STUDIES ON A CI ENGINE USING ORANGE SKIN POWDER DIESEL SOLUTION WITH DIFFERENT FUEL NOZZLE OPENING PRESSURE

by

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Experiments have been conducted to study the effect of nozzle opening pressure on the combustion process and exhaust emissions of a direct injection diesel engine fueled with orange skin powder diesel solution (OSPDS). Earlier investigation by the authors revealed that 30% OSPDS was optimum for better performance and reduced emissions. In the present investigation the nozzle opening pressure was varied with 30% OSPDS and the combustion, performance, and emissions characteristics were studied and compared with those of diesel fuel. The different nozzle opening pressures studied were 215 bar, 235 bar, and 255 bar. The results showed that the cylinder pressure with 30% OSPDS at 235 bar fuel nozzle opening pressure, was higher than that of diesel fuel as well as at other nozzle opening pressures. Similarly, the ignition delay was longer with shorter combustion duration in case of 30% OSPDS at 235 bar nozzle opening pressure. The brake thermal efficiency was higher at 235 bar than that of other fuel nozzle opening pressures with OSPDS and lower than that of diesel fuel. The NO_x emission was higher and HC and CO emissions were lower with 30% OSPDS at 235 bar. The smoke emission was marginally lower at 235 bar and marginally higher at 215 bar than diesel fuel. The performance of the engine at 235 bar nozzle opening pressure was better with reduction in emissions except NO_x than other nozzle opening pressures.

Key words: *diesel engine, orange skin powder diesel solution, combustion, performance, emission, nozzle opening pressure*

Introduction

Diesel engines have been widely used as a power source of engineering machinery, water pumping, shipping, and automobiles because of their excellent driveability and thermal efficiency. At the same time, diesel engines are the major contributors of various types of air pollutants such as smoke, particulate matter (PM), oxides of nitrogen (NO_x), and other harmful compounds. Hence stringent norms are imposed on exhaust emissions from automobiles to protect the environment. Many research works are in progress to develop newer technologies to reduce emissions.

In the last 20 years, considerable attention has been focused on biomass fuel. Biomass represents a renewable energy source that has not yet been harnessed to its full potential. Biomass cannot be fermented for generation of biogas which can be pulverized and mixed with a carrier medium like oil, water or methanol to form a slurry fuel. Many investigations have been carried out in several countries in the development of coal slurry fuels, especially COS (coal-oil solutions), CWS (coal-water slurry) or their use in medium diesel engine [1]. Experiments were

carried out using pulverized coconut shell – high speed diesel oil slurry as an alternative fuel [2]. However, because of the high viscosity fraction of coconut shell powder, the loading was limited to 25%. In addition the problem of erosion was observed.

India is an agriculture based country and biomass like orange skin is available in large quantities. Hence the disposal of solid waste becomes a problem. An attempt was made for fuel modification to use the orange skin powder directly, in the form of a solution, as a fuel in a CI engine [3]. In this technique, dry orange skin powder was mixed with diesel fuel and a solution was obtained. In this investigation, it was observed that the engine using orange skin powder diesel solutions (OSPDS) (10, 20, and 30%) gave satisfactory performance. It was proved that orange oil can be used as an alternative fuel in a CI engine by adopting the fumigation technique [4].

One of the operating parameters affecting the performance and emissions of a diesel engine is the fuel nozzle opening pressure [5-7]. Most of the nozzle opening pressure studies has been conducted on direct injection (DI) diesel engines using various alternative fuels. Higher nozzle opening pressures generate faster combustion rates, resulting in high cylinder gas temperatures. This tends to vaporize the fuel spray so quickly that the fuel cannot penetrate deeply into the combustion chamber. Therefore, the initial combustion with the spray is restricted to a small region near the injector, and the flame spreads around the chamber through slow propagation [8].

High nozzle opening pressure was used to reduce particulate emissions significantly with no change in HC emissions and increasing NO_x emissions marginally [9]. Increasing the nozzle opening pressure of the engine running with ethanol-diesel emulsions, reduces CO and smoke emissions, while increasing the NO_x [10]. Other methods used to reduce the NO_x and PM were using cetane number improver in ethanol-diesel blend [11] and diesel-oil emulsion [12].

