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Performance, Emission and Combustion Characteristics of a Variable Compression Ratio Diesel Engine Fueled with Karanj Biodiesel and Its Blends

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Abstract**—**The use of biodiesel in conventional diesel engines results in substantial reduction of unburned hydrocarbon, carbon monoxide and particulate matters. The performance, emission and combustion characteristics of a single cylinder four stroke variable compression ratio engine when fueled with Karanja (Pongamia) methyl ester and its 10-50 % blends with diesel (on a volume basis) are investigated and compared with standard diesel. The suitability of karanja methyl ester as a biofuel has been established in this study. The useful brake power obtained is similar to diesel fuel for all loads. Experiment has been conducted at a fixed engine speed of 1500 rpm, variable load and at compression ratios of 17.5:1 and 18.5:1. The impact of compression ratio on fuel consumption, combustion pressures and exhaust gas emissions has been investigated and presented. Optimum compression ratio which gives best performance has been identified. The results indicate longer ignition delay, maximum rate of pressure rise, lower heat release rate and higher mass fraction burnt at higher compression ratio for pongamia oil methyl ester when compared to that of diesel. The brake thermal efficiency for pongamia oil methyl ester blends and diesel has been calculated and the blend B20 is found to give maximum thermal efficiency. The blends when used as fuel results in reduction of carbon monoxide, hydrocarbon and increase in nitrogen oxides emissions. PME as an oxygenated fuel generated more complete combustion, which means increased torque and power. This is also supported with higher thermal efficiencies of the PME blends. NO_x is slightly increased due to the higher combustion temperature and the presence of fuel oxygen with the blend at full load. PME as a new Biodiesel and its blends can be used in diesel engines without any engine modification.

Keywords**—**Variable compression ratio CI engine, performance, combustion, emissions, biodiesel.

I. Introduction

THE progress of bio fuels can be traced back to early 19th century. In fact, the development of diesel engines and bio fuels has simultaneous history of technological advancements and economic struggle. Bio fuels offer enhanced employment opportunities and livelihood generation, leading to regional as well as national self sufficiency. The conservation of energy and reduction in atmospheric pollution had become a matter of serious concern in the last few decades. In order to improve air quality, especially in urban areas, increasingly stringent emission regulations are being imposed. Environmental concern and depletion in petroleum resources have forced researchers to concentrate finding renewable alternatives to conventional diesel fuels. A great deal of research and development on internal combustion engines has taken place not only in the design area but also in finding an appropriate fuel [1].

Many researchers have concluded that biodiesel holds promise as an alternative fuel for diesel engines, since its properties are very close to diesel fuel. The fuel properties of biodiesel such as cetane number, heat of combustion, gravity, and viscosity influence the combustion and so the engine performance and emission characteristics because it has different physical and chemical properties than petroleum-based diesel fuel [2]-[5].

The objective of current research work is to investigate the usage of Biodiesel and reduce the emissions of all regulate pollutants from diesel engines. A single cylinder, water-cooled constant speed direct injection variable compression ratio diesel engine was used for experiments. HCs, NO_x, CO, CO_2 of exhaust gas were measured to estimate the emission, various engine performance parameter such thermal efficiency, brake specific fuel consumption, peak pressure etc was calculated.

II. Karanja (Pongamia) Methyl Ester as a Diesel Fuel Substitute

Biodiesel, which is synthesized by transesterification of vegetable oils or animal fats sources, is a realistic alternative of diesel fuel because it is produced from renewable resources and involves lower emissions than petroleum diesel [6]. In addition, it is biodegradable and contributes a minimal amount of net greenhouse gases or sulfur to the atmosphere. The transesterification process combines the oil with an alcohol. The most common form of biodiesel is made with methanol and vegetable oils in the presence of a suitable catalyst. Additionally, the process yields glycerol. It is derived from crushing the karanja seed and by using large mechanical expellers. It is also important to note that most of the experiments conducted on biodiesel are mainly obtained from refined edible type oils only. The price of refined oils such as sunflower, soybean oil and palm oil are high as compared to that of diesel [7]. This increases the overall production cost of the biodiesel as well. Biodiesel production from refined oils would not be viable as well as economical for the developing countries like India. Hence, it is better to use the non-edible

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type of oils for biodiesel production. In India, non-edible type oil yielding trees such as linseed, castor, karanja, neem, rubber, jatropha and cashew are available in large number. The production and utilization of these oils are low at present, because of their limited end usage. Utilization of this oils/biodiesel as fuels in internal combustion Engines are not only reducing the petroleum usage, but also improve the rural economy. Efforts are made here to produce biodiesel from refined Karanj seed oil and to use it as the fuel in diesel engines [8]-[12].

