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A comprehensive study on performance, emission, and combustion characteristics of a dual-fuel engine fuelled with orange oil and Jatropha oil

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Abstract: Performance of a single-cylinder, water-cooled, direct-injection diesel engine on dual-fuel operation with Jatropha oil (JO) as pilot fuel and orange oil as primary fuel was evaluated. Constant load test at different power outputs was conducted at the rated speed of 1500 r/min with varying orange oil quantities. The loads were fixed as 20 per cent, 40 per cent, 60 per cent, 80 per cent, and 100 per cent.

In dual-fuel operation with orange oil induction, the thermal efficiency of JO was increased mainly at high power outputs. Maximum thermal efficiency with JO was found as 29 per cent at 31 per cent of orange oil induction at 100 per cent load. Smoke was reduced significantly with all orange oil induction rates at all power outputs in dual-fuel operation with JO. It was reduced from 4.4 to 3.3 BSU (Bosch Smoke Units) with JO at the maximum efficiency point at 100 per cent load. HC emissions were increased further at all power outputs in the dual-fuel mode with all rates of orange oil induction. Dual-fuel operation increased the ignition delay of JO. However, peak pressure and energy release rates were improved in the dual-fuel operation with orange oil induction. In general, dual-fuel operation with orange oil as inducted fuel with JO as pilot fuel showed inferior performance and emissions at part loads.

It is concluded that the JO as pilots fuel and orange oil as the inducted fuel could be used in diesel engines with reduced smoke levels and improved thermal efficiencies with no major detoriation in performance.

Keywords: biofuels, dual-fuel engine, Jatropha oil, orange oil, performance, emissions

1 INTRODUCTION

Energy is an important resource for development of a country. World energy survey reports that the consumption of the energy is increasing steadily every year by 2 per cent in the decade of 1990–2000 and it is projected that the world energy demand will be more than double by 2050 from now and triple by the end of the century [**1**]. Hence, there is an urgent need for alternative energy sources. Automobiles and industrial sectors are the major consumers of energy, and diesel engines are mainly used in industrial, transport, and agricultural applications due to their high efficiency and reliability. The increase in price, reduced availability, and issues on global warming

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due to harmful gases from automobiles and industrial sectors pay more attention to the search for alternative fuels [**1**]. It is commonly accepted that clean combustion in diesel engines can be achieved only if engine development with fuel reformulation and the use of alternative fuels are implemented.

Animal fats, vegetable oils, alcohols, liquefied petroleum gas, hydrogen, etc. are considered as potential alternative fuels for internal combustion engines [**2–6**]. Among these fuels, animal fats and vegetable oils have come across as good choice for diesel engines. As a compression ignition engine fuel, animal fats and vegetable oils have most of the properties (cetane number, lower heating value, density, etc.) comparable to diesel [**2**, **3**] (Table 1). Investigations have reported that animal fats and vegetable oils used as fuel in diesel engines produced the same power output but with reduced thermal efficiency, increased emissions, lubricant contamination, and formation of carbon deposits due to their high viscosity and density [**6**, **7**]. A number of methods, such as blending vegetable oils and fats with alcohols and diesel, transesterification of fats and vegetable oils, emulsification of oils and combustion chamber modifications, etc., are found as some of the methods to improve the performance of high-viscous fuels in diesel engines [**8–10**]. However, preparation of emulsion involves external mixing devices such as stirrer, heater, and vessels. Surfactants used for making the emulsions are very expensive as compared to vegetable oils. Stability of the emulsions for long periods is still an open issue [**11**]. In addition, high viscosity of emulsions makes difficult to be handled by the conventional fuel injection system of the diesel engine to inject the emulsified fuel. Though transesterifcation of vegetable oils is quite attractive, transesterification requires complex chemical reactions. It is a time-consuming process and it results in certain by products, such as glycerin and fatty acids, which cannot be used as fuel though they have some other use [**12**]. In addition, esterified biodiesel always results in higher NO emissions.

Dual-fuel operation in diesel engines has become attractive due to the advantage of significant reduction of smoke emissions [**13**, **14**]. Dual-fuel mode with fumigation technique offers the advantage of easy conversion of the normal diesel engine to work on volatile fuels with high-viscous vegetable oils as pilot fuels. The dual-fuel engine is a multifuel engine that can operate effectively on a wide range of fuels while maintaining the capacity for operation as a conventional engine. The dual-fuel engine needs a fuel with a high octane number as primary fuel and high cetane number as pilot [**13**]. Since vegetable oils have higher cetane numbers, they can be used in dual-fuel operation as pilot fuels. The dual-fuel operation can produce very less smoke and hence it is a method to improve the vegetable oil engine. In a dual-fuel engine, the primary fuel with a high octane number is carbureted along with the intake air stream. The resulting homogeneous mixture is compressed to a temperature below the self-ignition point. The pilot fuel (with a high cetane number) is injected through the standard injection system. This self-ignites and initiates the combustion in the primary fuel air mixture. The primary fuel supplies most of the energy. It has been reported that the dual-fuel operation at light load is slightly less efficient [**14**]. However beyond half load, the efficiency of dual-fuel engine is improved sufficiently to surpass that of the conventional fuel with major reduction in smoke emissions [**15**]. Orange oil extracted from the peel of orange fruits has most of its properties very close to gasoline and hence it is a good alternative fuel for spark ignition engines [**16**]. It was also used as diesel engine fuel. It has been reported that the performance of a diesel engine was superior with orange oil as compared to diesel [**17**]. Since orange oil is highly volatile (and has high octane number of 106), it can be used as the inducted fuel and Jatropha oil (JO) can be used as pilot fuels for ignition.

