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Experimental investigation into the effect of magnetic fuel reforming on diesel combustion and emissions running on wheat germ and pine oil

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ABSTRACT

The present study aims to explore the effect of fuel ionisation on engine performance, emission and combustion characteristics of a twin cylinder compression ignition (CI) engine running on biofuel. Wheat germ oil (WGO) and pine oil (PO) have been identified as diesel fuel surrogates with high and low viscosities, respectively. High viscosity biofuels result in incomplete combustion due to poor atomisation and evaporation which ultimately leads to insufficient air-fuel mixing to form a combustible mixture. Consequently, engines running on this type of fuel suffer from lower brake thermal efficiency (BTE) and higher soot emission. In contrast, low viscosity biofuels exhibit superior combustion characteristics however they have a low cetane number which causes longer ignition delay and therefore higher NO emission. To overcome the limitations of both fuels, a fuel ionisation filter (FIF) with a permanent magnet is installed upstream of the fuel pump which electrochemically ionises the fuel molecules and aids in quick dispersion of the ions. The engine used in this investigation is a twin cylinder tractor engine that runs at a constant speed of 1500 rpm. The engine was initially run on diesel to warm-up before switching to WGO and PO, this was mainly due to poor cold start performance characteristics of both fuels. At 100% load, BTE for WGO is reduced by 4% compared to diesel and improved by 7% with FIF. In contrast, BTE for PO is 4% higher compared to diesel, however, FIF has minimal effect on BTE when running on PO. Although, smoke, HC and CO emissions were higher for WGO compared to diesel, they were lower with FIF due to improved combustion. These emissions were consistently lower for PO due to superior combustion performance, mainly attributed to low viscosity of the fuel. However, NO emission for PO (1610 ppm) is higher compared to diesel (1580 ppm) at 100% load and reduced with FIF (1415 ppm). NO emission is reduced by approximately 12% for PO+FIF compared to PO. The results suggest that FIF has the potential to improve diesel combustion performance and reduce NO emission produced by CI engines running on high and low viscosity biofuels, respectively.

Keywords: Wheat germ oil; pine oil; fuel ionisation filter; biofuels; NO emission; electrochemical conversion

Nomenclature

ATDC	After top dead centre	HC	Hydrocarbon
BP	Brake power	LVLC	Low viscosity and low cetane index
BSEC	Brake specific energy consumption	NO	Nitrous oxide
BTDC	Before top dead centre	PO	Pine oil
BTE	Brake thermal efficiency	ppm	Parts per million
CAD	Crank angle degree	rpm	Revolutions per minute
CI	Compression ignition	SCR	Selective catalytic reduction
CO	Carbon monoxide	SOC	Start of combustion
CO2	Carbon dioxide	TDC	Top dead center
DPF	Diesel particulate filter	TFC	Total fuel consumption
EGT	Exhaust gas temperature	WGO	Wheat germ oil
FIF	Fuel ionisation filter		

1. Introduction

The global challenge facing automotive manufacturers is to develop an engine capable of meeting stringent emission standards without penalties in engine performance and driver experience. The latest aftertreatment technology to reduce NO and soot emissions are selective catalytic reduction (SCR) and diesel particulate filter (DPF), respectively [1]. These aftertreatment devices significantly lower NO and soot emissions produced by compression ignition (CI) engines. However, both systems suffer from inherent limitations, SCR can only operate in a small temperature window and DPFs require regular regeneration which leads to penalties in fuel economy [2]. In addition, the capital, operational and maintenance costs are amongst the major drawbacks of both systems. The limitations in the available fossil fuel resources and the global concern over energy security have led to the development of low carbon and sustainable fuels as an alternative to diesel for CI engine applications. Vegetable oil is an attractive option due to lower emission levels and widespread availability. In particular, vegetable oil reduces the CO2 emission, a significant contributor to global warming, since the plant feedstock absorbs CO2 from the atmosphere during its growth [3].

Numerous vegetable oils including jatropha, karanja, neem and mahua have been previously investigated [4-7]. The experimental studies revealed that vegetable oils reduce HC, CO and CO2 emission due to fuel-borne oxygen in their structure. However, the primary drawback of vegetable oil is its high viscosity and poor volatility which affects fuel atomisation, evaporation and air-fuel mixing [8]. Therefore, use of vegetable oil in CI engines can lead to poor engine performance, piston ring sticking and injector clogging [9]. To overcome these issues, various strategies such as transesterification [10], fuel preheating [11] and use of hydrogen [12] and oxygenates [13] have been developed and tested by other researchers to improve the performance of vegetable oil in CI engines. Among these methods, preheating and transesterification alter the chemical/physical properties of the fuel prior to combustion whereas hydrogen and oxygenates are used as additives to improve the engine performance. Martin et al. [14] compared various methods to improve the performance of cottonseed oil in a single cylinder CI engine at full load condition. Addtional methods such as transesterification, fuel preheating and diethyl ether blending were studied with cottonseed oil as base fuel. They reported brake thermal efficiency (BTE) improvement of 9% with ethyl ester as an oxygenate and 3% improvement with preheating at 110^{o}C compared to base vegetable oil. Transesterification and fuel preheating led to increase in NO emission with reduced HC, CO and soot emissions compared to base vegetable oil. They concluded that among the tested methods, transesterification resulted in the maximum engine performance improvement.

Electrochemical conversion of fuel through magnetic ionisation has been tested in this study as a pre-combustion treatment system. Although limited studies have previously investigated the effect of magnetic field on fuel atomisation, its effect on engine performance is yet to be fully explored. Fuel ionisation enhances the mixing of air and fuel molecules thus reducing fuel consumption and CO2 emission [15-18]. Previous studies incorporated a permanent magnet in the fuel line/pump to reform the fuel prior to injection to improve the combustion. Patel et al. [15] and Okoronkwo et al. [16] studied the effect of electromagnetic field on the ionisation and combustion of fuel in a single cylinder CI engine. The permanent magnets were placed in the fuel line with diesel as base fuel. They observed a significant reduction in harmful emissions such as HC, CO and soot. They also reported a decrease in fuel consumption due to improved mixing of ionised fuel molecules with air. Thiyagarajan et al. [18] conducted experiments with fuel magnetic ioniser in a single cylinder CI engine with ternary blends of karanja oil methyl ester, orange oil and methanol along with exhaust aftertreatment system. They observed that ionisation of fuel blends aids in performance improvements, resulting in higher BTE and reduced soot, HC and CO emissions. The results demonstrated that CO2 emission could also be reduced with this technique.

In this study, wheat germ oil (WGO) and pine oil (PO) with high and low viscosities, respectively, were utilised to assess the effect of magnetic ionisation on diesel engine performance running on biofuel. WGO is extracted from the germ of wheat kernels by cold pressing and supercritical CO2 extraction [19]. Wheat germ is a by-product of wheat obtained from milling industries, which belongs to the Gramineae family [20]. WGO is also used in cosmetics and for medicinal purposes. Wheat germ represents 2-3% of the entire wheat grain and it contains between 8% and 14% oil [21]. Although the physiochemical properties of WGO suggest that it can serve as a diesel fuel surrogate, its potential as an alternative low carbon fuel in CI engines is yet to be explored.

Recently, researchers have changed focus to using fuels with lower viscosity and lower cetane index (LVLC) in CI engines due to improved fuel atomisation, evaporation and air-fuel mixing characteristics [22]. Various LVLC fuels such as eucalyptus oil, PO, camphor oil and orange oil have been tested [23]. Several researchers have investigated PO as a potential diesel surrogate fuel [24-26]. The calorific value of PO is comparable to diesel, making it suitable for CI engine operation without conversion to biodiesel. Vallinayagam et al. [24] examined the combustion and emission performance of a 4-stroke single cylinder direct injection diesel engine running on PO and its blends of 25%, 50% and 75% with diesel. The experimental results indicated that PO has the potential to replace diesel in CI engines without modification. The authors reported improved BTE and fuel consumption for PO compared to diesel at all load conditions with 5% improvement at full load condition. The maximum heat release rate of the engine at maximum power output was approximately 27% higher compared to diesel. Furthermore, CO, HC and soot emissions were reduced with PO compared to diesel at higher load conditions. However, at full load, the results showed 25% increase in NO emission with PO compared to diesel.

