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ONE-DIMENSIONAL MODELING PREDICTIONS ON THE CHARACTERISTICS OF A CI ENGINE WITH DIESEL AND BIODIESEL BLEND

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Highlights

- D theoretical model for a direct injection compression ignition engine with multiple fuel options.
- Single integrated model for combustion, performance, and emission characteristics of a CI engine.
- Validation of the parameters of the modelled engine with those of the experimental engine.
- Similar trend of variation in characteristics of the modelled engine with the experimental engine.
- Prediction of heat release rate and brake specific fuel consumption with marginal error.

Abstract

The current study focused on utilizing an advanced modeling technique to create a one-dimensional model to analyze the full-cycle calculation of a compression ignition (CI) engine fuelled with a biodiesel blend to explore its performance and emission characteristics. The developed model is to solve the equations that govern conservation of energy and mass, which is used to find the rapid changes of gas exchange and combustion process as a function of crank angle degree. The combustion chamber was created as a one-dimensional model by simulation software, comprising two distinct zones: burned and unburned gases. Heat release and emission predictions were derived using simplified reaction mechanisms. The developed combustion model was integrated as a sub-model within the cylinder element of the one-dimensional solver, enabling the exchange of critical parameters such as cylinder pressure, mass fraction burned, and heat release rate at each crank-angle increment. The work focused on comparing the peak pressure, heat release rate, oxides of nitrogen, smoke, carbon monoxide, and brake specific fuel consumption of the created model with those of the results obtained from the experimental engine. By comparing the theoretical and experimental results, it was inferred that the theoretical model resulted in comparatively higher peak pressure (41%), higher heat release rate (10.38%), lower carbon monoxide (31%), and lower brake specific fuel consumption (4.7%) than that of the experimental engine for the biodiesel blend. The results of the current study showcased the feasibility of utilizing biodiesel in CI engines with better characteristics, which address the sustainable development goals.

Keywords: one-dimensional engine simulation model; biodiesel; diesel engine; emission; combustion.

SCIENTIFIC PAPER

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INTRODUCTION

Diesel engines were utilized by heavy, medium, and commercial vehicles for the transportation of goods and passengers, but the toxic emissions evolved from the engine impact the environment, which urged the researchers to find the most efficient combustion technology to control the emissions. The emission testing methods of diesel engines are more complicated in finding toxic emissions evolved at the outlet. Enhancing the recent technologies developed by the market producers leads to higher costs and time to calibrate the engine model for actual road conditions. Hence, an appropriate theoretical model was required for diesel engine testing methods, which would be favorable to

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the researchers and industries in patenting diesel engines with less pollution. The engine simulation software links the theory of diesel engine combustion behavior with the computer-modeled thermodynamics and also determines the gas combustion conditions within a small interval of time. Modeling an engine involves a wide range of technologies to design an engine using fluid dynamics and finite element analysis. Many automotive manufacturers use their own thermodynamic simulation in their research and development, which makes it easy to follow. The simplicity of engine modeling plays an important role and also saves the expenditure involved in real time testing bed.

Several investigations were carried out to simulate and model a diesel engine to predict the performance, emission, and combustion characteristics of the engine [1]. In a research work carried out in a variable compression ratio (VCR) diesel engine fueled with butanol biodiesel blends reported that the experimental results were similar to those of the theoretical results [2]. Investigations were carried out to compare the computational and experimental results of performance and emission of a diesel engine fueled with soybean-biodiesel blends. The results showed 48.23% lower smoke and 14.65% higher brake specific fuel consumption (BSFC) and a reasonable agreement between the theoretical and experimental results [3]. Ethanol, diesel, and biodiesel fuel blends were tested in a diesel engine, and the results were evaluated by a theoretical method. A marginal deviation of 5% was observed, and increasing biodiesel content in the blend resulted in considerable changes in the combustion characteristics of the engine [4]. The influence of natural gas on the combustion stage was quantitatively analyzed by using computational fluid dynamics (CFD) simulation software, and the same was compared with that of experimental data. It was reported that an excess air fuel ratio was the reason for higher heat release and oxides of nitrogen (NO_x) emissions [5]. Quasi-dimensional was utilized in an engine model to predict the engine characteristics fueled with neat diesel. The results showed that the predicted emissions were 2.5-20% lower than the experimental value [6]. A zero-dimensional multi-zone model was used in a study to evaluate the energy balance in a compression ignition (CI) engine at various conditions. The model was used to find combustion and emission parameters of the engine [7].

In all the earlier investigations, a separate model was developed to predict the performance, emission, and combustion parameters of the engine. It was also reported that developing a single model to predict performance, emission, and combustion parameters of the engine will be challenging [8]. A multi-zone combustion model for different parameters like fuel injection, fuel spray, and in-cylinder gas mixture was developed using the thermodynamic equations, and work was simulated by fueling Ethanol-diesel blends. It was reported that the developed model emits less smoke at all loads [9]. In developing a model, fuel vaporization, ignition delay, initial turbulence, and air swirl were the parameters considered, and the algorithm was coded as a program to find the result on brake thermal efficiency (BTE), NO_x, and carbon monoxide (CO) emissions [10].

