Experimental analysis and theoretical validation of C.I. engine performance and combustion parameters using zero-dimensional mathematical model fuelled with biodiesel and diesel blends

Puneet Singh Gautam1,2 Pradeep Kumar Vishnoi1, Peeyush Maheshwari1, Tarun Singh Samant1, V. K. Gupta3

1 Research Scholar, Department of Mechanical Engineering, G. B. Pant University of Agriculture & Technology, Pantnagar, Uttarakhand, India
2 Professor, Department of Mechanical Engineering, G. B. Pant University of Agriculture & Technology, Pantnagar, Uttarakhand, India

Abstract: The present work aims to analyze the experimental and numerical investigation on performance such as brake power (BP), brake specific fuel consumption (BSFC), brake thermal efficiency (BTE), and combustion parameters (in-cylinder pressure, heat release rate) on a 4.4 kW single-cylinder four-stroke C.I. engine running on biodiesel extracted from tamanu oil and diesel. A two-zone, zero-dimensional mathematical model was developed by using thermodynamic laws and cylinder volume, Wiebe’s co-relation, in-cylinder pressure, heat transfer model, engine friction model was developed to validate the above engine performance parameters and analyze the combustion parameters comprising of in-cylinder pressure, in-cylinder temperature, net heat release rate at different loads with single varying-parameter i.e. equivalence ratio while other parameters engine speed, compression ratio, injection timing were kept constant. It was found that the theoretical results were in good agreement with the experimental results for straight diesel and its blends with biodiesel.

Keywords: Biodiesel, Diesel engine combustion, Two-zone combustion model, Heat release rate

Nomenclature

| Symbol | Description                        | Units |
|--------|-----------------------------------|-------|
| A      | Instantaneous heat transfer area   |       |
| B      | Cylinder bore (mm)                |       |
| a      | Crank radius (mm)                 |       |
| hc     | Convective heat transfer (W/m²-K) |       |
| hr     | Radiative heat transfer coefficient (W/m²-K) |       |
| s      | Distance Between Crank, Piston Axes |       |
| Up     | Mean piston speed (m/s)           |       |
| Tw     | Cylinder wall temperature (K)     |       |
| Tb     | Burned temperature (K)            |       |
| l      | Connecting rod length (mm)        |       |
| dP     | Rate of change in pressure        |       |
| dθ     | Rate of change in temperature     |       |
| m_a    | Mass of air                       |       |
| m_f    | Mass of fuel                      |       |
| V_C    | Clearance volume (mm³)            |       |
| V_S    | Swept volume (mm³)                |       |
| W      | Work transfer                     |       |
| χ      | Mass fraction burned              |       |
| η_c    | Combustion efficiency             |       |
| Y      | Specific heat ratio               |       |
| λ      | Excess air coefficient            |       |
| θ      | Crank angle (in degree)           |       |

2 Corresponding author;
Email Id: psgautam51@gmail.com
1 Introduction

Modeling in science and engineering might be for the most part viewed as the way toward depicting the physical phenomena in a specific system with the assistance of mathematical equations (with reasonable assumptions) and having a solution by either discretizing or by any other method to see more about such complex phenomena. As a rule, building models help in designing better devices by seeing more about the complex physical processes happening in that [1,2]. In such a manner, there is great importance in modeling engine combustion processes. zero-dimensional models are the simplest and easy to use and observe the variation of the theoretical results in the engine operating parameters on accumulated heat release rates, cylinder pressure, cylinder temperature, etc. The zero-dimensional models mean that they don’t include the flow field dimensions, rather all the processes are instantaneous. Single zone models could be considered a thermodynamic system in which combustion takes place in the engine which exchanges heat (energy) and mass transfer with the surroundings and the first law of thermodynamics could be applied to determine heat transfer from the system [1]. These models have been approached used in two different directions: Both of these models could be used to anticipate the cylinder pressure with respect to crank angle by following energy release or Wiebe’s mass fraction burned profile. Another utilization with these models could be in determining the heat transfer rate with respect to crank angle by determining in-cylinder pressure data experimentally [1,2]. There has been a large focus by the researcher in recent decades in modeling the engine combustion for designing better engines that produce less pollution [3-9]. In this present investigation first, the traditional model is used because there was no pressure trace device available in the engine set up hence in-cylinder pressure data is predicted with the help of a single-zone mathematical model thereafter the pressure data was used with a two-zone model to further obtain burned and unburned mass and numerical results such as heat release rate power, torque, work done, etc.

