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Optimising Injection Strategies and EGR in Modified Piston Diesel Engines Fuelled with Waste Plastic Oil

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Abstract

Growing concerns over plastic pollution and fossil fuel depletion have driven research toward alternative fuels and engine optimisation strategies. The transition from fossil-based fuels to alternative fuels is imperative for reducing greenhouse gas emissions and addressing plastic waste. This study investigates the synergistic effects of injection pressure, ignition timing and exhaust gas recirculation on a single-cylinder diesel engine with modified piston bowl geometry using waste plastic oil-diesel blends. Waste plastic oil, derived via pyrolysis of municipal plastic waste, was blended with diesel in varying proportions (25–100%). Tests on engines were performed with varying injection pressures (200–500 bar), ignition timing (17°−23° before top dead centre) and exhaust gas recirculation rates (0–9%). Results reveal that the D25WP75 blend at 500 bar and 23° before top dead centre offers a peak brake thermal efficiency, minimum brake specific fuel consumption, and reduced emissions of CO, HC, NO_x and smoke opacity. Optimal exhaust gas recirculation at 6% further reduced NO_x. The maximum cylinder pressure and heat release rate are obtained at 73.02 bar and 52.66 J/deg. The improvement percentage in brake thermal efficiency at 500 bar for the D25WP75 blend compared with D100 is 10.8%, with a reduction in brake specific fuel consumption of 11.76%. It is also observed that NO_x reduces by 5.46%, CO by 44%, HC by 8.82% and smoke by 35.38%. The combined impact of piston geometry modification and injection strategies improved combustion uniformity and emission control. The findings suggest that integrating fuel modification with combustion optimisation offers a viable pathway to cleaner diesel engine operation.

Keywords: Injection pressure; Injection timing; Piston bowl geometry modification; Exhaust gas recirculation, Emissions; Waste plastic oil

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1. Introduction

Diesel engines are playing a significant role in the industrial, power, automotive and agricultural sectors because of their heavy-duty application and high thermal efficiency. However, with the increase in population, global energy demand and strict emission regulations, it is mandatory to improve fuel economy and reduce the emission concentrations evolving from diesel engines. Also, the challenge of plastic waste accumulation has become a global concern. One promising solution lies in the conversion of municipal plastic waste into usable liquid fuel via pyrolysis. Transforming waste plastic into fuel through pyrolysis provides a dual advantage, such as waste reduction and the production of energy-dense alternative fuels [1]. Recently, many

Nomenclature

Abbreviations and Acronyms

BSFC– brake specific fuel consumption
BTE – brake thermal efficiency
bTDC– before top dead centre
CA – crank angle
CO – carbon monoxide
CP – cylinder pressure
CR – compression ratio
D25WP75 – 25% diesel + 75% waste plastic oil
D50WP50 – 50% diesel + 50% waste plastic oil
D75WP25 – 75% diesel + 25% waste plastic oil
D100 – 100% diesel
EGR – exhaust gas recirculation
HC – hydrocarbon
HRR – heat release rate
IP – injection pressure
IT – ignition timing
NO_x – oxides of nitrogen
PPO – plastic pyrolysis oil
VCR – variable compression ratio
WPO – waste plastic oil
WP100 – 100% waste plastic oil

researchers have concentrated on the conversion of waste plastic into biodiesel because of increased utilisation of plastic and its waste across the globe [2]. Also, waste plastic causes pollution, which affects the environment by contaminating land and oceans mainly. It is estimated that 396 million tonnes of plastic waste were produced globally in the year 2018, which is expected to double in the next 20 years. The method known as pyrolysis is employed to convert waste plastic into plastic oil [3]. Waste plastic oil (WPO) is derived through pyrolysis, a thermochemical process in which plastic waste is decomposed at elevated temperatures in an inert atmosphere, typically in the absence of oxygen. This method breaks down long-chain polymer structures into shorter hydrocarbon molecules, yielding liquid fuel, gases and a minor amount of solid residue. The overall efficiency and product yield of the pyrolysis process can be significantly improved by introducing suitable catalysts, which promote cracking reactions and influence the composition of the resulting oil [4,5].

A comprehensive review by Pradeep et al. [6] examined the application of WPO as a fuel in diesel engines, focusing on its impact on engine performance, combustion efficiency and emissions. Their study compared thermal pyrolysis with catalytic pyrolysis and emphasised how different plastic feedstocks and catalyst types affect the quality and suitability of the oil for engine use. The review highlighted the potential of WPO as a viable alternative to conventional diesel, especially when produced and utilised under optimised conditions. Mangesh et al. [7] examined diesel engine performance using pyrolysis oil-diesel blends (5%, 10%, 15%). It was observed that the increased heat release rate (HRR) and ignition delay are attributed to alkenes in the oil. However, the blends showed a lower brake thermal efficiency (BTE) due to higher viscosity and reduced calorific value. NO_x emissions were found to be lower. In contrast, Janarthanan et al. [8] reported higher NO_x emissions with WPO, due to elevated adiabatic flame temperatures.

Pal et al. [9] tested blends from 25% to 100% plastic oil, noting a higher in-cylinder pressure (CP) and HRR, which led to increased NO_x emissions. However, BTE dropped by up to 6.8%, while CO and smoke emissions decreased by 21.8% and 4.47%, with 100% plastic oil at full load. Yaqoob et al. [10] evaluated diesel engines using 5–15% plastic pyrolysis oil (PPO) blends. The 5% blend achieved the highest thermal efficiency at 51.6%, outperforming pure diesel (47.44%), and showed reduced CO emissions. Similarly, Sudalaimani et al. [11] explored the use of WPO blends in a variable compression ratio (VCR) diesel engine. They observed that increasing the compression ratio by 12–20% improved BTE across all blends. The blends showed a slightly higher HRR and elevated combustion temperatures, leading to increased NO_x emissions. The study suggests using higher compression ratios in compression ignition (CI) engines running on plastic oil blends to optimise both BTE and brake specific fuel consumption (BSFC). Previous studies have shown mixed results for engines using WPO blends. Lower blend ratios often exhibit comparable performance to diesel, but higher WPO ratios may reduce brake thermal efficiency and increase NO_x emissions. To address these limitations, researchers have explored strategies such as adjusting injection pressure (IP), ignition timing (IT), and using exhaust gas recirculation (EGR).