Table 1 Properties of fuels

Properties	Diesel	JO	Orange oil
Formula	$C_{13}H_{24}$	$C_{20}H_{22}O_3$	$C_{10}H_{16}$
Density (kg/m^3)	840	918.6	820
Calorific value (kJ/kg)	42 490	39 774	42 000
Viscosity (CST)	4.59	49.93	0.95
Cetane number	45–55	40–45	10
Autoignition temperature (°C)	280	340	–
Vapour pressure at 20 °C	10 mmHg	–	4 mmHg
Oxygen (wt%)	0	15.4	0
Stoichiometric A/F ratio (kg/kg)	14.6	10.8	14.21
Flash point (°C)	75	240	46.1

Though several research projects have been carried out on a number of alternative fuels in diesel engines, not much data are available on the performance of constant speed stationary diesel engine fuelled completely with vegetable oil in dual-fuel operation. Since low-horsepower stationary diesel engines are commonly used in agricultural and transport sectors, there is a need to study their performance using alternative fuels. The results can be further extended to high power output multicylinder engines also. Hence it was proposed to use a small quantity of light, volatile vegetable oil to control smoke in the JO-fuelled compression ignition engine by adopting the dual-fuel mode. Orange peel oil (called as orange oil) was used as the inducted vegetable oil. An experimental study was undertaken with the objective of finding out the performance of a diesel engine operated on the fuels completely obtained from renewable energy sources such as JO and orange oil. The effect of orange oil induction on the performance, emissions, and combustion characteristics of a diesel engine fuelled with JO was studied experimentally using dual-fuel mode of operation with JO as pilot fuel and orange oil as inducted fuel.

Table 2 Engine details

Name of the engine	Kirloskar AV1
General details	Single cylinder, four stroke, CI, water cooled.
Bore and stroke	$80 \times 120\ \text{mm}^2$
Compression ratio	15:1
Rated output	3.68 kW at 1500 r/min
Fuel injector opening pressure	170 bar
Injection timing	27° Before TDC (Diesel) 29° Before TDC (JO)

2 EXPERIMENTAL SETUP AND EXPERIMENTAL PROCEDURE

A single–cylinder, four-stroke, water-cooled diesel engine developing 3.7 kW at 1500 r/min was used for the research work. Engine details are given in Table 2. The schematic of the experimental setup is shown in Fig. 1. An electrical dynamometer was used for loading the engine. A turbine-type meter connected to a large tank was attached to the engine to make air flow measurements. A carburetor fitted on the intake manifold of the engine was used to supply orange oil. An optical shaft position encoder was developed and used to give signals at TDC. A photo sensor along with a digital rpm indicator was used to measure the speed of the engine. The fuel flowrate was measured on the volumetric basis using a burette and a stopwatch. Chromel alumel thermocouples in conjunction with a digital temperature indicator were used for measuring the exhaust gas temperature. A high-speed digital data acquisition system in conjunction with a piezoelectric transducer was used for the measurement of cylinder pressure history. An infrared

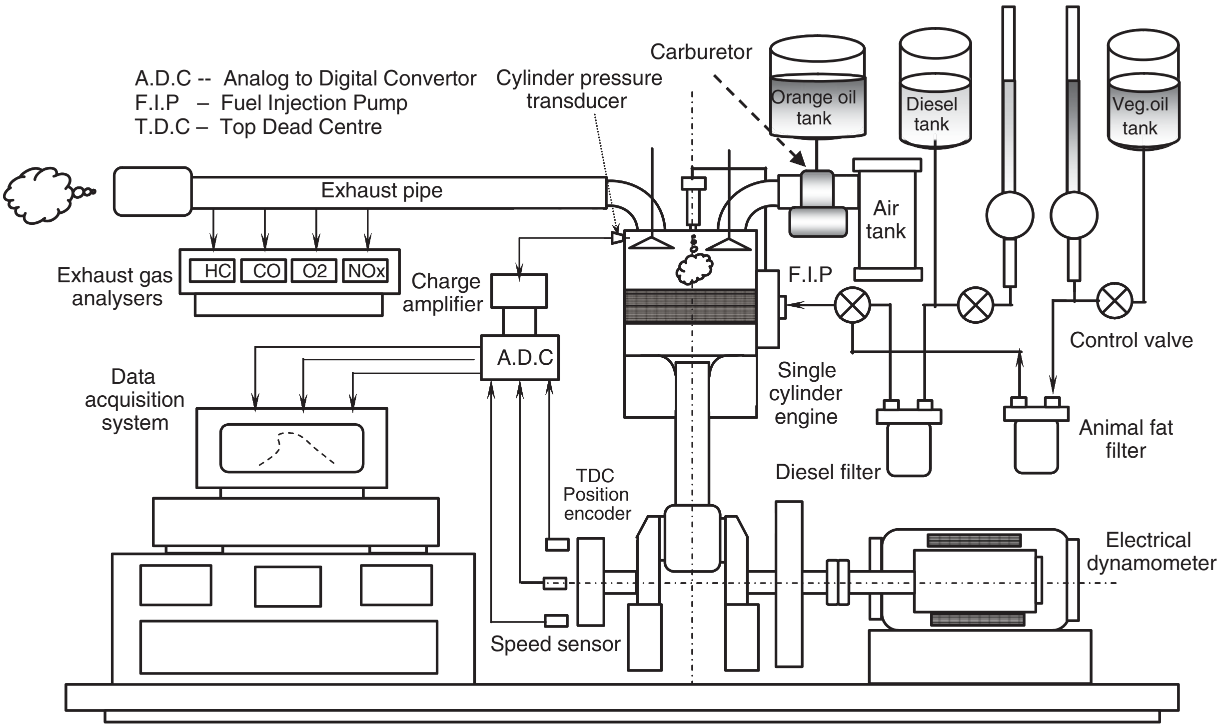

Fig. 1 Experimental setup

exhaust analyser was used for the measurement of HC/CO in the exhaust. For measuring NO, a chemiluminescent analyser was utilized. Smoke levels were obtained using a standard Bosch system.

Experiments were initially carried out on the engine using diesel as the fuel in order to provide baseline data. During the entire investigation, the injection timing was optimized and set at 27° before TDC which was the optimum for diesel operation. The cooling water temperature at the outlet was maintained at 70 °C. The engine was stabilized before taking all measurements. The injection timing for JO was optimized and set at 29° before TDC. The orange oil admission was extended up to 40 per cent of the energy share at maximum power output. However, it was extended up to 74 per cent of the energy share at 20 per cent load. In all cases, pressure crank angle data were recorded and processed to get combustion parameters.

3 RESULTS AND DISCUSSION

Results of the performance, emission, and combustion parameters of JO as pilot fuel and orange oil as the primary inducted fuel have been discussed in the following section. In all figures, single-fuel operations with neat diesel and neat JO have been drawn to have baseline reference for comparison with the dual-fuel operation.