In the earlier investigations, a two-zone model was created by employing 3D CFD software to predict soot emission of the engine [11,12]. A one-dimensional model was developed for a two-stroke engine by considering trapping efficiency, scavenging efficiency, and the height of the inlet and exhaust ports of the engine. The developed model was loaded in AVL boost simulation software to obtain BSFC and BTE of the engine [13]. In an investigation with multi cylinder engine, combustion models were developed by using AVL Boost software and optimized the engine load and drive train transmission ratio [14]. The feasibility of hydrogen in a dual fuel mode was investigated through a one-dimensional model, and the engine performance and emission characteristics were predicted through the developed model [15]. It was reported that a single model is necessary to predict the performance, emission, and combustion characteristics of the engine by varying both fuel and design factors of the engine [16-18]. Based on the earlier investigations, it was inferred that different two-dimensional models were created for the CI engine to predict its characteristics. The extensive literature study also exposed a gap between the utilization of renewable fuel and the need for creating a simulation model, offering real-time solutions in the reduction of toxic emissions without compromising engine performance.

To overcome this research gap, an integrated study that addresses the dual challenges of creating a simulation model for a diesel engine and also testing the model fueled with alternative fuels for different design and engine operating conditions is necessary. Two- and three-dimensional models are currently available in the development of a theoretical engine model, and to create a one-dimensional model that addresses the drawbacks of both the mentioned models is the highlight of the current research work.

This study leverages advanced one-dimensional simulation software that allows precise customization of engine components and accurate modeling of biodiesel combustion with minimal calibration. Unlike prior work on theoretical models and various fuel blends, our approach uniquely integrates gas dynamics, spray, and particle simulations for realistic and efficient engine performance analysis. Earlier studies on biodiesel blends (B20-B100) highlight viscosity's impact on fuel atomization and emissions trade-offs. B30 biodiesel blend was employed as the test fuel, consisting of 30% by volume of biodiesel produced from *Jatropha curcas* oil and 70% by volume of conventional petroleum-based diesel. The B30 blend balances these effects well, influencing key combustion metrics. In India, E20 ethanol blends are commonly used with minimal engine modifications, underscoring the relevance of the developed model.

The one-dimensional model used in the present work provides extensive customization capabilities for engine components and processes, enabling precise adaptation to complex engine configurations. Its modular design allows targeted modifications to individual elements without rebuilding the entire model. Additionally, the software

includes comprehensive libraries for alternative fuels, supporting accurate and detailed emissions predictions. The one-dimensional model also offers a multi-cylinder engine configuration for different testing conditions, such as engine modification, exhaust gas recirculation (EGR), and after-treatment process. A one-dimensional model is compatible with any design created by drafting software, and the specific dimensions will be calculated based on the uploaded design. If multiple fuels are used, then the fuel properties and various mixing ratios can be given as input in the fuel tab available in the software. To predict the thermodynamic process during combustion, data such as in-cylinder temperature, carbon, hydrocarbon, and oxygen content can also be modified in the software. Further, the software enables the simulation of turbocharged or supercharged engines with EGR, operating on multi-fuel blends, for analysis of performance and emission characteristics. The proposed model integrates biodiesel-specific properties into combustion sub-models, enhancing predictive accuracy with minimal empirical calibration. The model uses sub-models to maintain accuracy while improving computational efficiency, enabling fast simulations suitable for parametric analysis and real-time engine control.

Hence, to explore the combustion, performance, and emission characteristics of a single cylinder CI engine, a one-dimensional theoretical model was developed. The results were compared between the simulated model and experimental engine to comprehensively analyze the viability of employing biodiesel in diesel engines for enhanced performance and emission characteristics.

MATERIALS AND METHODS

Fundamentals of energy equations for Theoretical engine modeling

The essence of simulation is to determine the engine gas conditions for each crank angle rotation. Pressure, temperature, mass-energy flow, and gas composition are the parameters used to estimate the diesel engine performance in the model. Cylinder volume (CV) is the core in modeling for any equation, and the Euler integration method shown in Eq. (1) was used to predict the CV values [19]. The CV varies for every time step during the combustion that utilizes the energy equation to find the rate of change of temperature and pressure values.

$$P_{n+1} = P_n + \tau \frac{\left[\left(dP/dt\right)_n + \left(dP/dt\right)_{n+1}\right]}{2} \qquad (1)$$

The equations of the perfect gas law and the first law of thermodynamics were employed in finding the diesel engine performance. The theoretical model was developed from the energy equation, which is a function of CV [20], and the same is given in Eq. (2).

$$\frac{d(uM)}{dt} = \sum_I h_i m_i - \frac{dQ}{dt} - P\left(\frac{dV}{dt}\right) \qquad (2)$$

where dQ/dt is the net heat transfer, M is the mass, and P is the pressure, dV/dt is the rate of change of volume, and $h_i m_i$ is the gas work carried out to the surroundings in a CV process. The left-hand side of Eq. (2) shows an expanding process; the simplest expansion assumes to starts when the air-fuel mixture in the CV is homogeneous. The internal energy (u) in Eq. (2) depends upon the gas composition and temperature, and it is usually expressed as air fuel ratio. Eq. (2) becomes a simplified energy equation for a homogeneous mixture of air and burnt fuel at CV, and at higher temperature the dissociation of gas molecule effects can be neglected. The simplified energy equation for modeling a diesel engine is given in Eq. (3) [21].

$$M\left(\frac{\partial u}{\partial T}\right)T + M\left(\frac{\partial u}{\partial \varphi}\right)\varphi + uM = \sum_1 h_i m_i - Q - PV \qquad (3)$$

where T is the temperature, ϕ is the equivalence ratio, Q is the heat transfer, V is the volume, h_i is the specific enthalpy of the ith species, and m is the mass of the ith species.

The gas properties for an engine simulation process obtained from the above equation and the dissociation of molecules are required in the design of computational methods.

