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Research Paper

A numerical analysis of the characteristics of diesel-powered engines working on waste plastic oil blends: Base compromising

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ABSTRACT

The current study's objective is to use Diesel-RK simulation software to numerically assess the effects of employing waste plastic oil blends on the thermal characteristics of diesel engines. Each autonomous zone's governing equations are solved using the multi-zone combustion model. Six distinct volumetric mixes were used to analyze the engine's characteristics of waste plastic oil (10%, 20%, 30%, 50%, 70% and 100%) as a comparison to the standard diesel case. The data collected showed a slight decrease in pressure and heat release for all waste plastic oil blends compared to pure diesel fuel. The Sauter mean diameter of droplets increased by 0.91%, 1.85%, 2.68%, 4.35%,5.74% and 7.1% for 10%, 20%,30%, 50%, 70%, and 100% waste plastic oil (WPO), respectively. The slightly lower cetane number of WPO compared to fossil diesel resulted in a longer ignition delay, resulting in a slightly later combustion start. Brake specific fuel consumption (BSFC) increased by 2.2%, 4.1%, 5.86%, 9.23%, 15.14% and 15.960% for 10%, 20%,30%, 50%, 70% and 100% WPO, respectively, because of differences in density, viscosity, and heat content, respectively. A significant drop in the Bosch Smoke Number (BSN) was observed with 30% and 50% WPO, reporting reductions by 2.86% and 4.55%, respectively. The lowest increase in particulate matter was 1.96% and 2.87% for 30% and 50% WPO biodiesel blends. A higher biodiesel content resulted in lower NOx emissions compared to diesel. The findings suggest that 30% WPO is the optimal blend recommended for use in a diesel engine without alteration, which aligns with the results from other studies.

1. Introduction

Researchers explore alternative fuels due to the high emissions from fossil fuels, which contribute to environmental issues such as global warming, in addition to the fact that relying on fossil fuels leads to resource depletion [1,2]. Furthermore, the use of conventional fuels in transportation, industry, and agriculture impacts negatively on human health, causing heart problems, respiratory ailments, and higher mortality rates. To lower greenhouse gas emissions, save money, and diminish the environmental impact of gasoline and diesel, it is essential to use biofuels and other sustainable energy sources. [2–4]. Applying renewable fuels becomes more favored over technological modifications to address the environmental problems brought on by fossil fuel combustion. Sustainable and alternative energy sources can help reduce NOx, greenhouse gases, and particulate emissions from diesel engines. Biodiesel is one of the most promising alternative fuels because it is recyclable, non-toxic, sulfur-free, environmentally friendly, and more reliable. Research has demonstrated that biodiesel share many chemical and physical characteristics with diesel fuel, making it a viable substitute [5]. Its growing global popularity is driven by its fuel properties and compatibility with existing diesel engines [6]. Additionally, biodiesel reduces the impact of greenhouse gases since it does not increase the atmospheric CO2 levels [7]. It contributes to lower emissions of CO, HC, and smoke [8,9], while producing fewer contaminants. Biodiesel doesn't include cancer-causing substances such as polyaromatic hydrocarbons and nitrous polyaromatic substances. Like diesel, biodiesel has several benefits, including availability, lower exhaust emissions, renewability, and improved lubrication. However, it also has a number of drawbacks, such as higher density and viscosity, increased NOx emission, and lower heat content [10,11]. Biodiesel is produced from different feedstocks, such as neem oil, palm oil, waste frying oil, vegetable oil, animal fat, microbial oil, etc., so its location and quality are dependent on its feedstock, which are crucial variables [12,13]. Several studies have examined emissions from different renewable energy sources, namely biodiesel made from waste plastic [14–16]. Plastic is becoming more important in the current world, as evidenced by the investigation that shows that biofuel as a result of waste plastic has qualities that are similar to those of diesel [17].The data collected revealed that while waste plastic oil is a major carbon source, its emissions are lower than those from fossil fuels [18–22]. The recovered biofuel was mixed with diesel to evaluate various properties like density, fire point, and flashpoint. When used as an additive to compression ignition engine, the biofuel's properties aligned with ASTM specifications. Research investigated how performance, emissions, and combustion characteristics of waste plastic biodiesel were affected [23]. The blends were prepared by mixing WPO samples with pure diesel. For both sets of samples, the maximum cylinder pressure ranged between 67 and 71 bar. Pure diesel had the lowest rate of heat release than other WPO blends. A slight rise in fuel consumption and a minor decline in efficiency were observed when WPO was used. However, a decrease in NOx emissions and an increase in CO_2 emissions were noticed due to the presence of oxygen. Emissions from burning various diesel-biodiesel mixtures including ultra-low sulfur diesel blended with cooking oil and fat in 10, 20, 30, and 50% combinations were tested [24]. The findings revealed that NOx emissions from burning diesel and biodiesel were greater than those from burning sulfur-containing diesel.

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Nomenclature

A_o,A_2,A_3	Constants.	Vc	Cylinder volume m^3
CO	Carbon monoxide % (vol)	Vom	Average speed of the nozzle spray (m/s)
DF	Diesel fuel -	X	Fraction of heat release
D_n	Nozzle diameter (mm)	x_o	Fraction of the fuel vapor formed during ignition
SMD	Sauter mean diameter (mm)	*Greek Symbols*	
d_o	Current fuel droplet diameter micron	σ	Fraction of the fuel injected into the cylinder %
d_k	Initial fuel droplet diameter m	σ_f	Coefficient of the surface tension of the fuel
b_m	The forward spray depth (m)	σ_s	Fractions of fuel injected into cylinder at time τ_s
$[C]$	Soot concentration in the cylinder g/m^3	σ	Fractions of fuel injected into cylinder at time τ_k
$°C$	Degree centigrade	σ_μ	Fraction of the vapor formed during the ignition period. %
CN	Cetane number of fuel	τ	Delay period second
CPM	Empirical factor for PM emission	τ_k	Travel period for the EFM to amount to the length 1
CNO	Empirical factor for NOx emission	τ_m	Travel period for the EFM to amount to the spray's front
E_a	Apparent activation energy for the auto ignition process $kJ/kmol$	τ_s	Current time from fuel injection at the beginning second
EFM	Elementary fuel mass kg	τ_u	Current time when the droplet is inside the combustion
l	Current distance between the EFM and the nozzle injector m	$\mu\varepsilon$	Fuel dynamic viscosity $(Pa.s)$
l_a	The length of the spray at the initial phase m	θ	Crank angle (degree)
l_b	The length of the spray at the main phase m	ϕ	Equivalence ratio
EGT	Exhaust gas temperature K	φ	Dimensionless parameter
EGR	Exhaust gas ratio%	γ	Adiabatic exponent of exhaust gas
E_k	Experimental coefficient	γ_a	Cone angle for the spray during the initial stage degree
E_s	Experimental coefficient	γ_b	Cone angle for the spray during the basic phase degree
h_{wfr}	Height of the dense front of the NWF m	εb	Efficiency of air used. %
K	Constant of evaporation	EGT	Exhaust gas temperature
K_T	Evaporation constant	$BSFC$	Brake specific fuel consumption
I_m	Penetration length of EFM till the end in front of a spray m	WPO	Waste Plastic oil
$ASTM$	American Society for Testing and Materials.	BSN	Bosch smoke numberb
mf	Mass of fuel (kg)	CP	Cylinder pressure
NWF	Near wall flow	SMD	Sauter mean diameter
NO_x	Nitrogen oxides (ppm)	$BTDC$	Before top dead centre
P	Current pressure in the cylinder (Pa)	BTE	Brake thermal efficiency
PM	Particulate matter $g/kW.h$	CI	Compression ignition engine
rpm	Revolution per minute $(1/min)$	CR	Compression ratio
r_v	Rate of the relative evaporation (%)	DF	Diesel fuel
T_{wi}	Rate of the relative evaporation within various areas of the flow	$10\%WPO$	Diesel fuel 90% and Waste plastic oil 10%
S	The stroke of the piston m	$20\%WPO$	Diesel fuel 80% and Waste plastic oil 20%
T	Temperature in the cylinder K	$30\%WPO$	Diesel fuel 70% and Waste plastic oil 30%
T_k	Temperature in the related zone K	$50\%WPO$	Diesel fuel 50% and Waste plastic oil 50%
T	Zonal temperature K	$70\%WPO$	Diesel fuel 30% and Waste plastic oil 70%
t	Time s	$100\%WPO$	Diesel fuel 0% and Waste plastic oil 100%
TDC	Top dead center	HC	Hydrocarbon
V	Current speed of the EFM (m/s)	ICE	Internal combustion Engine
V_o	Initial speed of the EFM at the nozzle injector (m/s)	NO_x	Oxides of nitrogen
V_m	Velocity of the spray's front (m/s)	PM	Particulate matter
V_{ma}	Spray head speed is calculated in the initial stage (m/s)		
V_{mb}	Spray head speed is calculated in the basic stage (m/s)		