3.1 Performance parameters

Figure 2 shows the variation of brake thermal efficiency with orange oil energy share while using JO as pilot fuel. It can be seen that the thermal efficiency is lower with JO as compared to diesel during single-fuel operation (i.e. at 0 per cent orange oil energy share) at all power outputs. This is due to poor mixture formation as a result of low volatility, higher viscosity, and density of the JO. The brake thermal efficiency of JO is about 27.2 per cent when used as neat fuel, where as it is about 30 per cent with conventional diesel. In case of dual-fuel operation with JO as pilot fuel, the brake thermal efficiency increases mainly at higher power outputs such as at 100 per cent, 80 per cent, and 60 per cent loads. The brake thermal efficiency with JO increases from 26.9 per cent to 28.2 per cent at an orange oil energy share of 38 per cent, which is the best efficiency point at 80 per cent load. It is 29.4 per cent at 100 per cent load at an energy share of about 31 per cent. The rapid flame propagation through the orange oil air mixture increases the heat release rate as compared to the neat JO mode, which is the reason for the increase in thermal efficiency. However, beyond 31 per cent of orange oil energy share, engine operation becomes erratic and noisy. This condition was noticed from very high rate of pressure rise as a result of engine knock. This phenomenon of knocking in dual-fuel engine at very high rates of inducted fuel at high power outputs was also noticed by the earlier studies [**18**]. Hence, the orange oil inducted was stopped beyond 40 per cent of the energy share in order to avoid mechanical damage of the engine parts. At part loads (40 per cent and 20 per cent loads), orange oil induction shows inferior performance due to poor combustion of the inducted mixture.

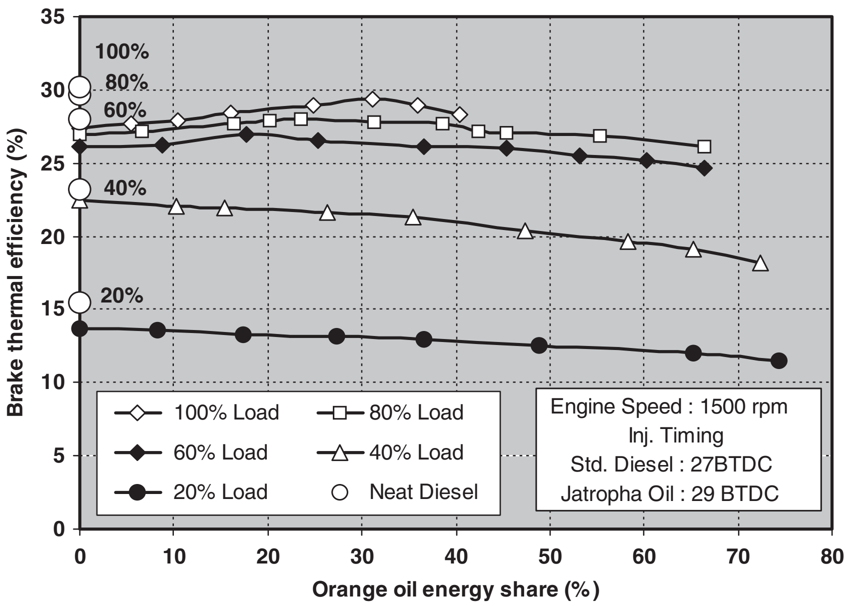

Fig. 2 Variation of brake thermal efficiency with orange oil energy share

Figure 3 shows the variation of volumetric efficiency of JO oil in dual-fuel operation with orange oil induction. The volumetric efficiency with JO is lower than diesel at 0 per cent orange oil admission (i.e. with neat JO mode) at all power outputs. This is because of the temperature of the retained exhaust gases, which depends on the exhaust gas temperature. The retained exhaust gas heats the incoming fresh air and lowers the volumetric efficiency. Diesel has the lower exhaust temperature and JO has the highest exhaust temperature. The volumetric efficiency is reduced further with JO when orange oil is inducted at all loads and all orange oil energy shares as seen in Fig. 3. This is because the orange oil vapour displaces air that is admitted in to the cylinder. In addition at high rates of orange oil admission, there is an increase in exhaust gas temperature which increases the temperature of the incoming air and hence the trend.

Exhaust gas temperature as shown in Fig. 4 is higher for JO as compared to diesel at 0 per cent orange oil admission for all power outputs. This is