The equation requires combustion product species in fractions to determine engine emission and performance. Kreiger and Borman's methods were the two methods used to find the actual combustion behaviour for a wide range of fuels, and it is possible based on JANAF table properties [22,23]. Eq. (4) is used to predict the combustion behaviour.

$$\varepsilon\varphi C_\alpha H_\beta + (0.21O_2 + 0.79N_2) \rightarrow b_1 CO_2 + b_2 H_2O + b_3 N_2 + b_4 O_2 \qquad (4)$$

The theoretical investigation for the diesel engine was carried out on one-dimensional engine simulation software. The software has flow elements (cylinder, duct, injector, etc.), control elements (actuator, sensor, thermocouple, etc.), and components, such as network, silencer, and tags to create an engine model.

Eqs. (1-3) show the relation of mass, energy, and pressure-volume for a diesel engine combustion process, and the simulation solver numerically solves each value for every crank angle movement of the engine.

Eq. (4) is dynamically updated within the simulation solver, which is coupled with the built-in sub-model. The elemental composition, specifically the percentages of carbon, hydrogen, and oxygen, along with the heating values of the diesel and biodiesel blend, were manually specified as input parameters. These fuel properties provide critical inputs for the solver to accurately perform combustion and emission calculations. Emission predictions utilize the extended Zeldovich mechanism for NO_x formation, while simplified reaction mechanisms are employed to estimate CO and HC emissions that approximate partial oxidation and incomplete combustion processes.

Creation of a diesel engine model

In this study, the dimensions of a chosen single-cylinder diesel engine were employed to create a theoretical engine model. Performance emission and combustion behaviour of the theoretical model were obtained from the quasi-steady-state process. This section elaborates on the methodology and significance of various elements employed during the creation of the diesel engine model. Figure 1 shows the flow chart of the creation of a

diesel engine model and the process in an experimental engine.

Ambient flow elements

This component is used to specify the connection to an infinite reservoir for the rated pressure, temperature, and air-fuel compositions. The element represents the ambient condition as well as the specification of the duct element. Atmospheric pressure and temperature were set as 1 bar and 300 K, respectively. To maintain uniformity in the testing conditions across the globe, 300 K was chosen as the standard ambient temperature to develop a theoretical model for the single-cylinder diesel engine. It was inferred that maximum engine efficiency occurs at an ambient temperature around 300 K. The standard temperature reduces the complexity of the simulation and ensures uniformity in aligning thermodynamic equations. Nozzle diameter and discharge coefficient values of the engine were fed as input to the model.

Duct

This element was used to connect the orifice and the ambient, and it describes the flow of air to the inlet condition. In most of the designs, the material used is to damp out the frequencies and dissipate the energy in another form. Many sub-models were utilized to find the temperature distributions over the inlet and outlet ducts. Dimensions of the inlet and outlet ducts of the engine, shown in Table 1, were employed during the model creation.

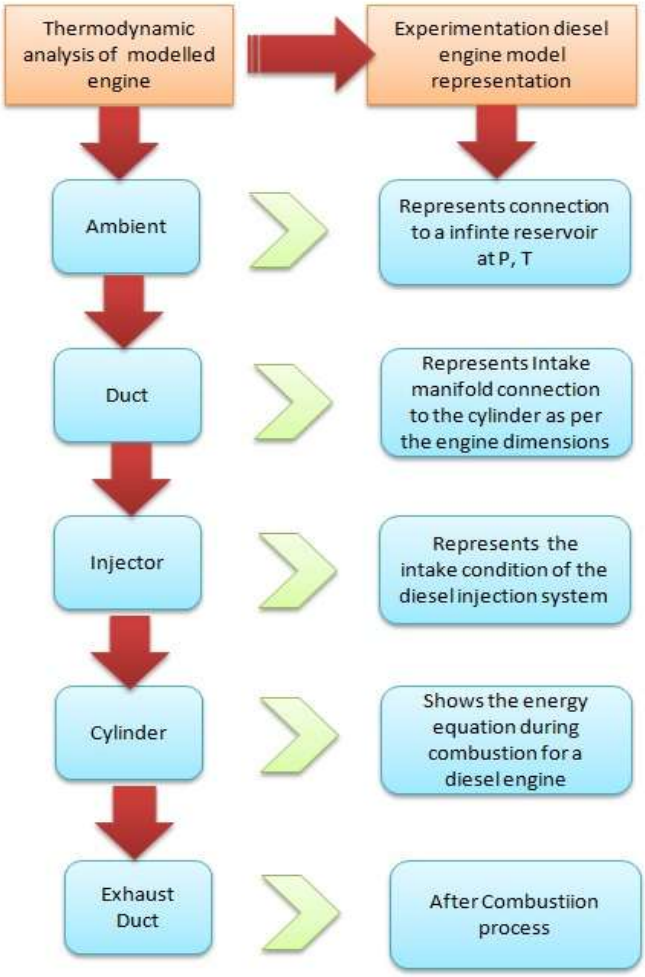

Figure 1. Flow chart of the thermodynamic model in a diesel engine.

Table 1. Duct dimensions of the engine.

Design	Shape	Duct 1 mm	Duct 2 mm	Duct 3 mm	Duct 4 mm
Left diameter	Circular	36	36	28	40
Right diameter	Circular	36	35	40	40
Discretization length	Circular	30	25	25	25
Overall length	Circular	100	50	90	150

Orifice and Injector

The orifice in the flow element was employed to connect the pipes of varying dimensions, which have zero volume and length. The orifice diameter should be less than the connected diameter of the ducts, and the orifice is equipped with a sensor. In the modeling, the injector element was placed at the centre of the cylinder. Fuel-air ratio, start of fuel injection, mixture temperature, nozzle diameter, spray spread angle, and fuel injection pressure were given as input to the software.

