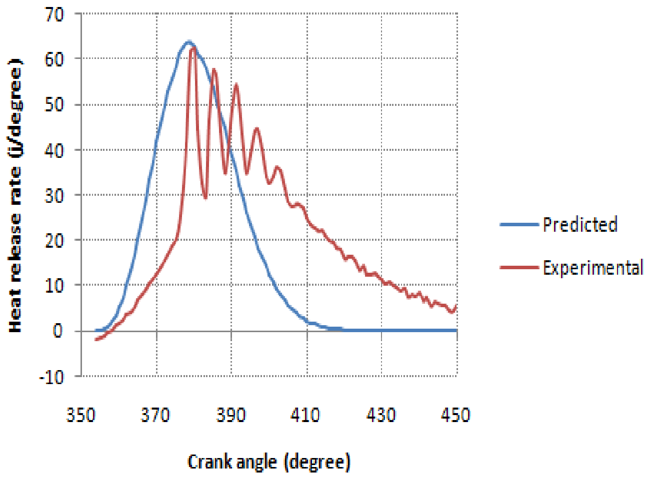


Figure 5: Comparison of experimental and predicted heat release rate

5.3 Heat transfer

Heat loss through cylinder wall occurred via radiation and convection. Hohenberge's correlation is applied to calculate heat loss through cylinder wall. The variation of engine heat transfer with crank angle is shown in figure 6. It has been observed that heat transfer through cylinder wall for diesel is more as compared to B20 due to more heat released. This is due to the high pressure and temperature generated during combustion which results in more heat release rate, hence more heat loss through cylinder wall. Heat transfer is increased at higher loads due to the higher heat release rate.

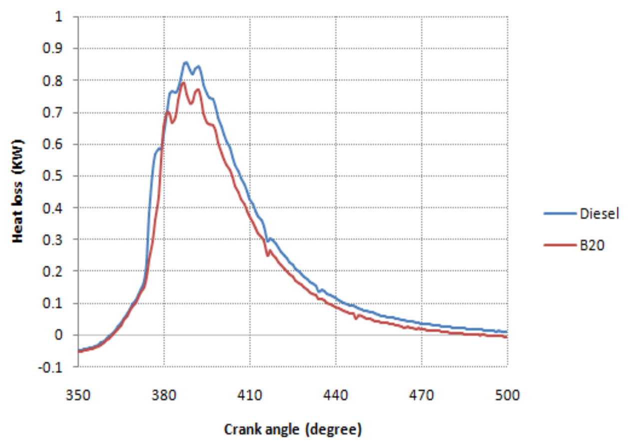


Figure 6: Heat loss vs. crank angle

5.4 In-cylinder temperature

The temperature distribution inside the engine cylinder at different crank angle position is shown in figure 6. High pressure results in high temperature. High peak pressure is obtained for pure diesel as compared to biodiesel. Instead of that, temperature recorded for diesel and biodiesel is near about similar. It occurred because of oxygen molecules in biodiesel which results in better combustion.

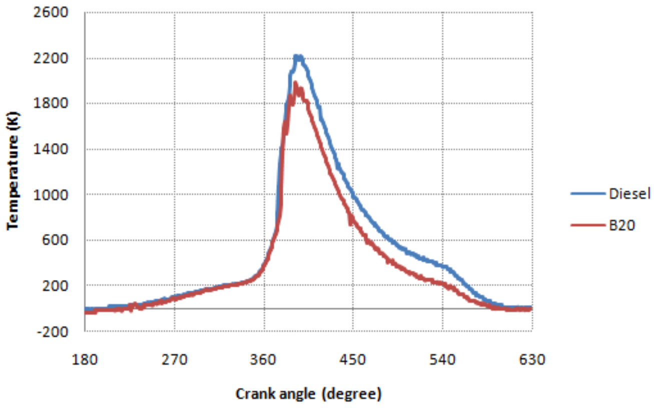
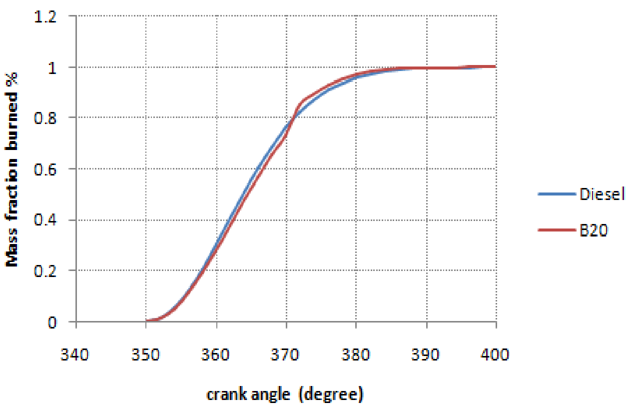


Figure 6: Temperature vs. crank angle

5.5 Mass fraction burned

Figure 7 shows variation of mass fraction burned with crank angle. Mass fraction burned is modelled with Wiebe correlation.