Many researchers conducted experiments on mathematical modeling either single-zone or two-zone zero-dimensional models to validate the experimental results or analyzing the effects of engine parameters such as compression ratio, ignition timing (injection pressure, start of injection), equivalence ratio, etc. of which few of them are discussed below.

Patil studied mathematical modeling for the prediction of performance parameters on CI engines running on biodiesel and biodiesel-diesel blends. The predicted BTE and peak pressure were in closer approximation with that of experimental results [8]. Potdukhe et al. studied a modeling and energy analysis, zero-dimensional single zone combustion model simulation to predict the single-cylinder constant speed diesel engine performance. The theoretical model obtained the pressure and temperature variations inside the cylinder by using the combustion correlations which were in closer simulated results with experimental model. The results of the present models were well in agreement with the experimental result [9]. Hariram et al. developed a comprehensive zero-dimensional model to estimate the in-cylinder pressure, the rate of heat release, and rate of pressure rise in a single-cylinder four-stroke CI direct injection engine using conventional diesel and blends of beeswax biodiesel. In this study, the thermodynamic approach was utilized to anticipate the in-cylinder pressure, and Wiebe’s and Wolfer’s relation was applied to investigate the heat release correlations and ignition delay respectively. At no load, the cylinder pressure for conventional diesel was found to be 48 bar and 53 bar by experimental and simulated studies respectively. The heat release rate for conventional diesel, 10% and 20% beeswax biodiesel (BWB10) at 0%, 50%, and 100% load shows an increasing trend between the values of 20 J/°CA and 58 J/°CA which was also supported by the simulated results with a deviation of 7.2%. The rate of pressure rise obtained from simulation of conventional diesel, 10%, and 20%
beeswax biodiesel indicated 8.3% higher as compared to the experimental data throughout all loading conditions [10].

In this present work a two-zone, zero-dimensional mathematical model was developed by applying the first law of thermodynamics and discretizing the same and empirical correlations with a forward difference method and solving the same in matrix laboratory (MATLAB) to validate the performance parameters such as brake specific fuel consumption, brake thermal efficiency with the experimental results running on 4.4kW single cylinder four-stroke C.I. engine. Other important theoretical results such as bulk in-cylinder pressure, in-cylinder temperature, heat release rate, power, torque, work done, etc. at different loads with varying equivalence ratios are analyzed. The engine parameters that was constant like engine speed, compression ratio, injection timing, inlet temperature, and pressure.

2 Materials and methods

In this investigation, biodiesel was obtained from tamanu oil also known as Calophyllum Inophyllum, and blends were prepared with conventional diesel. The investigation of these different blends has been studied by measuring its properties and performing these blends on a 4.4 kW/6HP, single-cylinder four-stroke diesel engine of which specification is shown in Table 2 and schematic diagram of the engine test rig is shown in Fig. 2. As there were no measuring devices available such as pressure transducer, dynamometer, crank angle sensor, a data acquisition device, fuel flow transmitter, airflow transmitter, exhaust gas analyzer, etc. Thus in-cylinder pressure was predicted using a single zone model thereafter heat release rate, power, torque, work done, etc. were analyzed. The blends were prepared once finished biodiesel was obtained with diesel. The blends B20D80 (20% biodiesel and 80% diesel) were prepared and the important fuel properties for the blend were determined and is tabulated in Table 1.

2.1 Mathematical Modelling of Engine Combustion

In this investigation mathematical modeling of zero-dimensional was developed with the help of empirical relation suggested by many authors and researchers [10,11,12] to determine cylinder pressure, cylinder temperature, heat release rate, etc. theoretically that was further utilized after discretizing the empirical relations to solve it through MATLAB and validate the results obtained experimentally.

