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Theoretical Investigation of Combustion and Performance Analysis of Diesel Engine under Low Load Conditions

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Abstract. Diesel engine is using prominently in islands and remote areas due to its reliability and stability for power generation. In recent years, most of the isolated power systems (e.g., islands and remote areas) have integrated renewable energies to reduce both the cost and pollution in diesel power generating system. However, due to intermittent and stochastic behaviour of renewable sources (e.g., solar and wind), it is unable to eliminate diesel generation entirely. In that case, low-load diesel operation (operation < 30% of maximum rated load) is particularly relevant for its ability to support higher levels of renewable penetration. In this paper, a thermodynamic model was developed using MATLAB for diesel engine combustion and performance. This model includes sub models such as heat release rate, heat transfer, double-Wiebe function, and ignition delay correlation. Engine thermal efficiency (TE), brake power (BP), indicated mean effective pressure (IMEP) and brake specific fuel consumption (BSFC) has been taken into consideration for performance analysis. The simulation results show that at 25% load, in-cylinder pressure and temperature are 168 bar and 2300 K which are the cause of lower heat release rate (74 J/deg) and longer ignition delay (0.25 ~ 0.5 ms higher than that of conventional mode) and significantly responsible for lower efficiency (18%), brake power (4kW) and higher brake specific fuel consumption (1.2 g/kWh).

1. Introduction

Nowadays, diesel engines are widely used in power generation and heavy-duty transportation sectors for its high efficiency and fuel economy, larger power range and longer lifetime[1]. For a number of years now, diesel engine driven generators have traditionally been used to supply electricity relatively small power networks those are associated with industrial complexes, marine applications, remote and island communities [2].

Most of the remote areas and islands communities are usually dependent on conventional diesel generation due to its reliability, low fuel cost and operational simplicity [3]. More recently, some technologies have been introduced in remote areas and islands power systems via renewable energy penetration to minimise dependency of diesel. The most common and abundant renewable sources i.e. solar and wind are employed for significant reduction of fuel consumption, however, they are uncertain and discontinuance, not able to remove diesel generation completely [4]. Moreover, the possibilities of highest renewable integration in hybrid systems can reduce the engine load dependency which lead to decrease fuel economy as well as system cost. In this case, low load (below 30% of maximum rated load) diesel operations are most relevant and flexible for maximum renewable energy utilization.



In terms of engine operation flexibility, diesel engine load limits and combustion efficiency are the main issues that arise in hybrid diesel systems during conventional mode of diesel operation [5-7]. Usually, operating uninterruptedly under low load mode can lead to ignition problems, increase lubricant oil consumption, and fuel dilution. The emergence and persistence of residue has a negative impact on the functional behaviour and on the lifetime of the engine. In addition, when a conventional engine is operating in low load mode, it cools down and due to low temperature in chamber, the fuel is partially burned, which can in turn produce a white smoke with high hydrocarbon emissions. The percentage of unburned fuel caused incomplete combustion and poor engine performance [8]. As low load mode shares a similar model to that of the conventional diesel mode [9].