Ignition delay

The ignition delay is predicted by using the in-cylinder temperature, pressure, and fuel cetane number. The relationship between these parameters is calculated from the given Eqs. (5-8):

$$\Delta\theta_{delay} = 323\exp^{\min(2100°C/T_{sum},80.0)/P_{sum}} \quad (5)$$

$$C = \frac{67}{(25+\text{Cetane})} \quad (6)$$

$$T_{sum} = \sum_n \frac{T_c^n + T_c^0}{2} * \frac{\Delta\theta_n}{\theta_{n+1} - \theta_o} \quad (7)$$

$$P_{sum} = \sum_n \frac{P_c^n + P_c^0}{2} * \frac{\Delta\theta_n}{\theta_{n+1} - \theta_o} \quad (8)$$

where *cetane* is the user-entered cetane number, T_c^n and P_c^n are the current cylinder temperature and pressure, respectively, T_c^0 and P_c^0 are the cylinder temperature and pressure at the start of injection, respectively, $\Delta\theta_n$ is the time step size, and θ_0 is the crank angle at the start of injection.

Engine

Creation of a single-cylinder diesel engine was initiated by selecting an engine tab in the software. By choosing the appropriate tab provided in the software, the type of combustion (SI or CI) and various emission parameters planned to measure were chosen in the software. In addition to that, various operating parameters were entered into the software. Table 2 shows the specifications of the chosen diesel engine, which has to be modeled, and the same was entered into the software with the help of an appropriate tab provided for that purpose.

Engine cylinder

A cylinder element is normally employed to create the cylinders of a conventional internal combustion engine in the system.

The energy equations, Eqs. (9-12) for the single zone model, are given as follows:

$$\Delta(mu) = \sum_{i=1}^{nvalues} m_i h_i - Q - P\Delta V \quad (9)$$

where Δmu represents the enthalpy flow due to flame propagation to the unburned zone in the combustion process.

In the unburned zone, the energy equation for the two-zone model is given as:

$$m_{u1}u_{u1} - m_{u0}u_{u10} + P(V_{u1} - V_{u0}) + Q_u - \Delta m_{ui}h_{ui} = 0 \quad (10)$$

where m_u is the unburned mixture, u_u is the specific internal energy, V_u is the volume occupied by the unburned zone, m_i is the mass, h_i is the specific enthalpy, Q is the heat transfer, P is the pressure, ΔV is the change in cylinder volume, and i is the index for species

Similarly, the equation for the burned zone is shown below:

$$m_{b1}u_{b1} - m_{b0}u_{b0} + m_{b1}R_{b1}T_{b1} - PV_{b0} + Q_b - \Delta m_{bi}h_{bi} = 0 \quad (11)$$

where m_b is the mass of the burned gases, u_b is the specific internal energy of burned gas, R_b is the gas constant, T_b is the temperature of burned gas, V_b is the volume of burned gas, Q_b is the heat transferred from the burned gas, P is the pressure, and i is the mass flow index.

The total cylinder volume is the sum of the volumes of the burned zone and the unburned zone:

$$m_{u1}R_{u1}T_{u1} + m_{b1}R_{b1}T_{b1} - PV_c = 0 \quad (12)$$

where R_u is the gas constant, T_u is the temperature, and V_c is the clearance volume.

Table 2. Specifications of the experimental engine.

Make and model	Kirloskar, TAF1 make
Number of cylinders	1
Bore, mm	87.5
Stroke, mm	110
Connecting rod length, mm	220
Swept volume, cm^3	661
Clearance volume, cm^3	36.87
Compression ratio	17.5:1
Rated power, kW	4.4
Rated speed, rpm	1500
Injection type	Direct Injection
Injection pressure, bar	210
Fuel injection timing, °CA bTDC	23
Displacement	0.661
Wrist pin offset	0.1
Clearance height, mm	1.5

Heat transfer and combustion model

The diesel engine model was selected for heat transfer and combustion, and the data about opening and closing of the intake valve, piston top surface area, cylinder head surface area, and average surface temperature for the cylinder regions were fed into the software [24]. In addition to that, reference speed, ignition delay, and premix fuel fractions were also entered. Diesel Wiebe combustion implies a non-predictive curve fit-correlation. The reference speed and cetane number were entered in the software to

enable a semi-predictive combustion model to predict the ignition and corresponding changes in the air-fuel mixture conditions and engine speed [25].

Combustion model

The mass fraction burned fuel is determined from the following equation:

$$W = \frac{premix}{p_f\left\{1-\left[1-(0.75\tau)^2\right]^{5000}\right\}} + \frac{diffusion}{d_f\left\{1-\left[1-(c_{d3}\tau)^{1.75}\right]^{5000}\right\}} + \frac{tail}{t_f\left\{1-\left[1-(c_{t3}\tau)^{2.5}\right]^{5000}\right\}} \quad (13)$$

where p_f, d_f, and t_f are the mass fractions of the premix, diffusion, and tail burn fuel, respectively, c_{d3} and c_{t3} are the burn duration coefficients for the diffusion and tail burn fuel, respectively, and τ is the burn duration term.