The findings of the previous studies demonstrate that biofuel viscosity has an impact on engine performance and emission levels. High viscosity biofuels result in poor combustion due to inadequate fuel atomisation while low viscosity biofuels lead to increased NO emission and could cause cold start issues due to low cetane index. Although the use of magnetic fuel ionisation as the pre-combustion system is a promising approach to tackle the above-mentioned shortcomings of high and low viscosity biofuels, limited research has been previously carried out by other researchers and its effect is yet to be fully explored. This study intends to ascertain the effect of magnetic fuel ionisation on engine performance and emission levels of a CI engine running on WGO and PO, high and low viscosity biofuels, respectively. The novelty of this study is the investigation of the effect of ion exchange in CI engines running on biofuel. Although PO has been previously explored as a potential diesel fuel surrogate, WGO is yet to be fully explored. This study aims to contribute to the knowledgebase by investigating the effect of magnetic ionisation on WGO in CI engines to overcome the fuel atomisation issues associated with this type of fuel, making it a viable low carbon alternative fuel for diesel-powered vehicles.

2. Methodology

2.1. Test fuels

In this study, WGO and PO with high and low viscosities, respectively, were used as test fuels while diesel was utilised as base fuel. Table 1 shows the ASTM standards used to determine the fuel properties and Table 2 depicts the fuel properties of diesel, WGO and PO. It is evident that the viscosity of WGO is almost four times higher than diesel and the heating value is lower for WGO with similar cetane index compared to diesel. The former causes poor fuel atomisation which leads to delayed start of combustion (SOC) [27]. In contrast, PO has very low viscosity and similar heating value compared to diesel. However, cetane index of PO is markedly lower compared to the base fuel which results in longer ignition delay, leading to higher NO emission and also cold start issues [25].

Table 1
ASTM standards used for the measurement of fuel properties

Property	Test standards	Biodiesel standards		Method
		ASTM D6751 -02	EN 14214/14213	
Kinematic viscosity, cST @ 40^{o}C	ASTM D445	1.9-6.0	3.5-5.0	Redwood viscometer
Density @ 15^{o}C, g/cm^3	ASTM D1298	-	0.860-0.900	Pycnometer
Lower heating value kJ/kg	ASTM D240	-	Min 35000	Bomb calorimeter
Cetane index	ASTM D976	Min 47	Min 51	Calculated based on API gravity & mid-boiling point
Flashpoint, ^{o}C	ASTM D93	Min 93	Min 120	Pensky martens apparatus [closed cup]

Table 2
Properties of diesel, WGO and PO

Properties	Diesel fuel	WGO	PO
Kinematic viscosity, cST @ 40^{o}C	3.6	17.1	1.3
Density @ 15^{o}C, g/cm^3	0.840	0.957	0.875
Lower heating value kJ/kg	42700	35017	42800
Cetane index	45-55	54	11
Flash point, ^{o}C	74	197	52

2.2. Test engine

The engine used in this investigation is a Simpson S217 twin cylinder CI engine, employed in tractors. The engine delivers maximum power output of 12.4 kW at constant speed of 1500 rpm and is coupled to an eddy current dynamometer. The compression ratio of the engine is 18.5 and a hemispherical type combustion chamber is used to create the necessary swirl. The engine specifications are listed in Table 3. A mechanical type fuel injection system with a pump and a 3-hole nozzle injector assembly is used. The fuel consumption is determined using a burette and a stopwatch; measuring the time taken to consume 10 cc of fuel. The density of both WGO and PO was measured in g/cc using a pycnometer based on ASTM D1298 standard. The following expression is used to convert the measured fuel volume flow rate to mass flow rate;

$$TFC = \frac{10}{t} \times \rho \times 10^{-6}$$

Where TFC is the total fuel consumption (kg/s), t is the time taken for 10cc of fuel consumption (s) and ρ is the density of fuel (kg/m^3).

Table 3
Engine specifications

Model	Simpsons S 217 tractor engine
Rated power	12.4 kW @ 1500 rpm
Type/Configuration	Vertical in-line diesel engine
Bore x Stroke	91.44 mm x 127 mm
No. of cylinders	2
Displacement	1670 cc
Compression ratio	18.5:1

Fig. 1 shows the schematic diagram of the experimental setup. Engine combustion analyser model ECA 1.0.1 is used to record the combustion parameters. The in-cylinder pressure is recorded using a Kistler HSM111A22 piezoelectric pressure transducer while the engine timing is measured through a shaft encoder which generates 360 pulses per revolution. Therefore, the data is recorded at one degree crank angle intervals. The pressure data is recorded for 100 consecutive cycles and the measured values are analysed using ECA software to obtain heat release rate and ignition delay. Gaseous emissions such as carbon monoxide (CO), carbon dioxide (CO2), hydrocarbon (HC) and NO are measured using AVL 5 gas analyser based on non-dispersive infrared (NDIR) technique. CO and CO2 emissions are measured in terms of percentage volume whereas HC and NO emissions are measured in terms of parts per million (ppm). An AVL 432C smoke meter, operating on light extinction principle, is used to measure smoke intensity in terms of percentage opacity.

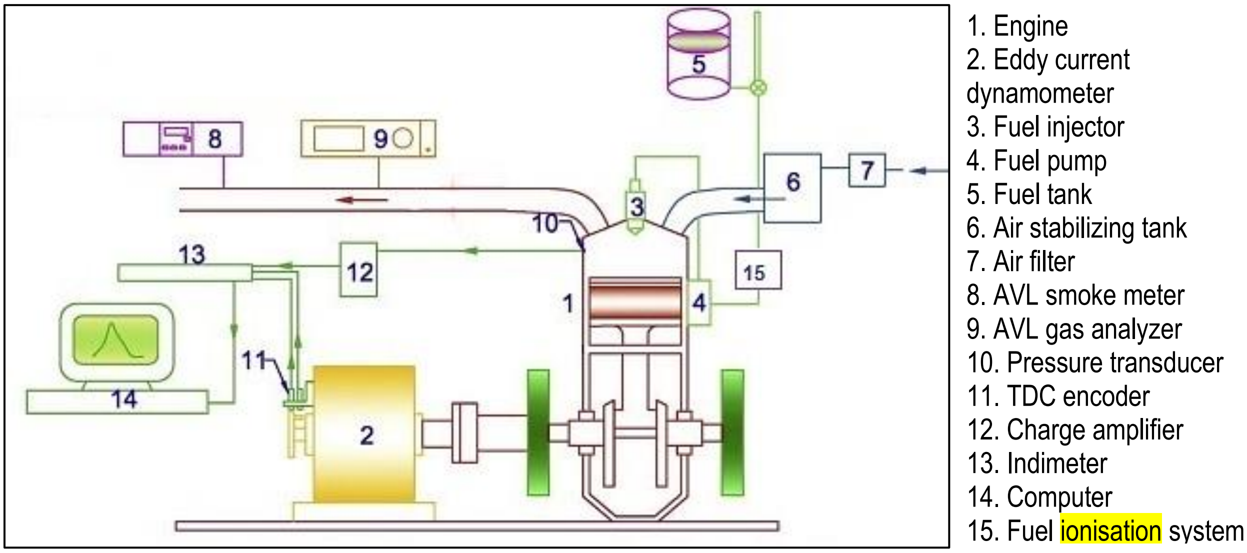

Fig. 1. Schematic diagram of the experimental setup

2.3. Fuel ionisation system

A fuel ionisation filter (FIF), shown in Fig. 2, which uses the principle of electrochemical liquid vortex ionisation, is designed, manufactured and fitted upstream of the fuel injection pump.

