Experimental Analysis of Partially Premixed Charge in a Diesel Engine with Jatropha Oil Methyl Ester and Diesel Blends

S. Sendilvelan and K. Bhaskar

ABSTRACT

Homogeneous charge compression ignition (HCCI) is an alternative combustion process which received considerable interest to meet the stringent emission norms. One such technology developed is partially premixed charge compression ignition (PCCI) combustion mode a variant of HCCI combustion mode. In this work an attempt has been made to experimentally analyze the performance and emission characteristics with Premixed Charge Compression Ignition (PCCI) mode assisted with pilot injection. Experiments were conducted in a modified single cylinder water-cooled diesel engine employing a conceptual system known as transient state fuel induction with different modes such as diesel and biodiesel conventional mode, diesel PCCI–diesel, biodiesel PCCI-diesel and biodiesel PCCI-biodiesel modes. In the present work, diesel and jatropha methyl ester was used premixed fuel along with conventional injection of either diesel or 20% jatropha oil methyl ester blend with diesel. The results indicate that brake thermal efficiency decreases marginally as the methyl ester in the conventional and PCCI modes and HC, CO emissions are higher and soot levels are lower in the case of biodiesel PCCI-biodiesel mode.

Keywords: Jetropha Methyl Ester, Diesel Engine, Emissions, Oxides of Nitrogen, Soot, Premixed Charge Compression Ignition.

INTRODUCTION

To minimize the magnitude of deviation from the ideal cycle and to comply with future emission norms the best features of spark ignition and compression ignition engine combustion can be coupled to obtain a lean burning hybrid combustion mode known as homogeneous charge compression ignition (HCCI) in which the combustion takes place spontaneously and homogenously. It is a form of combustion process which capitalizes upon the advantages of higher compression ratio, lean homogenous air fuel mixture, lower combustion temperature and instantaneous combustion to achieve higher thermal efficiency, higher fuel conversion efficiency, very low concentrations of pollution components, high efficiency during part and full load operating conditions and low specific fuel consumption and also shorter combustion duration [1].

In HCCI combustion well mixed fuel and oxidizer (typically air) are compressed to the point of auto-ignition controlled by means of a combustion initiator. As in other forms of combustion, this exothermic reaction releases chemical energy into a sensible form that can be translated by an engine into work and heat [2] [3]. Since obtaining very lean homogenous mixture is hard, it becomes difficult to sustain HCCI mode over the operating range of varying speeds and loads, to effectively control the HCCI combustion over the operating range a modified form of homogenous charge compression ignition mode combustion known as Premixed Charge Compression Ignition-Direct Injection (PCCI) can be used [4]. The major advantage of PCCI mode combustion over that of HCCI mode combustion is that, after achieving homogenous charge PCCI mode combustion employs a pilot injection as the combustion initiator there by establishing a effective control over the combustion in variable load and speed [5]. Although all the inherit characteristics of HCCI mode combustion cannot be obtained in this mode of combustion, it still performs better than the conventional mode of combustion.

PREMIXED CHARGE COMPRESSION IGNITION DIRECT INJECTION (PCCI-DI) MODE

The main purpose of the PCCI mode is to establish a bridge which effectively interfaces the in-cylinder condition with the fuel there by establishing a more effective means to achieve sustainable combustion in a controlled fashion. It is known that obtaining homogenous charge with uniform air fuel ratio is crucial to control the combustion; this system aims to obtain homogenous charge by induction the fuel in a transient state. Since air and vapor can form perfectly homogenous mixture, in this PCCI mode the fuel is injected in to the manifold at a temperature close to the flash point of the fuel being used in a stream of heated air being inducted the fuel instantly vaporizes and produces a homogenous charge [6] [7].

One of the important advantages that can be obtained by employing this PCCI mode is achieving sustainable combustion using lean air fuel mixture, and the advantages of having lean air fuel mixture are: (1) if the engine is operated with an equivalence ratio of 0.8 the specific fuel consumption reduces greatly, (2) the fuel conversion efficiency increases greatly as the leanness of the mixture increases, (3) as the mixture is lean the peak combustion temperature decreases which reduces the formation of NO_x pollutants, and (4) the uniformity and proper mixing of air fuel ensures complete combustion and more efficient operation of the engine [8].