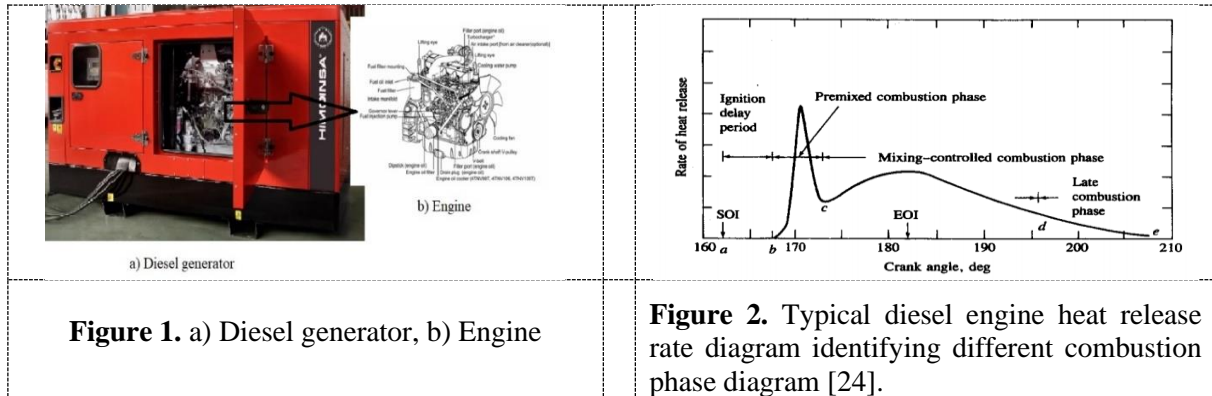
Basically, diesel engine performance is influenced by in-cylinder mechanisms because it contributes to the air-fuel mixing and control of fuel burnt rate. Understanding and reliable prediction of the combustion characteristics and key parameters such as ignition delay, cylinder temperatures and pressures, characteristics of fuel, mixture of fuel-air ration are required for diesel engine modelling and optimal performance prediction [10, 11]. The objective of this paper is to investigate low load diesel combustion profiles and performance characteristics using combustion-oriented engine modelling and simulation. Generally, diesel engine models are categorised into three types i.e., multi-dimensional, zero-dimensional, and quasi-dimensional based on the performance, combustion, and emissions [12, 13]. Multi-dimensional models are used to optimize injection timing, design of combustion chamber and swirl-ratio. This model can predict engine combustion, performance, and emissions precisely [14-16]. The zero-dimensional models are used to predict engine combustion, performance including fuel economy. Besides, zero-dimensional models are divided by two types. First being MZ model that coupled with combustion inputs, air get in into cylinder and evaporation of fuel. Multi-zone model can predict combustion profiles of burn and unburn zone, engine performance and emissions [17, 18]. Second is the single-zone model which is a simplified thermodynamics model which normally can predict the engine combustion profiles and engine performance [19, 20]. Basically, there is two way in modelling combustion phase in diesel engines: one is the Arrhenius type equations considering mixture density, oxygen mass fraction and vapor of fuel present in combustion chamber [21]. Another type is the Wiebe law that predict HR for combustion phases in consideration of ID, cylinder pressure and injection pressure and temperature, mass of fuel exist during combustion [22, 23].

In this study, the simplified single-zone thermodynamic model was developed and simulated in MATLAB to analyse engine combustion performance considering different loading conditions especially under low load. A double Wiebe model implementation has been considered to predict the premixed and diffusion phase of diesel engine combustion. The single-zone model predicts engine combustion profiles such as in cylinder pressures and temperatures, HRR and also mass of fuel and ignition delay. Another aim is to find out the key mechanisms which are responsible for low load operations and analyse engine performance under that (low load) conditions.

2. Theoretical Study

2.1 Description of the Model

The combustion phase starts where air is admitted into cylinder through the intake stroke which depends on the ambient condition imposed. In adiabatic compression phase, mass of air is compressed to high temperature and pressure after closing the intake valve. Gas mixture pressure can be modelled considering the adiabatic compression theory i.e., $PV^\gamma = \text{constant}$, where γ is the specific heat ratio. The time between the fuel injection and fuel burnt starts, is called ignition delay which significantly depends on fuel injection timing and temperature, engine load and speed and fuel cetane number. Combustion starts after ID and an overlap phase (compression and combustion) has risen cylinder pressure and temperature to high. In the expansion phase, piston shifts from Top Dead Centre to Bottom Dead Centre. As the exhaust valve opens, emission gases are released through the exhaust manifold. Figure 1 shows the picture of diesel engine. All input parameters and baseline conditions for combustion model simulation are given in table 1.

**Figure 1.** a) Diesel generator, b) Engine**Figure 2.** Typical diesel engine heat release rate diagram identifying different combustion phase diagram [24].**Table 1:** Engine specifications and baseline conditions for model simulations

Engine	4-stroke diesel, dry heavy-duty, direct-turbo
Bore, Stroke	84 mm, 90 mm
CR	18.9
Rated speed/power	1500 rpm/19.1kW
Injection timing	20° BTDC
Combustion duration	40° BTDC
Inlet manifold temperature	300 K
Inlet manifold pressure	1.5 bar

2.2 Mathematical Representation of the Model

The purpose of single zone combustion model is to consideration the process which happen during a diesel combustion cycle. A double Wiebe law and sub-models are used to simulate the HRR between the time of IVC to EVO.