Fig. 2. Fuel ionisation filter

A commercially available fuel filter is modified and converted to ionisation filter by placing three permanent magnets on the walls opposite to each other. Neodymium magnet with 2000 gauss is selected for this study. The magnets are positioned in such a way that the incoming fuel forms a vortex due to the magnetic field and becomes ionised. Metallic compounds such as copper, aluminum and galvanised iron wires are placed inside the filter, filling one third of the volume of the filter. These materials can donate electrons to the incoming fluid they come into contact with in the presence of a magnetic field. Therefore, when the fuel is atomised and sprayed into the combustion chamber, the fuel particles become more dispersed since particles with the same charge polarity repel, resulting in improved mixing with oxygen molecules in the combustion chamber [28, 29]. The FIF is positioned close to the fuel injection pump since the ionised fuel was highly unstable.

2.4. Experimental procedure

All tests are performed at a constant engine speed of 1500 rpm from low to high loads, corresponding to the brake power (BP) of 3.1 kW, 6.2 kW, 9.3 kW and 12.4 kW. The fuel injection timing remains constant at 23 crank angle degrees (CAD) BTDC and a constant injection pressure of 200 bar is maintained throughout the experiments. All the experiments are carried out under steady state conditions without modifying the test engine. All measurements are repeated five times and the average value is used to determine the derived parameters. PO has lower cetane index and when operated as neat fuel, the engine is deemed to encounter cold start problems. Therefore, initial tests are performed with diesel as the base fuel at various load conditions to attain warm-up conditions. Once the engine reaches sufficient temperature, PO is injected to avoid cold start issues. Engine performance, emission and combustion parameters of WGO and PO are measured and compared to diesel at different load conditions.

2.5. Error analysis

Uncertainty is an indicative parameter corresponding to the deviation of measured values which can be determined by uncertainty analysis. The uncertainties in experiments may arise due to instrument type, operating conditions, environmental factors and other unaccounted random sources. The accuracy of the instruments used are identified and their corresponding uncertainty values are determined as illustrated in Table 4.

Table 4
Uncertainty of various instruments and parameters

Measurement	Accuracy	% Uncertainty	Measurement technique
Load	± 0.1 kg	± 0.2	Strain gauge type load cell
Speed	± 10 rpm	± 0.1	Magnetic pickup type
Burette fuel measurement	± 0.1 cc	± 1	Volumetric measurement
Time	± 0.1 sec	± 0.2	Manual stopwatch
Manometer	± 1 mm	± 1	Principle of balancing column of liquid
CO	± 0.02 %	± 0.2	NDIR principle
HC	± 20 ppm	± 0.2	NDIR principle
CO2	± 0.03 %	± 0.15	NDIR principle
NO	± 10 ppm	± 1	Electrochemical measurement
Smoke	± 1 % opacity	± 1	Opacimeter
EGT indicator	± 1 ^{o}C	± 0.15	K-type thermocouple
Pressure pickup	± 0.5 bar	± 1	Piezoelectric sensor
Crank angle	± 1^{o}	± 0.2	Magnetic pickup type

The uncertainty of various parameters such as TFC, brake specific energy consumption (BSEC), BTE and exhaust gas temperature (EGT) are determined by the method proposed by Holman which is based on the uncertainty of the equipment used [30]. The uncertainty associated with each measurement is used to identify the total uncertainty in the measurements, this shows the maximum possible uncertainty within the presented results. The total uncertainty of the experiment is computed by the following expression;

$$\begin{array}{c} Total \\ experimental \\ uncertainty\ (\%) \end{array} = \sqrt{\left\{\begin{array}{c} (\text{uncertainty of TFC})^2 + (\text{uncertainty of BP})^2 + (\text{uncertainty of BSEC})^2 + \\ (\text{uncertainty of BTE})^2 + (\text{uncertainty of CO})^2 + (\text{uncertainty of HC})^2 + \\ (\text{uncertainty of NO})^2 + (\text{uncertainty of } CO2)^2 + (\text{uncertainty of smoke})^2 + \\ (\text{uncertainty of EGT indicator})^2 + (\text{uncertainty of pressure transducer})^2 \end{array}\right\}}$$

$$\begin{array}{c} Total \\ experimental \\ uncertainty \end{array} = \sqrt{\left\{\begin{array}{c} (1)^2 + (0.2)^2 + (1)^2 + (1)^2 + (0.2)^2 + (0.1)^2 + (0.2)^2 + \\ (1)^2 + (1)^2 + (0.2)^2 + (1)^2 \end{array}\right\}} = 2.48\%$$

The repeatability of the results is found to be within 3% based on the Gaussian distribution.

3. Results and discussions

3.1. Combustion characteristics

Fig. 3 illustrates the in-cylinder pressure data for diesel, WGO, WGO+FIF, PO and PO+FIF at 100% load condition. The fuel is directly injected into the combustion chamber which atomises, evaporates and mixes with air during the mixing phase. Subsequently the mixture reaches the autoignition temperature and combusts, producing mechanical work due to sudden release of energy. The timing and amplitude of the maximum pressure is mainly dependent on the amount of premixed fuel and air which is formed during the ignition delay period [31]. The results show that the peak in-cylinder pressure at 100% load condition is lower for WGO compared to diesel, 79.8 bar and 85.6 bar, respectively. The reduction in peak pressure is due to poor atomisation and evaporation of WGO compared to diesel due to higher viscosity and density, resulting in less premixed air-fuel mixture formation during the ignition delay period. Similar combustion characteristics have been reported with high viscosity biofuels [32]. The peak in-cylinder pressure for diesel occurs at TDC while for WGO it occurs at 6 CAD ATDC, this shift in due to higher ignition delay with WGO. Consequently, large quantity of fuel is burnt in the later part of combustion (i.e. diffusion combustion) which results in higher EGT and lower BTE. However, with the use of the FIF the heavier droplets of WGO can be converted to ions due to the magnetisation effect, improving the atomisation and mixing characteristics during the ignition delay period [18]. The Peak in-cylinder pressure for WGO+FIF is 83.3 bar and occurs at 3 CAD ATDC. The early occurrence of peak in-cylinder pressure for WGO with FIF compared to WGO is due to reduced ignition delay period owing to the quick dispersion of the ions, leading to improved combustion. In contrast, PO has a low cetane number which prolongs the ignition delay period, resulting in more profound premixed combustion and hence higher peak in-cylinder pressure due to greater air-fuel mixing [33]. The peak in-cylinder pressure for PO is 88.4 bar and occurs at 4 CAD ATDC. However, with the use of the FIF, the peak pressure of PO is reduced to 84.6 bar and occurs earlier, this may be attributed to earlier SOC caused by the ionisation of PO. The results demonstrate that FIF has minimal effect on low viscosity biofuels compared to high viscosity biofuels.