In the present work, diesel and biodiesel was used as premixed fuels along with conventional injection of either diesel or jatropha oil methyl ester blends with diesel with premixed ratio of 25%. The oil obtained from the Jatropha seeds is trans-esterified to get biodiesel. In the transport sector it can be used blended with diesel fuel [9]. The biodiesel obtained has good ignition ability in engine due to its relatively high cetane number compared to that of conventional diesel fuel [10].

The oxygen component in the biodiesel fuel has the effect of reducing the pollutant concentration in exhaust gases due to better burning of the fuel in the engine [11]. But the viscosity of biodiesel is very high compared to diesel and this affects its atomization and penetration characteristics and hence it is blended with diesel while being used as an alternate fuel in diesel engines [12]. Premixed charge was prepared partially in the inlet manifold and the remaining quantity of fuel injected into the cylinder by conventional mode. Premixed charge was injected into the manifold using a solenoid operated injector controlled by electronic control unit (ECU). Experiments were conducted to study the performance, emission characteristics of various modes such as: diesel, biodiesel conventional mode, diesel PCCI-diesel, diesel PCCI-biodiesel and biodiesel PCCI-biodiesel modes.

EXPERIMENTAL SETUP

The engine used for testing purpose was a single cylinder DI engine (Agricultural type water cooled) which is fitted to a brake drum dynamometer as shown in Figure 1. The engine specifications are, it is a water-cooled, vertical, 4 stroke cycle, direct injection, naturally aspirated as given in Table 1. The fuel is heated by means of a 1000 watts water bath provided with an electrical thermostat which enabled us to maintain the fuel at the desired temperature. The fuel is injected in to manifold using an electronic fuel pump through a fuel injector mounted on the inlet manifold whose spray angle is 30°, the fuel line pressure is maintained at 6 bars. The current rating of the fuel injection pump used is 20 ampere and that of the fuel injector is 0.3 ampere.

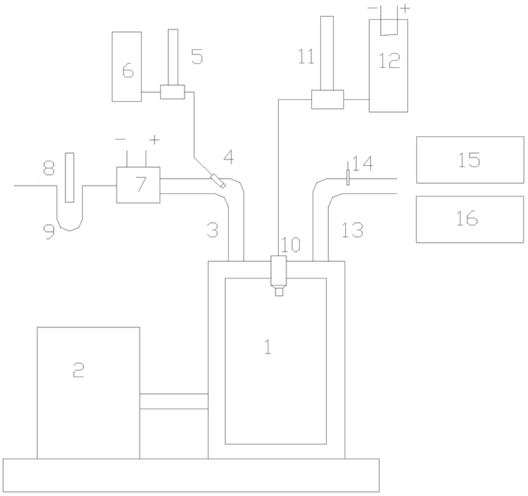
Table 1. Engine Specifications

Number of cylinder	One
Bore x Stroke	80 x 110 mm
Cubic Capacity	0.553 lit
Compression Ratio	16.5 : 1
Rated Output as per BS5514/ISO 3046/IS 10001	3.7 kW (5.0 hp) at 1500 rpm.
SFC at rated hp/1500 rpm	245 g/kWh(180 g/bhp-hr)
Engine Weight(dry) w/o flywheel	114 kg
Weight of flywheel	33kg - Standard

The injection is controlled by electronic circuit having a limit switch with frequency of about 750 cycles per minute. The limit switch is actuated by means of a bolt attached to the inlet valve rocker, effectively utilized the 8mm travel of the rocker arm for generating the electrical signal for initiating the injection during the suction stroke.