The main aim of the present work is to use a solution obtained from orange skin powder mixed with diesel and to determine the combustion, performance, and emission characteristics of a compression ignition engine at different fuel nozzle opening pressures and determine the optimum nozzle opening pressure.

Fuel preparation

The orange skin powder mixed with diesel in the form of a slurry or in the form of a solution can be used as an alternate or partial substitute to diesel. The process of making orange skin powder involves 3 steps:
(1) wet orange skin was taken in the form of a hemispherical cup; this was cut into 4 pieces for solar drying,
(2) orange peel was dried for about 7 to 10 days (the weight of orange skin reduced by 60%), and
(3) the dried orange skin was ground in a flourmill to make a fine powder; the powder was sized to less than 100 microns.

A solution was prepared from de-moisturized and finely ground orange skin powder mixed with diesel fuel on mass basis. 10% of orange skin powder was mixed with 90% diesel (*i. e.* 100 g powder + 900 g diesel fuel). After 15 to 20 days, the mixture was filtered in the form of OSPDS and only 93% (930 g) of OSPDS was obtained. Similarly, 20% blend (200 g powder + 800 g diesel fuel) and 30% blend (300 g powder + 700 g diesel fuel) gave 85% (850 g) and 76% (760 g) OSPDS, respectively. Diesel loss increased when more dry orange skin powder was mixed with diesel fuel because the diesel fuel absorbs the powder in the form of a precipitate. Precipitated diesel fuel was squeezed and added to the OSPDS. The OSPDS available in the form of a solution contains no solid particles. In the study the maximum quantity of orange skin powder used was

30% and the problems of plugging in the fuel filter, fuel injection pump and injector were not observed. Diesel fuel No. 2 was used for baseline data generation.

It can be observed from tab. 1 that orange skin powder consists of moisture, mineral matter, nitrogen, and oxygen. These properties make the powder to react with diesel fuel when orange skin powder is mixed with diesel fuel. A comparison of some properties of 30% orange skin powder diesel solution with diesel fuel and the ultimate analysis of orange skin powder is shown in tab. 1.

Table 1. Comparison of properties of OSPDS with diesel fuel and ultimate analysis of orange skin powder

Comparison of properties of OSPDS with diesel fuel			Ultimate analysis of orange skin powder	
Properties	Diesel	30% OSPDS	Components	Percentages
Calorific value, [kJkg^{-1}]	43.000	40.992	Moisture	7.43
Density, [kglit^{-1}]	0.8284	0.8515	Mineral matter	3.87
Viscosity at 40 °C, [cSt]	2.7	4.1	Carbon	44.44
Flash point, [°C]	52	70	Hydrogen	3.80
Fire point, [°C]	65	83	Nitrogen	1.73
			Sulphur	1.94
			Oxygen	36.79

Experimental setup and procedure

Tests were carried out on a single cylinder, air cooled, direct injection stationary diesel engine that had a bore of 87.5 mm, a stroke of 110 mm, and displacement of 661 cm^3. The schematic diagram of the experimental setup is shown in fig. 1. The engine (1) had a rated output of 4.4 kW at 1500 rpm with a compression ratio 17.5:1 and nozzle opening pressure of 215 bar at pump valve opening timing 23° before top dead centre (bTDC), as specified by the manufacturer. The engine was coupled to an electric dynamometer (2) to provide the brake load with an electric panel (3). A fuel switching circuit (7) was used to change over from one fuel to another while the engine was running. The fuel consumption was measured with the aid of a burette (5, 6) and a stop watch arrangement. The exhaust gas temperature was measured by a thermocouple (10).
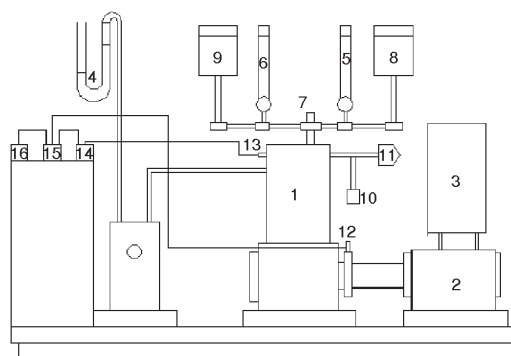
Figure 1. Experimental setup