2.1.1 Cylinder volume

The fundamental geometry of the reciprocating piston engine is shown in Fig. 1. Using the engine kinematics model volume every crank angle (θ) can be calculated by solving [1, 2]

\[ V(\theta) = V_c + \frac{\pi B^2}{4} [l + a - s] \]  

Where, \( V_c \) = clearance volume, \( B \) = cylinder bore, \( l \) = connecting rod length, \( a \) = crank radius, \( s \) = distance between the crank axis and the piston pin axis and is given by

\[ s(\theta) = a \cos(\theta) + (l^2 - a^2 \sin(\theta^2))^{1/2} \]

![Fig. 1 An engine kinematics model [12]](image)
Table 1 Fuel Properties of all the Blends

| Fuels samples | Density (gm/cc) @ 15°C | Cloud Point (°C) | Pour Point (°C) | Kinematic viscosity (mm²/s) @ 40°C | Flash Point (°C) | Fire Point (°C) | Calorific Value (MJ/kg) |
|---------------|-------------------------|------------------|-----------------|-----------------------------------|-----------------|-----------------|-------------------------|
| ASTM standards | 0.860-0.900             | -3 to 12         | -15 to 10       | 1.9-6.0                           | -               | -               | -                       |
| Diesel        | 0.823                   | -2               | -10             | 2.51                              | 60              | 63              | 45.86                   |
| B20D80        | 0.833                   | -2               | -7              | 3.24                              | 68              | 73              | 44.34                   |

Table 2 Specification of the engine test rig

| Engine Characteristic | Specifications |
|-----------------------|----------------|
| Make/Model            | Kirloskar TAFI |
| Maximum Power         | 4.4 kW @ 1500 RPM |
| Injection type        | Direct Injection |
| Number of cylinders   | Single         |
| Cylinder bore/Stroke  | 87.5 /110 mm   |
| Compression ratio     | 17.5.:1        |
| Fuel Injection timing | -24 CAD a TDC |
| Fuel Injection pressure | 170 to 250 bar |

Fig. 2 A schematic diagram of engine set-up

2.1.2 Wiebe co-relation

Single-zone models typically use the Wiebe co-relation to represent the chemical, or gross, energy release as a function of crank angle [3,4]. The Wiebe co-relation has a characteristic “S-shape” and is defined as follows:

\[
\chi(\theta) = 1 - \exp\left[-a(\theta - \theta_i)\right]^{k+1}
\]

(2)

Where \(a\) and \(k\) are adjustable constants (\(k= 5\) and 2 are commonly used), \(a=6.908\) \(b=3\) was used in this case, \(\theta = \) instantaneous crank angle, \(\theta_i = \) start of combustion, and \(\theta_d = \) burn duration. The burn profile is engine-specific, and the constants and \(k\) can be adjusted to tune the profile to a specific engine or application.

The ideal gas law is defined as:

\[
P V = m R T
\]

(3)

Where, \(P=\) pressure of an ideal gas, \(V=\) volume of the gas, \(m=\) mass of the gas, \(R=\) universal gas constant and \(T=\) mean gas temperature

Differentiating equation (3) with respect to \(\theta\)

\[
\frac{dP}{d\theta} = \left(\frac{P}{T}\right)\frac{dT}{d\theta} + \left(\frac{-P}{V}\right)\frac{dV}{d\theta}
\]

(4)

Where, \(P, T, \) and \(V\) are instantaneous values that are modeled with respect to the engine’s crank angle.
2.1.3 In-cylinder temperature

The same process can be applied to the first law of thermodynamics, for the four-stroke engine energy balance [2] which is expressed as:

\[ \Delta U = Q - W \] (5)

Where, \( Q \) = total energy transferred into the system, \( W \) = work transferred out of the system, \( \Delta U \) =is the change in internal energy within the system. In differentiating equation (5) with respect to \( d\theta \), equation (6) can be obtained

\[ \frac{dU}{d\theta} = \frac{dQ}{d\theta} - \frac{dW}{d\theta} = mC_v\frac{dT}{d\theta} \] (6)

Where, \( C_v \) =the specific heat of the combustion chamber gas.