The pilot injector is used to supply secondary fuel works by means of the signal from the rocker ram. The opening and closing of the pilot injector for supplying the secondary fuel depends upon the position of the rocker arm, which depends upon the position of the piston inside the combustion chamber. The air is heated to a temperature of 80°C before it is sent into the combustion chamber. The primary fuel is supplied to the combustion chamber without heating. In the case of the secondary fuel supply, the fuel is heated before entering the inlet manifold. The heating is done by an electrical heating arrangement. The secondary fuel is supplied into the combustion chamber in the suction stroke. The exhaust gas is analyzed by means of a five-gas analyzer to measure the HC, CO, and NO_x present in the exhaust gas. The exhaust gas temperature and smoke contents are measured by means of the thermocouple and smoke meter respectively. The primary fuel, secondary fuel, and air quantity are measured before they enter the combustion chamber. The experiments have been conducted in the following modes:

1. Diesel conventional mode
2. Bio-diesel conventional mode
3. 25% D PCCI - D mode: 25% Diesel (manifold injection)—Diesel (main injection)



Legend

1. Engine	2. Electrical Dynamometer
3. Inlet Manifold	4. Pilot Injector
5. Pilot Injector Fuel Measurement	6.Pilot Fuel Tank (with heating and thermostat arrangement)
7. Air Tank	8. Orifice Meter
9.Manometer	10. Main Fuel Injectior
11. Main Fuel Measurement	12. Main Fuel Tank
13. Exhaust Manifold	14. Exhaust Gas Analyzer
15. Smoke Meter	

Figure 1. Experimental Setup

4. 25% D PCCI and 20% JOME mode: 25% Diesel (manifold injection) - 20% Jatropha Methyl Ester and 80% Diesel blend (main injection)
5. 25% JOME PCCI - JOME: 25% Bio-diesel (manifold injection) - 20% Jatropha Methyl Ester and 80% Diesel blend (main injection)

RESULTS AND DISCUSSION

Specific Energy Consumption (SEC) and Brake Thermal Efficiency (BTE)

Figures 2 and 3 shows the variation of specific energy consumption and brake thermal efficiency respectively with brake power for conventional diesel, biodiesel, 25% diesel PCCI-diesel mode, 25% diesel PCCI-biodiesel mode and 25% biodiesel PCCI-bio-diesel mode. While calculating SEC and brake thermal efficiency, premixed fuel energy and main fuel energy both were taken into account. Specific energy consumption and brake thermal efficiency are inversely proportional to each other due to the part of the premixed fuel injected into the intake port is accumulated in the premixed chamber in a liquid state at low temperature and does not enter into the combustion chamber [13].

SEC is the input energy required to develop unit power output and by comparing methyl esters with diesel, methyl esters are show-