However, a higher blending ratio of biodiesel resulted in lower CO_2 emissions. The study concluded that biodiesel blends produced from plastic waste could serve as a viable alternative fuel, reducing diesel fuel consumption and reduce emissions from conventional diesel-powered vehicles. Research was conducted to examine emissions from mixtures of diesel and biodiesel with waste plastic and waste cooking oil (D80+B20, D70+B20+P10, and D60+B20+P20) [25]. A decrease by 23.6% in CO_2 emissions and by 22.7% in UBHC emissions was observed for the blend with 20% WPO, along with reduced NOx and smoke emissions compared to diesel. Thus, WPO and waste cooking oil blends offer a low-emission alternative fuel with better performance than pure diesel. Although there are several oils that burn with varying amounts of energy released, most of them are unsuitable for diesel engines due to the high viscosity and lower stability compared to pure diesel. Consequently, the performance of a compression engine and its emissions were analyzed for different diesel and biodiesel mixtures (such as waste plastic oil) [26]. The data indicated that diesel produced higher emissions than biodiesel. An experimental investigation on a light-duty CI engine was conducted utilizing diesel-WPO blends [27]. A decrease in smoke and HC emissions, while an increase in NOx emissions was observed, compared to diesel. Further research on WPO blends examined combustion properties [28]. A study was performed on a diesel engine operating at 1500 rpm using a mixture of (80%, 60%, and 40% of diesel) with (20%, 40%, 60% WPO) [29]. When compared to other plastic oil blends, WPO20 demonstrated the best performance and emission characteristics, with a maximum BTE of 28.12% which was sufficiently close to that of diesel (28.25%). Out of all the blends, WPO20 blends displayed the best emission characteristics. A diesel-RK simulation was used to investigate the performance characteristics of different biodiesels obtained via catalyst pyrolysis in diesel engines [30]. The findings demonstrated that utilizing biofuel improved performance and reduced harmful emissions. Recirculating exhaust gases at 3.5%, 7%, 10.5%, and 14% rates increased the emissions such as CO_2, PM, and BOSCH smoke number but lowered NOx emissions. Another study utilized a dual-fuel with waste plastic oil and biogas to investigate its impact on engine performance, combustion, and exhaust gas generation [31]. The results indicated lower torque, power, brake thermal efficiency, NOx, and smoke than the results of diesel, while a single biofuel was comparable to WPO20%. A recent experiment was implemented on a four-stroke single-cylinder diesel engine fueled by 20% WPO with different proportions of nonmetallic nano additive to enhance the characteristics of performance and emissions [32]. The study revealed that the W20 combination without additives reduced brake thermal efficiency by 3.5% and increased brake specific fuel consumption by 5.7% under full load conditions. The 20NA75 blend, however, increased BTE by 2.6% and decreased fuel consumption by 3.1%. The W20NA75 blend, with 75 ppm rice husk nanoparticles, reduced emissions of hydrocarbon, CO, and smoke opacity, while increasing NOx emissions by 14.1% compared to diesel. Most research on WPO use in diesel engines has been experimental, with limited numerical studies [33]. Additionally, studies primarily focused on 20% or 30% biodiesel-diesel blends rather than a range of ratios. The ongoing study aims to fill this gap by using Diesel-RK software to analyze the effects of blending pure diesel with 10%, 20%, 30%, 40%, 50%, 70%, and 100% WPO at different loads. The study examines how these blends impact diesel engine

Table 1. Diesel and WPO blends characteristics [14, 16, 34–44].

Properties	Unit	Diesel	WPO 10%	WPO 20%	WPO 30%	WPO 50%	WPO 70%	WPO 100%	ASTM #
C	%	87.00	86.89	86.77	86.66	86.44	86.21	85.87	—–
H	%	12.60	12.71	12.71	12.93	13.15	13.38	13.71	—–
O	%	00.40	00.402	00.41	00.41	00.41	00.415	00.42	—–
LHV	MJ/kg	45.84	45.29	44.74	44.19	43.09	41.99	40.35	D240
CN	—	53.40	53.06	52.72	52.38	51.70	51.02	50.00	D976
Density 323K	kg/m^3	830.83	830.50	831.00	831.50	832.50	833.50	835.00	D1298
Kinematic viscosity	mm^2/s	02.70	02.68	02.67	02.65	02.61	02.573	02.51	D445
Molecular mass	g/mol	190.00	194.68	199.40	204	213.40	222.70	236.80	—–

emissions, performance, and combustion characteristics.

2. Materials and Methods

This study focuses on the process of producing biofuel components from plastic waste. In order to obtain the required oil, the primary waste products are collected from dumps and then heated to extract the required oil, using the pyrolysis method. A dedicated section of computer code has been developed for the Diesel-RK software that allows the physical properties of all diesel oil mixtures, including Rapeseed Methyl Ester (RME) and Soybean Methyl Ester (SME), and Waste plastic oil (WPO), etc., to be calculated. Furthermore, the user interface in the software has been modified to allow users to customize their preferred biofuel-diesel blends. Each value fell within an acceptable range, indicating its potential as a viable substitute for conventional diesel.

3. Biodiesel preparation and engine test

The current study investigates the diesel engine's thermal characteristics using seven test conditions of 0%, 10%, 20% 30%, 50%, 70% and 100% WPO, blended with 100%, 90%, 80%, 70%, 50%, 30%, and 0% diesel, respectively, using pure diesel as the baseline for comparison. Diesel-RK, a numerical simulation program, serves as the foundation for the inquiry. The properties of diesel and biodiesel under study are listed in Table 1. The engine used is a Kirloskar model with a displacement volume of 661.45 cm^3. The engine's specifications are shown in Table 2.

Table 2. Details about the engine [34].