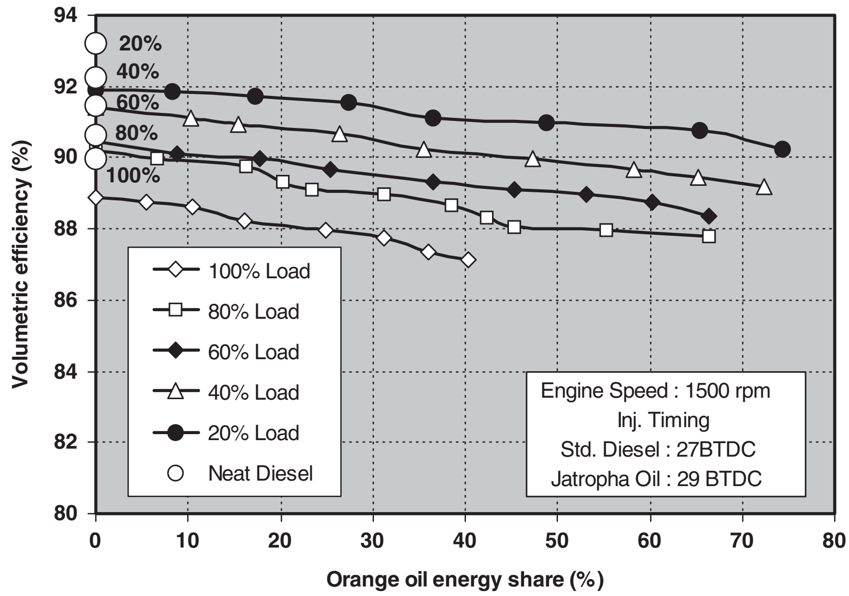

Fig. 3 Variation of volumetric efficiency with orange oil energy share

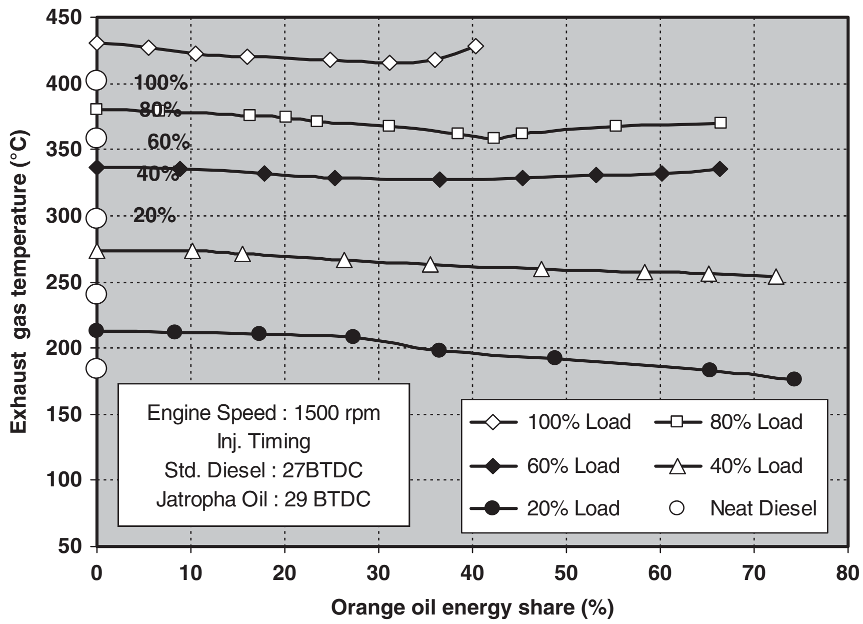

Fig. 4 Variation of exhaust gas temperature with orange oil energy share

due to slow combustion of the injected fuel. The poor volatility and high viscosity of JO are responsible for this trend of high exhaust gas temperature. The maximum temperature of exhaust gas at peak load is 430 °C for the JO and 406 °C for diesel. With the increase in proportion of orange oil, the exhaust gas temperature gradually drops at all loads due to reduction in the charge temperature. The drop in exhaust gas temperature is 380–364 °C with JO at 80 per cent load at the maximum efficiency point of 31 per cent orange oil energy share. At higher rates of orange oil admission, the exhaust temperature is increased due to rapid combustion. However, the exhaust gas temperature is lower at part loads at all orange oil admission rates.

3.2 Emission parameters

The variation of smoke emission of JO with orange oil energy share is shown in Fig. 5. With neat JO operation, smoke emission is very high at all power outputs as compared to neat diesel. This is due to the poor

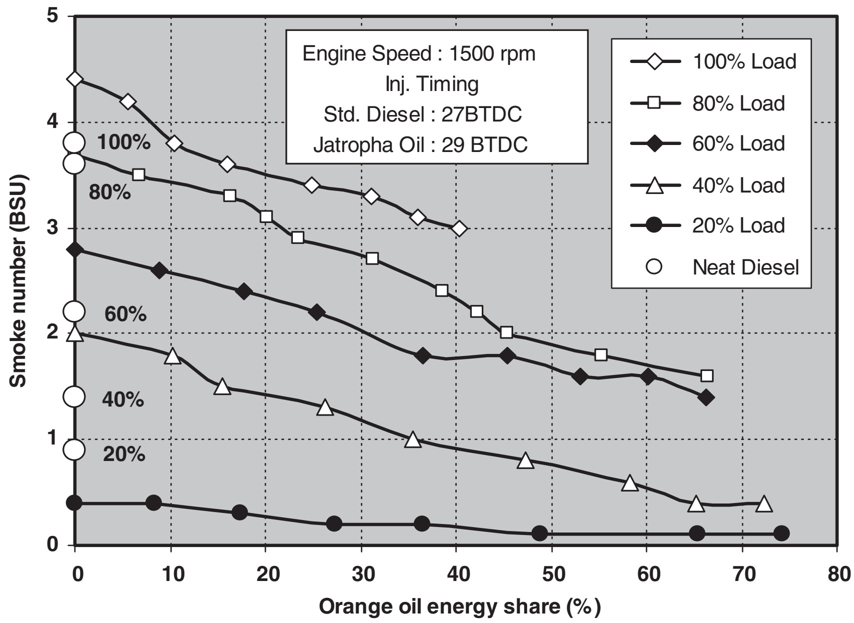

Fig. 5 Variation of smoke number with orange oil energy share

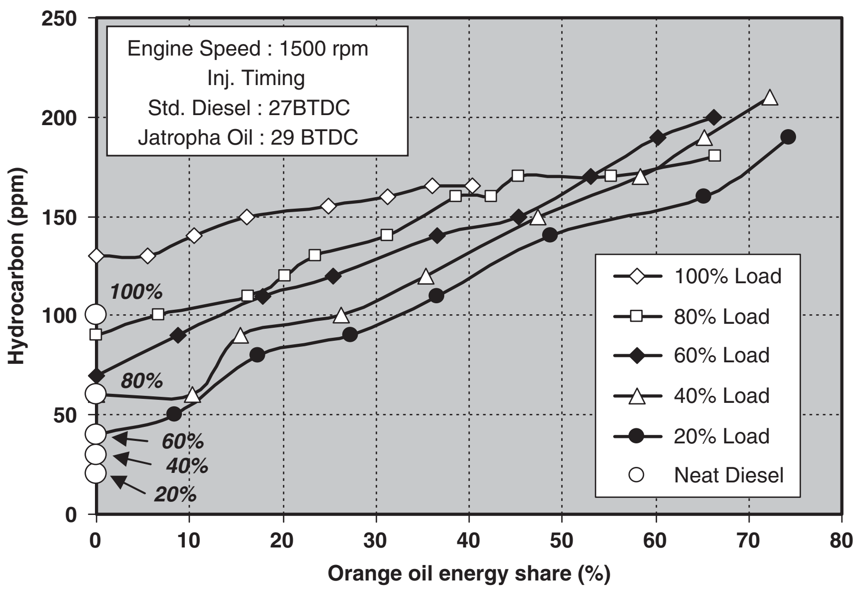

Fig. 6 Variation of hydrocarbon emissions with orange oil energy share

atomization of the fuel. Smoke level at maximum power is 3.8 BSU with diesel and 4.4 BSU with pure JO in the single-fuel mode. In dual-fuel operation, with JO as pilot fuel there is a significant reduction in smoke emission at all loads. Smoke emission is reduced to 2.4 BSU with JO at 80 per cent load. It is reduced from 4.4 to 3.3 BSU at an energy share of 31 per cent at peak load. This is due to rapid combustion of the orange oil in the premixed phase and reduction in the quantity of the pilot injected fuel, which forms smoke.