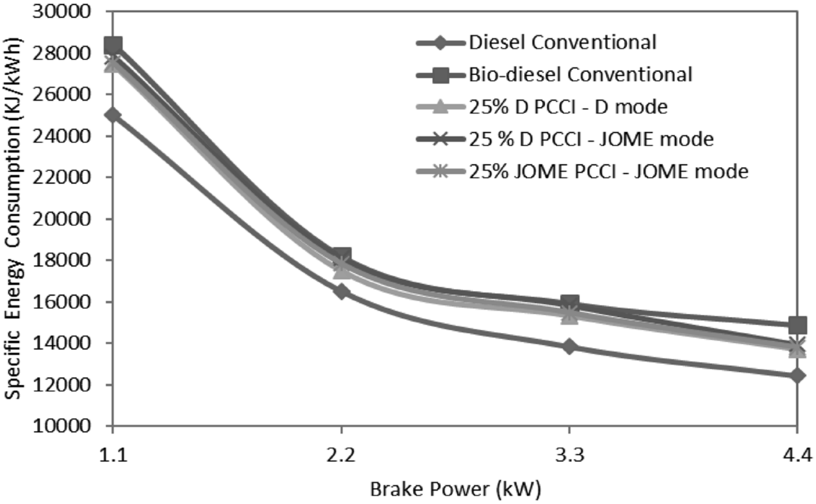


Figure 2. Variation of Specific Fuel Consumption with Brake Power

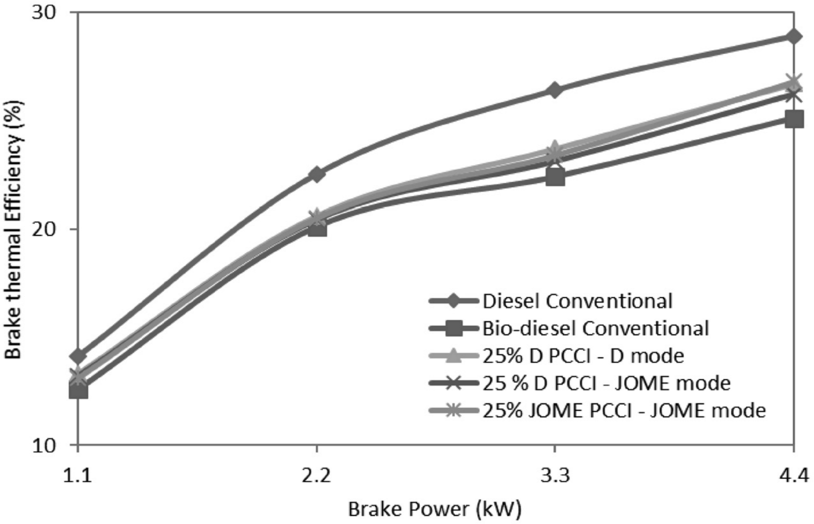


Figure 3. Variation of Brake Thermal Efficiency with Brake Power

ing comparatively higher values of SEC. SEC is higher and hence brake thermal efficiency is lower for the methyl ester and diesel PCCI mode at all power outputs compared to diesel. At full load the SEC of various modes tested varies from 12432 kJ/kWh to 14875 kJ/kWh. Higher Kinematic viscosities of biodiesel blends compared to that of diesel results in poor atomization, vaporization and dispersion of fuel in the combustion chamber. Lower calorific value of biodiesel result in higher fuel consumption compared to the biodiesel conventional and biodiesel and biodiesel modes of PCCI.

Exhaust Gas Temperature (EGT)

The exhaust gas leaving the cylinder determines the extent of temperature reached during the combustion process. With increase in load, the temperature of the exhaust gases increases for all the modes.

It is also observed from the Figure 4, that the exhaust gas temperature increases with methyl ester fuels for all the loads. This may be due to the oxygen content of the methyl esters, which improves combustion. Also the poor fuel atomization and vaporization due to higher viscosity of the methyl esters and their blends results in late burning of injected fuel and results in higher exhaust gas temperature [14]. At full load the exhaust gas temperature of various modes tested varies from 435°C to 470°C.

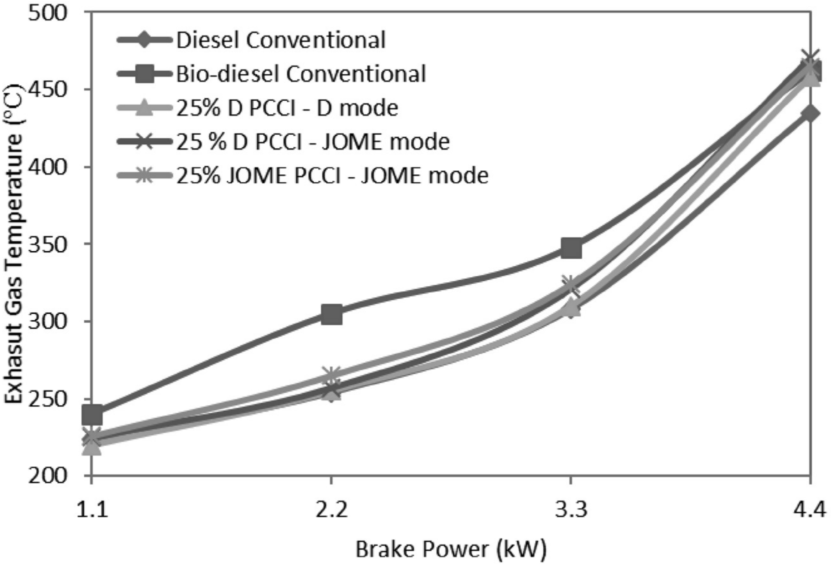


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Figure 4. Variation of Exhaust Gas Temperature with Brake Power