Youssef et al. [12] researched the impact of varying compression ratio (CR) and IT, affecting emissions of NO_x in engines powered by diesel, biodiesel and butanol blends. They observed that advancing IT from 15° to 5° before the top dead centre (bTDC) significantly lowered NO_x emissions, a trend also noted by Munivenkaeshappa et al. [13]. Similarly, Khoa et al. [14] achieved NO_x reduction through increased EGR. Ashok et al. [15] studied diesel engines running on pure calophyllum inophyllum biodiesel under various ITs and EGR levels. While retarding injection timing reduced NO_x by 11%, it compromised engine performance. Advancing timing led to a 21% rise in NO_x. EGR was more effective, with NO_x emissions dropping by 51%, 54% and 57% at increasing EGR rates under full load. Table 1 shows a brief description of the effects of EGR and WPO biodiesel blends on diesel engines.

Barik et al. [24] also reported improved performance and lower hydrocarbon (HC) emissions when IT was advanced from 23° to 24.5° bTDC, compared to standard timing and diesel. Agarwal et al. [25] evaluated the cumulative impact of IT and IP on spray behaviour, combustion and emissions using a B20 blend derived from karanja oil. It has been noticed that there is a reduction in NO_x emissions. Hirkude et al. [26] examined the influence of IT, IP and CR on diesel engine performance using a blend of WPO and diesel. The study revealed an increase in BTE and a reduction in brake specific fuel consumption (BSFC). However, raising the compression ratio from 18 to 19 slightly reduced BTE and increased BSFC – an expected engine behaviour. An optimal performance was achieved at 27° bTDC IT, 250 bar IP, and a CR of 18. EGT rose with increases in IP, IT and CR, while smoke opacity declined. Shrivastava et al. [27] conducted an experimental evaluation of roselle biodiesel blends under varying IP and engine load conditions. Their findings indicated that at an IP of 220 bar, the RB20 blend exhibited a marginal increase in CO emissions by approximately 1.6%, relative to conventional diesel. However, this was accompanied by a reduction in smoke and NO_x emissions by 2.2% and 3.18%, respectively, suggesting that optimised injection parameters can enhance the combustion characteristics of biodiesel while mitigating key exhaust pollutants. Teoh et al. [28] used the grey-Taguchi method to optimise diesel-coconut oil blends based on parameters such as blend ratio, engine speed and load. The method identified the best blend for minimising emissions and maximising performance at 3850 rpm and 25% load.

Table 1. The effect of EGR and WPO biodiesel blends on diesel engines.

Paper	Focus	Key findings
Chaitanya et al. [16]	1-hexanol/WPO blends, EGR	Higher EGR rates reduce NO_x but decrease brake thermal efficiency; higher alcohol blends can mitigate some emission issues.
Kumar et al. [17]	WCO+WPO blends, EGR, optimisation	Optimal blend and EGR settings improve efficiency and emissions; RSM is used for optimisation.
Saha et al. [18]	EGR, nano-additives, WPO-diesel-water emulsion	5% EGR offers the best trade-off: lower NO_x, minimal penalty on CO, HC, smoke, and fuel consumption.
Kaewbuddee et al. [19]	n-butanol/WPO blends	Alcohol addition reduces efficiency, increases HC/CO; optimisation needed for best trade-off.
Rajendran et al. [20]	Plasto-oil/diesel blends, PCCI, EGR	EGR reduces NO_x, but can increase carbon-based emissions; blends outperform diesel alone.
Saravanan et al. [21]	WPO, EGR, nano-coated chamber	EGR reduces NO_x, but increases CO/HC; nano-coating improves combustion performance.
Jayanth et al. [22]	WHDPE oil/diesel blends, EGR, injection strategy	10% EGR with optimized injection improves BTE, reduces NO_x and smoke.
Balaji et al. [23]	WPO/diesel blends, EGR	EGR (up to 15%) reduces NO_x, increases smoke; blends operate without engine modification.

Kumar et al. [29] explored the effects of EGR and IT on the performance and emission behaviour of CI engine fuelled with biofuels-diesel blends. At the optimisation point, NO_x was reduced to 12.4% and BSFC was increased to 2.9% when compared to that of neat diesel. Hirkude et al. [30] examined the combined effects of CR, IT and IP on the performance and emission behaviour of a CI engine operating on a blend of waste fried oil biodiesel blends. Their experimental analysis identified optimal operating parameters at a CR of 18, IT of 27° bTDC and an IP of 250 bar. Adjustments to both IT and CR were found to significantly enhance overall engine efficiency while concurrently reducing harmful exhaust emissions, thereby confirming the importance of integrated tuning for biodiesel applications.

The earlier research has shown that the modification of injection parameters with biodiesel blends has shown a significant effect on diesel engine working characteristics. However, studies often examine these variables in isolation, without considering the synergistic effects of modifying the combustion chamber geometry, specifically the piston bowl shape. In view of this, further studies are needed to study the impact of different injection strategies on a modified piston diesel engine with waste plastic oil.

Therefore, the current work made an attempt to change the piston geometry, and in addition to this, the injection pressure and injection timing of the engine were varied to study the effect on the modified piston geometry engine with varied proportions of WPO. The study is novel in its integrated approach, specifically the use of waste plastic oil biodiesel with a modified piston bowl and adjusted injection parameters. Further, the cooled EGR has been mixed with the intake air. The waste plastic oil biodiesel blends were used as a fuel for a single cylinder common rail direct injection (CRDI) VCR diesel engine. To the best of the author's knowledge, no study has been conducted on the modification of piston bowl geometry with different injection strategies and waste plastic oil as a fuel. The gap has been filled with the present study.

The current research work is split into three stages. In the first stage, the biodiesel was produced from waste plastic by the pyrolysis process. In the second stage, piston geometry modification was carried out to find out the influence on the engine working characteristics. In the third stage, the effects of injection pressure, injection timing and EGR on the modified piston diesel engine have been investigated. The present investigation has been carried out at a constant speed of 1500 rpm.