Input parameters for engine simulation

Fuel parameters for the engine simulation have to be fed into the software. The ideal gas option was chosen to select air as a mixing gas with the chosen fuel. By choosing the ideal gas option, the composition and properties of air will be taken as input for further calculation. For the modeled engine, diesel will be the default fuel as per the software design. At the same time, it will be possible to select a new fuel for the simulation of the modeled engine. Based on the chosen fuels, their composition of carbon, hydrogen, oxygen, and various properties, such as lower heating value and specific heat, were given as input to the software. Table 3 shows the key fuel properties of petroleum diesel and biodiesel blend (B30) employed in this present study. To simulate the modeled engine, the simulation duration in terms of cycles and related angles was fed into the software.

Table 3. Fuel properties of diesel and B30.

S. No.	Properties	Diesel	Biodiesel blend
1	Density (kg/m^3)	830	850
2	Viscosity (mm^2/s)	3.0	3.5
3	Cetane number	51	55
4	Lower heating value (LHV) MJ/kg	45	44
5	Oxygen content (wt%)	<1	2

Estimation of parameters of Theoretical model

By following the procedure explained in the previous sections, a one-dimensional diesel engine model was created, and the layout of the model is shown in Figure 2.

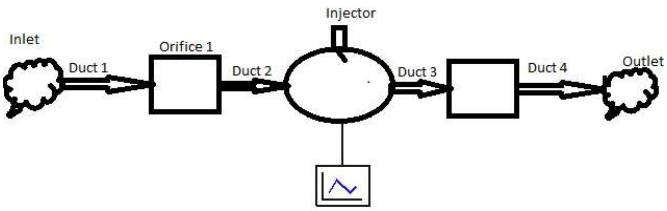

Figure 2. Layout of the created theoretical model for the diesel engine.

Initially, this engine was allowed to run as per the given input parameters, fueled with diesel. The model regenerates the steps for several cycles to estimate the performance, emission, and combustion parameters of the engine. By modifying the fuel properties, the engine was made to run with a biodiesel blend, and various parameters were estimated.

Cylinder Pressure and HRR

During the simulation of the modeled engine, pressure and temperature of the mixture before the start of combustion were estimated using Eq. (2). Data such as mass of fuel burnt, ignition delay, and start of injection were entered in the energy equation to find the cylinder pressure and temperature. A minimum delay period of 3 crank angle degrees was provided in the software, and the same can be increased further. The start of combustion will be estimated by the software for the given start of injection timing along with the corresponding cylinder pressure and temperature. From Eq. (2), the rate of fuel burnt and internal energy with respect to time was estimated. As a result of combustion, pressure, temperature, and equivalence ratio of the products will change, which will be utilized to calculate the appropriate new values of pressure and temperature using Eq. (3) [25]. Eq. (2) is used to find the heat release rate (HRR) by applying the condition that the inlet valve and exhaust valve are closed. The rate of heat transfer for the chosen cylinder geometry and dimensions shall be evaluated from the empirical assumptions. The HRR for the chosen fuel will be obtained from its burning rate and calorific value. With the chosen assumptions and the relationship between the mass and equivalence ratio, different HRR was obtained through integration [26].

Calculation of emissions Parameters

Since the emissions are dependent on temperature, a rich equivalence ratio arrives at the transient condition for diesel. The cylinder volume is calculated according to the energy Eq. (3), and the various emission parameters were obtained from Eq. (4). Based on the concentration of the molecules, the dissociation of fuel particles at higher temperature were calculated and entered to balance the atoms [27,28].

VALIDATION OF MODEL THROUGH EXPERIMENTAL PROGRAMME

To validate the various performance, emission, and combustion parameters obtained from the theoretical investigation, it was planned to compare the parameters of the modeled engine with those of the parameters obtained from the experimental investigation. The experimental investigation was carried out on a chosen diesel engine fueled with a diesel and biodiesel blend. Figure 3 shows the layout of the experimental setup, and the engine is coupled with an electrical dynamometer to load the engine. The engine was tested at a constant speed of 1500 rpm at full load conditions. Initially, the engine was made to run on diesel, and various parameters were recorded. This procedure was repeated when fueled with a biodiesel blend.

An AVL364 angle encoder and AVL GH12D pressure sensor were used to identify the crank angle position and cylinder pressure, respectively. The measured pressure signal was processed through AVL3066A02 charge amplifier and sent to an AVL615 Indimeter for data acquisition. Fuel consumption was measured using a 0.1 g sensitivity electronic balance, and a stopwatch with 0.01 s resolution recorded time. An AVL Di-gas 444 analyzer was employed to measure the exhaust emissions, such as CO and NO_x. Smoke opacity was recorded using an AVL 437C smoke opacimeter. The specifications and measurement uncertainties of the instrumentation are shown in Table 4.

Comparison of Theoretical and Experimental parameters

Performance, emission, and combustion parameters obtained from the modeled engine were compared with those of the experimental engine. While validating, it is necessary to calculate the difference in magnitude between the theoretical investigation and experimental work as an error. Percentage Error is calculated as per Eq. (5), which is the ratio of the difference in magnitude between theoretical and experimental data to the theoretical data.

$$Perentage\ Error = \frac{Theoretical - Experimental}{Theoretical} \times 100 \quad (13)$$

RESULTS AND DISCUSSIONS

In this section, various parameters obtained from the modeled engine fueled with diesel and biodiesel blend were presented by comparing them with those of the experimental engine.