(1) Test engine, (2) Dynamometer, (3) Panel board, (4) Orifice flow meter, (5) Diesel burette, (6) Orange skin powder diesel solution burette, (7) Fuel shifting valve, (8) Diesel tank, (9) OSPDS fuel tank, (10) Exhaust gas thermocouple, (11) Five gas analyzer, (12) Crank angle encoder, (13) Piezo electric transducer, (14) Pressure charge amplifier, (15) CRO, (16) Printer

The in-cylinder pressure was measured by installing a KISTLER type 7063-A, sensitivity 7A Pc/bar, water cooled piezo-electric pressure transducer (13) into the cylinder head, which was connected to the charge amplifier (14), and a TDC encoder (12) was fixed on the flywheel. The analogue signals from these were converted to digital signals and fed into a cathode ray oscilloscope (CRO) (15). A printer (16) was used to get the signal output of CRO. For emission measurements, a five gas analyzer (11) was used to measure the level of HC, CO_2, CO, O_2, and NO_x. The specifications of the measuring devices with their accuracy are given in tab. 2.

Table 2. Specification of measuring instruments

No.	Name of the measuring instrument	Make	Range	Accuracy
1	Pressure charge amplifier	KISTLER Instruments AG, Switzerland	12 stages graded pC 10…5000 1:2:5 and stepless 1 to 10	3%
2	Piezo-electric transducer	KISTLER type 7063-A	Sensitivity 7A Pc/bar	< 0.5%
3	Cathode ray oscilloscope	Make: Hewlett Packard, HP54600B Series	2 mV/div to 5V/div	< 1.5%
4	Exhaust gas analyzer	Make QRO 401	CO – 0 to 9.95% HC – 0 to 9999 ppm O_2 – 0 to 25% CO, HC, CO_2 – NDIR method NO_x, O_2 Electrochemical method	0.01% 1% 0.01%
5	Smoke meter	Type and make: TI diesel tune, 114 smoke density testers TI	0-10 Bosch smoke number	0.2

Experiments were carried out initially using diesel fuel to generate the base line data. The OSPDS was stored in a separate fuel tank as shown in fig.1. The engine was started with diesel fuel and then switched over to 10, 20, 30, and 40% OSPDS for the experimental investigation with fuel nozzle opening pressure of 215 bar at pump valve opening timing of 23° bTDC. A standard mechanical type fuel injection pump was used for supplying OSPDS. After experimentation with OSPDS, the engine was run for some time with diesel to enable trouble free operation later on.

Knocking and misfiring were observed when the engine was fueled with 40% OSPDS. It was also noticed that OSPDS quantity decreased when orange skin powder was added with diesel fuel. For the above reasons, 30% OSPDS was considered as optimum. From the experimental results, it was observed that 30% OSPDS showed lesser emissions however with a reduction in brake thermal efficiency compared to 10 and 20% OSPDS. Some of the important results are given in tab. 3. The brake thermal efficiency could be improved and emissions with 30% OSPDS could be reduced by adopting higher nozzle opening pressures.

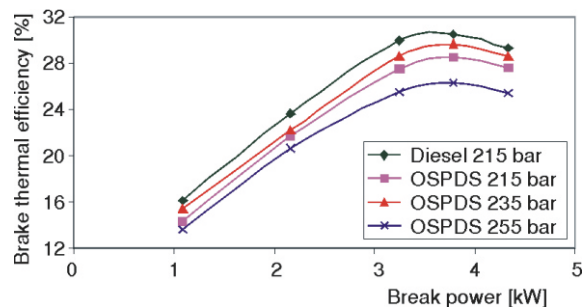**Table 3. Salient results of OSPDS compared to diesel fuel**

Parameters	Peak Cy. Pr [bar]	η_{bth} [%]	NO$_x$ [gkW^{-1}h^{-1}]	CO [gkW^{-1}h^{-1}]	HC [gkW^{-1}h^{-1}]	Smoke [BSU]
Diesel fuel	71	29.3	14.9	0.87	0.06	0.85
10% OSPDS	71.2 (+0.2)	28.5 (–0.8)	15.8 (+0.8)	0.37 (–0.4)	0.04 (–0.02)	0.76 (–0.9)
20% OSPDS	72 (+1)	28 (–1.3)	16.8 (+1.9)	0.31 (–0.56)	0.03 (–0.03)	0.71 (–0.14)
30% OSPDS	73 (+2)	27.6 (–1.7)	17.7 (+2.8)	0.3 (–0.57)	0.02 (–0.04)	0.68 (–0.17)

Experiments were conducted with diesel fuel and optimum 30% OSPDS from no load to full load with the original nozzle opening pressure of 215 bar, and then changed to 235 bar and 255 bar. All the experiments were conducted at the rated engine speed of 1500 rpm, and constant injection timing of 23° bTDC.