2. Materials and methods

2.1. Waste plastic oil production

In this investigation, WPO has been chosen as a biodiesel source for blending with conventional diesel fuel. The production of WPO was carried out using the pyrolysis technique, a thermochemical method that decomposes plastic waste into usable liquid fuel in an oxygen-free environment. The conversion process was conducted using a dedicated pyrolysis unit, which is illustrated in Fig. 1.

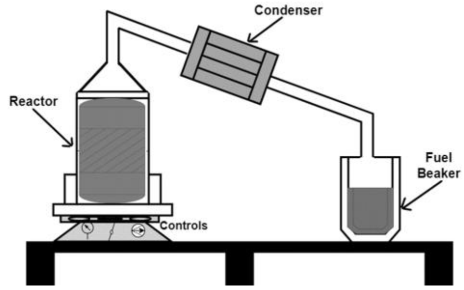


Fig. 1. Plastic oil production from waste plastic (pyrolysis).

This setup enabled the recovery of liquid hydrocarbons suitable for engine testing after appropriate condensation and collection. For the purpose of this experimental investigation, waste plastics were obtained from the municipal solid waste management department in Kurnool, Andhra Pradesh, India, and then pulverised into small pieces with sizes ranging from 0.50 to 1.5 cm^2 in area. To remove dust, dirt and unwanted particles from the plastic, it was washed with water, and a filter was used. It was washed over and over again to get rid of dirt, and then it was dried out in an electric oven set to 150°C to get rid of water. As shown in Fig. 1, the pyrolytic reactor was fed with 6 kg of waste plastic, 0.60 kg of coal (10% by weight of plastic waste), and 0.8 kg of aeolite as a catalyst in a single batch. The pyrolytic reactor consists of a heating chamber, an exhaust chute and a control unit. The sliced plastic was fed to the reactor in a sequential manner based on the batch size. The reactor operates in the range up to 900°C. A thermocouple was used to measure the temperature, and it is digitally controlled. The process followed to produce the waste plastic oil is that the crushed waste plastic is heated in a heating chamber within the temperature range of 600°C to 800°C for thermal cracking of waste plastic. The temperature was maintained for 3 hours. The produced gases due to thermal cracking of waste plastic occupied the chamber's uppermost layer before being moved to the cooling area (condenser) through the exhaust chute so that they could be condensed. For the condensation process, the cooling chamber was filled with water, which condenses the gases and forms the pyrolytic oil. The produced pyrolytic oil is collected in a beaker as illustrated in Fig.1. As shown in Fig. 2, the produced WPO was mixed with diesel fuel in various ratios. Table 2 lists the results of the test samples' properties in compliance with the American Society for Testing and Materials (ASTM) standard procedure.

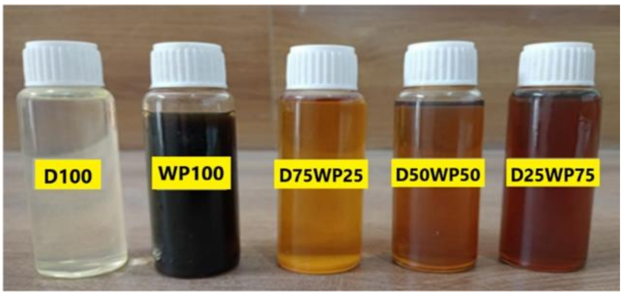


Fig. 2. Physical appearance of diesel, plastic oil and their blends.

Table 2. Fuel properties of different blends.

No.	Characteristics	WP100	WP75–D25	WP50–D50	WP25–D75
1	Kinematic viscosity at 40°C, cSt	1.45	1.55	1.57	1.47
2	Flash point, °C	Nil	27	28	35
3	Fire point, °C	32	30	34	40
4	Gross calorific value, kJ/kg	42 763	42 697	42 631	42 565
5	Density, kg/m^3	813	822	826	837

2.2. Experimental setup

To conduct the experimental analysis, a single-cylinder, water-cooled diesel engine equipped with a common rail direct injection (CRDI) system was employed. A schematic diagram of the experimental setup is presented in Fig. 3, while detailed engine specifications are listed in Table 3.

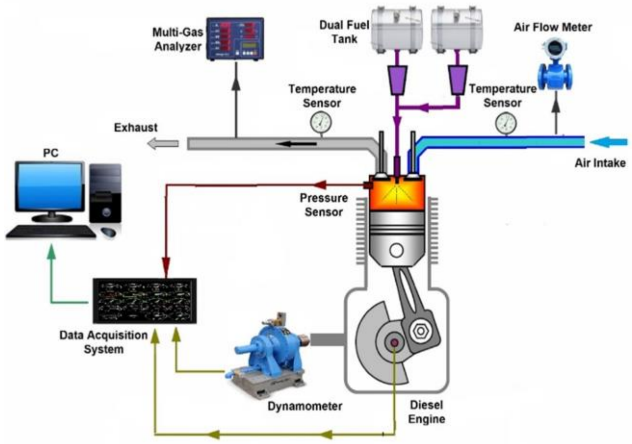


Fig. 3. Schematic experimental setup for the present study.

Table 3. Description of diesel engine.

Description	Specifications
Make	Kirloskar
Type	4 stroke single cylinder CRDI diesel engine
Compression ratio	12–18
Bore × stroke	87.5 mm × 110 mm
Swept volume	661cc
Rated power	3.5 kW
Length of connecting rod	234 mm
Rated speed	1500 rpm

Engine loading was managed using an eddy current dynamometer, which enabled precise application and control of load conditions. A proximity sensor has been employed to record engine speed, and calibrated flow transmitters were employed to determine the air and fuel flow rates. All test data, including combustion and performance parameters, were acquired and processed using a computerised engine analysis system. Exhaust emissions, specifically NO_x, HC and CO, were continuously monitored using a multi-gas analyser. The CRDI system featured electronically controlled fuel injection, allowing for independent adjustment of IP irrespective of engine speed. An external EGR loop was integrated into the intake system to reduce NO_x emissions. The recirculated exhaust gases were passed through a water-cooled heat exchanger to lower their temperature before mixing with the intake air.

2.3. Engine piston bowl geometry modification

For this experimental work, instead of the traditional hemispherical cavity combustion chamber (HCP), the piston bowl shape was modified to create a toroidal cavity combustion chamber (TCP). Surface grinding was used to remove material until the desired form was obtained. The bowl capacity was maintained constant for both combustion chamber arrangements, such that the compression ratio matched that of the typical engine. Figure 4 depicts the cross-sectional and photographic views of the two pistons used in this study.

