

EXPERIMENTAL INVESTIGATION ON A DIESEL ENGINE FUELLED WITH NEEM OIL AND ITS METHYL ESTER

by

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Fuel crisis and environmental concerns have led to look for alternative fuels of bio-origin sources such as vegetable oils, which can be produced from forests, vegetable oil crops, and oil bearing biomass materials. Vegetable oils have energy content comparable to diesel fuel. The effect of neem oil and its methyl ester on a direct injected four stroke, single cylinder diesel engine combustion, performance and emission is investigated in this paper. The results show that at full load, peak cylinder pressure is higher for neem oil methyl ester; peak heat release rate during the premixed combustion phase is lower for neat neem oil and neem oil methyl ester. Ignition delay is lower for neat neem oil and neem oil methyl ester when compared with diesel at full load. The combustion duration is higher for neat neem oil and neem oil methyl ester compared to diesel at all loads. The brake thermal efficiency is slightly lower for neem oil at all engine loads, but in the case of neem oil methyl ester slightly higher at full load. It has been observed that there is a reduction in NO_x emission for neem oil and its methyl ester along with an increase in CO, HC, and smoke emissions.

Key words: *neem oil, neem oil methyl ester, combustion, performance, Diesel engine*

Introduction

The world has been confronted with energy crisis due to the decrease of fossil fuel resources and the increase of environmental restrictions. Therefore attention has been focused on developing the renewable or alternate fuels to replace the petroleum based fuels for transport vehicles. There are several alternative sources of fuel like vegetable oils, biogas, biomass, primary alcohols which are all renewable in nature. Among these fuels, vegetable oils appear to have an exceptional importance as they are renewable and widely available, biodegradable and non-toxic, and environmental friendly. In agriculture-based country, like India, the use of vegetable oils has to be identified and initiated in order to prevent environmental degradation and reduce dependence on imported fossil supplies by partially replacing them with renewable and domestic sources. A great deal of research has been

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conducted on their feasibility and the researchers have concluded that neat vegetable oils hold promise as alternative fuels for Diesel engines for short-term use [1-4]. However researchers have reported that the use of neat vegetable oil causes engine-related problems. High viscosity, low volatility, and poor cold flow conditions of these fuels cause severe engine deposits, injector coking, piston ring sticking, and difficulty in starting especially in cold weather. There are four ways to use neat vegetable oils in diesel engine [5-8]: (1) direct use or blending in diesel fuel, (2) micro emulsions in diesel fuel, (3) thermal cracking of vegetable oils, and (4) transesterification. It has been reported that dilution of vegetable oils with such materials as diesel fuels, a solvent or ethanol produced heavy carbon deposit on the inlet valves and showed a considerable top ring wear. Thermal cracking refers to a chemical change caused by the application of thermal energy in the presence of air or nitrogen sparge. The viscosity of pyrolysed vegetable oil is reduced, but still exceeds the specified value of 7.5 cSt. The pyrolysed vegetable oils contain acceptable amounts of sulphur, water and sediment, but unacceptable ash, carbon residue amounts and pour point. Engine testing on pyrolysed oil has been limited to short-term tests. It has been reported that the engine tests on microemulsion consisting of vegetable oil: methanol: 2-octanol: cetane improver indicate the accumulation of carbon around the orifices of the injector nozzles and heavy deposits on exhaust valves. It has been reported that dilution of vegetable oils with such materials as diesel fuels, a solvent or ethanol produced heavy carbon deposit on the inlet valves and showed a considerable top ring wear. Hence, out of these, transesterification is the most popular and best way to use neat vegetable oils [8].

Sahoo *et al.* [9] have experimented with jatropha, karanja and polanga biodiesel in a Diesel engine. They reported higher peak cylinder pressure and shorter ignition delay for all biodiesels when compared with diesel. Banapurmath *et al.* [10] have experimented with methyl esters of honge (HOME), jatropha (JOME), and sesame (SOME) in a single cylinder, four stroke, direct injection compression ignition (CI) engine and reported a higher emission of CO, HC, and smoke and lower NO_x as compared to that of diesel. Edwin *et al.* [11] have studied the combustion process of rubber seed oil (RSO) and its methyl ester (RSOME) and also reported higher emissions of CO, HC, and smoke and lower NO_x as compared to that of diesel. Balusamy *et al.* [12] have experimented with methyl ester of *Thevetia Peruviana* seed oil (METPSO) and reported a lower emission of CO, HC, and a higher NO_x as compared to that of diesel. Qi *et al.* [13] have compared the combustion characteristics of diesel and biodiesel from soybean oil in a single cylinder, naturally aspirated Diesel engine and concluded that the peak cylinder pressure of biodiesel is close to that of diesel. They also reported that the peak rate of pressure rise and peak heat release rate during premixed combustion phase are lower for biodiesel.