Figures 6 and 7 show the variation of HC and CO emissions with orange oil energy share. It is seen from the Fig. 6 that neat JO leads to higher unburned fuel emission compared to neat diesel operation. Due to poor mixing of the fuel with the air, the hydrocarbon emission is higher with JO. Hydrocarbon level is found as 120 ppm for JO and 100 ppm for the diesel during single-fuel operation at 100 per cent load. JO also leads to higher CO emissions at peak power output than with diesel as seen in Fig. 7. This is due to the fuel richness (rich fuel air mixture) with JO as

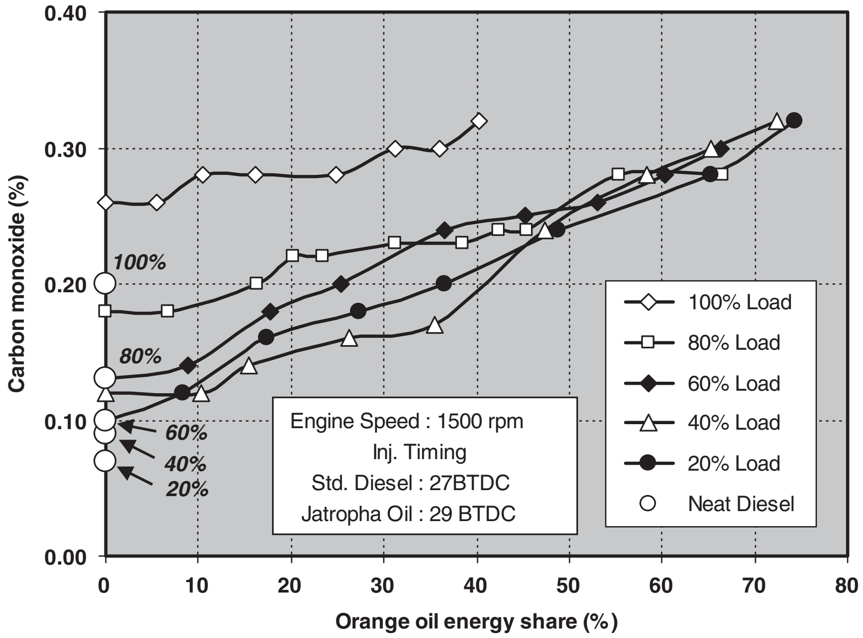

Fig. 7 Variation of carbon monoxide with orange oil energy share

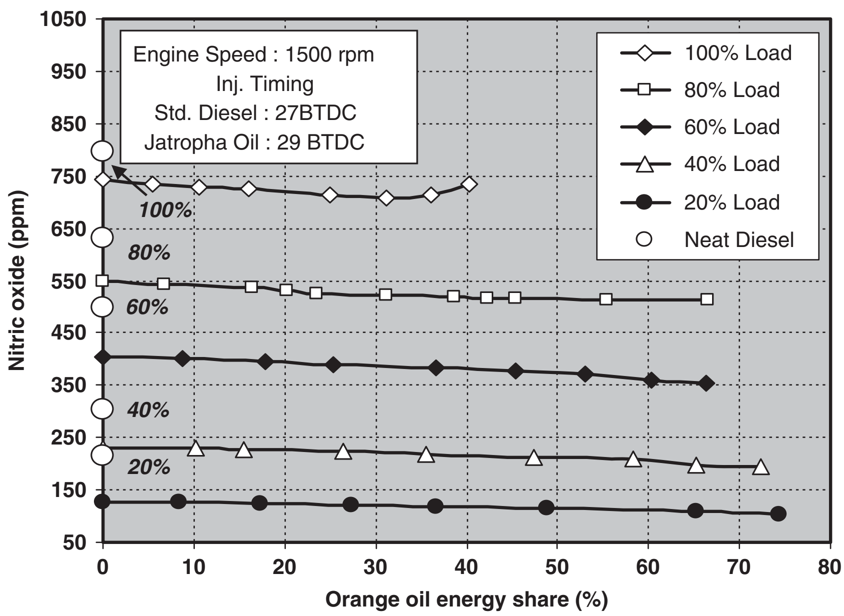

Fig. 8 Variation of nitric oxide with orange oil energy share

pilot fuel. It may be noted that a lower thermal efficiency with JO will lead to injection of higher quantities of fuel for the same engine power output condition. Since the amount of injected JO was not adjusted to maintain the power output, the power output remains approximately constant since a decrease in efficiency is compensated by an increase in mass of the injected fuel. In dual-fuel operation with orange oil carburetion, there is a further increase in hydrocarbon emission with JO as seen in Fig. 6. CO emission (Fig. 7) also raises with JO with all orange oil admission rates at all loads. It must be noted that the combustion process in this dual-fuel engine is not purely diesel combustion. Combustion when orange oil is inducted approaches constant volume combustion similar to a spark ignition engine combustion. Since HC and CO levels are generally more in spark ignition engines as compared to compression ignition engines, the HC and CO levels are found higher in dual-fuel mode even with the increased thermal efficiency. This increase in hydrocarbon level in dual-fuel mode is due to the presence of orange oil in the quench layer, which does not participate in combustion and is exhausted. The same behaviour of HC has been observed in dual-fuel operation with other inducted liquid fuels also in the earlier studies [**13**, **14**]. The rate of increase of hydrocarbon and carbon monoxide emissions is higher at lower loads, particularly at higher orange oil energy shares. This is because the injected fuel, which is the ignition source, is weak in this case.