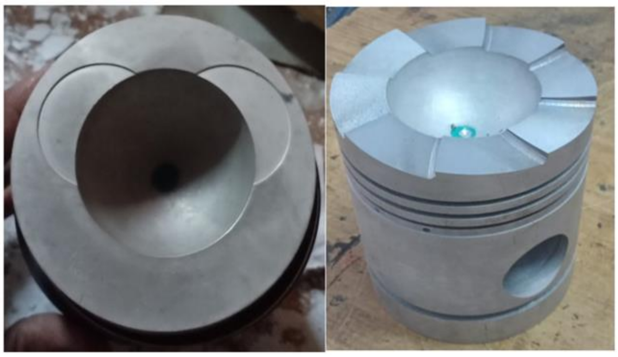


Fig. 4. Real view of unmodified and modified piston geometry.

Diesel engine combustion is influenced by the combustion chamber's form and the fluid dynamics present inside. The geometry of the combustion chamber must be thoroughly studied since the combustion chamber shape can act as a guide for the turbulence. The parameters used to choose the design of the combustion chamber will produce a strong squish and air movement, comparable to that of the well-known smoke ring. In the combustion chambers, the coupling between the swirl, squish, bowl shape, and turbulence is even more prominent. The quality of air fuel mixing and squish action is determined by the cavity created by the modified piston.

2.4. Injection pressure variation

The variation in injector opening pressure was achieved by manually adjusting the spring preload within the injector assembly. This was done by tightening or loosening the adjustment screw located at the top of the injector housing, thereby altering the force required to lift the nozzle valve. Experimental trials were subsequently conducted at four distinct injection pressures, such as 200 bar, 300 bar, 400 bar and 500 bar, using the engine equipped with the modified piston bowl geometry. These settings were selected to evaluate the impact of IP on fuel atomisation, combustion behaviour and emission characteristics under controlled test conditions.

2.5. Exhaust gas recirculation setup

This experiment uses the cold EGR approach since it lowers cylinder temperature and raises charge density. The inlet manifold was designed to receive the exhaust gases. Based on optimisation work done by the authors [31], the rate of recirculation was varied. The exhaust fumes were surrounded by a tube that circulates tap water, lowering the temperature of the gases to 350°C. A cold EGR with an orifice diameter of 4 mm was used in the setup.

$$\% \text{ of EGR} = (CO_2 \text{ at (inlet)})/(CO_2 \text{ at (outlet)}) \times 100.$$

The above equation shows the regulation of the exhaust gas flow by the control valve.

2.6. Modification in injection timing

The spill technique was applied in order to modify the test engines' initial IT. Finding the spill's position was made easier with the use of a specially designed adaptor with a hypodermic needle tip. By adjusting the quantity of shims inside the pump, static or original IT might be changed. The number of shims inside the fuel injection pump can be changed in order to alter the timing of the fuel injection.

Three shims were previously fitted in the injection pump when the engine was initially manufactured. A 0.3 mm thick shim was added to acquire the delayed IT, then the shim was removed to obtain the advanced IT. Depending on how many shims are in the pump, each shim has the capacity to deliver either advanced or retarded flow.

The time of injection was manually changed by employing the spill timing method by altering the number of shims within the fuel pump assembly. Each addition or removal of shims effectively advanced or retarded the injection timing, allowing for precise calibration of fuel delivery relative to the crank angle. The engine was significantly affected by the changes to the fuel injection timing brought about by [32].

3. Results and discussion

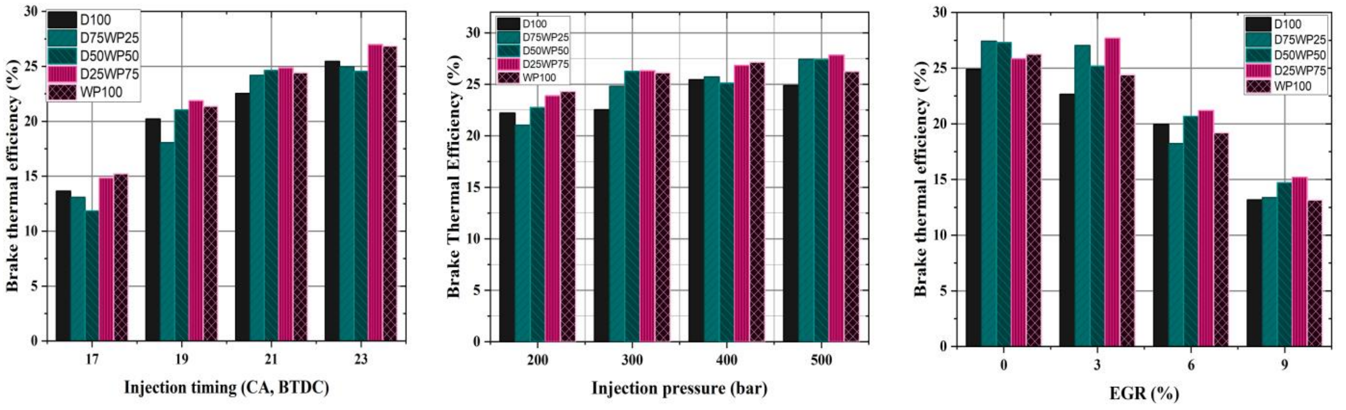
The performance, combustion and emissions characteristics of the engine with a modified geometry piston at different injection strategies (IT and IP) and variable EGR in percentage were determined and compared with pure diesel.

3.1. Brake thermal efficiency

Brake thermal efficiency (BTE) represents the effectiveness of converting chemical energy from the fuel into useful mechanical energy. This parameter also helps in finding the combustion behaviour of the engine [33]. In this study, a consistent increase in BTE was observed with increasing IP and advanced IT, whereas a reduction in BTE was noted with the increased EGR ratio. Figure 5(a, b and c) shows the variations in BTE of the modified piston geometry engine with neat diesel, plastic oil and their blends at different injection strategies (IT and IP) and various EGR percentages.