Research studies on the utilization of NeO as an alternative Diesel engine fuel are very little. Agarwal *et al.* [6] and Anjana *et al.* [7] have highlighted the suitability of NeO for Diesel engines. Narun *et al.* [14] have conducted experiment with diesel fuel and diesel NOME blends in a four stroke naturally aspirated Diesel engine. More research work on the engine performance, combustion and emission characteristics is required for complete evaluation of using NeO as an alternative Diesel engine fuel. The specific objective of the present work is to evaluate comparatively the performance, combustion and emission characteristics of a Diesel engine using neat NeO and its NOME, prepared by a method of two step "acid-base" process.

Experimental

Test engine and experimental procedure

Experiments have been conducted in a single-cylinder, four-stroke, naturally aspirated, direct injection Diesel engine (fig. 1). The specification of the engine is given in tab. 1. A mechanical unit pump of helical plunger type made by Bosch is used to deliver the fuel to the multi hole orifice. Two separate fuel tanks with a fuel switching system are used. The fuel consumption can be measured with the aid of an optical sensor. The fuel from the tank

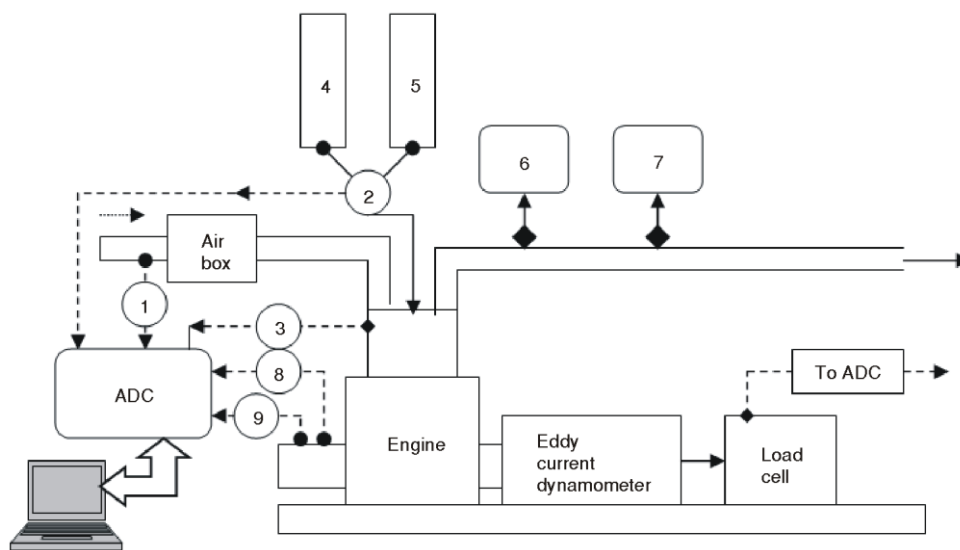


Figure 1. Experimental set-up

1 – air flow sensor, 2 – fuel flow sensor, 3 – pressure sensor, 4 – diesel tank, 5– biodiesel tank, 6 – five gas analyzer, 7 – smoke meter, 8 – speed sensor, 9 – crank angle encoder

Table 1. Engine specifications

Particulars	Specifications
Make & model	Kirloskar –TV1
BHP & speed	5 hp at 1500 rpm
Type of engine	Direct injection and 4 stroke
Compression ratio	16.5:1
Bore & stroke	80 mm and 110 mm
Type of loading	Eddy current dynamometer
Method of cooling	Water cooling
Inlet valve opening	4.5° bTDC
Inlet valve closing	35.5° aBDC
Exhaust valve opening	35.5° bBDC
Exhaust valve closing	4.5° aTDC
Injection timing	23° bTDC
Injection pressure	210 bar