Results and discussion

Brake thermal efficiency

The thermal efficiency of OSPDS at different nozzle opening pressure is compared with that of diesel fuel in fig. 2. The brake thermal efficiency at full load for OSPDS at 215 bar nozzle opening pressure is 27.6%, at 235 bar it is 28.7%, at 255 bar it is 25.3%, and for diesel fuel it is 29.3%. The brake thermal efficiency at 235 bar with OSPDS is about 1.1% higher than that of OSPDS at 215 bar nozzle opening pressure and is about 1.2% less than that of diesel fuel. This may be due to finer fuel spray and opti-

Figure 2. Variation of brake thermal efficiency

mum droplet size of OSPDS at 235 bar fuel nozzle opening pressure. The smaller droplets of OSPDS mix with air throughout the combustion chamber forming a homogenous change, thus resulting in complete combustion at 235 bar pressure.

Carbon monoxide emission

A comparison of CO emission with brake power for diesel and OSPDS at different nozzle opening pressures is shown in fig. 3. Generally, diesel engines produce lower CO due to a higher supply of oxygen. The CO emission at full load at 215 bar in the case of OSPDS is 0.33 g/kWh, at 235 bar it is 0.26 g/kWh, at 255 bar it is 0.58 g/kWh, and for diesel it is 0.38 g/kWh. It can be noticed that the CO emission decreases upto 235 bar nozzle opening pressure. This may
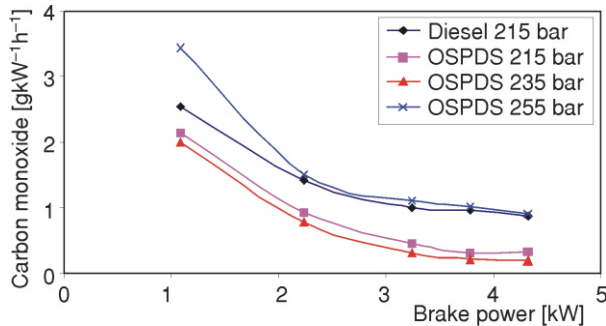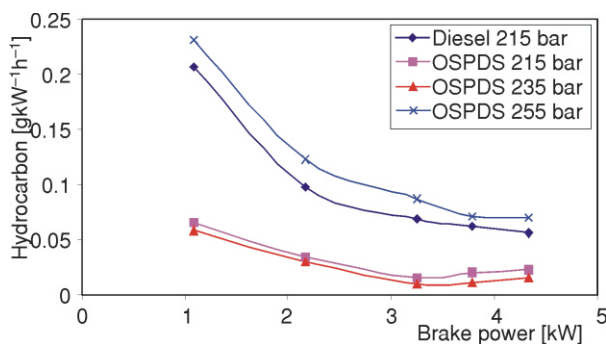
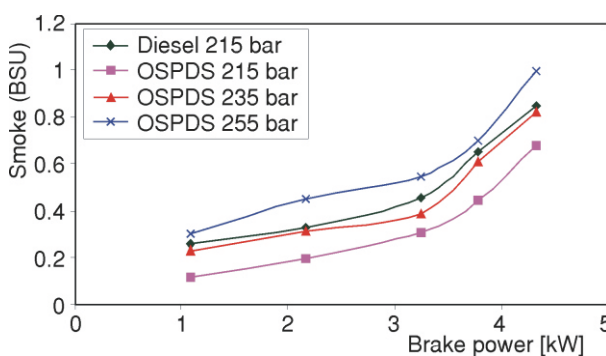**Figure 3. Variation of carbon monoxide**

Figure 4. Variation of hydrocarbon

be due to complete combustion, which results in a higher heat release rate and higher brake thermal efficiency at 235 bar pressure.