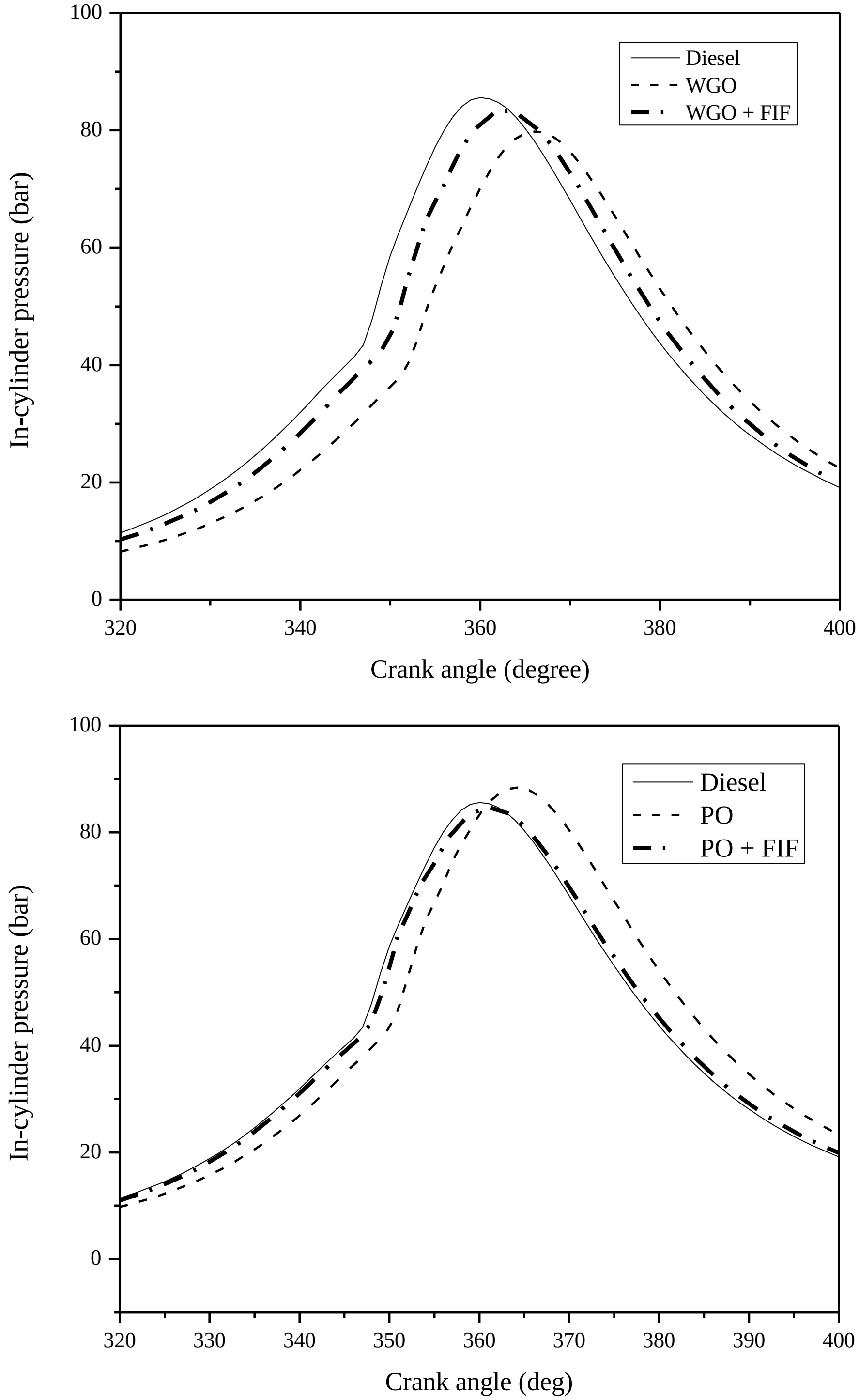

Fig. 3. In-cylinder pressure data at 100% load

Fig. 4 depicts the heat release rate data for diesel, WGO, WGO+FIF, PO and PO+FIF at 100% load condition.

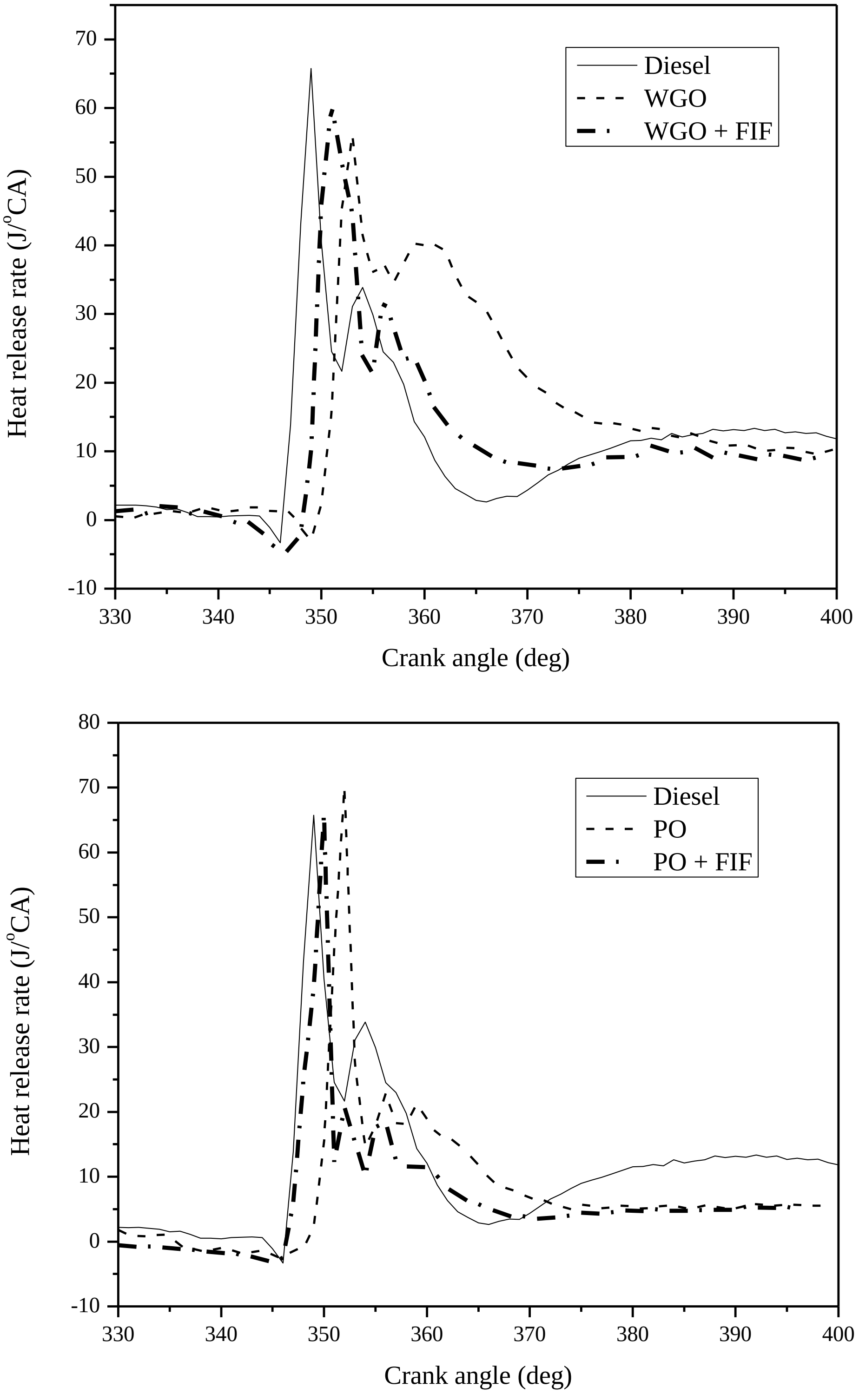

Fig. 4. Heat release rate data at 100% load

In general, high viscosity biofuels exhibit inferior combustion characteristics due to poor atomisation, evaporation and mixing with air during the ignition delay period [34]. The peak heat release for diesel and WGO are 65.7 J/^{o}CA and 56 J/^{o}CA, respectively. The peak heat release for

WGO occurs later in the cycle and shows more diffusion combustion compared to diesel, this is mainly due to poor atomisation, evaporation and air-fuel mixing resulting in longer ignition delay and more fuel burning later in the cycle. This has an adverse effect on engine performance and leads to higher soot emission due to the presence of fuel-rich combustion. Kasiraman et al. [35] investigated the combustion characteristics of cashew nut shell oil in a CI engine and reported similar findings. The peak heat release rate for WGO+FIF is 60.6 J/^{o}CA and occurs earlier in the cycle compared to WGO due to ionisation of heavy WGO fuel particles, enhancing fuel atomisation, evaporation and mixing processes during the ignition delay period. Consequently, less diffusion combustion occurs which reduces the soot emission and improves the engine performance. On the other hand, low cetane number of PO leads to longer ignition delay, resulting in the formation of more premixed air-fuel mixture [36]. Consequently, peak heat release of PO is 70 J/^{o}CA and occurs later in the cycle compared to diesel. However, unlike WGO, PO exhibits better combustion characteristics due to more profound premixed combustion. The ignition delay of PO+FIF is shorter, resulting in reduced peak heat release rate of 65.9 J/^{o}CA.