The highest BTE of 27.69% was achieved for the D25WP75 blend at 500 bar and 23° bTDC. This improvement can be attributed to several synergistic factors such as enhanced atomisation, intense swirl, turbulence and a higher calorific value of WPO. Moreover, the fuel blends D25WP75 and WP100 show a maximum BTE compared with the other fuel blends at all IPs and IT.

Notably, at lower IPs (200–300 bar), BTE values were significantly lower due to poor atomisation, especially for higher WPO blends, confirming that the injection pressure plays a dominant role in fuel-air preparation for viscous fuels.

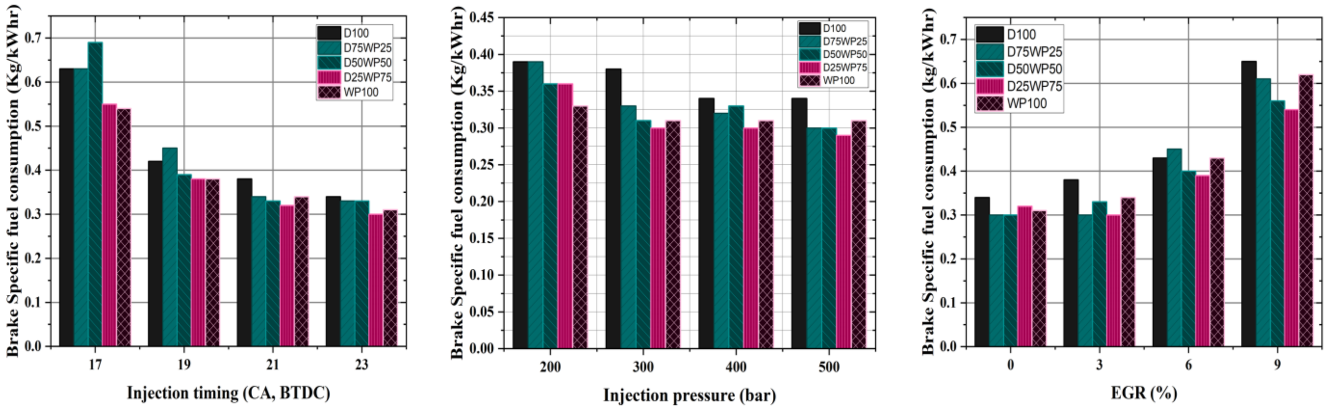


a) b) c)

Fig. 5. Change in BTE in relation to: a) IT, b) IP, c) EGR.

3.2. Brake specific fuel consumption

Brake specific fuel consumption **(**BSFC) is inversely related to engine efficiency, representing the amount of fuel consumed per unit of power output. Figure 6(a, b and c) depicts the variation in BSFC of the engine with modified piston head geometry, fuelled with neat diesel and its plastic oil blends, with respect to different injection strategies and EGR. A consistent decline in BSFC has been noted with increasing IPs and IT across all WPO blends. The lowest BSFC, 0.31 kg/kWh, was recorded for the D25WP75 blend at 500 bar and 23° bTDC. This was primarily due to efficient combustion facilitated by enhanced atomisation and optimal combustion phasing [34]. Figure 6a shows the plot of BSFC with respect to IT (CA, BTDC), and Fig. 6b illustrates the variation in BSFC with respect to IP, which shows that advanced IT and higher IPs also reduce BSFC for modified piston geometry because of better atomisation and vaporisation. It was also noted that BSFC increases with the increase in EGR percentage, and a minimum was found at 3% EGR for the blend D25WP75, as the carbon present in the exhaust gases leads to improper combustion of fuel.



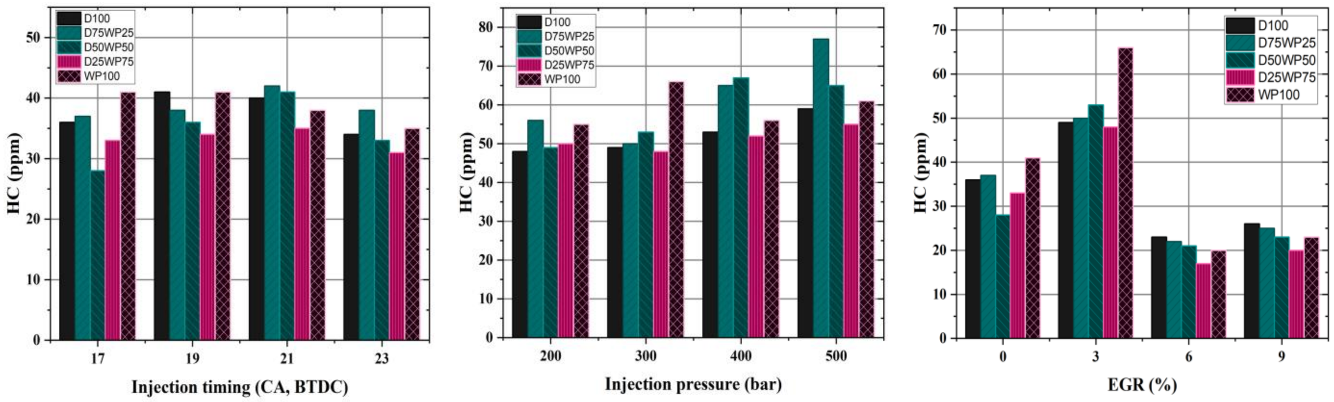
a) b) c)

Fig. 6. Change in BSFC in relation to: a) IT, b) IP, c) EGR.

3.3. Hydrocarbon emissions

Hydrocarbon (HC) emissions typically originate from flame quenching and crevice volume fuel deposition. The influence of IT, IP and EGR rate on HC emissions is illustrated in Fig. 7(a, b, c).