The CO emissions increase at 255 bar nozzle opening pressure due to longer duration of combustion, which results in lower peak heat release rate, and lower peak pressure [10].

Hydrocarbon emission

The variation of HC emission with brake power for OSPDS at fuel nozzle opening pressures 215 bar, 235 bar, 255 bar, and diesel is shown in fig. 4. The HC emission at full load at 215 bar fuel nozzle opening pressure is 0.023 g/kWh, at 235 bar it is 0.019 g/kWh, at 255 bar it is 0.035 g/kWh, and for diesel fuel it is 0.059 g/kWh. The HC emission reduces marginally at 235 bar and reduces significantly compared to diesel at 215 bar. This may be due to complete combustion of OSPDS and also the availability of fuel-air mixture at lean condition. Normally, oxygenated fuels give lower emissions of HC and CO [13].

When the nozzle opening pressure is increased to 255 bar, the HC increases due to improper mixing because, some fuel particles do not get burned due to higher spray angle and will not penetrate inside the combustion chamber. This may result in incomplete combustion that would increase the hydrocarbon emissions.

Figure 5. Variation of smoke

Smoke emission

Figure 5 shows the variation of smoke with brake power for diesel fuel and for OSPDS at fuel nozzle opening pressures of 215 bar, 235 bar, and 255 bar. Smoke varies from 0.26 BSU at low to 0.85 BSU at full load in the case of diesel fuel, for the nozzle opening pressure of 215 bar with OSPDS from 0.12 BSU at low load to 0.68 BSU at full load, for 235 bar with OSPDS it is 0.23 BSU at low

load to 0.82 BSU at full load, and for 255 bar it varies from 0.3 BSU at low load to 1.0 BSU at full load. It can be observed that the smoke level is lower at 235 bar nozzle opening pressure than that of diesel fuel. The promotion of fuel air mixing caused by the spray injected with high pressure results in shorter combustion duration and a reduction in local fuel rich region [9] which results in lower smoke level with OSPDS at 235 bar nozzle opening pressure. However, the smoke level increased with an increase in the nozzle opening pressure to 255 bar in the entire load spectrum. This is due to longer diffusion combustion phase, as the fuel does not burn completely [13].

Oxides of nitrogen

Figure 6 shows the variation of NO_x with brake power for diesel and OSPDS. The NO_x emission at full load at fuel nozzle opening pressure of 215 bar with OSPDS is 17.7 g/kWh, at 235 bar it is 20.2 g/kWh, at 255 bar it is 13.67 g/kWh, and for diesel fuel it is 14.9 g/kWh. The emission of NO_x is significantly influenced by the in-cylinder gas temperature and availability of oxygen during combustion

It can be noticed that the NO_x emission is higher for 215 bar OSPDS than that of diesel and OSPDS at other nozzle opening pressures. The probable reason for the increase in NO_x may be due to high intensity heat release in the premixed combustion phase [7] with OSPDS. It is also observed that the NO_x emission decreases when the fuel nozzle opening pressure increases. This may be due to corresponding changes in the in cylinder gas temperature and the low intensity heat release rate in the premixed combustion phase and longer combustion duration.

Figure 6. Variation of oxides of nitrogen

Pressure variation

Figure 7 shows the variation of cylinder pressure with crank angle for the engine running on OSPDS and diesel fuel at rated power. The cylinder pressure is 71 bar for diesel fuel and it is 73 bar, 75 bar, and 69 bar for OSPDS at 215 bar, 235 bar, and 255 bar nozzle opening pressures, respectively. It can be observed that the cylinder pressure for OSPDS is higher at 235 bar nozzle opening pressure. This may be due to the improved spray pattern of OSPDS. It can also be observed that the ignition delay reduces