Fig. 5 shows the ignition delay and combustion duration for WGO and PO with and without FIF compared to diesel at 100% load. Ignition delay occurs due to inherent physical and chemical properties of the fuel referred to as physical and chemical delays, respectively. The former is due to the time required for the fuel to atomise, evaporate and mix with air to form a combustible mixture while the latter is dependent on the cetane number of the fuel which promotes autoignition of the premixed mixture [37]. Ignition delay for diesel and WGO is 11 CAD and 14 CAD at 100% load. Prolong ignition delay of WGO is due to physical delay caused by high viscosity and density of the fuel, resulting in poor atomisation, evaporation and air-fuel mixing. Agarwal and Dhar [38] also reported longer ignition delay with karanja oil. The ignition delay for WGO+FIF is reduced to 12 CAD, this is attributed to the ionisation effect caused by the FIF system, promoting early onset of combustion. The ignition delay for PO is 13 CAD which is longer compared to diesel. Although physical delay with PO is lower compared to diesel due to lower viscosity and density, the chemical delay is significantly longer due to lower cetane number. However, ignition delay of PO+FIF is reduced to 10 CAD, this is due to further reduction in physical delay due to ionisation of fuel particles.

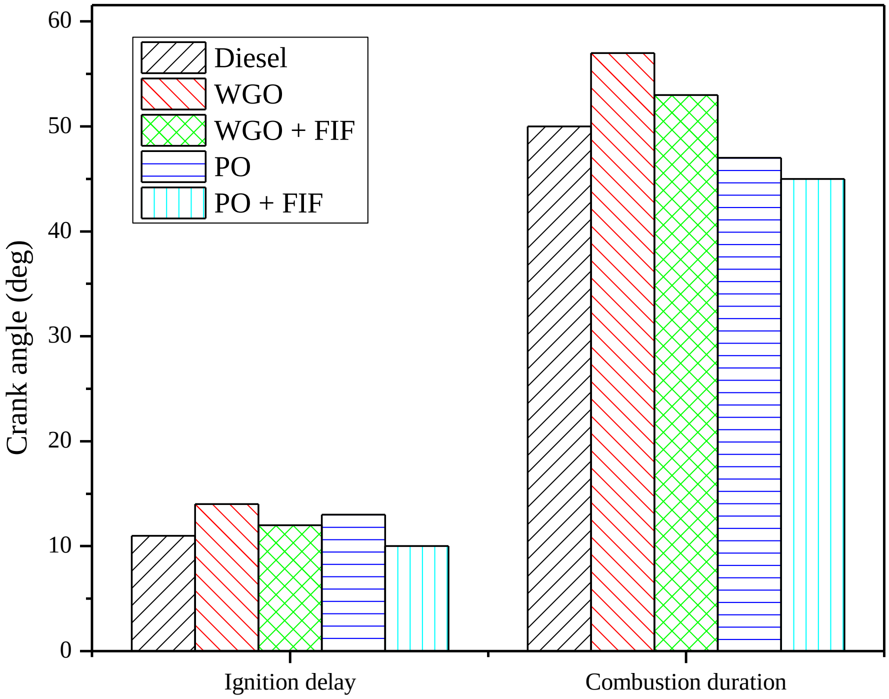

Fig. 5. Ignition delay and combustion duration characteristics at 100% load

Combustion duration is calculated as the time between the SOC and 90% mass fraction burnt. Combustion duration for diesel and WGO at 100% load is 50 CAD and 57 CAD, respectively. Longer combustion duration with WGO is due to delayed SOC for WGO, as evident in Fig. 4. The high viscosity and density of WGO lead to poor fuel atomisation and less premixed mixture formation which results in more diffusion combustion as shown in Fig. 4. Consequently, higher EGT and lower BTE are obtained with WGO compared to diesel. Enhancement in combustion is observed with FIF due to improved air-fuel mixing. Combustion duration for WGO+FIF is 53 CAD, shorter compared to WGO at 100% load. Combustion duration for PO and PO with FIF at 100% load is 47 CAD and 45 CAD, respectively, which is shorter compared to diesel due to improved combustion as a result of better atomisation and mixing with air owing to lower viscosity. The results indicate that FIF has minimal impact on combustion duration.

3.2. Engine performance

BTE is the ratio of brake power to fuel energy, indicating the conversion efficiency of fuel energy to mechanical work. Fig. 6 shows the variation of BTE for diesel, WGO, WGO+FIF, PO and PO+FIF at different load conditions.

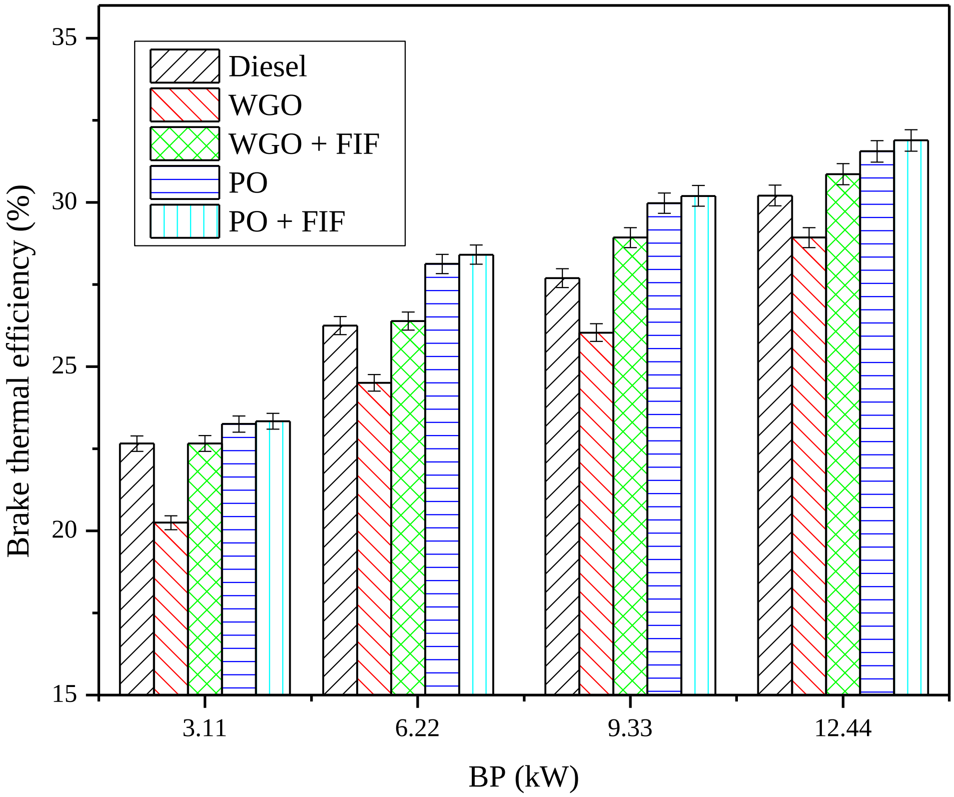

Fig. 6. Brake thermal efficiency at different load conditions

BTE increases with increase in load due to reduction in heat loss and friction. At 100% load, BTE for diesel and WGO is 30.2% and 28.9%, respectively. BTE is dependent on the amount of heat release during the premixed combustion phase [39], thus the reduction in BTE for WGO is attributed to poor mixture formation due to higher viscosity and density. This is confirmed by the heat release rate data shown in Fig. 4. BTE is increased to 30.85% for WGO+FIF compared to 28.9% for WGO at 100% load. BTE is improved with the use of FIF due to enhanced mixing of fuel and air in the combustion chamber owing to ionisation effects. The ionised cationic molecules (negatively charged) repel inside the combustion chamber, causing the molecules to disperse and mix with oxygen molecules [18]. This is also evident in Fig. 4 where WGO+FIF resulted in higher premixed combustion. The same trend is observed at other load conditions. BTE for PO and PO+FIF at 100% load is 31.55% and 31.88%, respectively. BTE is improved with PO compared to diesel at all load conditions. This is mainly attributed to better atomisation, dispersion, evaporation and mixing characteristics of low viscosity biofuels [24]. However, it is evident that ionisation of such biofuels has minimal impact on BTE.