The variation of NO emission with JO as pilot fuel in dual-fuel operation with orange oil as the primary fuel is shown in Fig. 8. It is seen that the JO emits lower NO levels as compared to standard diesel during single-fuel operation. The NO emission is observed as 795 ppm with neat diesel operation and 744 ppm with JO at the maximum load (i.e. 3.7 kW). This reduction in NO emission with neat JO is mainly associated with the reduced premixed burning rate following the delay period with JO. This is due to lower air entrainment and fuel air mixing rates with JO as compared to diesel. The heat release rate (will be explained later in Fig. 13) of neat JO clearly indicates reduced premixed combustion phase and increased diffusion combustion as compared to neat diesel operation. Thus, the peak temperature and NO levels are lower even with the increased exhaust gas temperature. In the dual-fuel mode of operation, NO is reduced further with JO at all loads. NO is reduced from 744 to 708 ppm at the maximum efficiency point of 31 per cent of orange oil energy share at maximum load. This reduction is mainly associated with the lowering of the intake charge temperature due to vaporization of the inducted orange oil. However, at full load and very high rates of orange oil admission, the NO emission is increased with JO due to high heat release rates resulting in increase in the cylinder gas temperature.

3.3 Combustion parameters

The variation of ignition delay is shown in Fig. 9. Ignition delay was calculated from the rate of pressure rise curve derived from the cylinder pressure crank angle diagram. Start of injection was obtained

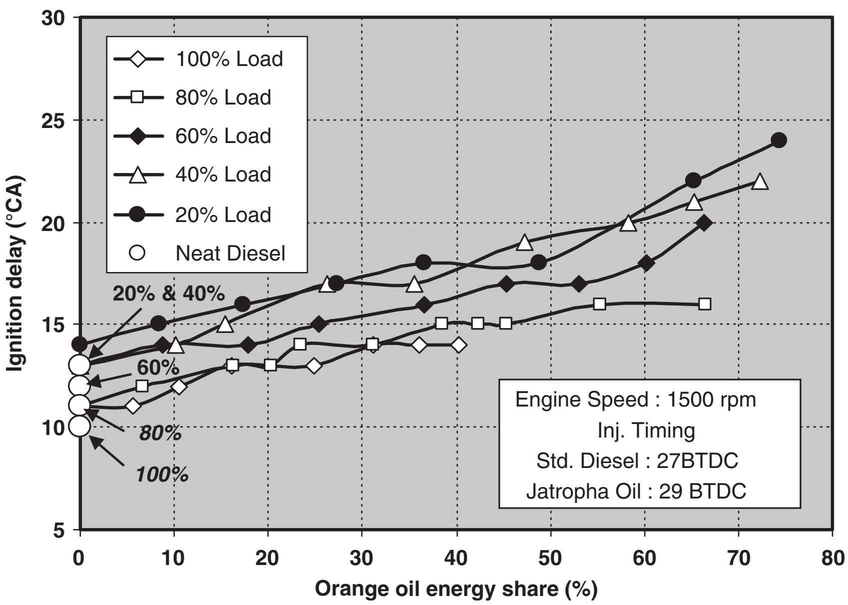

Fig. 9 Variation of ignition delay with orange oil energy share

from the needle lift sensor and start of ignition was found from the rate of pressure rise curve. Pressure crank angle diagram obtained from average of 100 cycles has been taken for getting one pressure crank angle curve.

JO showed longer ignition delays compared to diesel during single-fuel operation as expected. This increase is due to the poor atomization and mixture formation of the injected fuel. Ignition delay is further increased for JO with orange oil induction through carburetion. The increase in ignition delay is from 11° to 15° CA (CA, crack angle) with JO at the optimum energy shares at 80 per cent load. Reduction in the intake mixture temperature due to the vapourization of orange oil is the reason for this trend. The rise in the ignition delay is more pronounced at light loads.

The variation of peak pressure and the rate of pressure rise with JO for various rates of orange oil admissions are shown Figs 10 and 11. The peak pressure and maximum rate of pressure rise are lower for neat JO operation as compared to neat diesel. In a compression ignition engine, the peak pressure depends on the combustion rate in the initial stages, which in turn is influenced by the amount of fuel taking part in the uncontrolled combustion. The uncontrolled combustion phase is governed by the delay period and the spray envelope of the injected fuel. It is also affected by the mixture preparation during the delay period. Thus, the higher viscosity and lower volatility of the JO is the reason for this trend of peak pressure and maximum rate of pressure rise. With increase in the admission of orange oil, the peak pressure and rate of pressure rise increase mainly at 100 per cent and 80 per cent loads. The peak pressure increases from 59.4 to 62.6 bar with JO oil at 80 per cent load and from 62 to 66.5 bar at 100 per cent load at the maximum efficiency points. The higher peak pressure and maximum rate of pressure rise are due to rapid burning of the injected pilot fuel and orange oil entrained in the spray. It has been reported in the earlier studies that increase in load at constant speed with increase in inducted fuel mass cause an increase in ignition delay of pilot fuel which then autoignites and starts burning the inducted fuel at a higher rate of pressure rise [**19**]**.** In addition, long ignition delay also causes more amount of the injected fuel to participate in the premixed combustion phase.

The combustion duration presented in Fig. 12 is increased with JO as compared to neat diesel in neat fuel operation. This is due to higher quantity of fuel injected to maintain the power output as compared to neat diesel. With orange oil admission in dual-fuel operation, combustion duration is reduced at higher rates mainly at maximum load of 100 per cent. Increase in combustion rate due to combustion of the orange oil by flame propagation is the reason. However, at other loads and higher admission rates of orange oil, there is a notable increase in combustion duration.