And by applying \( \gamma = \frac{C_p}{C_v} \) and \( C_p - C_v = R \), and \( \frac{dW}{d\theta} = P\frac{dV}{d\theta} \) which gives us the change in temperature as a function of crank angle:

\[ \frac{dT}{d\theta} = T \frac{dQ}{d\theta} - \frac{T}{\gamma - 1} \frac{dV}{d\theta} \] (7)

2.1.4 In-cylinder pressure

Using \( \eta_c \) (the combustion efficiency) and LHV (the lower heating value of the supplied fuel) equation (8) was obtained as

\[ \frac{dQ}{d\theta} = \frac{dQ_{in}}{d\theta} - \frac{dQ_{loss}}{d\theta} \]

\[ \frac{dQ}{d\theta} = (\eta_c \times m_f \times LHV \times \frac{d\chi}{d\theta}) - \frac{dQ_{loss}}{d\theta} \] (8)

Lastly, the change in pressure was obtained by substituting equation (7) into equation (8)

\[ \frac{dP}{d\theta} = \frac{Y - 1}{V} \left( \frac{dQ}{d\theta} \right) - \frac{Y}{V} \frac{P}{V} \frac{dV}{d\theta} \] (9)

The equation (9) is the basis of the model which was used for engine simulation.

2.1.5 Heat transfer model (Annand’s heat release model)

This model incorporates all the processes taking place inside the cylinder for heat transfer calculations, i.e. air motion in the cylinder, fuel spray growth, and mixing sprays impact on the walls, turbulence, droplets evaporation, fuel ignition delay, and combustion process. Annand and Ma 1971 formulated the co-relation for heat transfer [1]

Annand’s heat release model

\[ \frac{dQ_{loss}}{d\theta} = (h_c(\theta) + h_r(\theta))A(\theta)(T_\theta - T_w) \left( \frac{1}{\omega} \right) \] (10)

Where, \( \omega \) = woschni’s factor (60/ (360*RPM), and \( r \) denotes the reference, \( h_c(\theta) \) = convective heat transfer coefficient and \( h_r(\theta) \) = Radiation heat transfer coefficient
2.1.6 Modeling engine friction

Friction losses vary significantly from engine to engine and can be introduced through bearing components and pistons, along with the process of driving engine accessories [2,3,5]. Engine friction losses could be very difficult to model without known engine data and can vary based on engine coolant and oil temperatures, ambient conditions, and engine speed and throttle settings. Heywood suggested that friction mean effective pressure (FMEP) could be approximated for a CI engine as:

\[ FMEP = C + 48 \times \left( \frac{\text{RPM}}{1000} \right) + 0.4 \times U_p^2 \]  

(11)

Where, \( C = 75 \text{kpa}, U_p = \text{Mean piston speed} \)

In choosing a PC program to execute the requirement of a two-zone model, Matrix Laboratory (MATLAB) was considered. It was considered due to its simplicity of reproduction and speed of validation. In light of keeping all limitations add up, the script was created in MATLAB for future utilization. The MATLAB code was set up using the script and all the script is separated into a few subsections. In the first section, the engine geometry and ambience inputs of known engine information sources such as bore, stroke, length of connecting rod, number of cylinders, compression ratio, and other operating parameters such as the timing of opening and closing of the valves in crank angle, the angle at the start of combustion, combustion duration, etc. was mentioned to the program as inputs for the validation. Given the information sources script would determine the area of the cylinder, clearance volume, cross-sectional piston area, and cylinder head piston area. The air inputs were set such as the ambient temperature 300K (room temperature) and atmospheric pressure as 1 atm. It was followed by Fuel input parameters, for example, the mass of the fuel, Calorific value, Lower Heating Values, and air-fuel ratio, equivalence ratio were taken according to the experimental model of the engine. The air-fuel ratio and lower heating value (LHV) to calculate the actual air-fuel ratio. Excess air coefficient or inverse of equivalence ratio was varied from 1.25 to 0.83. In the further section pre-allocation of arrays was done. This kept MATLAB from having to resize arrays or matrices in no. of iteration, reducing the overall computational time and space.