Fig. 7 shows BSEC for diesel, WGO and PO with and without FIF at different load conditions. At 100% load, BSEC for diesel, WGO and WGO+FIF is 11.91 MJ/kWh, 12.44 MJ/kWh and 11.66 MJ/kWh, respectively. Higher BSEC for WGO is attributed to poor combustion characteristics due to unfavourable properties such as higher viscosity and density and lower heating value. However, this trend is reversed with the use of FIF due to improved fuel atomisation, evaporation and air-fuel mixing. Previous studies reported that BSEC of low viscosity vegetable oil is lower compared to diesel [33, 35, 36]. BSEC for PO and PO+FIF at 100% load is 11.4 MJ/kWh and 11.28 MJ/kWh, respectively. BSEC for PO is slightly lower than diesel due to lower viscosity and density. In contrast to WGO, ionisation has almost no effect on BSEC of PO.

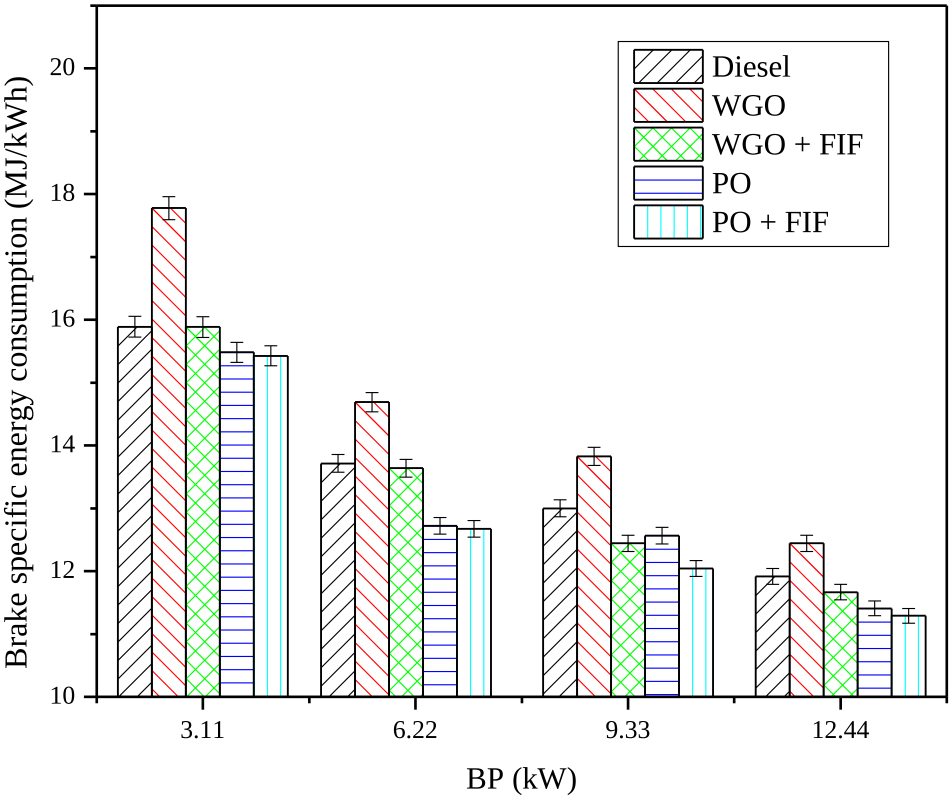

Fig. 7. Brake specific energy consumption at different load conditions

Fig. 8 shows EGT for diesel, WGO and PO with and without FIF at different load conditions. EGT for diesel, WGO and WGO+FIF is 485^{o}C, 555^{o}C and 536^{o}C, respectively. EGT for WGO is higher compared to diesel which is an indication of higher thermal losses due to slower combustion, leading to extended combustion phasing [40]. As evident in Fig. 4, the diffusion phase for WGO occurs later in the cycle compared to diesel which results in more heat rejection by the exhaust gases, reducing the BP output. However, EGT for WGO+FIF is lower compared to WGO at all load conditions due to shorter ignition delay, resulting in earlier onset of diffusion combustion phase. EGT for PO and PO+FIF is 475°C and 450°C, respectively. Although, ignition delay for PO is longer compared to diesel, combustion phasing is less due to faster combustion of PO [36] which results in reduced EGT and higher BTE. The use of FIF leads to earlier onset of combustion compared to PO, reducing EGT and the thermal losses.

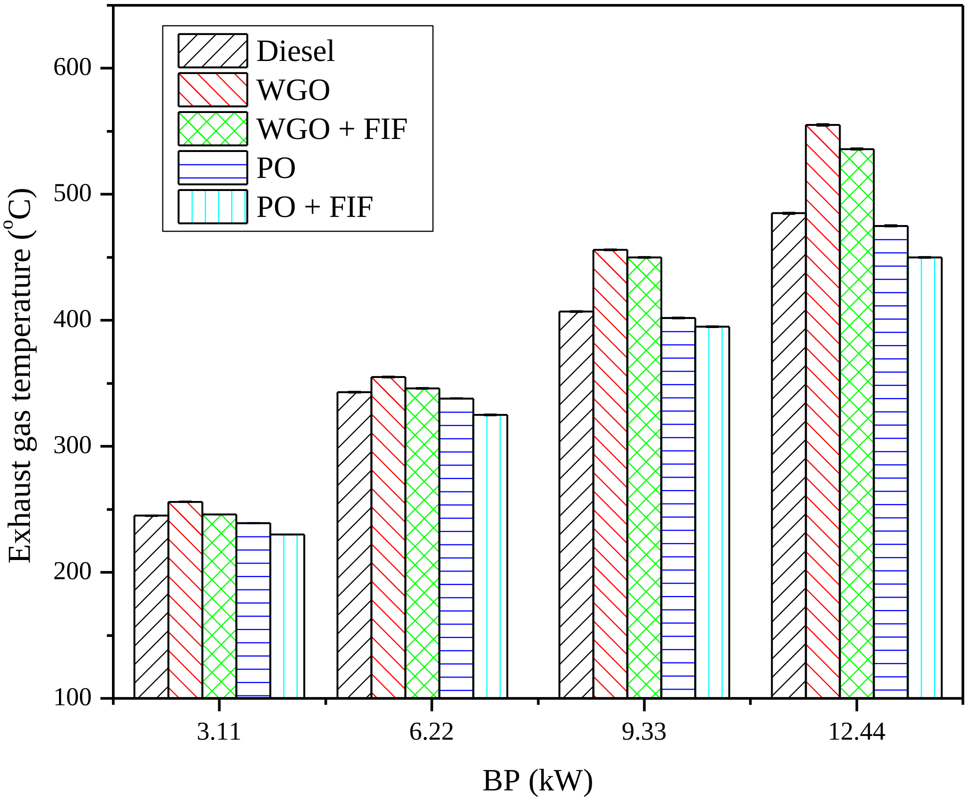

Fig. 8. Exhaust gas temperature at different load conditions

3.3. Emissions

Fig. 9 illustrates NO emission for diesel, WGO and PO with and without FIF at different load conditions. NO emission is formed due to high temperature combustion and presence of oxygen. NO emission in CI engines is primarily formed during the premixed combustion phase due to higher combustion temperature [41]. NO emission increases with increase in load due to higher combustion temperature. At 100% load, NO emission for diesel, WGO and WGO+FIF is 1580 ppm, 1529 ppm and 1575 ppm, respectively. NO emission for WGO is less compared to diesel at all load conditions due to less profound premixed combustion. However, with FIF the combustion is improved resulting in higher combustion temperature and higher NO emission formation. NO emission is generally higher for low viscosity biofuels due to lower cetane number. Low cetane number leads to longer ignition delay, resulting in higher heat release rate during the premixed combustion phase. NO emission for PO and PO+FIF is 1610 ppm and 1415 ppm, respectively. Higher NO emission is formed with PO due to longer ignition delay. However, NO emission is reduced by approximately 12% for PO+FIF compared to PO. The reduction in NO emission with FIF is due to reduced ignition delay which leads to less profound premixed combustion, lowering the combustion temperature. Magnetic field causes de-clustering which results in induced vibration in the fuel molecules [17], however, its effect on low and high viscosity biofuels is different. De-clustering has a more profound effect on WGO, resulting in heavy molecules to split into simple ions which enhances the combustion process. Consequently, high temperature combustion occurs, leading to higher NO emission. However, in the case of PO, the physical delay is reduced due to the magnetic effect, reducing the overall ignition delay period. This leads to shorter combustion duration, as shown in Fig. 5, precluding NO formation due to limited residence time available.