2.2.1 Engine Geometry. During combustion, piston moves from TDC to BDC, for which area of cylinder, volume, and stroke w.r.t. crank angle changes vice-versa. The combustion chamber volume, V can be found from the kinematic motion model. The following derivations are used for engine geometry calculations [25] to calculate engine geometry,

$$X(\varphi) = (l + R) - \left(R \cos(\varphi) + (l^2 - \sin^2(\varphi))^{1/2} \right) \quad (1)$$

$$A(\varphi) = \frac{\pi B^2}{4} + \frac{\pi B S}{2} (R + 1 - \cos(\varphi) + (R^2 - \sin^2(\varphi))^{1/2}) \quad (2)$$

$$V(\varphi) = V_c + \frac{\pi B^2}{4} X(\varphi) \quad (3)$$

where, R is crank radius ratio, the rod length is l (m). Cylinder bore and stroke are denoted by B and S , respectively and φ is the CA (deg.).

2.2.2 Cylinder Pressure. The in-cylinder and pressure with CA is calculated from the first law of thermodynamics. The pressure rate change against CA resolved and found as follows,

$$\frac{dP}{d\varphi} = -\gamma \frac{P}{V} \frac{dV}{d\varphi} + \frac{\gamma-1}{V} \left(\frac{dQ_w}{d\varphi} \right) + \frac{\gamma-1}{V} \left(\frac{dQ_{in}}{d\varphi} \right) \left(\frac{dX_b}{d\varphi} \right) \quad (4)$$

where, P is pressure and V is volume, γ is the specific heat ratio, Q_{in} is the HRR (J/deg.), Q_w is the cylinder wall heat loss rate (J/deg.).

2.2.3 ID Correlation. Ignition delay is typically defined as the time between start of injection and start of combustion. For this study, an ID correlation has been taken using charge pressure and temperature at TDC. It also takes the engine's mean piston speed into account and calculates the activation energy as a function of cetane number (CN). This expression can be written as [26],

$$ID(CA) = (0.36 + 0.22 \bar{S}_p) \exp \left(Ea \left(\frac{1}{R_u T} \right) - \left(\frac{1}{17190} \right) + \left(\frac{21.2}{P - 12.4} \right)^{0.63} \right) \quad (5)$$

The temperature and pressure in engine inlet manifold are assumed as proportional to the values when the piston is at BDC. When the piston is at TDC, the mean CR during the ID CR_{av-DP} is proportional to the CR when the piston is at TDC which can be written as $T = T_m \times CR^{c-1}$ and $P = P_m \times CR^c$, respectively. Here, c is the polytropic exponent of compression, T_m is the absolute inlet manifold temperature in K and P_m is pressure in bar. The CN is an indicator of combustion speed of diesel fuel which is determines the fuel quality. The following expression can be the substitute for the activation energy,

$$E_a = \frac{618840}{CN+25} \text{ (J/mol)} \quad (6)$$

where, CN is the cetane number.

2.2.4 Heat Release Rate model. The characteristics of diesel engine combustion are mainly determines through HRR that is found from the measured pressure data as a function of CA (ϕ) [27]. For combustion, the numerical formulation is given as summation of premixed and diffusion phase. The HRR varies with the in-cylinder pressure, volume, crank angle (ϕ) and specific heat ratio via equation (7),

$$\frac{dQ_n}{d\phi} = \frac{\gamma}{\gamma-1} \cdot P \frac{dV}{d\phi} + \frac{1}{\gamma-1} \cdot V \frac{dP}{d\phi} + \frac{dQ_w}{d\phi} \quad (7)$$

where, V is the cylinder volume, P is cylinder pressure, γ is the specific heat ratio ($\gamma = 1.35$ was assumed to be the average of γ values during the compression and expansion, respectively).

2.2.5 Wall Heat Transfer Model. According to the Newtonian model, heat transfers to the wall are calculated with the following equation:

$$\frac{dQ_w}{d\phi} = \frac{hA(\phi)(T_{gas}-T_{wall})}{6N} \quad (8)$$

where, T_{gas} is the temperature of gas and T_{wall} is the temperature cylinder wall. The instantaneous area of the cylinder is $A(\phi)$ and h is the coefficient of heat transfer which is determined by the models [14, 28-30]. In this study the rate of heat transfer during the burning process, calculated using the Woschni model [14] that has presented by below equation:

$$h = 3.01426B^{-0.2}P^{0.8}T_{gas}^{-0.5}v^{0.8} \quad (9)$$

where, v is the velocity of burned gas that is written as,

$$v(\phi) = 2.28\bar{U}_p + C_1 \frac{V_d T_{gr}}{p_r V_r} (p(\phi) - p_m) \quad (10)$$

The quantities T_{gr} , p_r and V_r are the references of state properties of inlet valve closing and p_m is the pressure at same time to obtained p (pressure without combustion) w.r.t. crank angle (ϕ). The constant C_1 represents for compression process, combustion, and expansion process. The diesel engine combustion phases can be found on the typical heat release-rate diagram that has shown in figure 2. This diagram shows the ignition delay period ($a-b$), combustion phases such as premixed ($b-c$), controlled ($c-d$) and late combustion ($d-e$) w.r.t. crank angle (deg.).