Keeping in mind the end goal that the fundamental program has separated into two sub-loops wherein the main loop with a predefined index (i =0:360, for compression and expansion only) computed instantaneous engine properties as said above. Notwithstanding the properties of immediate properties like volume, pressure, and temperature it has additionally scripted for to ascertain work done in a total cycle, indicated power, friction power, brake power, the correction factor for variable specific heat ratio were used. Variable specific heat ratio model was used from the derivation of the polynomial method.

3 Results and discussion

All the blends were performed on a constant speed 4.4 kW single-cylinder four-stroke of constant speed diesel engine and the engine performance test such as brake specific fuel consumption, brake power, brake thermal efficiency, exhaust gas temperature, and smoke density were determined.

3.1 Modeling Results

All the numerical results showed in this investigation were calculated by varying equivalence ratios at different loads (50%, and 100%) because assuming actual fuel-air and stoichiometry fuel-air ratio as equal that is \( \Phi = 1 \) might or might not fit the experimental results obtained for diesel, biodiesel and diesel-biodiesel blends. Hence for the blends running on biodiesel and diesel require that it is to be observed by varying equivalence ratio (\( \Phi \)) from 0.8 to 1.2 with the other parameters kept as constant which is nothing but the ratio of actual fuel-air ratio and stoichiometry fuel-air ratio for closer
approximation with the experimental results. The method to calculate the molecular weight of selected blends was taken from a reference [12]. Because of the complexity to approximately find the molecular weight and lower heating value (LHV) with the addition of ethanol thus in calculating the equivalence ratio so its ethanol blends were not considered. The experimental results that were validated were Diesel, B20D80.

3.1.1 Validation of brake specific fuel consumption and brake thermal efficiency

**Fig. 3** shows the validated results of brake-specific fuel consumption and brake thermal efficiency versus load (%) that were calculated theoretically using MATLAB. The brake-specific fuel consumption was obtained numerically and the brake thermal efficiency was calculated with varying equivalence ratios from 0.8 to 1.2 to best fit the results obtained experimentally. It was observed that theoretical results were found to be closer to the experimental results by varying the equivalence ratio from 0.8 to 1.2. It was also evident from the figures that the equivalence ratio for diesel fuel was found to be greater than one while as the percentage of biodiesel was increased it was approaching towards the equivalence ratio other than one. That line plotted from the theoretical model for brake specific fuel consumption which is closer to the experimental line has been shown by the bold black dotted line of corresponding equivalence ratio and accordingly for brake thermal efficiency has been shown by bold black dotted line for the same equivalence ratio as from brake specific fuel consumption as shown in **Fig. 4**.

**Fig. 3** Experimental and numerical brake specific fuel consumption with varying equivalence ratio for Diesel, B20D80

3.2 Numerical results

Other results plotted which were obtained numerically included cylinder pressure, cylinder temperature as a function of crank angle. The temperature of burned and unburned masses, heat release rate, torque, power, and work done calculated as a function of the crank angle at different loads (50%, and 100% load).

3.2.1 In-cylinder pressure

**Fig. 5** represents cylinder pressure as a function of the crank angle at various loads with a variation of equivalence ratio. As the equivalence ratio was varied from 0.8 to 1.2 the pressure was also increased at all the loads. At 50% load the peak pressure was 42, 45, 47 and 47.5 bar for \( \phi=0.8, 0.9, 1, 1.1, \) and 1.2 respectively. The same fashion was followed for other loads. The maximum peak pressure of 95 bar was observed at an equivalence ratio of 1.1 at 100% load.
Fig. 4 Experimental and numerical brake thermal efficiency with varying equivalence ratio for Diesel

Fig. 5 Variation of in-cylinder pressure as a function of the crank angle at 50% and 100% load