Engine Type	Kirloskar diesel engine
Type of product	TAF-1
Kind of engine	DI, 4-stroke, one-cylinder
Bore x Stroke	87.5 mm × 110 mm
Compression ratio	17.5
Rated power	4.41 kW
Cooling type	Liquid (water) cooling
Speed of the engine	1500 rpm
Injection type	Injecting directly
Injection pressure	160 bar
Diameter of nozzle	0.15 mm
Injection timing	20°BTDC

4. Numerical Analysis

Diesel-RK was developed in 1981–1982 by the Bauman Moscow State Technical University's Internal Combustion Engines (Piston Engines) department. With the use of sophisticated mathematical models and algorithms, this software allows researchers to improve the engine parameters, including opening of the inlet and exhaust valves, timing of the injection, spark timing, and other aspects. Additionally, it enables engine simulations under various conditions, such as dual-fuel, exhaust gas recirculation, and turbocharging. To model engines using various biofuels and their blends, the fuel library tab allows users to add other fuels beyond conventional gasoline and diesel. Diesel RK employs advanced combustion and emission models, such as the Zeldovich mechanism and detailed kinetic process for NOx formation, heat release rate graphs, $p-v$ diagrams, and $p-\theta$ diagrams. The feature of the 3D tool represents spray development on a flat diagram. Figure 1 demonstrates the operating flow diagram for the IC engine simulations. Initial engine settings include bore, stroke, compression ratio, speed, fuel injection pressure, fuel sort, fuel characteristics, and exhaust gas recirculation ratio. Results and graphs are produced by scanning the ICE simulation. Once the simulation is complete, users can easily access the required results and performance analysis. Figure 1 displays the operational structure for the Diesel-RK software. The gas exchange system and recirculation of exhaust gases, turbocharging, fuel choice, operating modes, basic engine parameters, combustion chamber design, fuel injection system, and simulation execution are covered in the simulation after engine design. A multi-zone combustion model is employed, where each zone functions as an open thermodynamic system.

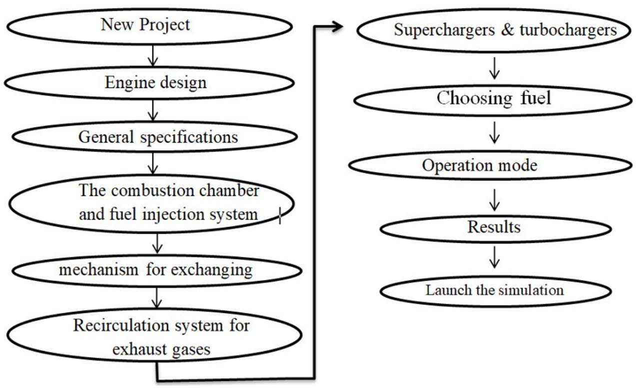

Figure 1. Operational flow diagram for Diesel-RK software.

4.1 Conservation equations

As displayed in Eq. 1 [45], an open system's overall mass flow is preserved, where m_i stands for the mass flow rate for each species.

$$\frac{dm}{dt} = \sum_i m_i \tag{1}$$

The equation defines the mathematical expression for species conservation Eq. 2.

$$Y_i = \sum_i \frac{m_i}{m} \tag{2}$$

Y_i stands for the mass fraction of each species. The following basic energy equation may be used to represent open thermodynamic systems, Eq. 3.

$$\frac{d(mu)}{dt} = -p\frac{dv}{dt} + \frac{dQ_{ht}}{dt} + \sum_i m_i h_i \tag{3}$$

Equation 3 displays the entropy flow, the rate of displacement work and heat transfer on the right side, and the energy change rate on the left.

4.2 The spray evaluation model

This study uses a prototype for multi-area combustion with fuel spraying in accordance with reference [46]. A Fuel is fed into the combustion chamber and separated by a number of distinct zones [47]. The following statement can represent an Elementary Fuel Mass (EFM) traveling in a short period of time step out of the injector to the end for spraying, Eq. 4:

$$\left[\frac{U}{U_0}\right]^{3/2} = 1 - \frac{l}{l_m} \tag{4}$$

Where:
U: An indicator for the elementary fuel mass current speed.
U_0: The starting speed of the nozzle injector of the elementary fuel mass.
l: The separation from the injector to the elementary fuel mass
lm: The elementary fuel mass's penetration length till it comes to a halt ahead

of a spray.

The following is a partial solution to the statement of mathematics Eq. 5. A simplified diagram of the spray is displayed in Fig. 2.

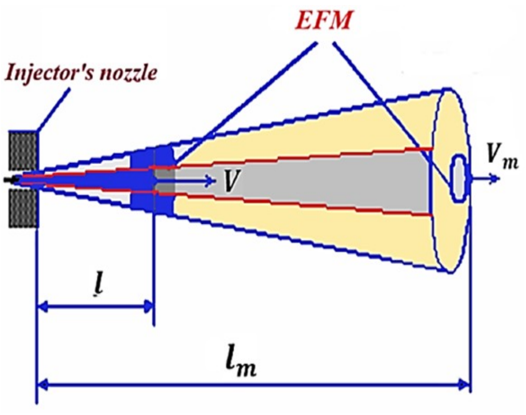

Figure 2. A spray nozzle design [47].

$$3l_m \times \left[-\left(1-\frac{l}{l_m}\right)^{0333} + 1 \right] - U_o\ \tau_k = 0 \quad (5)$$

Where:
τ_k: EFM's travel time proportional to the distance from the injector nozzle. As the EFM of a spray tip stops, $[\tau_k = \tau_m]$.
τ_m : The time required for EFM to reach the spraying head rather than the tip, Eq. 6.

$$l_m = U_0 \frac{\tau_m}{3} \quad (6)$$

Applying the Eq. 4 to Eq. 6. The distance and current speed of EFM are obtained as, Eq. 7 and Eq. 8.

$$U = U_0 \left(1 - \frac{\tau_k}{\tau_m}\right)^2 \quad (7)$$

$$l = \left[1 - \left(1 - \frac{\tau_k}{\tau_m}\right)^3\right] \times l_m \quad (8)$$

4.3 Fuel allocation for the spraying

As shown in Fig. 3, the Diesel-RK combustion model divides the spray into seven different zones, with different evaporation and combustion requirements for each zone. Only three zones must be taken into account prior to jet impingement during the free spray phase, as listed below:

- A compact conical core
- A strong leading face
- A thin outer layer
- An impenetrable (NWF) layer on the piston surface
- An inaccessible (NWF) layer in front
- An unbreakable (NWF) layer ahead
- A reduced topmost shell of the (NWF)

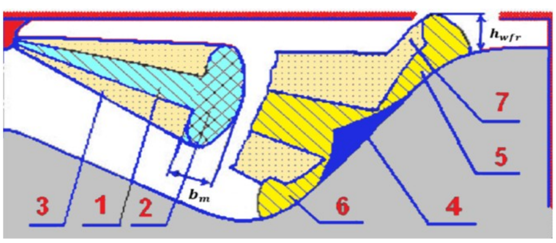

Figure 3. The spray fuel zones [48].

Near the wall, the flow is heterogeneous, making the calculations of vaporized fuel more difficult. As a result, free spray measurements may be utilized to differentiate between mass, pattern regions, and heat transmission characteristics in boundary flows. A new set of zones must be assessed because of the wall impact effects.