2.2.6 Wiebe Law. Normally Wiebe law is used to characterise fraction of mass burn or burnt rate of a combustion [31, 32]. In DI engines, there is always a premixed combustion phase, except for a diffusion combustion phase[33]. The rate of HRR in a diesel engine combustion can be predicted with relatively

simple way and reasonable accuracy using double-Wiebe function [23, 34]. The standard Wiebe function can be expressed as:

$$x_b(\varphi) = 1 - \exp \left[-a \left(\frac{\varphi - \varphi_0}{\Delta\varphi} \right)^{M+1} \right] \quad (11)$$

where, $x_b(\varphi)$ is the mass fraction burnt, φ is the crank angle, φ_0 is the crank angle at the start of combustion (SOC), $\Delta\varphi$ is the combustion duration defined as the difference between φ_0 and the end of combustion (EOC), M is the form factor which determines combustion process curve shape, and a is the efficiency process curve which controls the duration of the combustion process. In diesel engine there are two phases of combustions such as premix and diffusion. equation (12) can be considered for different combustion phases,

$$x_b(\varphi) = 1 - \sum_{k=1}^2 \beta_k \cdot \exp \left[-a_k \left(\frac{\varphi - \varphi_k}{\Delta\varphi_k} \right)^{M_k+1} \right] \quad (12)$$

where, β_k is fraction of burnt fuel in the k th is combustion phase.

3. MATLAB Script Procedure

Matrix Laboratory (MATLAB) language was used to develop engine combustion and performance model because of its suitability and ease of simulation of these parameters. The MATLAB script began with known engine inputs (bore, stroke, connecting rod length, number of cylinders, compression ratio) and operating conditions. Based on all inputs, the area of the cylinder, clearance volume, and surface area of the piston head were calculated. In this step, the atmospheric conditions were chosen, for example, the initial inlet temperature and pressure considered were 298 K and 1 atm, respectively. Fuel inputs such as mass of the fuel, calorific value, lower heating values (LHV) and air-fuel ratio were considered.

In the first loop of the program, a specified index was used to calculate instantaneous engine properties such as engine geometry, fuel properties etc. In addition, instantaneous properties like volume, pressure, and temperature were scripted for to calculate work done during the total range of the cycle. Indicated power, friction power, brake power, correction factor, etc, were used to calculate the brake specific fuel consumption. MATLAB script also has statements to cope up with heat release rate and wall heat transfer with Wischini model. Ignition delay was calculated from Hardenberg model.

In plot sections, each plot was sized and given a title based on the minimum and maximum variable values. MATLAB script was developed to have plots between all the performance parameters as a function of CA.

4. Simulation Results and Discussion

The effect of engine load on the combustion process, the in-cylinder pressure, temperature and heat release were predicted at different loading conditions at 1500 rpm speed operation.

4.1 Combustion Analysis under Low Load Conditions

4.1.1 Cylinder Pressure Figure 3 represents the cylinder pressure at full, 75%, 50% and 25% engine loading conditions w.r.t. crank angle using standard diesel fuel kinematic viscosity. It can be seen from figure 3 that between $5^\circ \sim 7^\circ$ ATDC range, the in-cylinder pressure seems peak for all loading conditions and the trends is to reduce in-cylinder pressure with decreasing engine loads. The possible reason is that it could be the quantity of fuel burned decreases with reducing engine load which caused a decrease in the energy releases, resultant in a decrease in peak cylinder pressure. The decrease in pressure profile for lower loading operations may be due to lean fuel-air ratio, which increases ignition delay.