was connected by way of a solenoid valve to a glass burette and the same is connected to the engine through a manual ball valve. The fuel solenoid of the tank will open and stay open for 30 seconds, during this time fuel is supplied to the engine directly from the fuel tank and also fills up the burette. After 30 seconds the fuel solenoid closes the fuel tank outlet, and then the fuel in the burette is supplied to the engine. When the fuel level crosses the high level optical sensor, the sequence running in the computer records the time of this event. Likewise when the fuel level crosses the low level optical sensor, the sequence running in the computer records the time of this event and immediately the fuel solenoid opens filling up the burette and cycle is repeated. Now, volume of the fuel between high level and low level optical sensors (20 cm^3) is known. The starting time of fuel consumption, *i. e.* time when fuel crossed high level sensor and the finish time of fuel consumption, *i. e.* time when the fuel crossed low level sensor gives an estimate of fuel flow rate *i. e.* $20 \text{ cm}^3/\text{difference of time in second}$. The air flow to the engine is routed through cubical air tank. The rubber diaphragm fixed on the top of the air tank takes care of neutralizing the pulsation for air-flow measurement. The inlet air tank is provided with an orifice. The differential pressure of air is measured in the computer using a differential pressure transducer calibrated to indicate volume air flow. The pressure ports are connected to instrumentation panel using smooth flexible hose. The pressure drop across the orifice is measured using a differential pressure transducer. The output of the differential pressure transducer is amplified using an instrumentation amplifier and fed to the data acquisition system. The engine is coupled with an eddy current dynamometer which is used to control the engine torque. Engine speed and load are controlled by varying excitation current to the eddy current dynamometer using dynamometer controller. A kistler piezoelectric transducer (water cooled type) is installed in the cylinder head in order to measure the combustion pressure. Signals from pressure transducer are fed to charge amplifier. A high precision crank angle encoder is used for delivering signals for top dead center (TDC) and crank angle. The signals from charge amplifier and crank angle encoder are acquired using Kistler data acquisition system (12 bit). In-cylinder pressure and TDC signal are acquired and stored on a high speed computer based digital data acquisition system. There are filters present in the pressure signal. The data from 100 consecutive cycles are recorded. These are processed with specially developed software to obtain the pressure crank angle data. A program has been developed to obtain the average pressure crank angle data of 100 cycles. An AVL exhaust-gas analyzer (Model: diGas 444) and AVL Smoke meter (Model: 437) are used to measure emission parameters CO, HC, NO_x , and smoke intensity, respectively. The specification and accuracy of gas analyzer and smoke meter is given in tab. 2. The uncertainties of some of the measured and calculated parameters are given in tab. 3. Loads, defined in terms of brake power (BP), are changed in five levels from no load, $\text{BP} = 0 \text{ kW}$, to the maximum load, $\text{BP} = 3.5 \text{ kW}$. The engine is operated at the rated speed *i. e.*, 1500 rpm for all the tests. For all the tests, the engine is started with diesel fuel and allowed to stabilize for 45 minutes. After the engine is warmed up, it is then switched to NeO/NOME. For each experiment, three measurements are taken to average the data so as to determine the repeatability of the measured data and have an estimate of measured accuracy. At the end of test, the fuel is switched back to diesel and the engine is kept running for a while before shutdown to flush out the NeO/NOME from the fuel lines and injection system. By doing this, the cold starting problems can be avoided to some extent. The performance parameter such as brake thermal efficiency (BTE) and brake specific energy consumption (BSFC), combustion parameters such as cylinder pressure, ignition delay, combustion duration, rate of heat release and rate of pressure rise, and emission parameters such as like smoke intensity, unburned HC,

CO, and NO_x emissions are measured for diesel fuel, neat NeO and NOME. Finally, the test results are analyzed and compared.

Table 2. Gas analyzer and smoke meter specifications

Gas analyzer model	AVL 437 DiGas 444	Accuracy
Pollutant	Range	
CO	0-10 vol.%	0.01
HC	0-20000 ppm	±10 ppm
NO _x	0-5000 ppm	±10 ppm
Smoke meter	AVL 437	Accuracy
Smoke intensity	0-100 opacity [%]	±1% full scale reading

Table 3. Uncertainties of some measured and calculated parameters

Parameter	Percentage uncertainties
NO _x	±0.1
CO	±0.01
HC	±0.1
Kinematic viscosity	±1.3%
BTE	±1%
BSFC	±1.5%
BP	±0.5%