Oxides of Nitrogen Emissions (NO_x)

The variation of NO_x emission with brake power for various fuels tested is shown in Figure 5. Oxides of Nitrogen (NO_x) consist of Nitric oxide (NO) and Nitrogen dioxide (NO_2). Nitric oxide is predominant in the oxides of nitrogen produced inside the engine cylinder. NO_2/NO in a diesel engine is approximately in the ratio of 0.1 to 0.3. Oxides of Nitrogen emissions (NO_x) emissions are higher for the methyl ester compared to diesel mode, but are considerable lesser compared to biodiesel mode at rated power output. For biodiesel conventional mode, the NO_x emissions vary from 250 ppm at 20% of load to 501 ppm at full load. For diesel PCCI-diesel mode, diesel PCCI-biodiesel mode and biodiesel PCCI-biodiesel mode variations are from 12.54 g/kWh to 9.47 g/kWh, 13 to 9.04 g/kWh, 13.1 to 8 g/kWh respectively.

The operating range of engines using PCCI mode is observed to be too narrow and good performance is obtained only from 25% to 75% of rated power output.

Carbon Monoxide Emissions (CO)

Figure 6 shows the variation of carbon monoxide emissions with brake power. It can be observed that CO emission is higher at all power

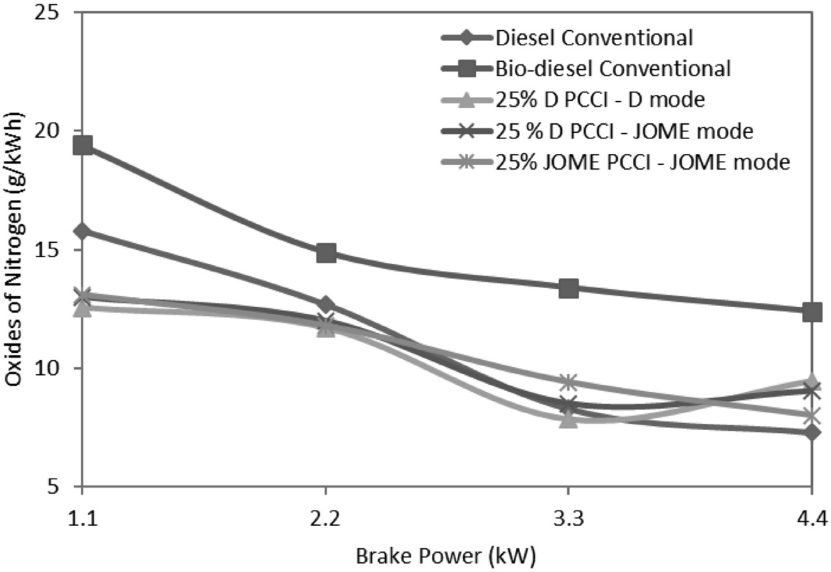

Figure 5. Variation of Oxides of Nitrogen with Brake Power

outputs for PCCI mode compared to base line diesel mode. In the case of PCCI modes the CO values of are higher compared to base line diesel mode. The fuel released from the crevices volume is not fully oxidized due to low in-cylinder temperature and gives rise to higher CO emissions. The OH radical concentration gets reduced due to incomplete combustion in intermediate temperature regions [15]. CO emission increases in PCCI combustion mode. Poor vaporization of premixed fuel injected in the manifold may also increase the CO emissions.

The CO emission for diesel PCCI-diesel mode, diesel PCCI-biodiesel mode and biodiesel PCCI-biodiesel mode varies from 32 g/kWh to 19 g/kWh, 25 to 18 g/kWh, 27 to 18 g/kWh respectively.