III. Transesterification Process

PME was synthesized in reactor vessel using both NaOH & KOH as a catalyst. The ester preparation involved a two step transesterification reaction followed by washing and drying. The two step reaction utilized a 100% excess methanol, or a total molar ratio of methanol-to-oil of 6:1 with methanol equally divided in two steps. 1000gm was placed dry flask equipped with a magnetic stirrer and thermometer. In another flask, approximately 300gm of methanol was mixed with 7gm of NaOH until all of the catalyst dissolved. This mixture was quickly added to the oil and stirred vigorously for 1 hr maintaining temperature 55-60degree Celsius. After 24 hr, ester layer is set up on upper part and glycerol is set up on lower part. Then using separating funnel separates glycerol and ester is poured into another flask. Finally the ester was dried by silica gel [13]-[15].

IV. Properties of Pure Biodiesel

TABLE I
Properties of Pure Biodiesel

Sr. No.	Fuel & Blends	Density (kg/m^3)	Calorific value (KJ/Kg)	Viscosity at 40^{0}C (CSt)	Flash point (^{0}C)	Cloud point (^{0}C)	Pour point (^{0}C)
1	Pure Diesel	850	44000	2.87	56	-16.5	-20.1
2	PME 10	860	43850	3.05	65	4.8	2.5
3	PME 20	868	43756	3.39	73	5.3	3.1
4	PME 30	870	42360	4.21	81	5.8	3.8
5	PME 40	871	41810	4.63	88	6.0	4.4
6	PME 50	873	41608	5.12	99	6.2	4.8
7	PME 100	876	38532	5.57	170	10.4	6.8

V. Engine Details

TABLE II
Engine Details

Engine	KIRLOSKAR
General Details	Single cylinder, Four Stroke, VCR, Water cooled, CI Engine TV1
Bore X Stroke	87.5mm X 110mm
Compression Ratio	17.5:1 - 18.5:1
Capacity	661cc
Rated Output	5.2kW at 1500 rev/min
Injection pressure	200 kg/cm^2
Dynamometer	Rope brake with Mechanical loading

VI. Experimental Setup

The experiment was set up using the following equipment. The engine used for experimentation is a Kirloskar make computerised diesel engine used in agricultural application. The piezo sensor is having a range of 5000 PSI. The crank angle sensor is having a resolution 1 Deg, speed 5000 rpm with TDC marker pulse. Engine indicator is used for data scanning and interfacing, with speed indicator. Rotameter is used for water flow measurement. Digital thermocouple type temperature sensors are used as temperature indicator.

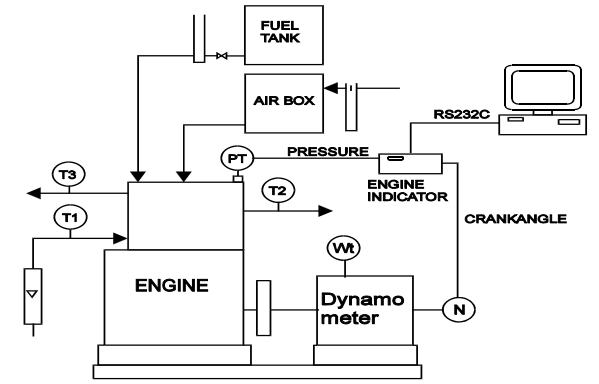




Fig. 1 Schematic of Experimental Setup and Engine arrangement

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VII. Methodology and Experimental Procedure

Tests have been conducted on a four stroke, direct injection; naturally aspirated single cylinder diesel engine is employed for the present study. The detail specification of engine is given in above Table I. To obtain the baseline parameters, the engine was first operated on diesel fuel. Performance and emission tests are carried out on the diesel engine using PME, and its various blends. The tests are conducted at the rated speed of 1500rpm at various loads and blends are prepared by volume basis i.e. PME10, PME20, PME30, PME40, PME50. Similar experiments were conducted over the same diesel engine. The experimental data generated are documented and presented here using appropriate graph.

In each experimental phase, engine parameter is related to thermal performance of the engine such as brake thermal efficiency; specific fuel consumption and exhaust gas temperature are measured. Mainly, at the given loading conditions, comparative analysis of the engine performance on the PME, and its blends with diesel and their emission were investigated. Load on the engine is steadily increased.