The variation of heat release rate with JO and diesel in single-fuel operation at maximum power output is shown in Fig. 13. It is seen that the premixed combustion phase is more pronounced with the diesel as expected. The diffusion-burning phase indicated

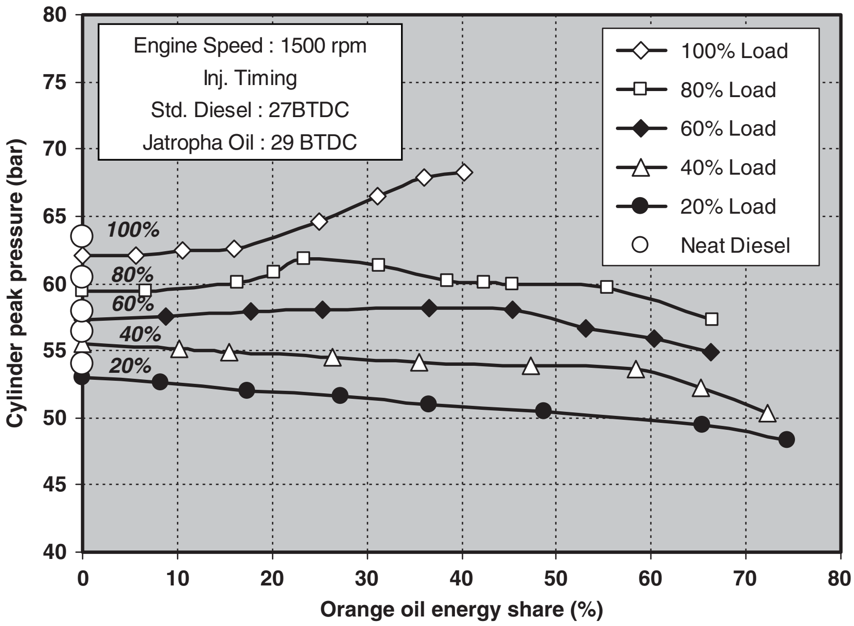

Fig. 10 Variation of cylinder peak pressure with orange oil energy share

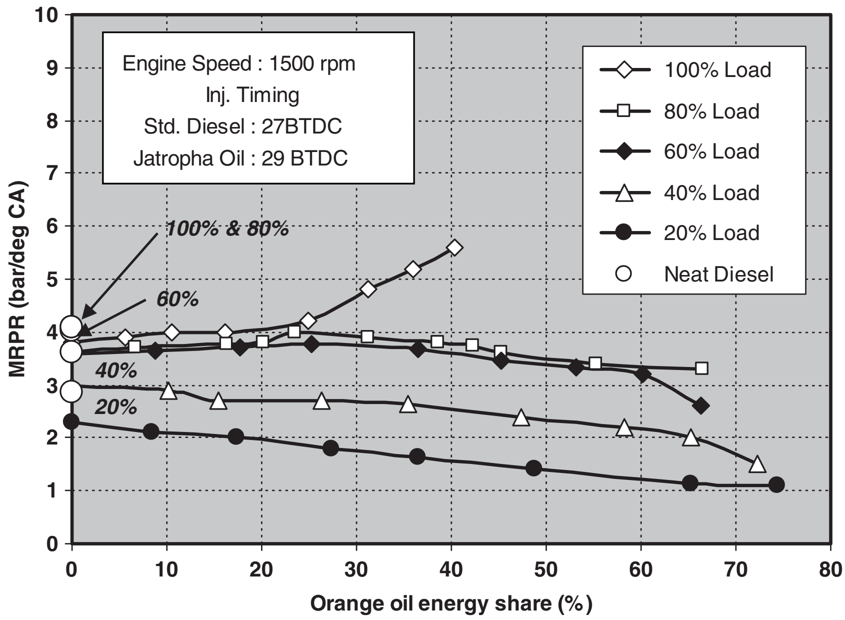

Fig. 11 Variation of maximum rate of pressure rise with orange oil energy share

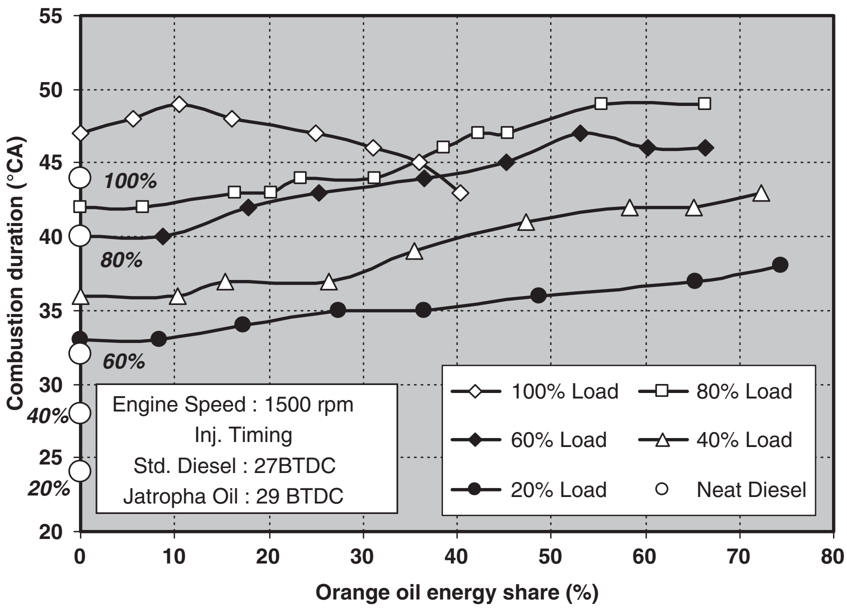

Fig. 12 Variation of combustion duration with orange oil energy share

under the second peak is greater for JO. This is consistent with expected effects of vegetable oil viscosity on the fuel spray, and reduction of air entrainment and fuel air mixing rates. At the time of ignition, less fuel air mixture is prepared for combustion with the JO. Therefore, more burning occurs in the diffusion phase rather than in the premixed phase. This is the reason for the lower brake thermal efficiency and lower NO emissions with neat JO as compared to neat diesel. However, in the dual-fuel operation with orange oil induction, there is an improvement in heat release rate with JO mainly at higher power outputs. There is no distinction between the premixed and diffusion phase as seen in Fig. 14. It can be explained that by increasing the admission of orange oil, the premixed phase of the heat release