4.4 Model of Heat Release

Fuel combustion occurs in four stages, each of which has unique physical and chemical properties that regulate the burning rate [49, 50].
a. Delay period,Eq. 9 .

$$\tau = \sqrt{\frac{T}{p}} \times e^{\left(\frac{E_a}{8312T} - \frac{70}{CN+25}\right)} \times (3.8 \times 10^{-6}) \times (n - 1.6 \times 10^{-4}) \quad (9)$$

b.Premixed combustion phase, Eq. 10.

$$\frac{dx}{dt} = \varphi_1 \times \left\{\frac{d\sigma_u}{d\tau}\right\} + \varphi_o \times A_o(0.1\sigma_{ud} + x_o)(\sigma_{ud} - x_o)\frac{m_f}{V_i} \quad (10)$$

c. Diffusion combustion, Eq. 11.

$$\frac{dx}{dt} = \left\{(\Phi - x)(\sigma_u - x) \times \frac{m_f}{V_c} \times A_2\right\} \times \varphi_2 + \frac{d\sigma_u}{d\tau} \times \varphi_1 \quad (11)$$

d. End of combustion, Eq. 12.

$$\frac{dx}{dt} = \varphi_3 A_3 K_T (1-x)(\varepsilon_b \phi - x) \quad (12)$$

where $\varphi_o = \varphi_3 = \varphi_2 = \varphi_1$, which is a function that characterizes the areas' overall fuel vapor combustion.

4.5 The growth of NOx Modeling

The formation of NOx is explained by the Diesel-RK software using the Zel'Dovich altered mechanism. Nitrogen dioxide and nitric oxide typically combine to form NOx emissions [51]. Diesel engines do not produce major emissions beyond NO, so only NO formation is investigated through thermal processes Eqs. 13, 14 and 15.

$$O_2 \leftrightarrow 2O \quad (13)$$

$$O + N_2 \leftrightarrow NO + N \quad (14)$$

$$O_2 + N \leftrightarrow NO + O \quad (15)$$

Equation 15 illustrates how the atomic oxygen concentration affects this process. The NO volume concentration was determined using the following Eq. 16, [52]:

$$\frac{d[NO]}{d\theta} = \frac{233 \times 10^7 P \times e^{-\frac{38020}{T_Z}} [N_2][O] \times e^{\left(1-\left(\frac{[NO]}{[NO_e]}\right)^2\right)}}{RT_Z\left(1 + \frac{2365}{T_Z} \times e^{\frac{3365}{T_Z}} \times ([NO]/[O_2]) \times e\right)} \times \frac{1}{rps} \quad (16)$$

4.6 Soot Model

As a result of the chemical reaction during combustion, soot particles form and oxidize. The concentration of soot particles has a major impact on environmental pollution Eq. 17, [47].

$$[C] = \int_{\theta_B}^{480} \frac{d[C]}{d\tau} \frac{d\theta}{6n} \left(\frac{0.1}{p}\right)^{\gamma} \quad (17)$$

$[C]$ represents the cylinder's current soot content, and $\gamma$is the exhaust gas's adiabatic factor. The following formula could be applied to determine particulate matter (PM) emissions utilizing the Bosch Eq. 18.

$$[PM] = 565 \times \left[ln \frac{10}{10 - Bosch}\right]^{1.206} \quad (18)$$

Additionally, the air pollutant emissions equation (SE) is another that combines both PM and NO_x pollutants [53].

5. Validation

Simulation results were validated by comparing them with those of M. Mani et al. [17] as depicted in Fig. 4 and 5. Similar engine data and operating conditions were used in the database for application software. Table 3 outlines the engine's operating conditions.

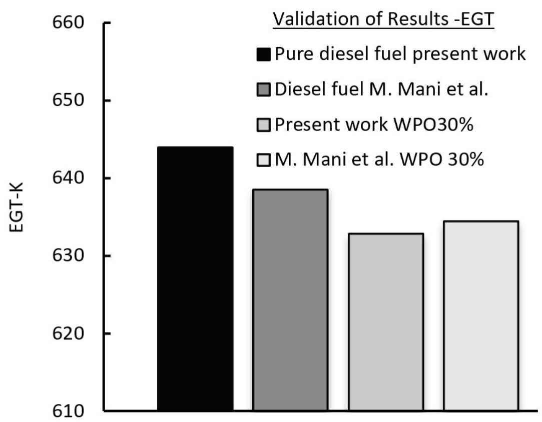

Figure 4. Exhaust gas temperature validation using additional research.

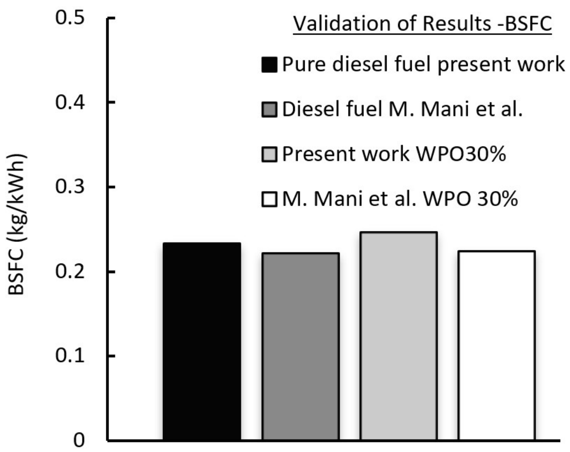

Figure 5. BSFC validation between the present study and another work.

Table 3. Characteristics of the fuels that were utilized for validation [17].

Characteristics	Waste plastic oil	Pure diesel
Density (Kg/m^3)	840	840 to 880
Calorific value (kJ/kg)	44340	46500
Kinematic viscosity (mm^2/sec)	2.52	2.0
Cetane number	51	55
Sulphur content (%)	<0.002	<0.035
Flash point (°C)	42	50
Fire point (°C)	45	56
Aromatic content (%)	55	20

Table 4. The specifications Four stroke engine, CI, air cooled, single cylinder, DI diesel engine), [17].

Engine characteristic	Details
Model's make	Kirloskar TAF1
Bore(mm)	87.5
Stroke (mm)	110
Compression ratio	17.5:1
Power rating at 1500 rpm (kW)	4.4
Nozzle opening pressure (bar)	200
Retarded injection timing (°CA)	17°BTDC

Table 5. Characteristics of the three engine configurations utilized for verification [11, 33, 54]

Test facility	Setup-1 Kirloskar	Setup-2 Kirloskar TV1	Setup-3 Legion Brothers
Kind of engine	4-stroke One-Cylinder, CI engine	4-stroke One-Cylinder, CI engine	4-stroke One-Cylinder, CI engine
Bore × Stroke (mm)	87.5 × 110	87.5 × 110	80 × 110
Cooling system	Ail cooled	Ail cooled	Water cooled
Compression ratio	17.5:1	17.5:1	17.5:1
Rated power output	7 BHP 1500 rpm	5.2 kW 1500 rpm	3.7 kW 1500 rpm
Injection pressure, (bar)	200	160	200

The program's fuel repository now includes these attributes and requirements as shown in Table 3. Table 4 presents the engine parameter. Fig. 4 displays the exhaust gas temperature data for diesel and 30% WPO under full load conditions. The variation in results is 0.84%, and 0.25% for regular diesel, and 0 30% WPO, respectively. Figure 5 displays the BSFC values for both pure diesel and 30% WPO under full load conditions, the results align with those of M. Mani et al. [17], with a deviation of only 5% for diesel and 9% for 30% WPO. The Diesel-RK software is a powerful and effective modeling tool that mimics a diesel engine's combustion process. To further validate the findings of other researchers, an additional set of validations is generated by the program. It employs identical configurations and operating conditions, which are compared with those produced in [11, 33, 54] under identical functional conditions. For the ongoing study, the same configurations are used in its databases. Table 5. presents the technical specifications of the engines utilized for validation. Figures 6 and 7 illustrate the pressure progression and the spray formation regarding the crank angle. The analogy shows a strong convergence with minimal deviation, with little discrepancy observed. The Diesel-RK's low variance confirms its reliability for simulating combustion in an IC engine.