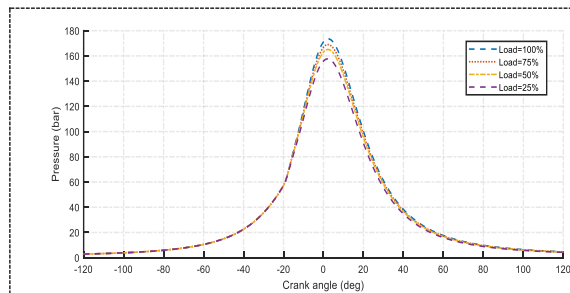


Figure 3. In-cylinder pressure vs CA at various loading operations

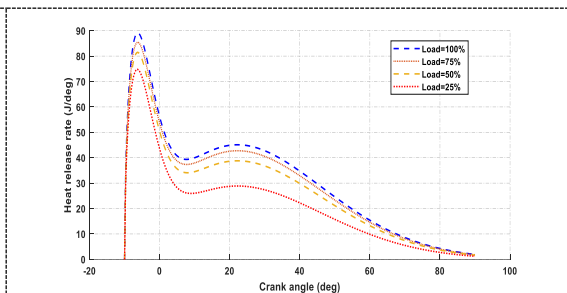


Figure 4. HRR vs CA at various loading operations

4.1.2 HRR and Cylinder Temperature. Figure 4 shows the HRR, as a function of CA at various engine loading conditions. It can be found from figure 4 that HRR decreases when engine load decreases. At 25% loading conditions, the peak heat release rate was achieved in engine powered by diesel and is equal to 74 J/deg CA. In full load conditions, the highest rate of heat release in the engine was achieved and the value is 89 J/deg CA. The possible reason is that less charge is delivered to the cylinder during the intake stroke at low load, hence lower temperature is reached at the time of ignition start. At full load to the cylinder, more air and gas mixture is supplied, and at the beginning of fuel injection higher in cylinder temperature is reached which is shown in figure 5.

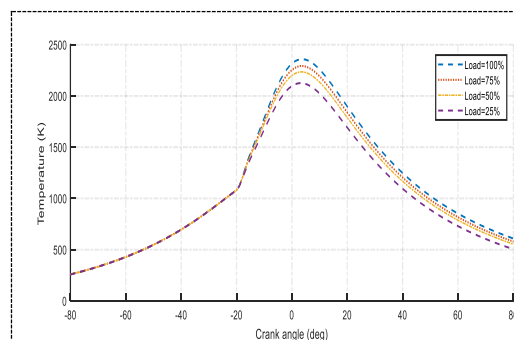


Figure 5. Cylinder temperature with CA at various loading operations.

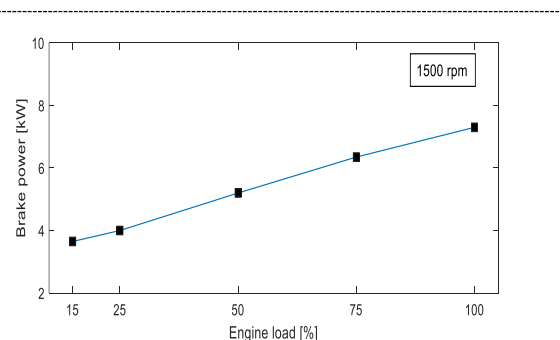


Figure 6. Engine brake power at different loading operations.

4.2 Performance Analysis under Low Load Conditions

4.2.1 BP. Figure 6 illustrates brake power for different loading conditions at 1500 rpm speed diesel operating condition. It can be seen from this figure that brake power reduces with reduction of engine load. As lower load leads the lean fuel-air ratio (i.e., in ignition chamber is being igniting with too much air and too little fuel), hence increase the heat, friction, and mechanical losses which reduce engine brake power. Poor atomization, vaporisation and mixing of the fuel droplet with air during combustion performance due to low engine loading range cause brake power reduction. In addition, at low load range operation causes poor combustion performance that result in lower value of brake power [35, 36].

4.2.2 BSFC. Basically, BSFC of a diesel engine depends on the brake power and mass of fuel flow at a specific engine speed and load [37]. Figure 7 illustrates the BSFC w.r.t. different engine loads at speed (1500 rpm). The trend of BSFC with load is that it decreases with engine load. The reason for increasing BSFC with decreasing load is that at lower loads, the in-cylinder temperature in combustion chamber treats low which is the main barrier for proper mixing and atomization, resulting in lower combustion

efficiency. In addition, at low load with little fuel i.e., lean fuel-air ratio, heat liberated is absorbed by a large air present, result in high BSFC.