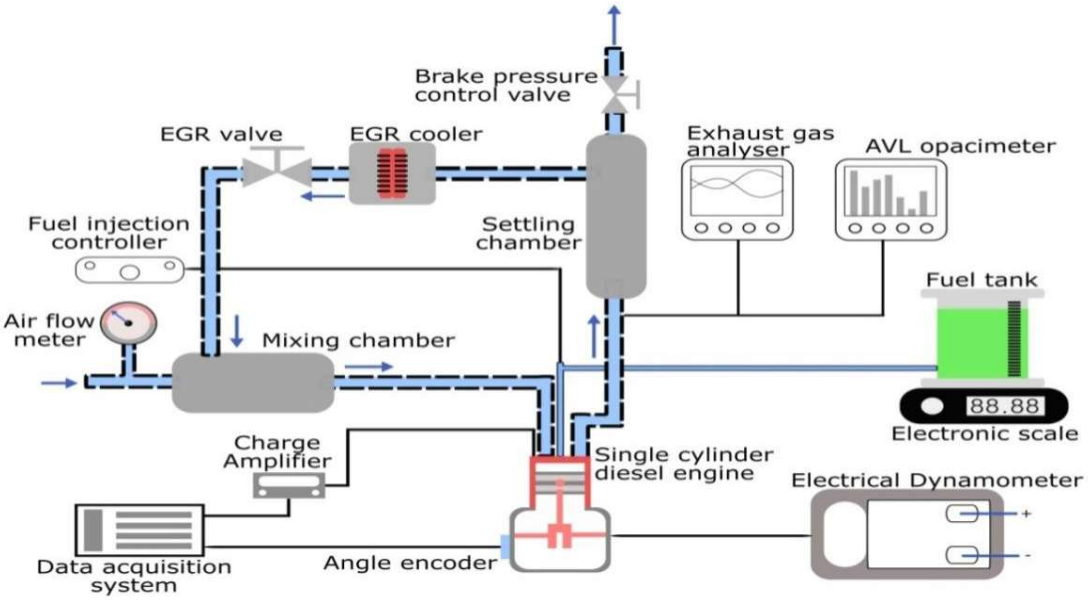

Figure 3. Experimental setup.

Table 4. Experimental uncertainty analysis.

Instrument	Measured quantity	Range	Accuracy	Uncertainties, %
AVL DIGAS 444 Exhaust gas analyzer	NO_x	0 - 5000 ppm	±10 ppm	0.2
AVL 437C Smoke meter	Smoke opacity	0 - 100%	±2 %	1.0

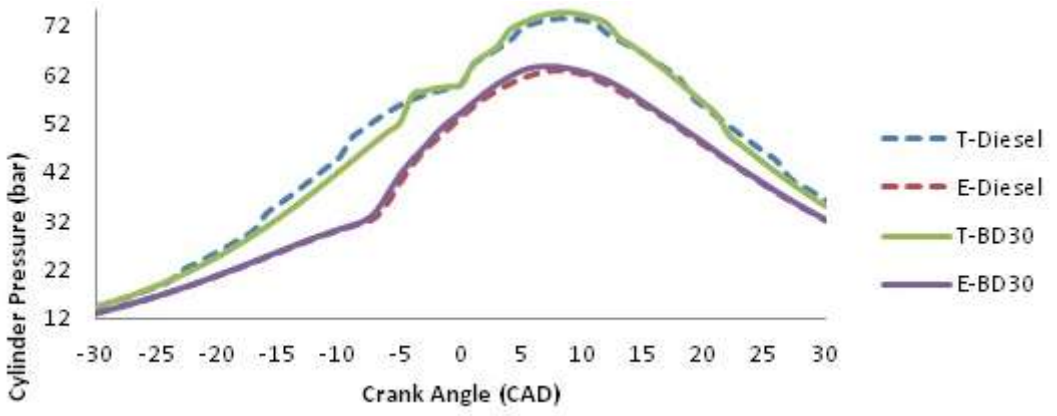

Figure 4. Cylinder pressure vs. crank angle for modelled and experimental CI engine fuelled with diesel and biodiesel at 1500 rpm, full load.

Comparison of Cylinder pressure

Figure 4 compares the variation of cylinder pressure with respect to crank angle for the modeled engine with that of the experimental engine when fueled with diesel and biodiesel. It was observed that the modeled engine follows a similar trend of variation in the cylinder pressure with higher magnitude and more fluctuations than those of the experimental engine. This may be due to uniform pressure, temperature, and equivalence ratio across the process at all times. This may also be due to the absence of heat losses in the modeled engine. As the heat losses were not taken into consideration, combustion products will have high heat energy and hence a higher magnitude of cylinder pressure.

The modeled engine pressure rise curve for diesel fuel depends upon the mass rate of ignitable mixture in the cylinder. In the modeled engine, the temperature of the inducted air and kinetic energy of fuel at different locations of the combustion chamber are evaluated theoretically as a function of crank angle degree, with an assumption that the air-fuel ratio of the mixture was uniform during the combustion process. This resulted in a higher rate of combustion and hence a shorter time for complete combustion. Whereas in the experimental engine, the fuel atomization, diesel chemical characteristics, and spray diffraction at each crank angle rotation were different, which produced a lower peak pressure when compared to that of the modeled engine.

In a single cylinder engine, the fuel is injected, and a peak pressure rise is obtained at the end of the compression stroke, approximately 10 degrees and 5 degrees before top dead centre. The peak pressure rise depends on the shorter ignition delay, inlet air pressure, and premixed combustion phase. The experimental and theoretical results of neat diesel show a gradual rise of pressure from 42 bar to 72 bar, and for biodiesel, it is from 22 bar to 52 bar, and the figure shows a similar trend for the blends.