Hydrocarbon Emissions (HC)

Figure 7 shows that the variation of HC emissions with brake power. The thermal boundary layer formed on the cylinder surface of PCCI engines may also be another source of HBHC emissions. The air fuel mixture in the boundary layer does not undergo complete combustion due to flame quenching by the cooler cylinder wall temperatures. Absorption and desorption of fuel vapour into oil layers on the cylinder wall also gives rise to higher UBHC emission [16] [17]. The effect is more

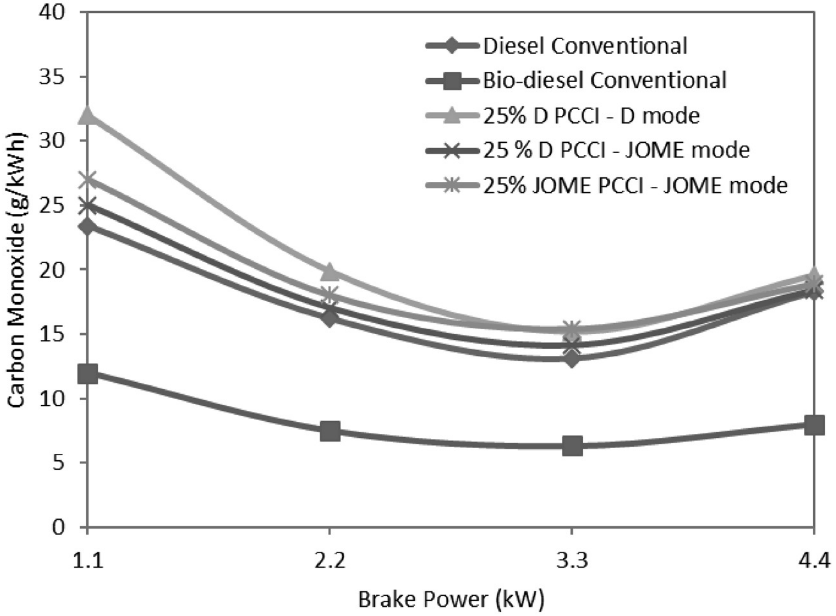

Figure 6. Variation of Carbon Monoxide with Brake Power

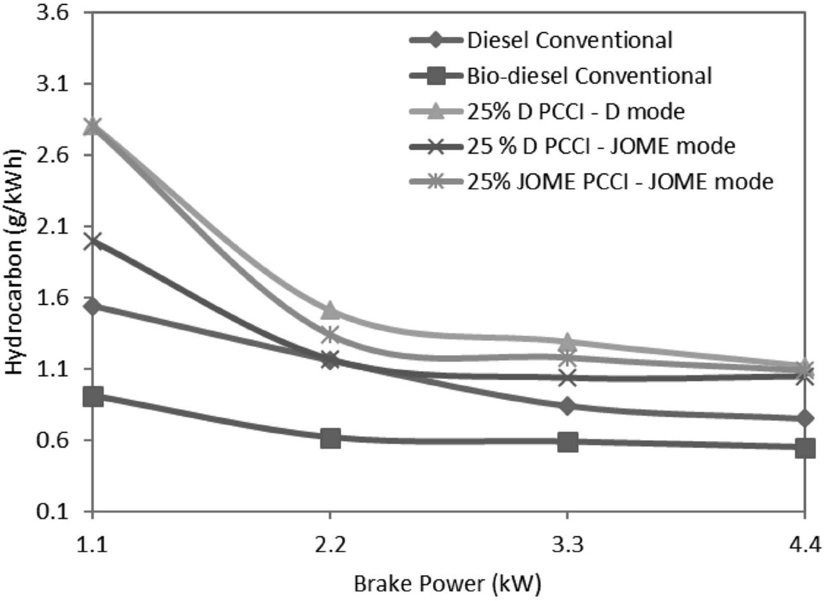

Figure 7. Variation of Hydrocarbon with Brake Power

pronounced at higher premixed ratios. The UBHC emission for diesel PCCI-diesel mode, diesel PCCI-biodiesel mode and biodiesel PCCI-biodiesel mode variations are from 2.81 g/kWh to 1.12 g/kWh, 2 to 1.05 g/kWh, 2.8 to 1.09 g/kWh respectively.