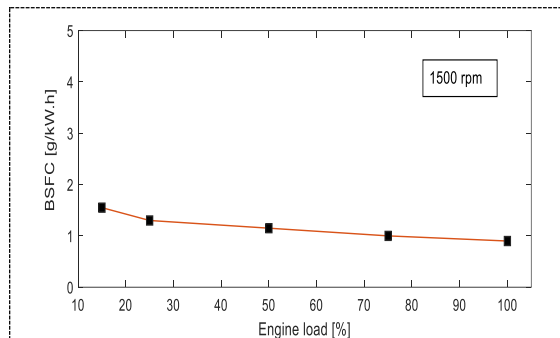


Figure 7. BSFC at different loading operations

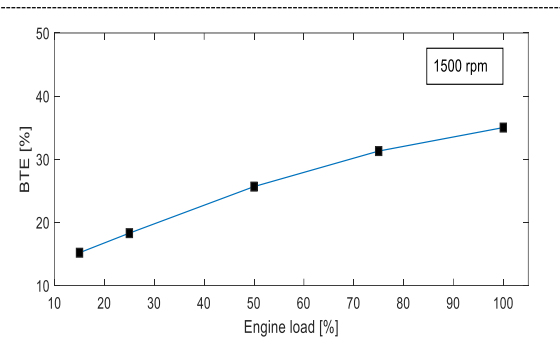


Figure 8. BTE at different loading operations

4.2.3 BTE. Figure 8 shows BTE as a function of different load for diesel engine run on the diesel fuel. For low load condition, the brake thermal efficiency is comparatively lower. This can happen due to increment of heat losses and reduction of power at reduced load. Low load ranges lead to lean air-fuel mixture, lower combustion, and longer ignition delay which can reduce efficiency.

4.2.4 Ignition Delay. Ignition delay time with crank angle has illustrated in figure 9. The trend shows that ignition delay (in milliseconds) increases when engine load decrease. The reason for that is, as load decreases the residual gas and cylinder wall temperature decreases, hence, result in lower air temperature at injection, which increase ignition delay time. Investigation has shown that lower load exhibited the larger ignition delay, due to poor mixture preparation and combustion result which impacts on engine response time.

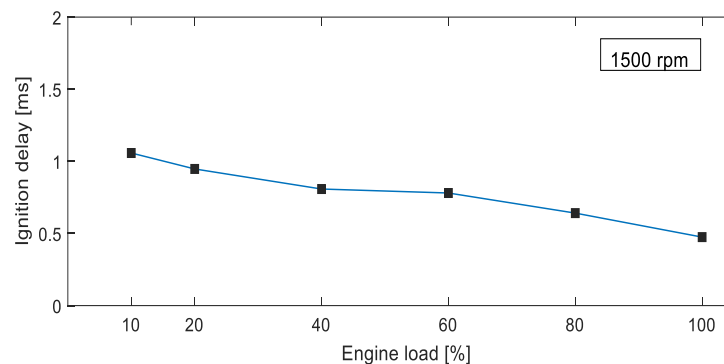


Figure 9. Ignition delay at different loading operations

5. Synthesis of Information

In this study, a combustion model for diesel engine has been developed using MATLAB. The study focused on the combustion and performance profiles of diesel engine at low load ranges. It was found that engine combustion characteristics (e.g., pressure, heat release rate and temperature) and the engine performance characteristics such as BP and BT decrease with decrease in engine load. On the other hand, BSFC is higher at lower engine load due to lean fuel air ratio which leads to an increase air pressure. Ignition delay result shows that ignition delay is longer at lower load because residual gas and cylinder wall temperature decreases at lower injection temperature at low load conditions.

6. Conclusions

Nowadays, diesel engines are widely used as a reliable source in remote and island areas power systems to provide electricity. In terms of flexibility and maximum renewable penetration, this system needs to reduce engine load for getting better fuel economy. In this study, a single-zone thermodynamic model was developed using MATLAB for analysing combustion and performance characteristics under low load conditions. The results show that at lower loading operations, engine brake power and thermal efficiency decreases in comparison with conventional mode operations due to lean air-fuel mixture, lower in-cylinder pressure and temperature. Furthermore, brake specific fuel consumption (BSFC) increases at low load condition due to decreasing brake power and mass of fuel flow. Compared with conventional model operations, low load diesel shows lower efficiency and longer ignition delay times due to lower injection temperature and pressure. The range of ignition delay value varies from 0.25~0.5 ms compared to that of conventional model operations (above 50% load of rated power) due to poor mixture preparation which drops engine efficiency.

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