Estimation of the experimental heat release rate

The combustion characteristic and heat release rate based on the data of the recorded cylinder pressure are analyzed. From the first law of thermodynamics:

$$\frac{dQ_g}{d\theta} = \frac{dQ_n}{d\theta} + \frac{dQ_w}{d\theta} \quad (1)$$

This can be rewritten as:

$$\frac{dQ_n}{d\theta} = \frac{dQ_g}{d\theta} - \frac{dQ_w}{d\theta} \quad (2)$$

Thus the net heat release rate is the difference between the heat released by combustion of fuel and the heat absorbed by cylinder wall. Using the first law of thermodynamics, the net heat release rate is calculated by eq. (3):

$$\frac{dQ_n}{d\theta} = \frac{\lambda}{\lambda-1} P \frac{dV}{d\theta} + \frac{1}{\lambda-1} V \frac{dP}{d\theta} \quad (3)$$

where θ is the crank angle and λ is the ratio of specific heats, C_p/C_v .

Transesterification

Transesterification is the process of reacting a triglyceride with alcohol in the presence of a catalyst to produce fatty acid esters and glycerol. It is difficult to produce ester from NeO using alkaline catalyst (NaOH/KOH) because of its high free fatty acid (FFA). Therefore, a two step transesterification process is chosen to convert the non-edible NeO to its methyl ester. The first step acid catalyzed esterification reduces the FFA value of the oil to about 2%. The second step, alkaline catalyzed transesterification process converts the products of the first step to its mono-esters and glycerol. In acid esterification, 1000 ml NeO is heated to about 50 °C, 250 ml methanol is added and stirred for a few minutes. With this mixture 2% H₂SO₄ is also added and stirred at a constant rate with 50 °C for one hour. After the reaction is

over, the solution is allowed to settle for 24 hours in a separating funnel. The excess alcohol along with sulphuric acid and impurities floats at the top surface and is removed. The lower layer is separated for further processing (alkaline esterification). In alkaline catalyzed esterification, the products of the first step are again heated to about 50 to 55 °C. With this mixture, 5 g KOH dissolved in 250 ml methanol is added and stirred for 60 minutes. After the reaction is over, the solution is again allowed to settle for 24 hours. The glycerin settles at the bottom and esterified NeO rises to the top. This esterified NeO is separated and purified with warm water. After washing the final product is heated up to 60 °C for 10 minutes. The esterified neem oil so prepared is referred as NOME.

Results and discussion

Characterization of biodiesel from NeO

The important physiochemical properties of diesel, NeO, and NOME are shown in tab. 4. The determination of specific gravity, calorific value, viscosity, flash point, and fire point are carried out, as per the ASTM standard, by using a hydrometer, a Bomb calorimeter, a Redwood viscometer, and Pensky-Martins closed cup apparatus. The present results show that transesterification process improved the fuel properties of the NeO. The kinematic viscosity of the NOME is found to be 4.5 cSt, which is approximately 15% higher than that of diesel. The specific gravity of NOME is 0.867, which is approximately 6.67% higher than that of diesel. The lower calorific value of NOME is 41.0 MJ/kg, which is 5% lower than that of diesel. Therefore more amount of biodiesel needs to be injected into the combustion chamber to produce the same amount of power. The flash and fire point of NOME are higher than that of diesel as seen in tab. 4. This is beneficial during transportation.

Table 4. Properties of diesel, NeO, NOME

Property	Diesel (D100)	NeO	NOME	ASTM code
Chemical formula	C ₁₄ H ₂₂	C ₁₈ H ₃₄ O ₂	C ₁₈ H ₃₄ O ₂	–
Calorific value, [MJkg ⁻¹]	43.2	39.0	41.0	D4809
Stoichiometric A/F	15	12.41	12.41	–
Oxygen content, [wt.%]	0	11.0	11.0	–
Specific gravity	0.823	0.926	0.867	D445
Kinematic viscosity (at 40 °C) [cSt]	3.9	38	4.5	D2217
Flash point, [°C]	56	245	152	D92
Fire point, [°C]	64	278	158	D92
Cetane number	48	47	51	–

Combustion analysis

Figure 2 shows the fuel-air equivalence ratio, Φ , *i. e.*, the actual fuel-air ratio (by mass) by its stoichiometric value, for diesel, NeO, and NOME. It can be seen that the fuel-air equivalence ratio curve lies little higher than the corresponding one for diesel fuel and

NOME. This shows that engine runs overall little “richer” conditions with NeO for the same engine load and speed conditions.