At each interval the readings are taken on the manual instrumentation or logged onto the computer analysis software; the variables gathered can then be used with the engine specifications to calculate characteristics which determine the performance of the fuel on the engine during operation.

VIII. Combustion Process and Combustion Rate in CI Engine

Combustion is the process of burning of the fuel in the presence of oxygen to produce heat. The formation of NO_x is dependent on the temperatures during the combustion, the amount of O_2 and N_2 in the charge, and the time available for them to react with each other in the combustion chamber. The combustion process in CI engines is mainly divided into three phases. The first phase of combustion is called as ignition delay, in which the tiny fuel droplets evaporates and mixes with high temperature (or high pressure) air. This period depends mainly on cetane number, and temperatures of fuel and air. The second phase of combustion is called as period of rapid combustion or premixed combustion. In this phase the air-fuel mixture undergoes rapid combustion, therefore the pressure rise is rapid and releases maximum heat flux. The third phase of combustion is called as period of controlled combustion. In this period, the fuel droplets injected during the second stage is burns faster with reduced ignition delay due to high temperature and pressure. In the third phase the pressure rise is controlled by the injection rate and the combustion is diffusive mode [16], [17].

The combustion rate has an effect on NOx production. More premixed combustion means a high initial rate of combustion which increases NOx. Premixed combustion corresponds to the fuel that is mixed with air and prepared to burn during the ignition delay period. When this fuel auto ignites it usually burns very quickly. Cetane number and fuel volatility are the two most important properties that determine the combustion rate. A biodiesel with a high cetane number is expected to shorten the ignition delay period and thus lower the amount of fuel that is involved with the premixed portion of the biodiesel combustion, thus lowering NOx emission [18], [19].

IX. Results and Discussions

A. Engine Performance and Emission Test Analysis

Engine performance characteristics are the major criterion that governs the suitability of a fuel. The following engine performance parameters are evaluated.

B. Brake Thermal Efficiency

The variation of brake thermal efficiency with respect to brake power for both fuels and its blends is as shown in Fig. 2. Brake thermal efficiency of PME and its blends is slightly higher as compared to that of diesel. It has been observed that the brake thermal efficiency of the blends is increasing with increase in applied load. It was happened due to reduction in heat loss and increase in power developed with increase in load. The maximum brake thermal efficiency at full load is 30.08% for PME20 at CR 18.5 which is 5-10% higher than that of diesel. By increasing the load of the engine, the brake thermal efficiency also increases for all the fuel type tested. The decrease in brake thermal for higher blends may be due to the combined effect of its lower heating value and increase in fuel consumption.

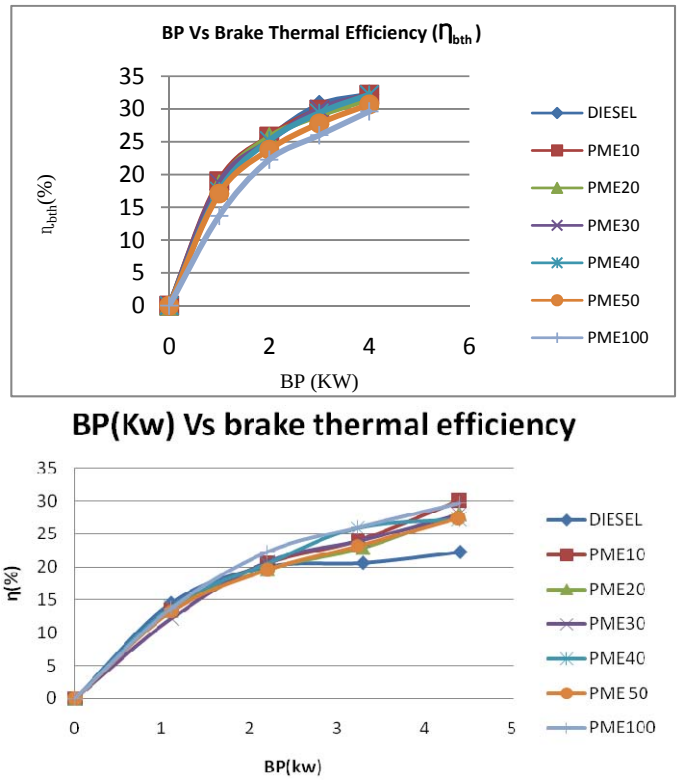


Fig. 2 Variation of brake thermal efficiency for PME, its Blends and diesel at CR 17.5 and 18.5

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C. Brake Specific Fuel Consumption

The variation of brake specific fuel consumption with respect to brake power for both fuels and its blends is as shown in Fig. 3. PME has lower calorific value than that of diesel. Hence the specific fuel consumption is slightly higher than that of diesel for PME and its blends. At higher percentage of blends, the SFC increases. This may be due to fuel density, viscosity and heating value of the fuels. B20 has higher energy content than other blends, but lower than diesel. Lesser values of SFC are apparently desirable.