Soot Emissions

Figure 8 shows the variation of soot emission with brake power. It is observed that biodiesel-biodiesel PCCI-DI modes are showing lower value of soot compared to diesel conventional mode. At full load the Soot emissions of various modes tested varies from 75 mg/m^3 to 163 mg/m^3.

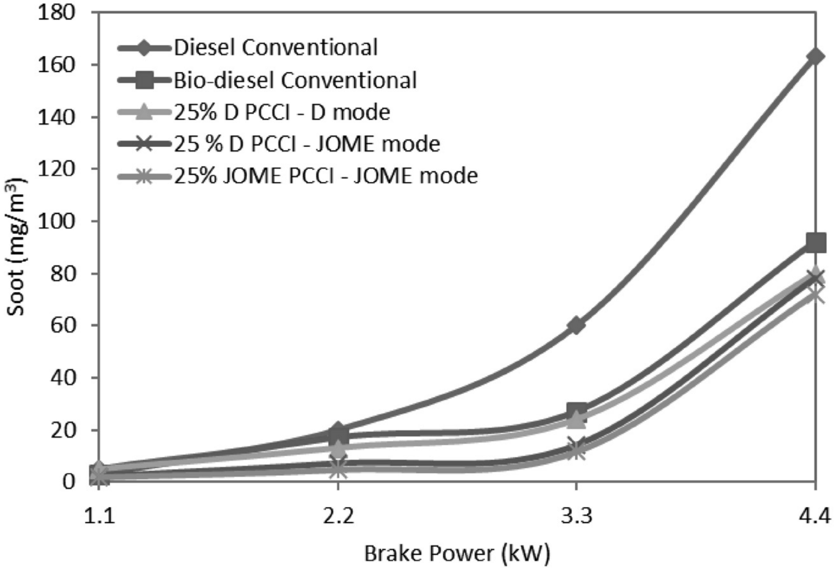

Figure 8. Variation of Soot with Brake Power

Diesel particulate matter consists of highly agglomerated solid carbonaceous material, ash, volatile organic and Sulphur compounds resulting from incomplete combustion of hydrocarbons present in fuel; it is also partly contributed by the burning of lubricating oil. Soot is formed in locally rich regions during combustion. Most of the soot formed is subsequently oxidized. A small fraction of the evaporated fuel and lubricating oil does not undergo oxidation and appear as soluble organic fractions (SOF) in the exhaust. The increase in soot emission for diesel fuel is also due to higher boiling point and thermal stability of aromatic hydrocarbons present in it. In particular, the presence of branched and ring (multi-ring or poly) structure of diesel fuel can increase the soot levels [18]. Higher boiling point paraffins or aromatics adsorbed from particulates emitted by the engine are reported to increase volatile organic fractions [19]. In the experimental test biodiesel PCCI -biodiesel mode shows better reduction in smoke values compared to other modes tested. In diesel PCCI-biodiesel mode soot varies from 2 to 78 mg/m^3 and in the biodiesel PCCI-biodiesel mode soot varies from 2 to 72 mg/m^3. Fuel rich regions inside the combustion chamber are eliminated in PCCI mode compared to conventional mode which results in lower soot emissions. Decrease in soot emission is also due to their intrinsic oxygen present in the methyl ester blends.

CONCLUSION

The engine performance and emission characteristics of diesel and biodiesel conventional, diesel PCCI–diesel, biodiesel PCCI-diesel and biodiesel PCCI-biodiesel modes were investigated. The results of this study are summarized as follows:

- Brake thermal efficiency decreases marginally as the methyl ester in the conventional and PCCI modes.
- HC, CO emissions of these PCCI modes are higher compared to the conventional mode.
- Exhaust gas temperature are observed to be higher for all modes compared to diesel convention mode.
- Oxides of nitrogen are observed to be higher at rated power output for all modes compared to diesel mode. Oxides of nitrogen are observed to be higher at all power output in the case of biodiesel conventional.
- The soot values are lower in the case of biodiesel PCCI-biodiesel and diesel PCCI-biodiesel modes.
- Specific energy consumption is observed to be higher in the case of biodiesel conventional mode, and it is minimum in the diesel conventional mode.

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