From Fig. 7a, it is observed that for conventional diesel, HC levels are relatively higher at an IT of 18° bTDC and decrease progressively as the timing is advanced to 23° bTDC. Notably, at 21° and 23° bTDC, higher concentration WPO-based blends emit lower hydrocarbons than diesel, likely due to more favourable combustion phasing and enhanced volatility of the blend. Figure 7b shows that IP exerts a notable effect on HC formation. For neat diesel, HC emissions tend to increase with pressure, peaking at 500 bar. However, under the same condition, WPO blends – particularly at higher concentrations – demonstrate reduced HC emissions. This contrast suggests that high-pressure injection enhances fuel atomisation more effectively in WPO blends, improving vaporisation and reducing incomplete combustion zones. As shown in Figure 7c, increasing the EGR rate up to 9% contributes to a further decline in HC emissions for all tested blends. This improvement is attributed to optimised in-cylinder thermal conditions that facilitate oxidation, alongside the dilution effect of recirculated gases, which moderates local flame quenching [35,36]. Additionally, the modified piston



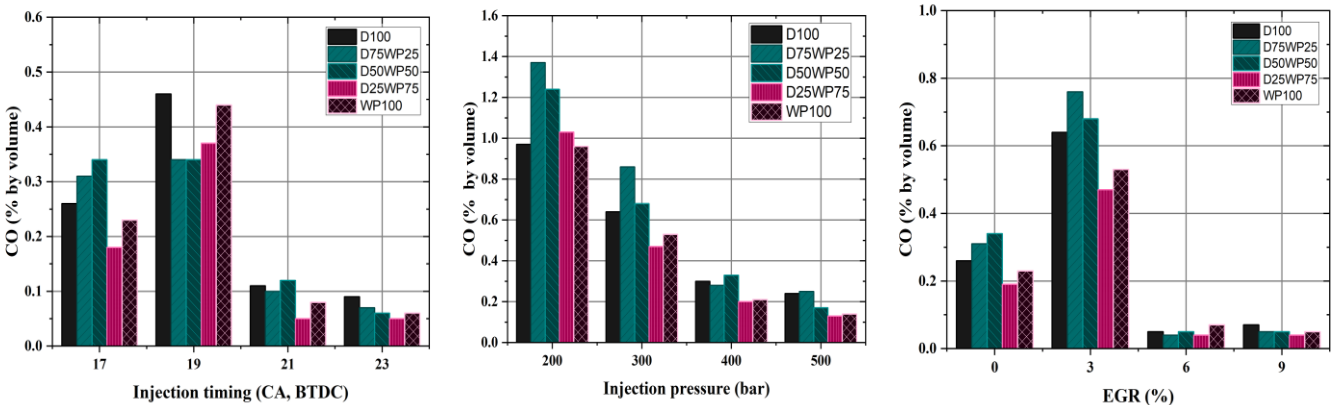
a) b) c)

Fig. 7. Change in HC in relation to: a) IT, b) IP, c) EGR.

bowl geometry contributes to this reduction by promoting efficient air-fuel mixing and limiting quench layer formation.

3.4. Carbon monoxide emissions

Carbon monoxide (CO) emissions arise from incomplete combustion, particularly in rich zones or under low oxygen availability [37]. Other factors that may affect CO emissions include delay duration, short residence time, insufficient or excessive equivalence ratios, and insufficiently high flame temperature [38]. The effects of IT, IP and EGR percentage on CO emissions for diesel, WPO and their respective blends are illustrated in Fig. 8(a, b, c).



a) b) c)

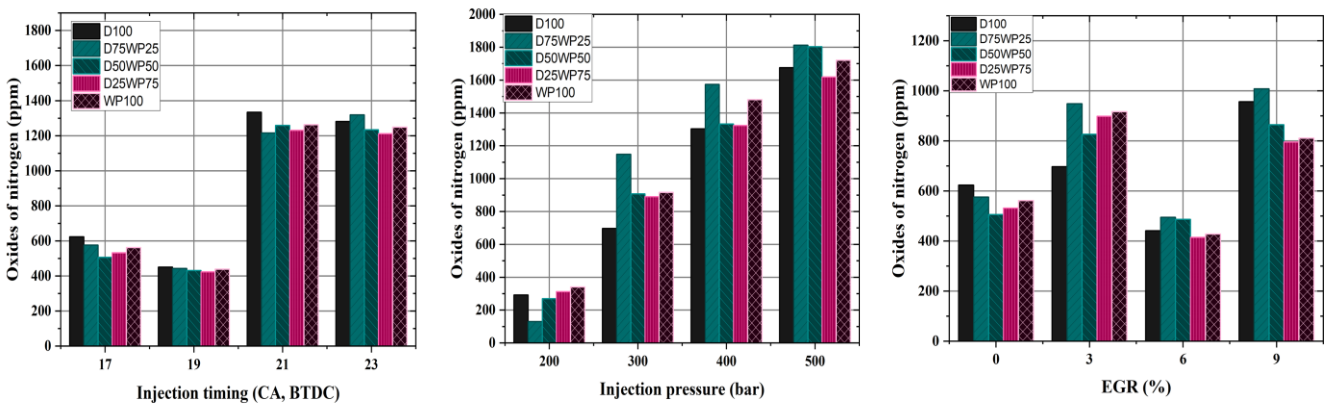
Fig. 8. Change in CO emission in relation to: a) IT, b) IP, c) EGR.

As shown in Fig. 8a, the pure WPO blend (WP100) consistently produced lower CO emissions than conventional diesel across all injection timings. Advancing IT from 19° to 23° crank angle (bTDC) led to noticeable reductions in CO emissions across the fuel samples – by approximately 8%, 16.5% and 10.3%, respectively. These reductions can be attributed to improved combustion phasing, which ensures that a higher fraction of the fuel undergoes oxidation within the optimal crank angle window. A similar trend is evident in Fig. 8b, where increasing the injection pressure resulted in a marked decline in CO emissions for all tested fuels. This behaviour can be linked to the finer fuel spray and enhanced atomisation achieved at higher pressures, which promote thorough mixing with air and more complete combustion.

Figure 8c presents the relationship between CO emissions and EGR percentage. It is evident that CO emissions decreased progressively with increasing EGR rates up to 9% across all fuel types. This trend can be explained by the dilution effect of EGR, which lowers combustion temperatures and facilitates more uniform in-cylinder combustion, thus minimising localised rich zones where CO typically forms. The piston bowl modification, which enhances in-cylinder swirl and promotes superior fuel vaporisation, likely contributed to this reduction as well. Additionally, the higher heat release rate (HRR) observed under optimal injection settings supports faster flame propagation and more complete fuel oxidation, further suppressing CO formation [39].