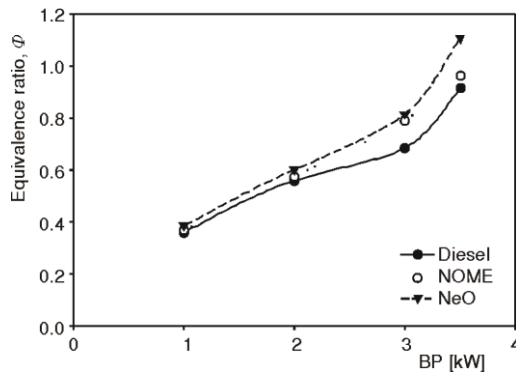


Figure 2. Equivalence ratio vs. BP

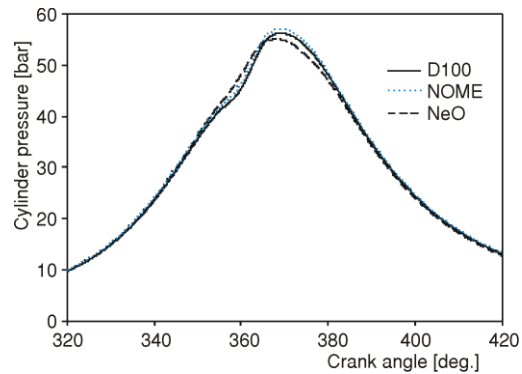


Figure 3. Cylinder pressure vs. crank angle at full load

The variation of cylinder pressure with crank angle at full load is shown in fig. 3. In a CI engine, the cylinder pressure characterizes the ability of fuel to mix well with air and burn. It is clear that the combustion starts earlier for NeO and NOME due to shorter ignition delay and advanced dynamic injection timing (because of higher bulk modulus and higher density of NeO and NOME). It can also be observed that NOME has a higher peak pressure than NeO and diesel. Due to longer ignition delay of NOME compared to NeO, more fuel is accumulated in the combustion chamber which leads to higher peak pressure at the time of premixed combustion stage. The higher peak pressure for NOME as compared to diesel may also be due to dynamic injection advance, which results in initiation of combustion before TDC and the pressure rises quickly. On the other hand, while running with diesel, due to longer ignition delay, the combustion starts later for diesel compared to NOME, which leads to lower peak cylinder pressure. The location of the peak pressures for NeO and NOME are comparable with that of diesel and are within 1-10 crank angle degree aTDC. The peak pressure for NeO and NOME is 55 bar occurring at 7 °CA aTDC, 57 bar occurring at 8 °CA aTDC, respectively, while in the case of diesel, it is 56 bar occurring at 9 °CA aTDC.

The variation of ignition delay with BP for NeO, NOME, and diesel is shown in fig. 4. The ignition delay in a Diesel engine is defined as the time between the start of fuel injection and the start of combustion. The start of fuel injection is usually taken as the time when the injector needle lifts off its seat. Since needle lift sensor is not available, the timing at which fuel injection line pressure reaches the injector nozzle opening pressure (210 bar) is taken as the start of injection. Hence, the fuel pump and injector setting are kept identical for all

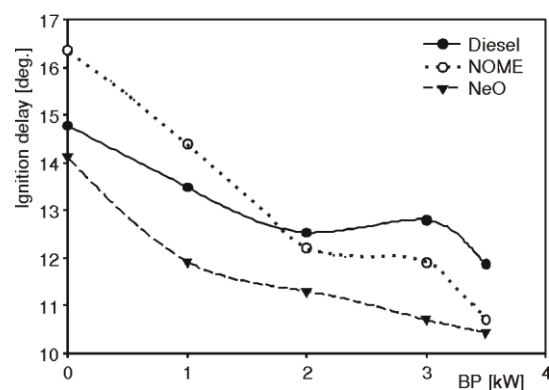


Figure 4. Ignition delay vs. BP

fuels. The start of combustion can be influenced by changes in fuel properties such as viscosity. The start of combustion is defined in terms of the change in slope of the heat release rate that occurs at ignition. Hence the ignition delays for the fuels are defined as an interval between 23 °CA bTDC (standard injection timing) and fuel ignition. The ignition delay is observed to be lower in case of NeO compared to diesel and NOME at all engine loads. The reason may be that a complex and pre-flame reaction takes place at higher temperatures. As a result of higher cylinder temperature existing during fuel injection, NeO may undergo thermal cracking and lighter compounds are produced, which might have ignited earlier, resulting in a shorter ignition delay. The increase in fuel viscosity, particularly for the fuels derived from petroleum products, results in poor atomization, slow mixing and reduced cone angle. These result in longer ignition delay. But the biodiesel is not derived from crude petroleum; the opposite trend is seen in case of neat oil and biodiesel.