The premix phase and diffusion burning phase are different for the modeled and experimental engine. In the modeled engine, the premix region was absent in the quiescent chamber; the generation of high swirl tends to combine the air-fuel mixture at a higher rate. In the present model, the combustion chamber is assumed as single homogeneous zone, and the equivalence ratio, temperature, and pressure of the mixture are assumed to be uniform across the chamber. Similarly, the rate of change of total mass at any point is the same, and the fuel injection pattern was ignored for the modeled engine. This resembles that as soon as fuel is injected, it gets burned, and hence, there is a high-pressure rise for the modeled engine.

Comparison of the heat release rate

Variation of heat release rate with respect to crank angle for the modeled engine fueled with diesel and biodiesel was shown in Figure 5 by comparing with those of the experimental engine.

The modeled engine shows a similar trend of variation in heat release rate with a lower magnitude of maximum heat release rate as those of the experimental engine. The rate of heat is calculated from Eq. (1) for the modeled engine, and the cylinder pressure from Figure 4 was taken as input. During the fuel injection period, the HRR is zero for the theoretical model, and for the experimental engine, the negative heat release was found due to the fuel vaporisation and endothermic reaction, which were absent and not included in the created theoretical model.

In the modeled engine, it could be observed that the period of rapid combustion time is longer due to the evaporation of fuel found around the air droplets, and takes time to form a combustible mixture. The experimental engine shows higher HRR than the modeled engine because the fuel injected during the second stage of controlled combustion will be mixed at a faster rate to reduce the ignition delay. The modeled engine has high HRR in the after-burning phase due to the rated velocity of diffusion of burnt fuel with air for the generated turbulent motion by the piston. In the experimental engine, the obtained heat during combustion was absorbed by the piston & cylinder wall surface, and the balancing of heat loss is to be considered during after burning process. The HRR is calculated from the first law of thermodynamics applied at the control volume for the modeled engine in the cylinder. The air-fuel mixture needs to react at equilibrium conditions for the obtained pressure rise. In the modeled engine, the valve overlap and blow - by gases are assumed to be zero, and the energy equation is simplified when compared with the experimental model.

The HRR depends on the cylinder pressure and temperature. Figure 5 shows a high heat release rate for a neat diesel fuelled engine, whereas for biodiesel, HRR starts at 5 to 3 degree before top dead centre and extends up to 20 degrees after top dead centre. The 30% biodiesel blend has higher viscosity, which leads to high ignition delay, and uncontrolled combustion phase was observed due to the heterogeneous mixture. The result shows similar trends for both the tested fuels.

Comparison of performance and emission parameters

Performance and emission parameters of the modeled engine fueled with diesel and biodiesel blend were shown in Table 5 and compared with those of the experimental engine.

The engine-out emissions, such as NO_x, smoke, and CO, are predicted using built-in empirical and semi-empirical sub-models integrated within the combustion framework. NO_x formation was modelled using the extended Zeldovich thermal mechanism, considering in-cylinder temperature, oxygen concentration, and equivalence ratio. Smoke emissions were estimated through empirical correlations that account for local air-fuel ratios, fuel properties, and combustion quality, particularly in fuel-rich regions where soot formation is prominent. CO emissions are modeled based on incomplete combustion behaviour, influenced by oxygen availability and local flame temperature. These models provide a balance between predictive accuracy and computational efficiency.

NO_x emission of the modeled engine was higher than that of the experimental engine due to the freezing of NO_2 levels by excess air at premixed equilibrium conditions.

During the Injection period, a lower rate affects the rate of heat release; hence, a lower HRR and higher cylinder pressure initiate a higher temperature to form higher NO_x for the modeled engine at any portion of the charge. It was observed that the smoke and CO emissions of the modeled engine were lower than those of the experimental engine when fueled with both diesel and biodiesel. It was inferred that the trend of variation of NO_x, smoke, and CO for the modeled engine was similar to that of the experimental engine. It was observed that the smoke emission of diesel and biodiesel blend of the developed model deviates from the experimental engine by 0.0047 and 0.006 g/kWh respectively. Smoke emission of the model was estimated based on fuel atomization and injection strategy, and the other factors affecting the smoke formation were not included during the development of the model. It can be seen that the BSFC of the modeled engine was lower than that of the experimental engine for both diesel and biodiesel. The modeled engine was tested at full load at constant speed. The conversion efficiency of the fuel in the modeled engine is lower since the uniform turbulence was assumed for the air.

In the experimental engine, the turbulence of the air was varied with respect to load. It can also be seen that the percentage error in predicting the performance and emission parameters of the modeled engine was marginal when fueled with diesel and considerably higher for biodiesel operation. This may be due to the change in fuel properties of biodiesel, which may not be revealed in the prediction of performance and emission parameters of the modeled engine fueled with biodiesel blend.

Table 6 summarises the key input parameters (fuel and engine) and output parameters (combustion) of the model.