3.5. Oxides of nitrogen emissions

The formation of oxides of nitrogen (NO_x) during combustion is primarily governed by three interrelated factors: oxygen concentration within the combustion chamber, peak flame temperature, and the residence time of the high-temperature gas. Figure 9(a, b, c) illustrates the variation in NO_x emissions for diesel and WPO-based blends under different injection strategies and EGR rates.



a) b) c)

Fig. 9. Change in NO_x emission in relation to: a) IT, b) IP, c) EGR.

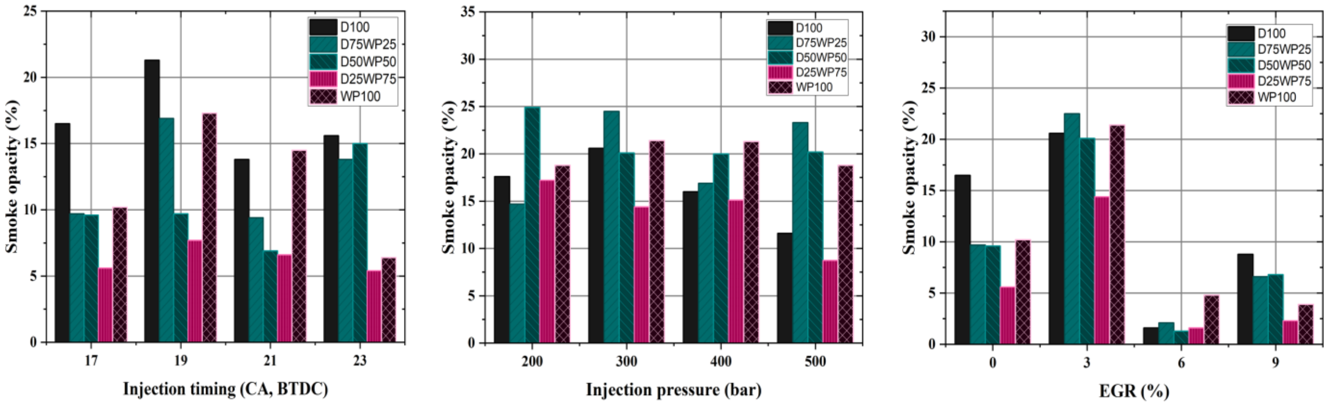
Figure 9a displays the effect of IT on NO_x generation across various fuel blends, including diesel, D75WP25, D50WP50, D25WP75 and WP100. The results indicate that NO_x emissions tend to increase when injection timing is advanced (i.e., closer to the top dead centre), due to the higher in-cylinder pressures and elevated flame temperatures resulting from increased premixed combustion. Conversely, retarding the injection timing to 19° bTDC significantly lowered NO_x formation across all fuel samples. This reduction is linked to delayed combustion phasing, which reduces peak temperature and shortens the duration available for thermal NO_x formation [40,41].

Figure 9b explores the influence of varying IP from 200 bar to 500 bar, on NO_x emissions. A clear trend emerges: as IP rises, NO_x emissions increase notably. This behaviour can be attributed to improved fuel atomisation and air–fuel mixing at higher pressures, which enhances combustion efficiency but also raises combustion temperature – a key driver of NO_x synthesis [42].

The effect of EGR percentage on NO_x mitigation is shown in Fig. 9c. Among the tested EGR rates, 6% emerged as the most effective in reducing NO_x emissions. Introducing EGR into the intake stream displaces a portion of the intake air with inert gases, which in turn lowers oxygen concentration and peak combustion temperature, both essential for limiting NO_x formation. Notably, beyond 6%, additional EGR did not yield significant further reductions and occasionally resulted in compromised combustion stability and elevated hydrocarbon emissions. These observations are consistent with findings by Leung et al. [43], who noted that advancing the injection timing increases peak pressure and temperature, thereby promoting NO_x formation.

3.6. Smoke opacity

The influence of IT, IP and EGR rate on smoke opacity for all tested fuel blends, using a modified piston bowl geometry, is presented in Fig. 10(a, b, c).



a) b) c)

Fig. 10. Change in Smoke Opacity emission in relation to: a) IT, b) IP, c) EGR.

Across all conditions, the D25WP75 blend consistently exhibited lower smoke in contrast to other blends and baseline diesel, demonstrating the effectiveness of the toroidal piston design in improving combustion efficiency. The reduction in smoke emissions is largely attributed to enhanced air motion within the cylinder, which promoted more effective air-fuel mixing and reduced the presence of fuel-rich zones that typically lead to soot formation.

Figure 10a shows the variation of smoke opacity with IT. At a timing of 23° bTDC, D25WP75 achieved the lowest smoke value of 5.3%, a marked improvement over other injection timings and all other fuel combinations tested. Damodharan et al. [44] also noted a similar kind of increment in their studies. This decrease is a result of more complete combustion achieved by better premixing and combustion phasing at this timing. Figure 10b displays smoke emissions as a function of injection pressure. At 500 bar, the D25WP75 blend recorded a smoke opacity of just 8.2%, which is significantly lower than that of diesel and other blends at the same pressure. In their studies, Saba et al. [45] also found that the rise in injection pressure lowers the smoke opacity. Figure 10c illustrates the impact of EGR on smoke emissions. At an EGR rate of 6%, the D25WP75 blend yielded the lowest smoke opacity of 1.1%, benefiting from moderated combustion temperatures and extended ignition delay, which allowed more time for fuel-air mixing. Collectively, these results show that the application of a modified piston bowl geometry, combined with optimised injection and EGR strategies, dramatically reduces smoke emissions. Specifically, in comparison to neat diesel, smoke opacity for the D25WP75 blend was reduced by approximately 62% at 23° bTDC injection timing, 44% at 500 bar injection pressure, and 32% at 6% EGR. These reductions confirm the synergy between in-cylinder flow enhancements and precise fuel delivery control in achieving cleaner combustion with a waste plastic oil blend.

3.7. Cylinder pressure

Figure 11 depicts the fluctuation in in-cylinder pressure versus crank angle under optimal conditions for various fuel blends. Several factors influence cylinder pressure development during combustion, including the energy density and volatility of the fuel, ignition delay and the extent of premixed combustion [46,47].