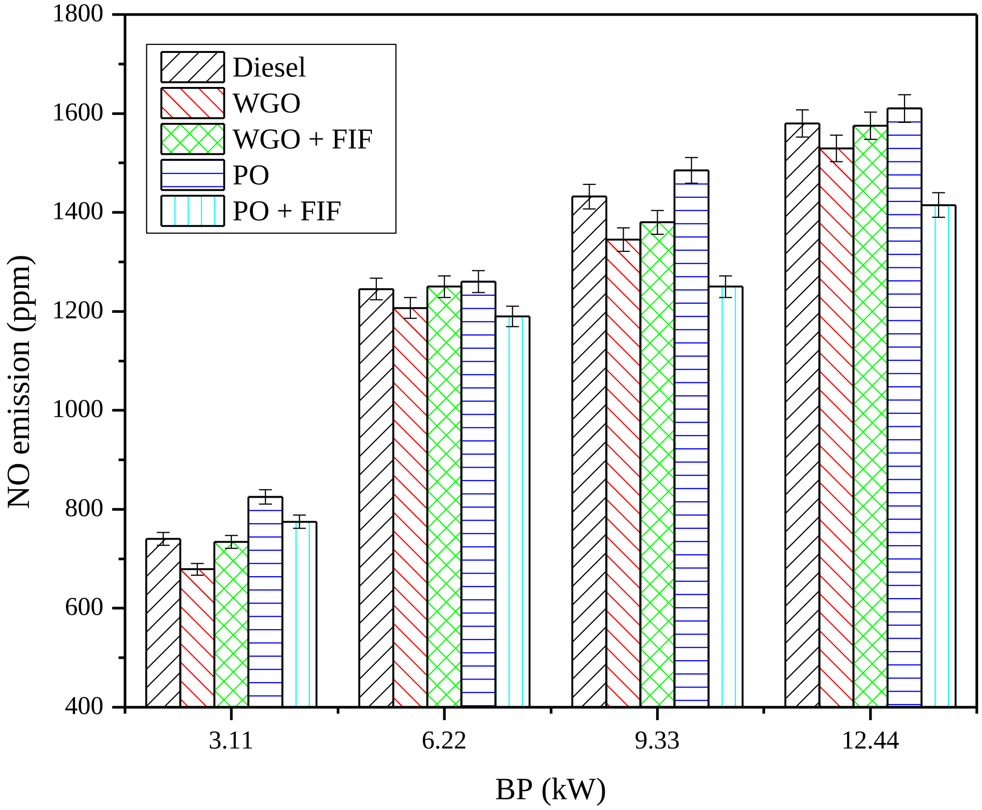

Fig. 9. NO emission at different load conditions

Fig. 10 shows smoke opacity for diesel, WGO and PO with and without FIF at different load conditions. Soot emission is formed during diffusion combustion and increases at higher loads due to increase in fuel injection quantity. At 100% load, smoke opacity for diesel and WGO is 55.6% and 62%, respectively. Higher smoke emission for WGO is due to more profound diffusion combustion caused by higher viscosity and density as shown in Fig. 4 [35]. The smoke opacity for WGO+FIF at 100% load is reduced to 57% due to enhanced atomisation, evaporation and air-fuel mixing, resulting in less diffusion combustion. The smoke opacity for PO and PO+FIF at 100% load is 54.8% and 50%, respectively. Smoke emission is lower for low viscosity biofuels due to improved atomisation and better mixing with air. FIF implementation aids in further enhancement of air-fuel mixing, resulting in further reduction of smoke emission. It is suggested that the presence of magnetic field helps break the bond between the molecules, improving the mixing process which reduces the formation of rich combustion. The magnetic field also reduces the viscosity of the fuel [42] which improves the atomisation and mixing processes, reducing soot formation. This is also observed in this investigation with both test fuels.

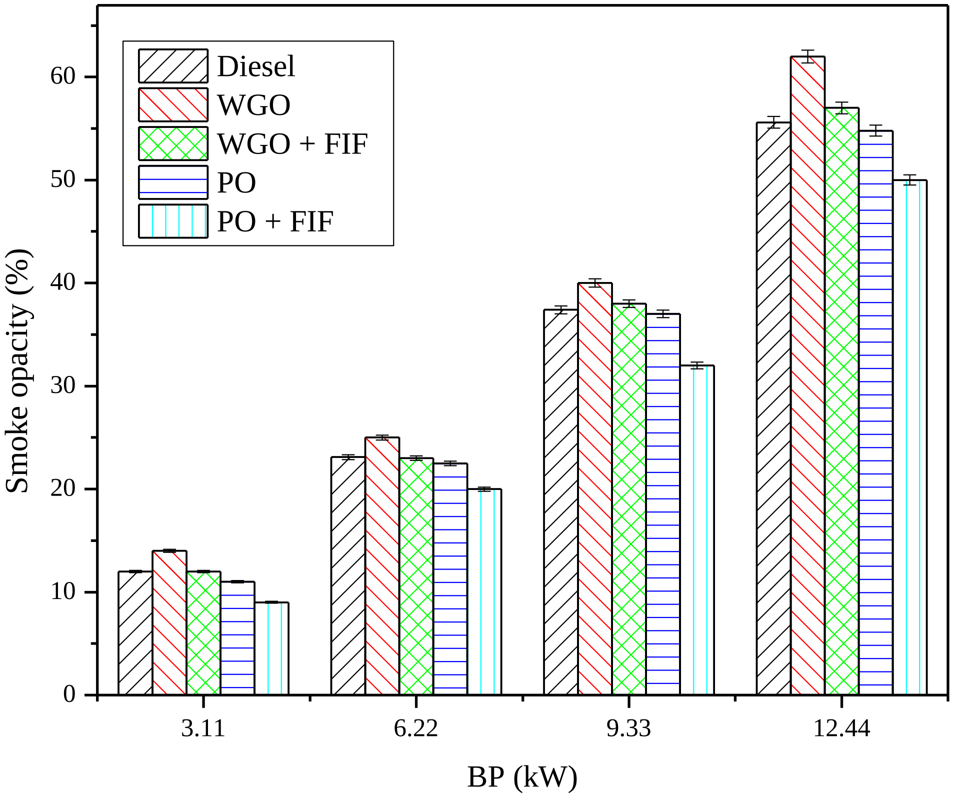

Fig. 10. Soot emission at different load conditions

Formation of HC emission in CI engines is mainly due to engine operating condition, fuel property, spray characteristic and air-fuel ratio [43]. Fig. 11 shows HC emission at different load conditions for the test fuels. At 100% load, HC emission for diesel, WGO and WGO+FIF is 5 ppm, 7 ppm and 6 ppm, respectively. Higher HC emission for WGO is due to poor spray characteristics owing to high viscosity of the fuel. This leads to lower combustion temperature, precluding the oxidation of HC. At low load conditions, HC emission is approximately 80% higher for WGO compared to diesel. This is due to poor fuel properties and significantly lower combustion temperature at low loads. With FIF, HC emission is slightly reduced due to improved fuel atomisation which improves the combustion, resulting in oxidation of HC at high temperatures. At 100% load, HC emission for PO and PO+FIF is 4 ppm and 3 ppm, respectively. This is due to superior fuel atomisation, evaporation and air-fuel mixing. HC emission is slightly lower for PO due to improved combustion and lower C-H ratio in the fuel structure. The results demonstrate that HC emission reduces with FIF at all load conditions. The uniform mixture formation of fuel ions and air molecules in the combustion chamber reduces the quench distance, resulting in lower HC emission formation. Subramanian et al. [18] also investigated the effect of fuel reforming on biofuel blends and reported similar findings.