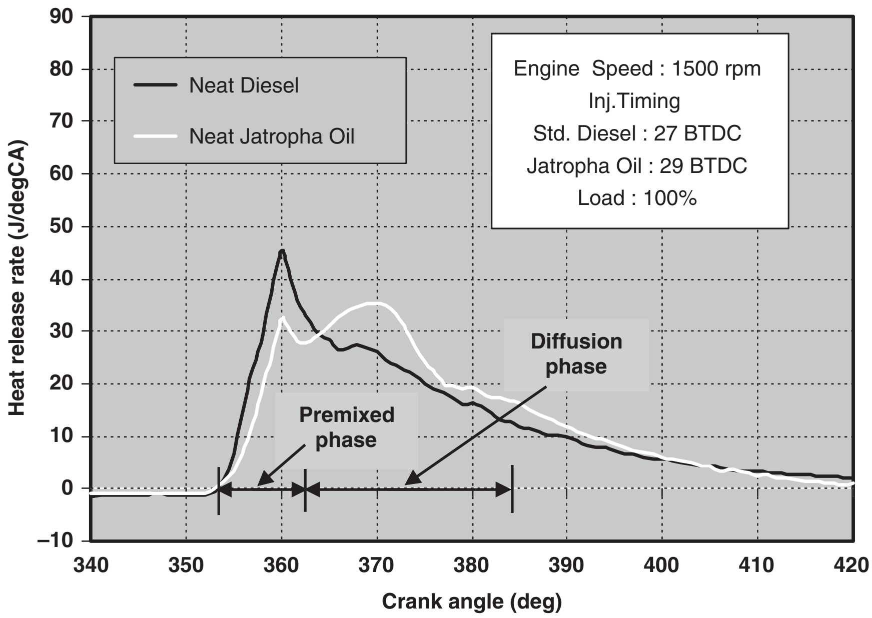

Fig. 13 Variation of heat release rate with neat diesel and neat JO at 100 per cent load

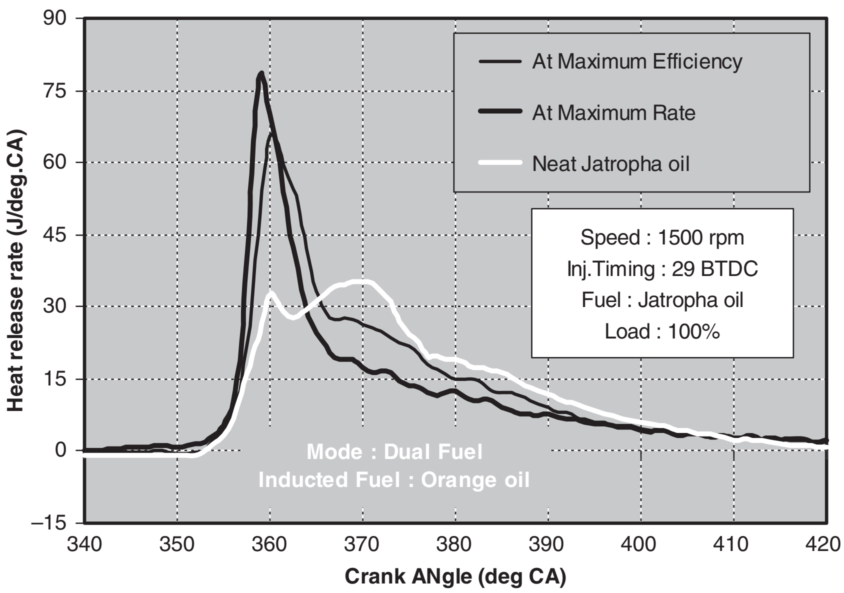

Fig. 14 Variation of heat release rate with JO in dual-fuel operation at 100 per cent load

curve is increased and the diffusion phase is reduced. This is because the orange oil burns with the pilot fuel in the early stages itself. Considerable portion of the heat release due to the flame propagation of orange oil results in improved heat release rate. At the highest orange oil admission point, the peak heat-release rate becomes very high due to very rapid combustion of orange oil. Similar trends can be seen in Fig. 15 also at 80 per cent load. However, the values of the maximum heat release are lower as compared to 100 per cent load for all orange oil admissions.

4 CONCLUSION

In the dual-fuel operation with orange oil induction, there is an appreciable increase in brake thermal efficiency compared to neat JO operation, mainly at high power outputs. Smoke is significantly reduced with orange oil induction at all power outputs. In addition,

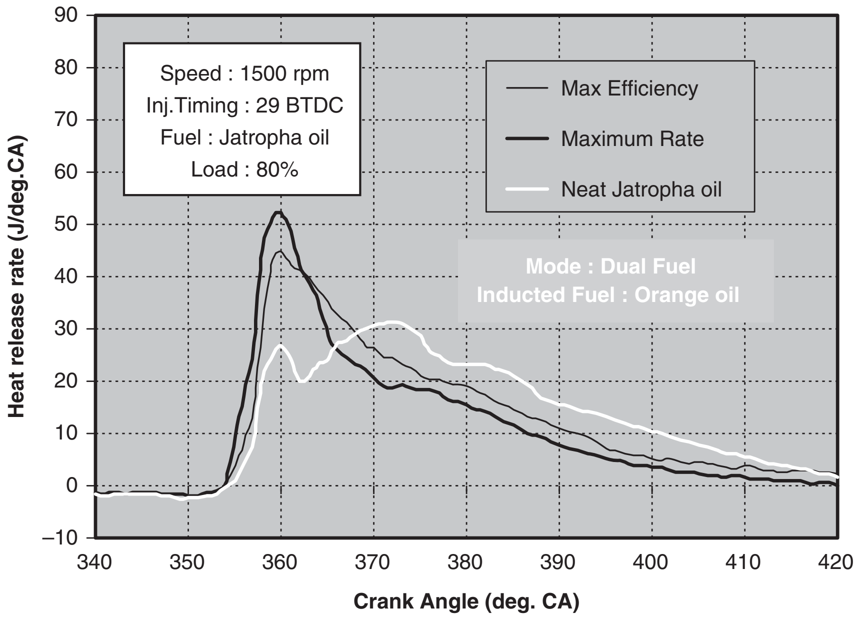

Fig. 15 Variation of heat release rate with JO in dual-fuel operation at 80 per cent load

NO emission is also reduced at all power outputs with orange oil induction. However, the decrease is observed as small. Hydrocarbon and carbon monoxide emissions are found to be higher. Ignition delay, peak pressure, rate of pressure rise, and heat release rate are higher at medium and high outputs. From the above results, it is concluded that orange oil can be used as the inducted fuel for reducing smoke and NO emissions, with improved brake thermal efficiency in a compression ignition engine fuelled with JO. The penalty will be increase in HC and CO emissions.

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