Figure 7. Variation of pressure with crank angle (color image see on our web site)

with OSPDS at 255 bar nozzle opening pressure, thus resulting in reduced cylinder pressure [7]. The lowest peak pressure with 255 bar nozzle opening pressure for OSPDS is due to a lesser heat release rate during combustion, owing to lesser spray penetration inside the combustion chamber.
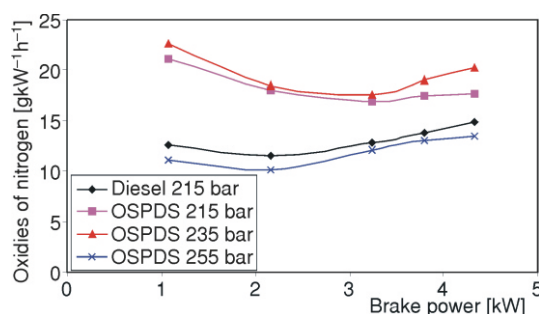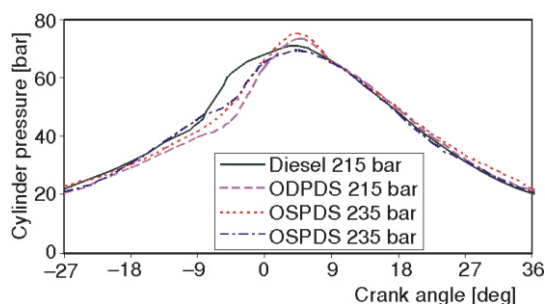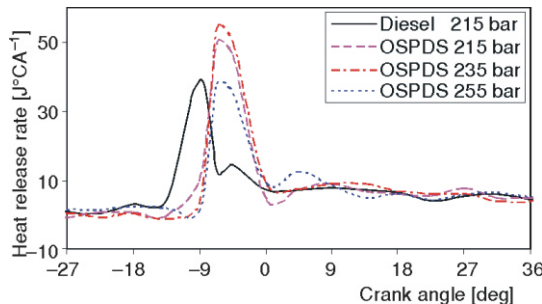
Figure 8. Variation of heat release rate with crank angle (color image see on our web site)

Heat release rate

The heat release rate for OSPDS at fuel nozzle opening pressures of 215 bar, 235 bar, 255 bar, and for diesel fuel at rated load is shown in fig. 8. It can be seen that the heat release rate for OSPDS at 235 bar nozzle opening pressure is higher than that of diesel fuel and other nozzle opening pressures. When the nozzle opening pressure is increased, the amount of fuel delivered during the ignition delay period increases because of the high injection rate. The mixing process of fuel with air is also enhanced by higher nozzle opening pressures.

As a result, a more combustible mixture is formed during the ignition delay, and higher initial combustion takes place. This is the reason for the higher heat release rate for OSPDS at 235 bar nozzle opening pressure. The peak heat release rate at full load for diesel fuel is 39 J/°CA, for OSPDS at 215 bar nozzle opening pressure it is 50 J/°CA, for OSPDS at 235 bar nozzle opening pressure it is 54 J/°CA and for OSPDS at 255 bar nozzle opening pressure it is 38 J/°CA. In the case of 255 bar, the heat release rate is lower due to lesser air entrainment. Since the fuel droplets are very fine they do not find air to form a homogeneous mixture [13].

Maximum rate of pressure rise

The variation of maximum rate of pressure rise with brake power for diesel fuel and OSPDS at different fuel nozzle opening pressures is shown in fig. 9. The maximum rate of pressure rise for diesel fuel varies from 3.4 bar/°CA at low load to 3.8 bar/°CA at full load; at 215 bar with OSPDS it is 3.5 bar/°CA at low load to 4.1 bar/°CA at full load, at 235 bar with OSPDS it is 3.7 bar/°CA at low load to 4.3 bar/°CA at full load and at 255 bar with OSPDS it is 2.8 bar/°CA at low load to 2.9 bar/°CA.

It can be noticed that the maximum rate of pressure rise is higher at 235 bar with OSPDS at all the loads. This is due to the finer spray formation resulting in better combustion [7]. In the case of 255 bar nozzle opening pressure, the rate of pressure rise is lesser compared to other nozzle opening pressures with OSPDS and diesel fuel. This may be due to lesser spray penetration with OSPDS, which results in a lower rate of pressure rise.

Figure 9. Variation of maximum rate of pressure rise

Conclusions

A single cylinder compression ignition engine was operated successfully using orange skin powder diesel solution (OSPDS) as an alternative fuel at different nozzle opening pressures. Tests carried out at 215 bar, 235 bar, and 255 bar nozzle opening pressure indicated that 235 bar is optimum for better performance and reduction in emissions. The following conclu-

sions are made based on the experimental results obtained with 235 bar nozzle opening pressure and compared to diesel operation.