3.2.2 Heat transfer (kW)

The heat transfer (or cumulative heat transfer) to the wall or heat loss is shown in Fig. 6 as a function of the crank angle at various loads with variation in equivalence ratio. The heat transfer at lower load and equivalence ratio was found to be lower. The heat transfer was increased as the equivalence ratio increased at all the loads. As soon as the compression process ends there was a sudden rise of heat release and reached a peak point during the combustion period due to heat being produced from the fuel thereby started decreasing from the point where the expansion process started taking place. The lowest heat transfer at the peak was 10.5 kW at 50% load at $\Phi=0.8$ and was found to be highest at 100% load as 21 kW.
Fig. 6 Variation of heat transfer as a function of crank angle with variation in equivalence ratio at 50% and 100% load

4 Conclusion

The performance evaluation of diesel engines on selected fuel blends was found acceptable based on brake power, brake specific fuel consumption, and brake thermal efficiency. The validation with the present model showed a closer approximation with the experimental results such as brake specific fuel consumption and brake thermal efficiency even with biodiesel blends with diesel with varying equivalence ratios. Other results obtained such as in-cylinder pressure, heat release rate, burned and unburned temperature, etc. from the two-zone zero-dimensional model showed closer approximation with the practical results. Hence this two-zone model could be used for the prediction of performance characteristics of CI engine when there are fewer data available for any type of hydrocarbon fuels. This model could also be used for optimization purposes such as by varying compression ratio, inlet pressure, inlet temperature, fuel injection timing, etc.

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References

[1] Ganesan V., 2000. “Computer Simulation of Compression Engine” 1st edition.
[2] Heywood, J.B., 1988. Internal combustion engine fundamentals (Vol. 930). New York: Mcgraw-hill.
[3] Blair, G.P., 1999. Design and Simulation of Four-Stroke Engines. Technology, 2011, pp.10-18.
[4] Cuddihy, J.L., 2014. A User-Friendly, Two-Zone Heat Release Model for Predicting Spark-Ignition Engine Performance and Emissions (Doctoral dissertation, University of Idaho).
[5] Hareesh, K., N Teja Rohith, . Reddy, B.Konda. 2014. Computer Simulation of Compression Ignition Engine through MATLAB. International Journal for Research in Applied Science & Engineering Technology (IJRASET), 2: 2321-9653
[6] Krieger, R. B. & Borman, G. L. 1966. “The Computation of Apparent Heat Release for Internal Combustion Engines,” ASME paper 66-WA/DGP-4,
[7] Patil, S., 2013. Thermodynamic modelling for performance analysis of compression ignition engine fuelled with biodiesel and its blends with diesel. *Int. J. Recent Technol. Eng.(IJRTE)*, 1, pp.134-138.

[8] Potdukhe, S.P. and Deshmukh, M.M., 2013. Modeling and Energy Analysis of a Diesel and Biodiesel Fuelled Engine. *International Journal of Science and Research (IJSR)*, 4(5):89-93

[9] Arun, B., 2014. A study on production, performance, and emission analysis of Tamanu oil-diesel blends in a CI engine. *International Journal of Science and Research*, 3, p.2319-7064.

[10] Hariram, V. and Bharathwaaj, R., 2016. Application of zero-dimensional thermodynamic model for predicting combustion parameters of CI engine fuelled with biodiesel-diesel blends. *Alexandria Engineering Journal*, 55(4), pp.3345-3354.

[11] Krieger, R. B. & Borman, G. L. 1966. “The Computation of Apparent Heat Release for Internal Combustion Engines,” ASME paper 66-WA/DGP-4.

[12] Dinh, Thi Luong, Hoang,Vu Nguyen. 2014. Determination of C/H/O fractions and lower heating values for diesel-biodiesel blends derived from Vietnam. *International Journal of Renewable Energy and Environmental Engineering*; 2(3): pp. 224-231.

[13] Bafghi, A.A.T. and Chegeni, F.K., 2014. An Experimental Study on the Equivalence Ratio of Biodiesel and Diesel Fuel Blends in Small Diesel Engine. *Bull. Env. Pharmacol. Life Sci*, 4, pp.40-44.