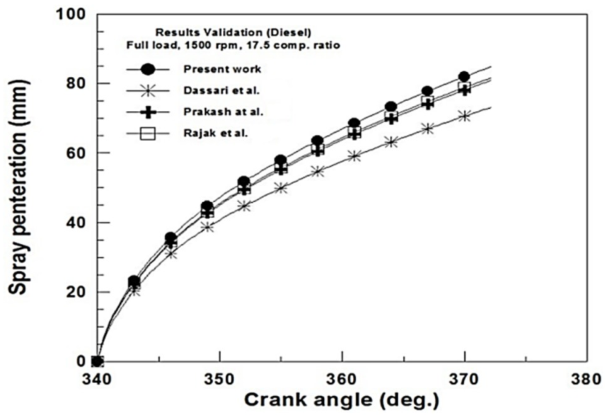

Figure 6. Spray dispersion versus crank angle validation.

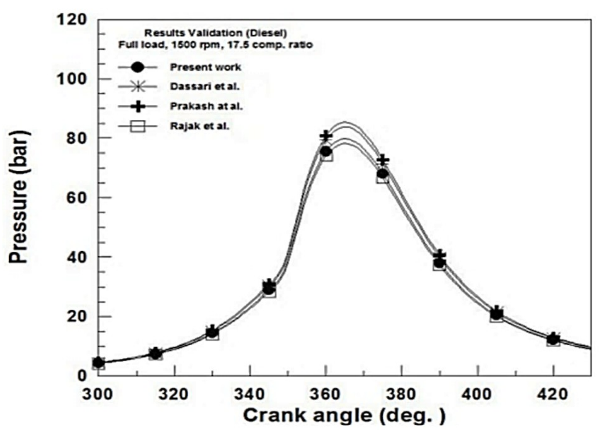

Figure 7. Cylinder pressure versus crank angle validation.

6. Results and Discussion

The full-load setting is selected due to engine's lowest air-to-fuel ratio, facilitating the comparison of different blends with respect to performance, combustion properties, and emissions of pollutants. Additionally, a variable load scenario is chosen for several parameters. The impact of waste plastic oil on performance, emission, and combustion parameters is discussed in the following sections.

6.1 Combustion Characteristics

Figure 8 shows the variation of cylinder pressure per crank angle, starting at 180°BTDC and ending at 180∘ ATDC. This 360°phase represents the power cycle, comprising compression, combustion, and expansion strokes. The primary difference observed is the variation in peak pressure magnitude and timing, which depends on the WPO percentage. The cylinder pressure for WPO fuel is lower than that of conventional diesel. The difference in lower heating values and oxygen concentrations of the tested fuels leads to a decrease in cylinder pressure as the WPO blending ratio increases. Peak pressure ranges from 10.07 MPa for pure diesel to 7.9 MPa for 100% WPO. The use of 10%, 20%, 30%, 50%, 70%, and 100% WPO reduces peak pressure by 1.09%, 1.5%, 2.08%, 3.25%, 7.26%, and 21.64%, respectively, compared to pure diesel. Figure 9 presents the variations of zonal combustion temperature for diesel and WPO blends with respect to crank angle.

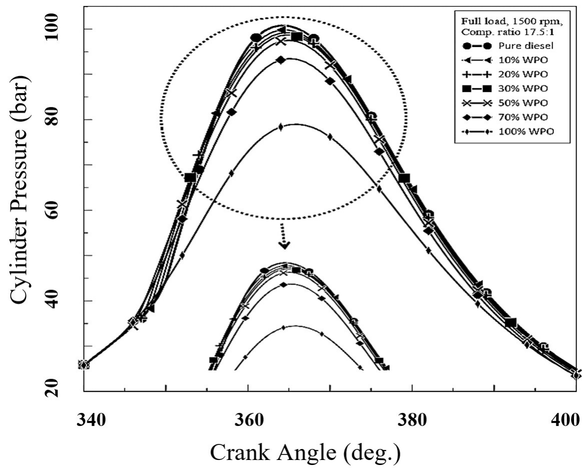

Figure 8. Cylinder pressure with crank angle.

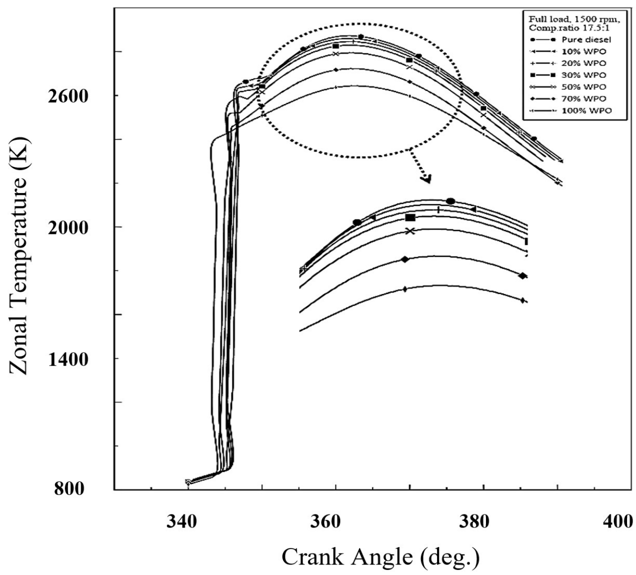

Figure 9. Zonal temperature vs. crank angle.

The combustion temperature decreases as the WPO blending ratio increases, caused by variations in oxygen content. In contrast to all WPO blends, diesel combustion starts earlier due to higher cetane number. Diesel has an average maximum temperature of 2873 K, whereas 10, 20, 30, 50, 70, and 100% WPO biodiesel had temperatures of 2860 K, 2846 K, 2830 K, 2795 K, 2723 K, and 2644 K, respectively. The heat release rate pattern provides critical insights into the onset of combustion and heat distribution during the combustion stroke [11, 55]. Figure 10 illustrates the variation in heat release within 340°–390°crank angle range. The computational step considered is 0.2°crankshaft angle because major changes occur within this interval. Compared to diesel, all WPO blends have a late ignition start. Figure 10 present the heat release rate fbleor different blends. The blending of WPO with diesel reduces the peak heat release rate from 68.8 (J/deg) at 350.4°of pure diesel to 66.93 (J/deg) at 351.8∘, 65.55 (J/deg) at 350.2°, 62.11 (J/deg) at 349.4°, 48.96 (J/deg) at 348.6°, 42.74 (J/deg) at 355.4°and 33.75 (J/deg) at 364.4°for 10%, 20%, 30%, 50% 70% and 100% of WPO respectively. This is attributed to variations in viscosity, cetane number, and heating energy. The study conducted by [53] captures similar results, where the Sauter Mean Diameter (SMD) variance for diesel and WPO blends is shown in Fig. 11.