The variation of combustion with brake power for NeO, NOME, and diesel is shown in fig. 5. The combustion duration is defined as duration between the start of combustion and 90% cumulative heat release. The combustion duration increases with increase in BP of the engine with all the fuels due to increase in the quantity of fuel injected. Longer combustion duration is observed with NeO than NOME and diesel due to the longer diffusion combustion phase. This again confirms the slower combustion characteristics of NeO.

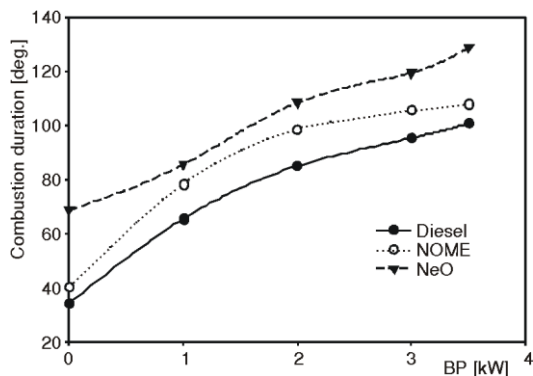


Figure 5. Combustion duration vs. BP

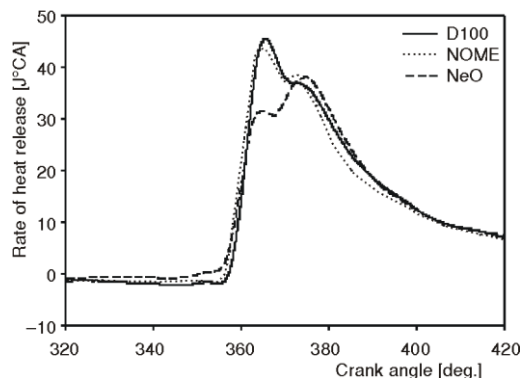


Figure 6. Heat release rate vs. crank angle at full load

The variation of heat release rate with crank angle at different loads for NeO, NOME, and diesel is shown in fig. 6. Due to heat loss from the cylinder and the cooling effect of the fuel vaporizing as it is injected into the cylinder, the heat release rate is slightly negative during the ignition delay period. After the combustion is started, this becomes positive. The initial phase of combustion, called the premixed combustion is very rapid because of the combustion of fuel that has mixed with during ignition delay. After this phase, the combustion continues slowly until most of fuel is burned. This phase of combustion is called mixing-controlled combustion. The final combustion phase is the late combustion, which continues until the end of the expansion stroke. It can be observed that peak heat release rate is higher for diesel than NOME and NeO. This may be due higher volatility and better mixing of diesel with air. Another reason may be, as a consequence of the longer ignition delay, the intensity of premixed combustion phase for diesel is more. On the other hand, the peak heat release rate is lower for NeO compared to NOME and diesel. This may be due to the lower volatility and higher viscosity of NeO, it leads to a reduction in air

entrainment and fuel-air mixing rates, resulting in lesser amount of fuel being prepared for premixed combustion stage during ignition delay. It can also be observed the diffusion burning indicated by the area under second peak is dominant for NeO. This is consistent with the expected effects of neat NeO viscosity on the fuel spray and reduction of air entrainment and fuel-air mixing rates [2]. At the time of ignition less air-fuel mixture is prepared for combustion with neat NeO. As a result more burning occurs in the diffusion burning combustion phase rather than in the premixed combustion phase.

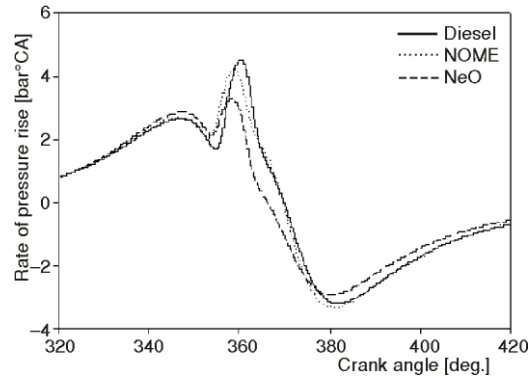


Figure 7. Rate of pressure rise vs. crank angle at full load

The variation of rate of pressure rise with crank angle for NeO, NOME, and diesel at full load is shown in fig. 7. The peak rate of pressure rise is lower for NeO and NOME compared to diesel at full load. This is because NeO contains heavier HC molecules which have higher viscosity and lower volatility. As a result, lower rates of heat release at the time of premixed combustion stage are produced for NeO and NOME compared to diesel.