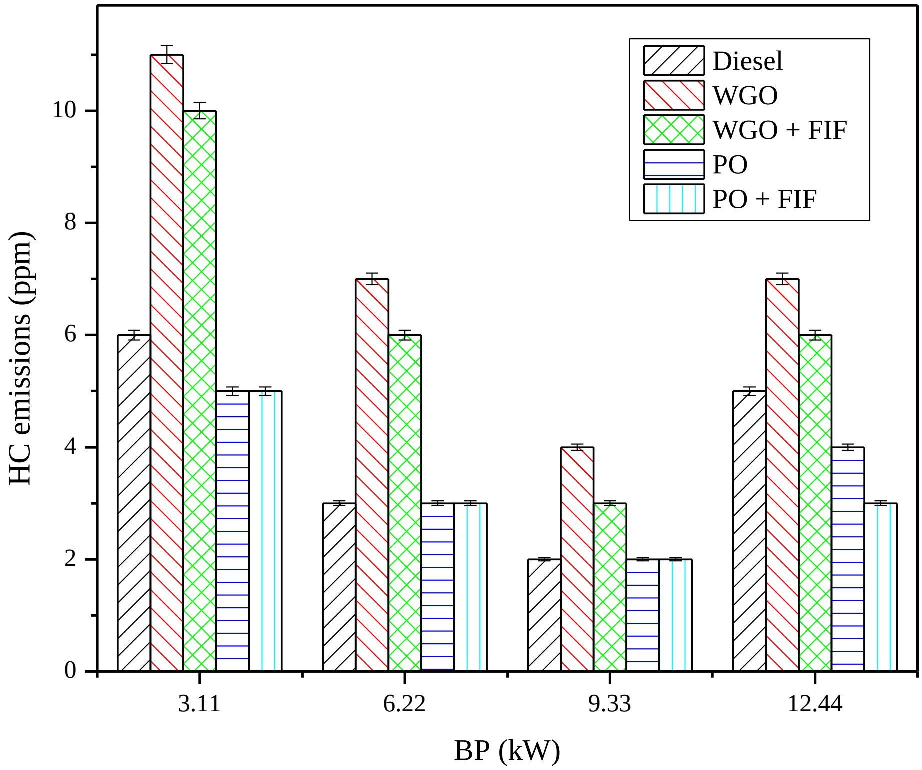

Fig. 11. HC emission at different load conditions

Fig. 12 shows the CO emission at different load conditions for the test fuels. At 100% load, CO emission for diesel, WGO and WGO+FIF is 0.03%, 0.05% and 0.04%, respectively. Higher CO emission for WGO is due to higher viscosity of WGO which leads to inefficient air-fuel mixing, promoting CO emission formation during the diffusion combustion phase where the combustion temperature is lower. CO emission is reduced with FIF due to improved fuel atomisation and dispersion, resulting in more complete combustion with less intermediate CO emission formation. CO emission for PO and PO+FIF at 100% load is 0.03% and 0.025%, respectively. Improved fuel atomisation due to lower viscosity reduces CO emission which is further reduced with the implementation of FIF. The reduction in CO emission with FIF is mainly attributed to enhanced combustion due to better fuel atomisation and improved mixing with air with both WGO and PO. Furthermore, improved mixture formation leads to a faster combustion, limiting the residence time for the formation of CO emission, in particular during the diffusion phase.

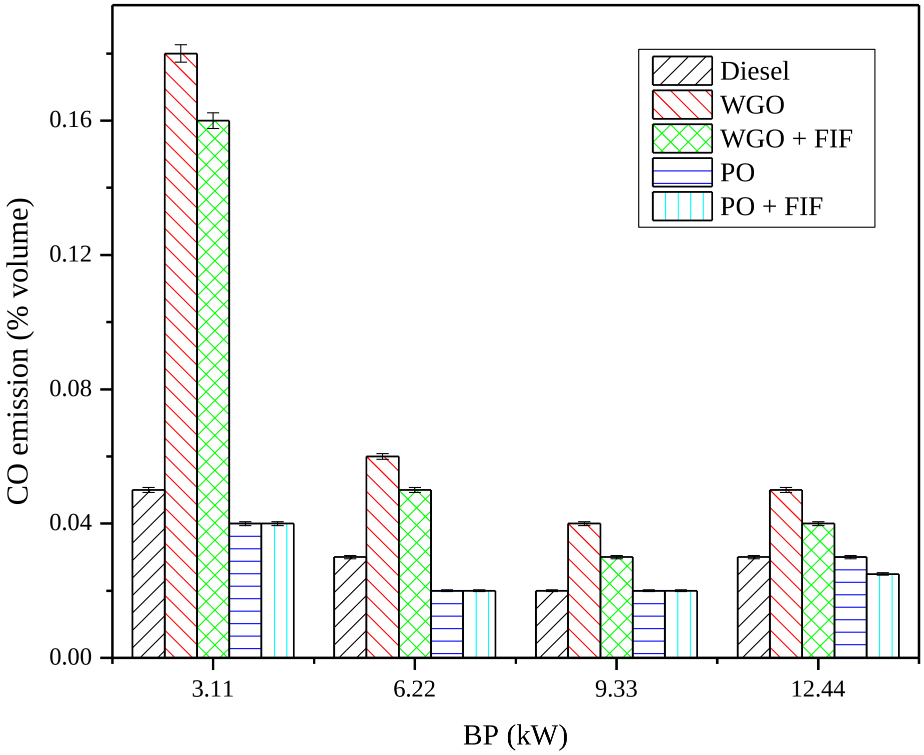

Fig. 12. CO emission at different load conditions

4. Conclusions

In this study the effect of high and low viscosity biofuels on diesel combustion and emissions were investigated in a twin cylinder CI engine. WGO and PO were tested at different load conditions as diesel fuel surrogates with high and low viscosities, respectively. WGO suffered from poor atomisation, evaporation and air-fuel mixing due to high viscosity which resulted in inferior combustion characteristics. In contrast, PO exhibited better atomisation and mixing characteristics due to lower viscosity, however, its low cetane number resulted in longer ignition delay and higher NO emission. A bespoke fuel ionisation system with a permanent magnet was fitted upstream of the fuel pump to ionise the incoming fuel prior to injection. The tests were carried out with the fixed injection timing of 23 CAD BTDC and at constant engine speed of 1500 rpm. At 100% load, BTE reduced from 30.2% to 28.92% for WGO compared to diesel and improved to 30.85% for WGO+FIF. Although, BTE for PO was higher compared to diesel due to lower viscosity, unlike WGO FIF had minimal impact on PO. NO emission was lower for WGO compared to diesel due to less profound premixed combustion caused by poor atomisation and mixing. However, NO emission increased with FIF due to improved premixed mixture formation which resulted in higher temperature combustion. NO emission for PO was higher compared to diesel due to longer ignition delay caused by low cetane number and reduced with FIF due to improved atomisation and mixing, resulting in shorter ignition delay. Higher HC, CO and soot emissions were observed with WGO compared to diesel, however, they reduced with FIF due to improved combustion characteristics. HC, CO and soot emissions were less with PO due to lower viscosity and further reduced with FIF due to ionisation of the fuel particles. Heat release rate and peak in-cylinder pressure were lower and occurred later in the cycle for WGO compared to diesel and improved with FIF due to improved fuel atomisation and mixture formation. Longer ignition delay for PO increased the heat release rate and peak in-cylinder pressure which were slightly reduced with FIF.

The use of high and low viscosity biofuels in CI engines can adversely affect engine performance due to poor mixture formation and longer ignition delay. However, FIF could alleviate the challenges associated high and low viscosity biofuels. The difficulty associated with high viscosity is addressed through fuel ionisation which improves fuel atomisation and air-fuel mixing while the ignition delay is reduced with low viscosity biofuels due to shorter physical delay which in turn reduces NO emission. The proposed method of fuel ionisation can be easily retrofitted to existing and future diesel vehicles, eliminating the need for complex and expensive transesterification processes.

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