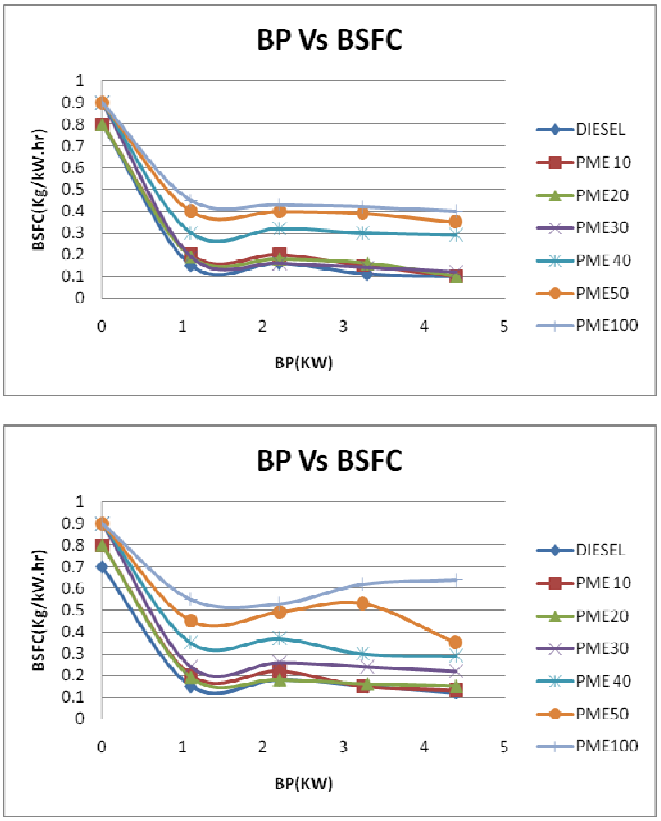


Fig. 3 Variation of brake specific fuel consumption for PME, its blends and diesel for CR 17.5 and 18.5

D. Indicated Power

The variation of indicated power with respect to brake power for both fuels and its blends is as shown in Fig. 4. The indicated power is slightly lower for PME blends than diesel. Thermal efficiency is defined as the ratio of the power output to the energy introduced through fuel injection; the later is the product of the injected mass flow rate and the lower heating value. Brake power decreases at higher compression ratio due to the conversion from the chemical energy to mechanical energy. Due to the lower heating value of the blends and unstable combustion the brake power decreases.

The biodiesel also contains some amount of oxygen molecule in the ester form. It is also taking part in the combustion. For PME20, this reveals that the effective combustion is taking place and there is saving with respect to exhaust gas energy loss. This fact is reflected in brake thermal efficiency and brake specific fuel consumption as well.