Performance analysis

Significant engine performance parameters such as BTE and BSEC are evaluated for NOME, NeO, and diesel. These results are represented graphically in figs. 8 and 9. The thermal efficiency is lower for NeO than that of NOME and diesel. It may be due to larger differences in viscosity, specific gravity and volatility between diesel and NeO. Poor spray formation and reduced spray angle causes reduction in air entrainment and fuel-air mixing rates [2]. The NOME has a lower viscosity, which results in better atomization that leads to complete combustion of the fuel as compared to NeO.

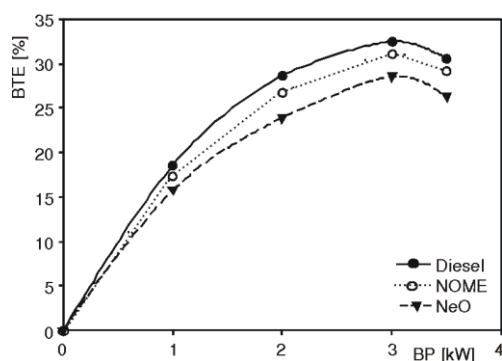


Figure 8. BTE vs. BP

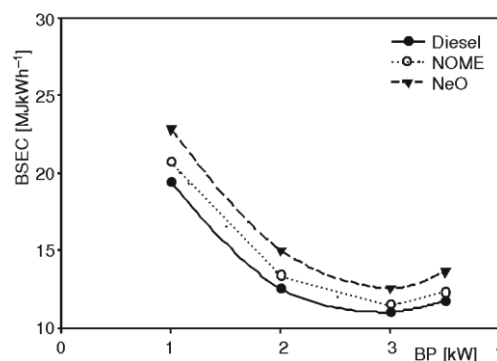


Figure 9. BSEC consumption vs. BP

The BSEC is an ideal parameter for comparing the engine performance of fuels having different calorific values and specific gravities. The BSEC is calculated as the product of brake specific fuel consumption and lower calorific value. The BSEC consumption of the

engine with NeO operation is higher compared to NOME and diesel at all loads. This may be due to the lower heating value, higher viscosity and density of NeO. Lower heating value of NeO requires larger fuel flow rates to maintain constant energy input to the engine [11].

Emission analysis

The variation of smoke intensity with BP for diesel, NOME, and NeO is shown in fig. 10. It can be observed that smoke emission is higher with NeO as compared to NOME and diesel due to poor mixture formation tendency. Due to heavier molecular structure and higher viscosity of NeO, atomization becomes poor which leads to sluggish combustion, leading to higher smoke emission. The NOME has a lower viscosity (which is about 4.5 cSt), which results in better atomization that leads to complete combustion of the fuel as compared to NeO. Hence, the smoke intensity value of NOME is lower compared to NeO.