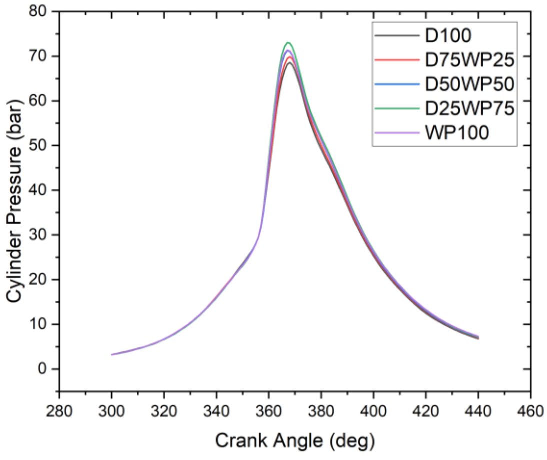


Fig. 11. CP variation in relation to crank angle.

The incorporation of WPO into diesel resulted in a noticeable rise in peak CP for blended fuels compared to neat diesel. Mangesh et al. [48] also noticed that the addition of WPO raises the cylinder pressure in their studies. Among the blends, the D25WP75 blend exhibited the highest CP peak. Quantitatively, the peak cylinder pressures for diesel, D75WP25, D50WP50, D25WP75 and WP100 were observed at 68.57 bar at 368° crank angle (CA), 69.47 bar at 367° CA, 71.21 bar at 367° CA, 73.02 bar at 366° CA and 71.33 bar at 368° CA, respectively. Compared to diesel, the percentage increases in peak pressure were 2.05% for D75WP25, 3.85% for D50WP50 and 4.02% for D25WP75. These results confirm that blending WPO with diesel enhances in-cylinder combustion characteristics, particularly when combined with geometrical and injection optimisations. The elevated peak pressures reflect a more rapid and complete combustion process, which is advantageous for thermal efficiency.

3.8. Heat release rate

Figure 12 shows how the heat release rate (HRR) varies with crank angle for different fuel mixes under optimal operating conditions. HRR is a vital combustion parameter that provides insight into the energy liberation characteristics of the fuel-air mixture within the cylinder. During the ignition delay phase, HRR registers negative values due to the latent heat absorbed during fuel vaporisation and thermal losses to the cylinder walls. Once auto-ignition occurs, HRR becomes positive and exhibits a steep rise, marking the commencement of premixed combustion [49].

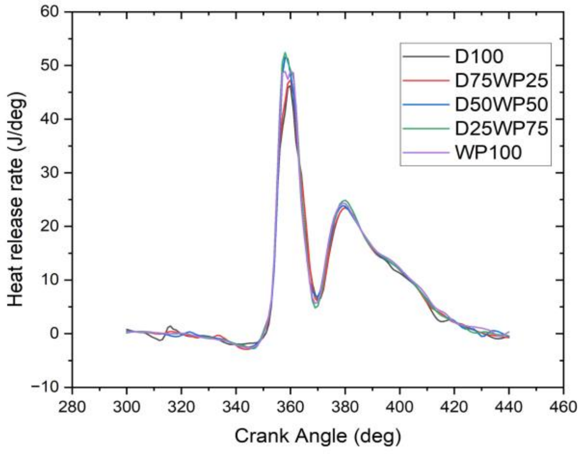


Fig. 12. HRR variation in relation to crank angle.

The experimental outcomes showed that increasing the concentration of WPO in the diesel blend enhances HRR, particularly during the premixed combustion phase. This enhancement is attributed to more favourable combustion conditions enabled by improved volatility and better in-cylinder mixing facilitated by the modified piston bowl geometry. Blends containing higher proportions of WPO exhibited a noticeable advancement in the CA position of peak HRR compared to neat diesel, reflecting a reduction in ignition delay and earlier combustion onset.

Zhang et al. [50] also noticed the same pattern in HRR by adding WPO to diesel blends. Among the tested fuels, the D25WP75 blend delivered the highest HRR peak of 52.66 J/deg at 357° CA, indicating the superior combustion intensity. In comparison, the peak HRR values for diesel, D75WP25, D50WP50 and WP100 were 46.25 J/deg at 360° CA, 47.27 J/deg at 359° CA, 52.48 J/deg at 358° CA and 48.94 J/deg at 360° CA, respectively. These results correspond to an increase of 2.2%, 13.5%, 13.85% and 5.81% in HRR for D75WP25, D50WP50, D25WP75 and WP100 blends relative to diesel. The shift in peak position to earlier crank angles with increasing WPO concentration underscores the role of pre-mixed combustion enhancement in accelerating the heat release process.

4. Conclusions

This experimental investigation explored the combined effects of WPO blending, injection strategies and EGR on the performance, combustion and emission behaviour of a diesel engine equipped with a modified piston bowl geometry. The findings provide compelling evidence that strategic integration of fuel composition and engine parameter optimisation can significantly enhance combustion efficiency and reduce pollutant formation.

Among the tested configurations, the D25WP75 blend (25% diesel and 75% WPO) demonstrated superior performance across all operating metrics. The following conclusions are drawn from the study:

- All WPO-diesel blends showed improved BTE relative to neat diesel, especially under higher IP and advanced IT with no EGR. The highest efficiency gain was observed for the D25WP75 blend.
- The incorporation of WPO led to a measurable reduction in BSFC compared to pure diesel. Specifically, at 23° CA injection timing, the D25WP75 blend exhibited a BSFC decrease of 8.81% relative to diesel.
- Emissions of CO and HC declined with increasing IP and IT for all blends. The D25WP75 blend consistently reported the lowest CO and HC levels across test conditions.
- A rise in NO_x emissions has been noticed with increased IP and advanced IT, due to elevated combustion temperatures. However, this effect was mitigated by introducing cooled EGR. A 6% EGR rate proved optimal in reducing NO_x without adversely affecting performance.
- WPO blends exhibited a pronounced reduction in smoke emissions across all injection settings. Notably, the D25WP75 blend at 23° CA reduced smoke opacity by approximately 64.1% compared to diesel, attributable to improved air-fuel mixing and in-cylinder turbulence.

The optimal engine configuration – D25WP75 blend, 500 bar injection pressure, 23° CA injection timing and 6% cooled EGR – achieved a balanced improvement in efficiency and emissions.

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