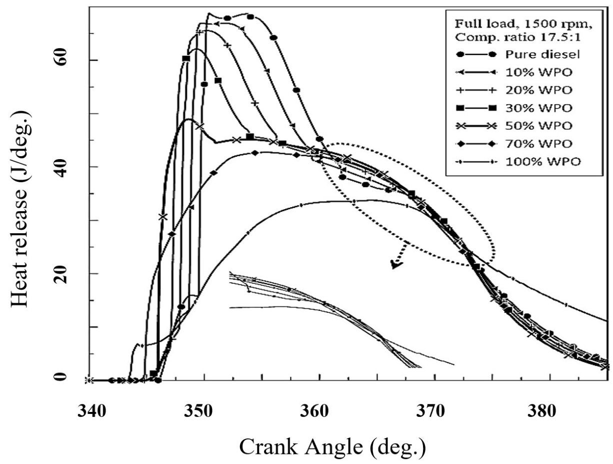

Figure 10. Rate of heat release rate vs. crank angle.

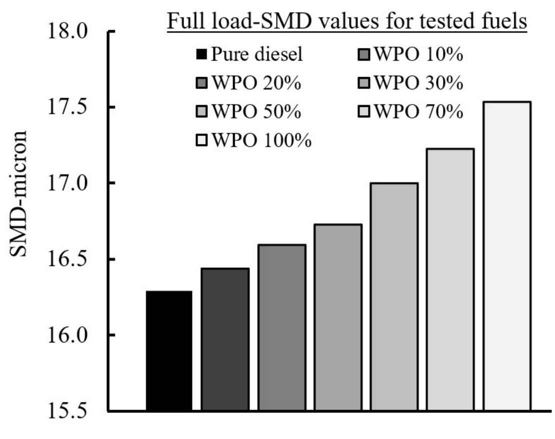

Figure 11. Variation of SMD values with WPO blends.

Compared to diesel fuel, where viscosity is a primary factor influencing the change in SMD, the mixes of WPO biodiesel have a slightly lower viscosity and a greater density and surface tension. As the size of the droplets increased, the fuel burning rate and velocity decreased, leading to a slower combustion process. As a result, the biodiesel combustion showed higher SMD than the neat diesel. The current results are consistent with findings from [56–58]. The SMD increases by 0.91 percent, 1.85 percent, 2.68 percent, 4.29 percent, 5.74 percent, and 7.63 percent for 10%, 20%, 30%, 50%, 70%, and 100% WPO, respectively. An important parameter to consider while researching combustion processes is ignition delay, particularly in gas turbines and internal combustion engines. It represents the time interval between the start of fuel injection and the onset of a discernible heat release from combustion. The ignition delay is composed of two main phases: physical and chemical delays.

The time required for the fuel to undergo fundamental physical mechanisms before its chemical reaction is considered as the physical delay, involving atomization of fuel (Fuel breaking down into little droplets), fuel vaporization (The transformation of fuel from liquid to vapor), and fuel vapor combining with the oxidizer (typically air). The duration of physical delay depends on the fuel properties (e.g., vaporization of latent heat and boiling point), pressure of injection, combustion chamber design, and environmental elements (e.g., pressure and temperature) [59]. The chemical delay is the time required for the chemical reactions leading to combustion. It begins when fuel molecules interact with an oxidizer, triggering chain reactions that result in an exothermic combustion process. It is affected by the fuel composition, temperature, pressure, and the presence of any catalysts or inhibitors [60]. The effects of the various WPO blends on the start of combustion (SOC) are depicted in Fig. 12.

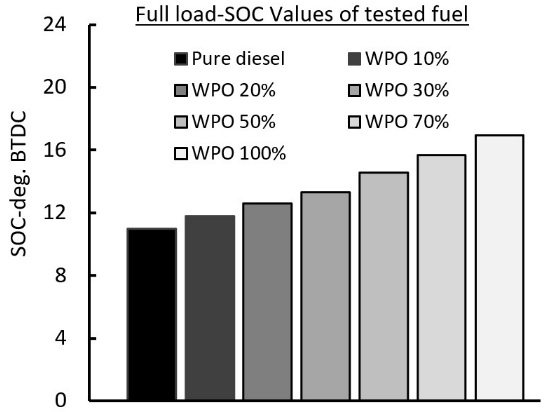

Figure 12**.** Variation of SOC values with WPO blends.

Cetane number frequently affects the ignition delay; blending WPO with diesel generally increases the delay period, retarding the onset of combustion. A high cetane number indicates a short delay period, meaning the alternative fuel blends burn after pure diesel. According to the results, combustion of neat diesel occurs at 10.97°BTDC, while WPO blends progress to 11.8°, 12.6°, 13.33°, 14.54°, 15.7°, and 16.95°, BTDC for 10%, 20%, 30%, 50%, 70%, and 100% of WPO, respectively.

6.2 Performance Characteristics

Figures 13 and 14. show the BSFC for diesel and WPO blends. As the WPO ratio increases, BSFC gradually rises BY 2.20%, 4.13%, 6%, 9.23%, 15.15%, and 16%, for 10%, 20%, 30%, 50%, 70%, and 100% of WPO, respectively, due to its higher density value compared to diesel, aligning with previous studies [61]. The higher fuel consumption rate in WPO blends is attributed to their lower heating value. The variation of brake specific fuel consumption (BSFC) with engine load at various WPO blending ratios is depicted in Fig. 14.

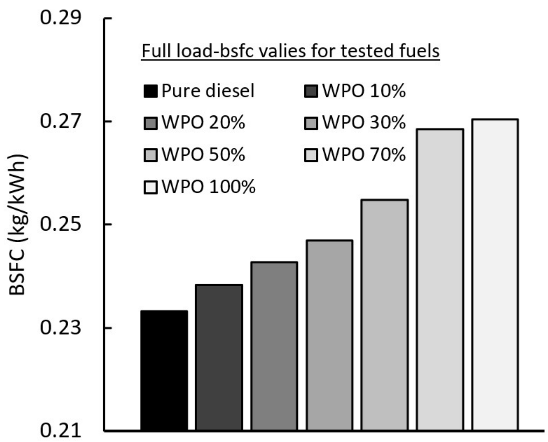

Figure 13**.** Variation of BSFC values with WPO blends.

BSFC dropped as engine load rose, which is attributed to the engine overloading, where brake power rises faster than fuel consumption. Furthermore, WPO burns more thoroughly due to its oxygenated nature, further reducing BSFC. However, as WPO blend ratios increased, BSFC increased as well since biodiesel has a higher density and a lower heating value than pure diesel. similar results were reported in [62]. The BTE values for the diesel blends are shown in Fig. 15 and 16. Due to variations in viscosity and heating values, which lowered atomization and vaporization, net diesel exhibited a higher BTE, which declined as the proportion of WPO increased. According to a study by [62, 63], BTE variations might also be examined by looking at the phases, time, and heat release rate during combustion.

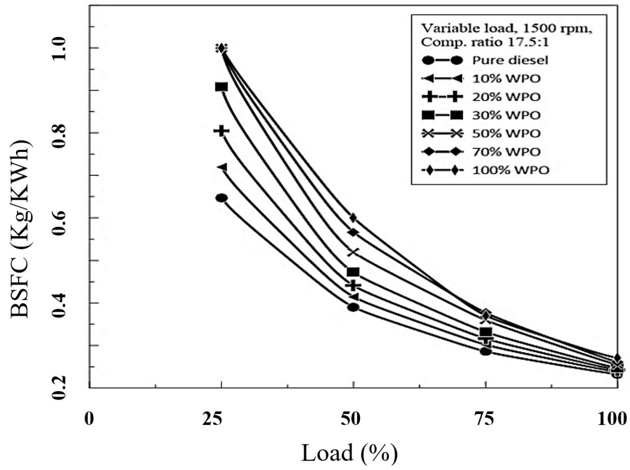

Figure 14**.** Variation of BSFC values with WPO blends at variable load.