It has been seen that biodiesel takes more time as compared to pure diesel in mixing/atomization due to lower volatility and higher viscosity. So that the burning rate of biodiesel is slower and takes more time. Therefore large amount of fuel needs to be injected taking longer injection time resulting inferior atomization. For high viscous blend, fuel should be injected at high injection pressure for proper air fuel mixing.

6. Conclusion

In the present study experimental calculation and simulation of rate of heat release and pressure have been carried out on the compression ignition (CI) engine fuelled with diesel and biodiesel (20% by mass). The single zone zero dimensional model for closed cycle combustion process has been successfully developed. This model predicted the combustion characteristics in closer approximation to that of experimental results; hence the developed mathematical model is suitable for prediction of combustion characteristics of CI engine. Combustion Characteristics showed smaller ignition delay and slower combustion result in longer combustion duration for biodiesel blends. Biodiesel blend B10 and B20 showed approximately the same results like diesel fuel so that it can be used as alternative fuel for CI engine. B30 and B40 showed low peak pressure and heat release rate as compared to diesel and other blends because of lower volatility, lower heating value and higher viscosity. In summary, karanja oil's higher concentration blends are not suitable as alternate fuels in unmodified engine.

References

[1] V. Gaba, P. Nashine, S. Bhowmick, "Combustion Modeling of Diesel Engine using Bio-Diesel as Secondary Fuel", International Conference on Mechanical and Robotics Engineering (ICMRE), May 26-27, 2012, Phuket.

[2] Lukas Lansky, "Diesel engine modelling and control", Master's thesis, Czech Technical University in Prague Faculty of Electrical Engineering Department of Control Engineering, 2008, pp. 1700-1745.

[3] Mohamed F. Al-Dawody, S. Bhatti, "Theoretical modeling of combustion characteristics and performance parameters of biodiesel in DI diesel engine with variable compression ratio" International Journal Of Energy And Environment, Volume 4, Issue 2, 2013, pp. 231-242.

[4] R. Prasath, P. Porai, Mohd. F. Shabir, "Two-zone modeling of diesel / biodiesel blended fuel operated ceramic coated direct injection diesel engine", International Journal of Energy and Environment, Volume 1, Issue 6, 2010, pp. 1039-1056.

[5] U. Santhan Kumar, K. Ravi Kumar, "Performance, Combustion and Emission Characteristics of Corn oil blended with Diesel", International Journal of Engineering Trends and Technology (IJETT) – Volume 4, Issue 9-Sep 2013.

[6] D. Saravanan, T. Vijaykumar, "Experimental analysis of Combustion and Emission characteristics of CI Engine Powered with Diethyl Ether blended Diesel as Fuel", Research Journal of Engineering Sciences, ISSN 2278–9472, Vol. 1(4), pp. 41–47, October 2012.

[7] M. Shahabuddin, A. Liaquat, H. Masjuki, M. Kalam, M. Mofijur, "Ignition delay, combustion and emission characteristics of diesel engine fuelled with biodiesel", Renewable and sustainable Energy Reviews 21(2013), pp. 623–632.

[8] L. Raut, "Computer Simulation of CI Engine for Diesel and Biodiesel Blends", International Journal of Innovative Technology and Exploring Engineering (IJITEE), ISSN: 2278-3075, Volume-3, Issue-2, July 2013.

[9] A. Agrawal, A. Dhar, "Experimental investigations of performance, emission and combustion characteristics of karanja oil blends fuelled with DICI engine", Renewable Energy, 52 (2013), pp. 283-291.

[10] VenkateswaraRao, Amba Prasad Rao, "Prediction of Heat-Release Patterns for Modeling Diesel Engine Performance and Emissions', International Journal of Advances in Engineering & Technology, ISSN: 2231-1963, March 2012.

[11] Jacob George, "Experimental Investigation and Modelling of a Four Stroke Single Cylinder DI Diesel Engine Under Various Injection Timings", International Journal Of Engineering Science And Technology (IJEST), Vol. 4 No.08 August 2012.

[12] B. Tesfa, R. Mishra, "Combustion Characteristics of CI Engine Running with Biodiesel Blends", International

Conference on Renewable Energies and Power Quality (ICREPQ' 11) Las Palmas de Gran Canaria (Spain), April 13-15, 2011.

[13] M. Venkatraman, G. Devaradjane, "Computer Modeling of a CI Engine for optimization of operating parameters such as Compression ratio, Injection Timing and Injection Pressure for Better Performance and Emission using Diesel-Diesel Biodiesel Blends", American Journal of Applied Science 8 (9): pp. 897-902, 2011.

[14] P. Rao, "Effect of Properties of Karanja methyl ester on combustion and NOx emission s of a diesel engine", Journal of petroleum Technology and Alternative Fuels Vol. 2(5), pp.63-75, May 2011.

[15] Yunus A. Cengel, Michael A. Boles, "Thermodynamics an Engineering Approach", Tata McGraw Hill Edition, 2006, pp. 205-300.

[16] John B. Heywood, "Internal Combustion Engine Fundamentals", Tata McGraw Hill Edition, 2012, pp. 25-90.

Author Profile

Tushar Naitam is PG Student, Department of Mechanical Engineering, Govt. College of Engineering, Amravati (M. S.), India.