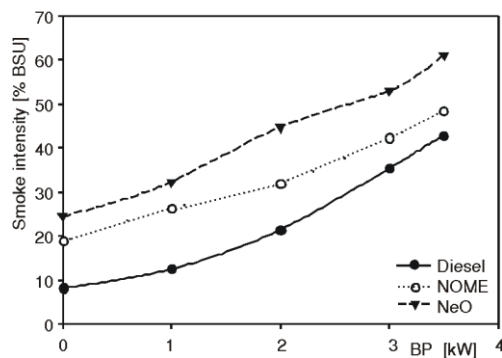


Figure 10. Smoke intensity vs. BP

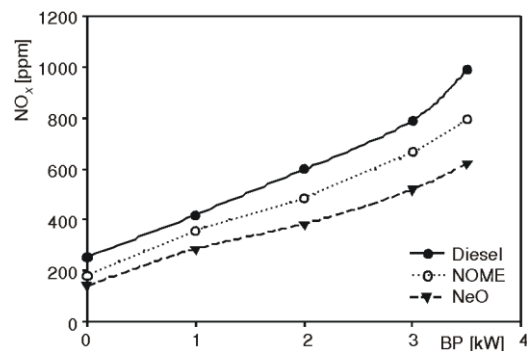


Figure 11. NO_x vs. BP

The variation of NO_x with BP for diesel, NOME, and NeO is shown in fig. 11. It can be observed that the NO_x emission is lower with NeO, as compared to diesel and NOME. The most important factors that cause NO_x emission are maximum combustion temperature and locally air-fuel mixture. The NeO and NOME have higher viscosity therefore the fuel droplet size in the engine is expected to be larger than the diesel. Larger droplets have longer combustion duration and they demonstrate significant energy release during later part of the combustion process. This suggests that the maximum combustion temperature is possibly lower (less intensity of heat release in the premixed combustion phase), which leads to lower emission of NO_x for NeO and NOME compared to diesel. This is the most important emission characteristic of NeO and NOME, as the NO_x emission is the most harmful gaseous emissions from the engines, the reduction of it is always the target of engine researchers and engine makers.

The variations in CO and HC emissions with BP for diesel, NOME, and NeO are shown in figs. 12 and 13. It can be observed that the CO and HC emissions are higher with NeO as compared to diesel and NOME. This may be due to relatively poor atomization and lower volatility of NeO, NOME compared to diesel. As a result, some of the fuel droplets may not get burned. When these unburned droplets mix with the hot combustion gases, oxidation reactions occur, but do not have enough time to undergo complete combustion. Comparing with neat NeO, lower CO, and HC emissions are observed with the NOME.

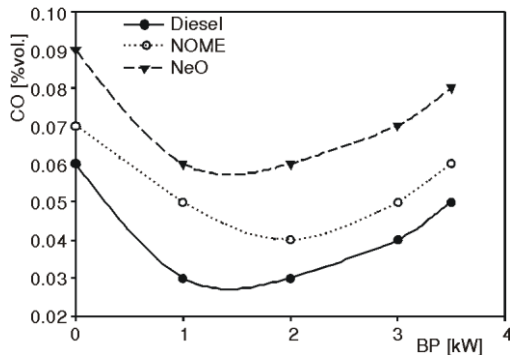


Figure 12. CO emission vs. BP

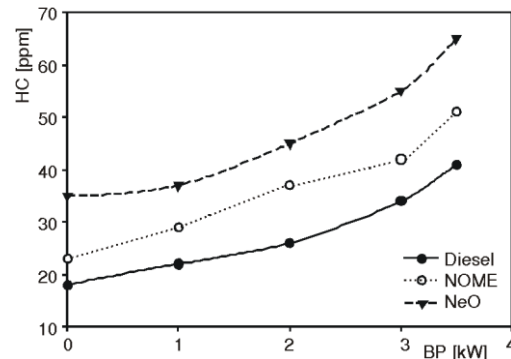


Figure 13. HC emission vs. BP

Conclusions

A single cylinder diesel engine is operated successfully on NeO and NOME. The following conclusions are drawn based on the experimental results.

- At full load, peak cylinder pressure for NOME is higher as compared to diesel and NeO. The peak heat release rate during the premixed combustion phase and peak rate of pressure rise is lower for NeO and NOME as compared to diesel. Ignition delay is observed to be lower for NeO when compared with NOME and diesel over the entire engine operating conditions. Combustion duration is higher with NeO compared to that of NOME and diesel.
- The BTE and BSEC for NeO and NOME is lower than that of diesel at all loads.
- From the emission analysis it is observed that there is a 37% reduction in NO_x emission for NeO and 19% reduction for NOME at full load. NO_x is the most important emission characteristic of NeO and NOME, as the NO_x emission is the most harmful gaseous emissions from the engines, the reduction of it is always the target of engine researchers and engine makers. This emission character for NeO and NOME is very useful character for the application of NeO and NOME to Diesel engines as a kind of alternative fuel for the petroleum based diesel fuel.
- The CO and HC emissions from NeO and NOME are higher than those of diesel. There is an increase in CO emission for NeO and NOME by 40% and 20% at full load respectively. It can also be seen that the engine emits very small CO emissions for all the fuels, which are less than 0.09%, so that a small change of the CO content will cause a big change in percentages. There is an increase in HC emission by 54% in case of NeO and 24% in case of NOME. The smoke emissions for NeO, NOME are higher than that of diesel by 46.8% and 16% at full load, respectively.
- It can be concluded that the performance, combustion and emission characteristics of NOME are better compared to neat NeO. Hence, biodiesel from NeO is quite suitable as an alternative to diesel.

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