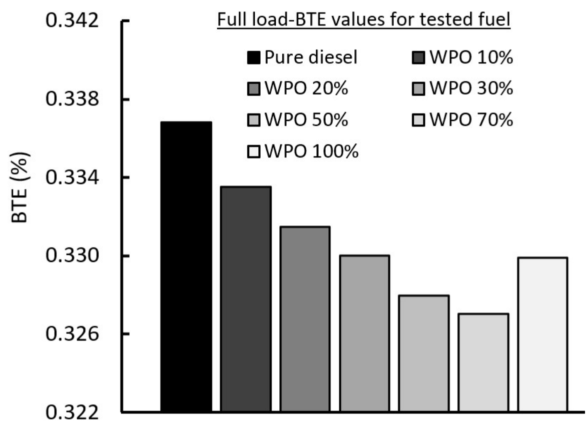

Figure 15**.** Variation of BTE values with WPO at full load.

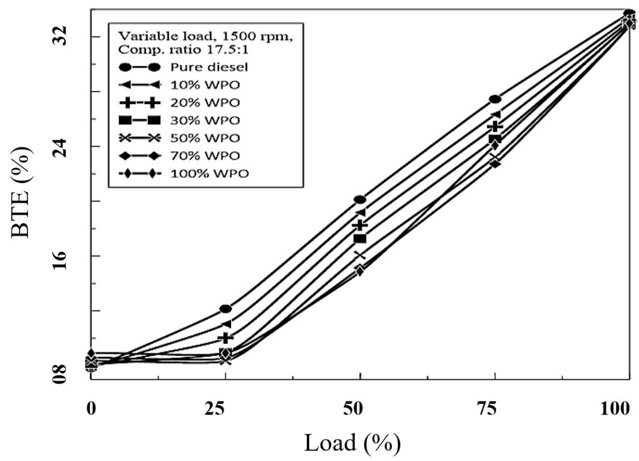

Figure 16**.** Variation of BTE values with WPO at variable load.

The use of 10%, 20%, 30%, 50%, 70% and 100% of WPO reduced the BTE by 0.99%, 1.58%, 2.03%, 2.62%, 2.89% and 2.05% respectively. The exhaust gas temperature (EGT) values for the fuel mixes were analyzed as shown in Fig. 17. Due to variations in viscosity and heating values, which inhibited atomization and vaporization, net diesel exhibited a higher EGT, which declined as the proportion of WPO increased. Research suggests that variation in EGT might also be investigated by looking at the phases, time, and heat release rate during combustion. The graphic shows how the EGT of the tested fuels varied. The use of 10% ,20%, 30%, 50%, 70% and 100% of WPO reduce the EGT by 0.64%,1.2%,1.73%,2.81%, 4.7%, and 1.24% respectively.

6.3 Emissions characteristics

Figure 18 shows the soot concentration for diesel, and WPO mixes throughout a 60°crank angle period. Diesel has a greater soot content (6.26 g/m^3), but incorporating 10%, 20%, 30%, 50%, 70% and 100% of WPO results in a reduction of 6.02%, 5.5%, 5.23%, 5.26%, 5.36% and 4.36% respectively. This is attributed to increased oxygen content, which helps fuel oxidation in the rich zone, leading to reduced smoke emissions. The oxygen presence in fuel reduces soot formation. Additionally, biodiesel's lower carbon-to-hydrogen (C/H) ratio compared to pure diesel contributes to reduced smoke emissions. Research of [63, 64] reports similar findings. The significant drop in the Bosch Smoke Number (BSN) is another advantage of using WPO biodiesel. As the 30% WPO reduced in BSN by 2.85%, compared to pure diesel fuel, while the other blends fluctuated, as seen in Figs. 19 and Fig. 20 at full load and variable load. The particulate matter (PM) emissions for diesel and WPO blends are displayed in Fig. 21.

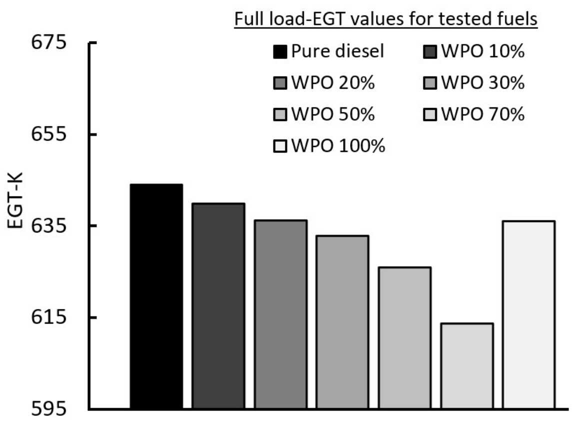

Figure 17**.** Variation of EGT values with WPO blends.

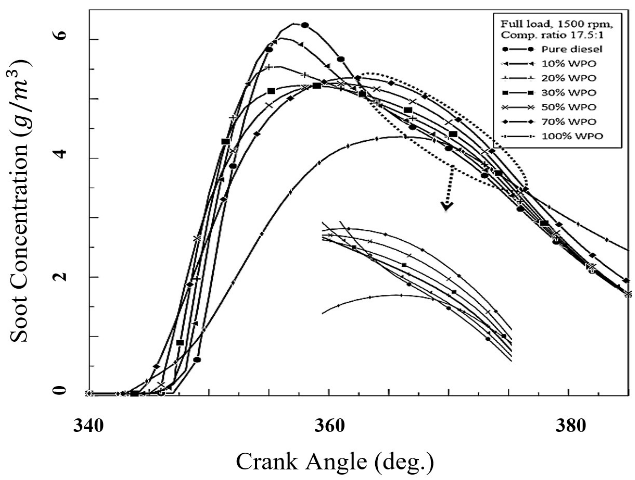

Figure 18**.** Changes in soot concentration according to crank angle.

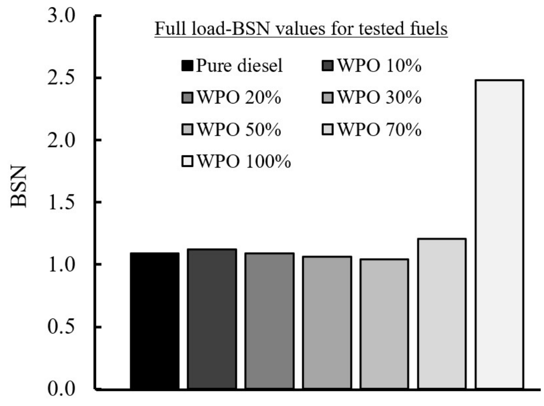

Figure 19**.** Variation of BSN values with WPO blend at full load.