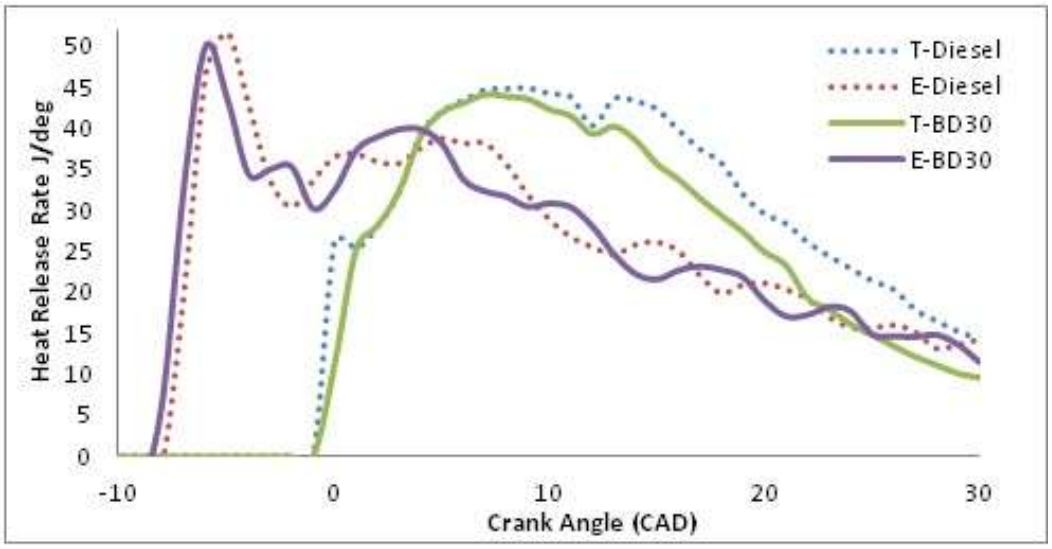

Figure 5. Heat release rate vs. crank angle for modeled and experimental CI engine fueled with diesel and biodiesel at 1500 rpm, full load.

Table 5. Comparison of performance and emission parameters.

Parameters	Diesel			Biodiesel blend		
	Theo.	Exp.	% Error	Theo.	Exp.	% Error
NO_x (g/kWh)	21.52	16.36	23.9	19.48	18.43	5.3
Smoke (g/kWh)	0.01	0.016	-41.6	0.0140	0.02	-42.8
CO (g/kWh)	2.93	3.01	-2.7	1.6	2.1	-31
BSFC (kg/kWh)	0.23	0.32	-38.1	0.29	0.3	-4.7

Table 6. Summary of input and output parameters of the model.

Input	Engine parameters	Compression ratio	17.5:1	
		Rated power, kW	4.4	
		Rated speed, rpm	1500	
	Fuel parameters		Diesel	Biodiesel blend
		Viscosity (mm^2/s)	3.0	3.5
		Cetane number	51	55
		Lower heating value (LHV) MJ/kg	45	44
Output	Combustion parameters	Peak cylinder pressure (bar)	71	70
		Heat release rate (HRR) J/deg	47	45
		Combustion duration (CAD)	-4 to 30	-3 to 30

CONCLUSION

In the present work, a simulation model was developed to accurately predict the combustion, performance, and emission characteristics of a single-cylinder compression ignition engine fueled with diesel and biodiesel. Parameters of the modeled engine were validated by comparing with the parameters of the experimental engine. From the comparison, it was inferred that the variation in absolute values between the chosen parameters of the theoretical model and experimental engine was marginal, which confirms the suitability of the developed model for the compression ignition engine. From the inferences, the following conclusions were drawn:

- One-dimensional modeling is an effective tool to simulate the engine characteristics by developing a prototype model, which enables critical analysis of the engine characteristics.
- The developed simulation model can be applied by the automotive industries to reduce the investigation time and cost in creating a real-time experimental engine that runs on alternative fuels. This helps to reduce the dependency on fossil fuels for the automotive sector and also significantly enhances the possibility of meeting the ever increasing energy demands, especially from the automotive and power sector, and also paves the way for a much greener and sustainable environment.

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NAUČNI RAD

PREDVIĐANJE KARAKTERISTIKA MOTORA SA KOMPRESIONIM PALJENJEM NA DIZEL GORIVO I MEŠAVINE BIODIZELA PRIMENOM JEDNODIMENZIONALNOG MODELOVANJA

Ova studija je bila usmerena na primenu napredne tehnike modelovanja radi razvoja jednodimenzionalnog modela za analizu proračuna kompletnog radnog ciklusa motora sa kompresionim paljenjem koji koristi mešavine biodizela, sa ciljem ispitivanja njegovih performansi i emisionih karakteristika. Razvijeni model rešava jednačine koje opisuju zakone održanja energije i mase, čime se omogućava praćenje brzih promena u procesu izmene gasova i sagorevanja u funkciji ugla kolenastog vratila. Komora za sagorevanje je u simulacionom softveru predstavljena kao jednodimenzionalni model koji se sastoji od dve odvojene zone: zone sagorelih i zone nesagorelih gasova. Oslobađanje toplote i predviđanje emisija određivani su primenom pojednostavljenih reakcionih mehanizama. Razvijeni model sagorevanja integrisan je kao podmodel unutar cilindarskog elementa jednodimenzionalnog rešavača, čime je omogućena razmena ključnih parametara, kao što su pritisak u cilindru, udeo sagorele mase i brzina oslobađanja toplote, pri svakom inkrementu ugla kolenastog vratila. Rad se fokusirao na poređenje vršnog pritiska, brzine oslobađanja toplote, oksida azota, dima, ugljen-monoksida i specifične potrošnje goriva za kočenje kreiranog modela sa rezultatima dobijenim eksperimentalnim motorom. Poređenjem teorijskih i eksperimentalnih rezultata zaključeno je da je teorijski model rezultirao uporedno višim vršnim pritiskom (41%), višom brzinom oslobađanja toplote (10,38%), nižom količinom ugljen-monoksida (31%) i manjom specifičnom potrošnjom goriva za kočenje (4,7%) u odnosu na eksperimentalni motor za mešavinu biodizela. Rezultati ove studije pokazali su izvodljivost korišćenja biodizela u motorima sa usisnivanjem sa boljim karakteristikama, što je u skladu sa ciljevima održivog razvoja.

Ključne reči: jednodimenzionalni model simulacije motora; biodizel; dizel motor; emisija; sagorevanje.