An increase in nozzle opening pressure of about 20 bar (235 bar) gives improved brake thermal efficiency of about 1.1% than that of 215 bar nozzle opening pressure with OSPDS.

CO emission reduced by about 39% of OSPDS is comparable to that of diesel.

HC emission is lesser by about 66%.

Smoke emission reduced by 27%.

NO_x emission is higher by about 26%.

The peak cylinder pressure and the peak heat release rate increased.

It is observed that the use of OSPDS in an engine at 235 bar nozzle opening pressure, increase the brake thermal efficiency than at 215 bar. At the same time, the smoke, HC and CO emissions are lower and the NO_x emissions are higher with OSPDS than that of diesel fuel.

Acknowledgment

References

[1] Udayakumar, M, Venkatachalapathy. S., Coal Fueled Diesel Engines: A Status Report, *Proceedings*, 17th National Conference on I. C. Engines and Combustion, Narosa Publishing House, New Delhi, 2000, pp. 146-153

[2] Shankapal, S. R., Lakshminarayan, H. V., Operation of Compression Ignition Engine Using Bio-Mass Based Slurry Fuel, *Proceedings,* National Conference on Energy, Anna University, Chennai, India, 1998, pp. 528-533

[3] Purushothaman, K., Nagarajan, G., Experimental Investigation on Performance and Emission Characteristics of Orange Skin Powder Mixed with Diesel as a Solution as Alternative Fuel in Compression Ignition Engine, *Proceedings*, 19th National Conference on I. C. Engine and Combustion, Allied publishers private Limited, Chennai, India, 2005, pp. 481-484

[4] Senthil Kumar, M, Ramesh, A., Nagalingam, B., Complete Vegetable Oil Fueled Dual Fuel Compression Ignition Engine. SAE International: 2001–28–0067, 2001, pp. 441-448

[5] Horrocks, R. W., Light-Duty Diesel- An Update on the Emissions Challenge, *Proceedings*, Institution Mechanical Engineers, Part D, *J. Automobile Eng., 208* (1994), 4 pp. 289-98

[6] Watanabe, H., *et al*., Study of the Effects in Exhaust Emissions in Direct Injection Diesel Engines: Effects of Fuel Injection System, Distillation Properties and Cetane Number, *JSME Rev., 19* (1998), 1, pp. 21-26

[7] Ieingur,Y., Altiparmak, D., Effect of Fuel Cetane Number and Injection Pressure on a DI Diesel Engine Performance and Emissions, *International Journal of Energy Conversion and Management, 44* (2003), 3, pp. 389-397

[8] Mbarawa, M., *et al*., Fuel Injection Characteristics of Diesel-Stimulated Natural Gas Combustion, *Ins. J. Energy Res., 23* (1999), 5, pp. 1359-1371

[9] Choi, C. Y., Reitz, R. D., An Experimental Study on the Effects of Oxygenated Fuel Blends and Multiple Injection Strategies on DI Diesel Engine Emissions, *Fuel, 78* (1999), 11, pp. 1303-1317

[10] Can, O., Celikten, I., Usta, N., Effect of Ethanol Addition on Performance and Emissions of a Turbocharged Indirect Injection Diesel Engine Running at Different Injection Pressures, *International Journal of Energy Conversion and Management, 45* (2004), 3, pp. 2429-2440

[11] Nagaraja, A. M., Prabhukumar, G. P., Characterization and Optimization of Rice Barn Oil Methylester for CI Engine at Different Injection Pressure, SAE paper 2004-28-0048, 2004, pp. 320-325

[12] Reddy, K. V., Babu, K. R., Ganesan, V., Effect of Injection Pressure on Diesel Engine Performance with Vegetable Oil Diesel Blends, *Proceedings*, 17th National Conference on I. C. Engine and Combustion, Narosa Publishing House, New Delhi, India, 2000, pp. 154-159

[13] Lu, X.-C, *et al*., Effect of Cetane Improver on Heat Release Rate and Emissions of High Speed Diesel Engine Fueled with Ethanol – Diesel Blend Fuel, *Fuel, 83* (2004), 14-15, pp. 2013-2020

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