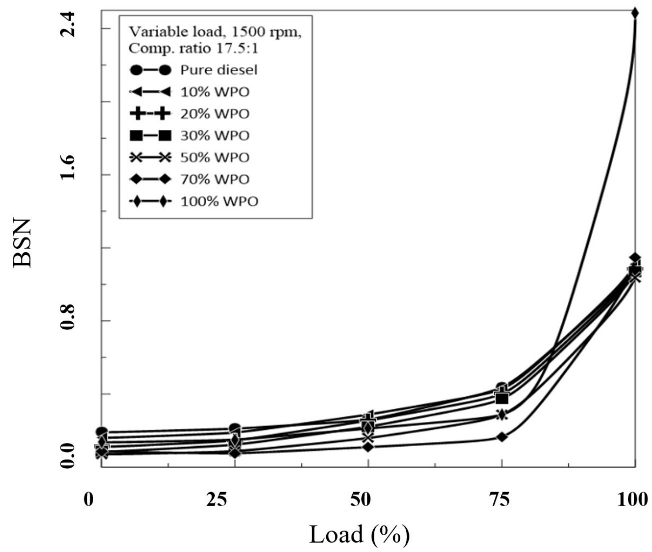

Figure 20**.** Variation of BSN values with WPO blend at variable load.

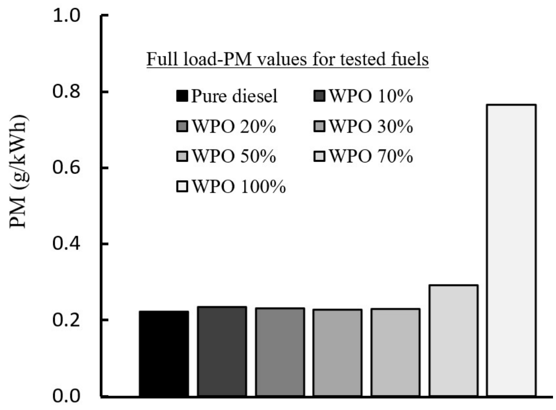

Figure 21**.** Variation of PM values with WPO blends at full load.

The use of WPO blends increases PM emission, with the lowest increase in WPO 30% by 1.960% compared to diesel. Figures 22 & 23 shows nitrogen oxide level (NO_x) emissions across a 60°crank angle from (340 to 400) degrees. A Crank angle caused the NO_x emissions to increase over time, peak, and then progressively decrease. It's observed that higher WPO blend ratios reduce NOx emissions by 2.47%, 5.15%, 8.61%, 18.10%, 43.09%, and 78.10% for WPO 10%, WPO 20%, WPO 30%, WPO 50%, WPO 70%, and WPO 100%, respectively. The decrease in nitrogen oxides is due to the decrease in temperatures [65].

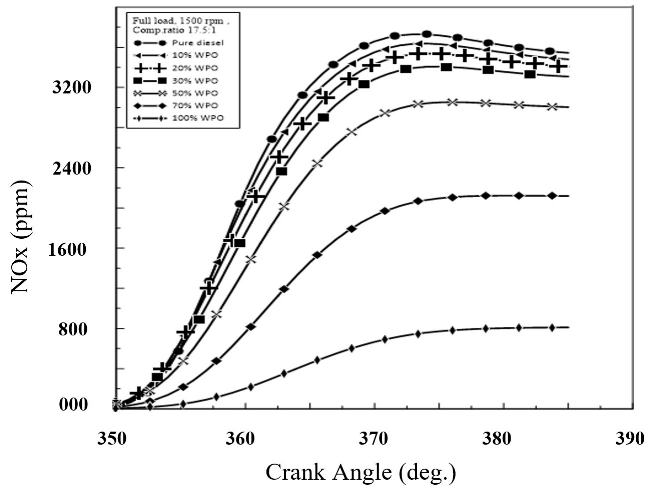

Figure 22**.** The change of NOx amounts with the WPO blends.

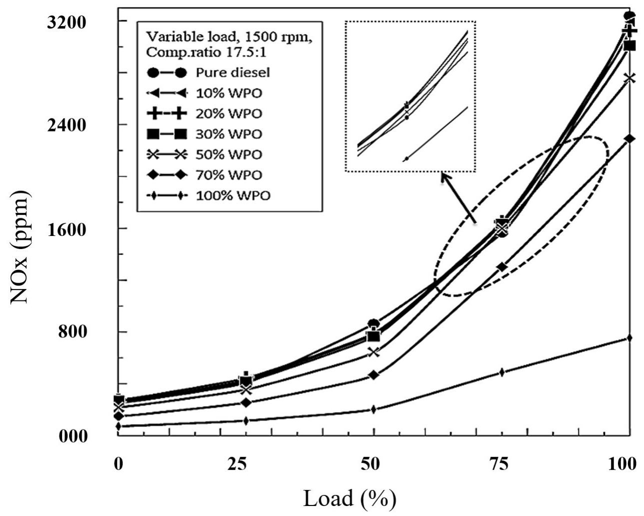

Figure 23**.** NO_x variation with WPO blends at variable load.

In conclusion, nitrogen oxides formation is influenced by three factors: high temperatures that provide the energy needed for the process; an oxygen-rich environment that provides the required reactant and sufficient time for the reaction to occur. Combustion processes are a key source of NO_x due to these conditions [66,67].The addition of WPO biodiesel reduces the NO_x emissions, compared to pure diesel. Higher content of oxygen has two functions: one is to reduce carbon emissions, and the other is to raise NO_x emissions. At full load, NO_x emissions are recorded as 3189.6, 3126, 3038.2, 2762.6, 1954.2 and 753.1 ppm, for 10%, 20%, 30%, 50%, 70% and 100% WPO biodiesel, respectively, compared to 3245.6 ppm for pure diesel.

7. Conclusions

The current study uses numerical analysis to explore the effects of seven distinct volumetric blends of waste plastic oil (WPO) biodiesel on diesel engine thermal parameters compared to conventional diesel fuel operations. The key findings are as follows: (1) One possible strategy to reduce diesel fuel usage and address the growing carbon emissions is the use of WPO biodiesel. (2) Blending WPO with diesel results in a slight decrease in cylinder pressure, temperature, and heat release rate. (3) The diesel-WPO blends cause a delayed start of combustion attributed to lower cetane number, which increases the ignition delay. (4) When using WPO blends instead of diesel, a higher sauter mean diameter (SMD) is observed compared to pure diesel. (5) The temperature of exhaust emissions is reduced by 0.64%, 1.20%, 1.73%, 2.81%, 4.7%, and 1.24% when using 10% ,20% , 30%, 50%, 70% and 100% WPO respectively, compared to pure diesel. (6) When WPO is added to diesel fuel, the reduced C/H ratio and oxygen content compared to diesel result in a significant drop in the soot concentration, with minor changes observed in BSN. (7) At 30% WPO, a drop in BSN by 2.855% is observed compared to pure diesel. (8) A sharp decrease in nitrogen oxide (NOx) emissions compared to diesel fuel, due to a decrease in temperatures, which helps prevent the decomposition of nitrogen. The rate of NOx reduction is 2.47%, 5.15%, 8.61%, 18.10%, 43.09% and 78.10% for WPO 10%, WPO 20%, WPO 30%, WPO 50%, WPO 70% and WPO 100%, respectively, compared with pure diesel. (9) A slight increase in PM was observed compared to diesel fuel, with the lowest increase seen in 30% WPO at 1.96%. (10) Further optimization research is recommended to examine how the designated mixes react to varying compression ratios, engine speeds, and injection timings. (11) Conducting experimental research is recommended to examine the practical impacts of WPO biodiesel blends on the characteristics of diesel engines.

Authors' contribution

All authors contributed equally to the preparation of this article.

Declaration of competing interest

The authors declare no conflicts of interest.

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This study didn't receive any specific funds.

Data availability

The data are available from the corresponding author upon reasonable request.

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