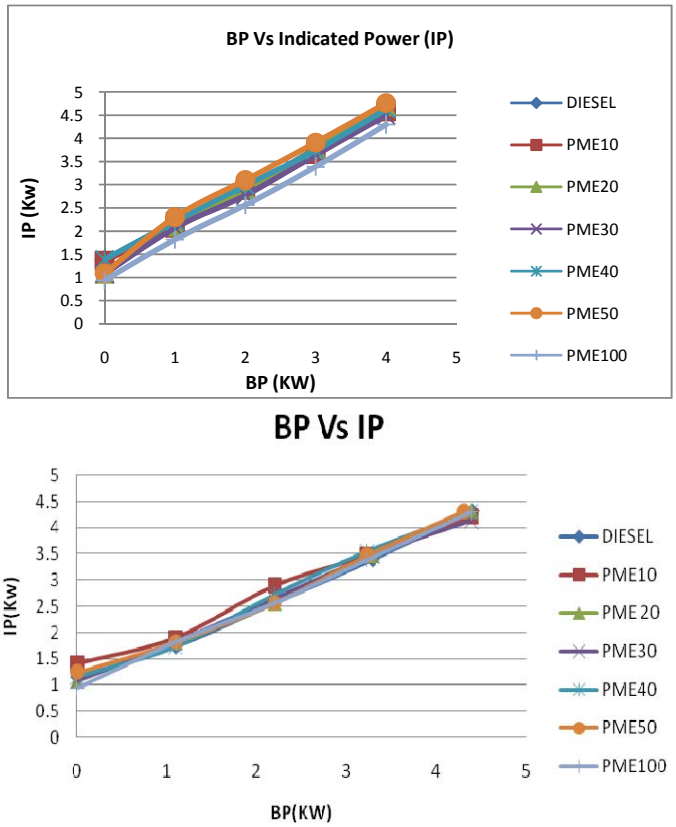


Fig. 4 Variation of indicated power for PME, its Blends and diesel for CR 17.5 and 18.5

E. Mechanical Efficiency

The variation of mechanical efficiency with respect to load for both fuels and its blends is as shown in Fig. 5. The mechanical efficiency for PME blends and diesel are close to each other. It has been observed that the mechanical efficiency of the blends is lesser in lower compression ratio and higher in higher compression ratio. The mechanical efficiency of the blend B20 increases with the increase in compression ratio, when compared to that of standard diesel. The maximum mechanical efficiency obtained from blend B20 for compression ratio 18.5 is 85.79%. Mechanical efficiency increases with increasing compression ratio for all the blends.

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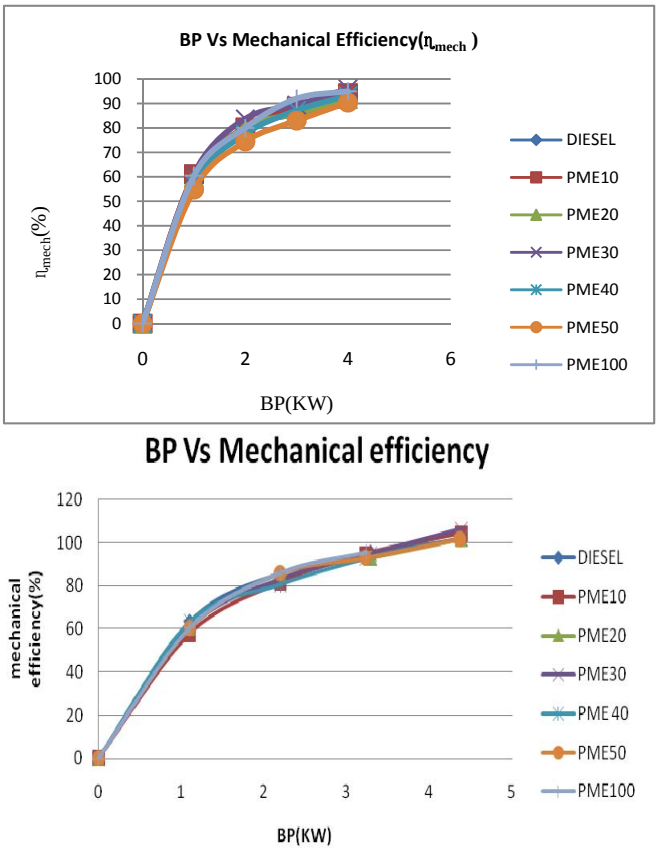


Fig. 5 Variation of mechanical efficiency for PME, its Blends and diesel for CR 17.5 and 18.5

F. Exhaust Gas Temperature

The variations of exhaust gas temperature for different compression ratio and for different blends are shown in Fig. 6. The result indicates that exhaust gas temperature decreases for different blends when compared to that of diesel. At lower compression ratio 17.5:1 the exhaust gas temperature of the blends are higher compared to that of standard diesel. As the compression ratio increases, the exhaust gas temperature of the various blends is lesser than that of diesel. The highest temperature obtained is 390^{0}C for standard diesel for a compression ratio of 18.5:1, whereas the temperature is only 320^{0}C for the blend B40.

G. Combustion Pressure

Variation of combustion pressure with crank angle for full loads and for different blends is shown in Fig. 6. It has been observed that PME20 gives higher combustion pressure compared to that of standard diesel due to longer ignition delay of PME. The peak pressure has been observed to be 52 bar and 49 bar for PME20 and standard diesel respectively at full load. This is happened due to rapid and complete combustion of fuels inside the combustion chamber.

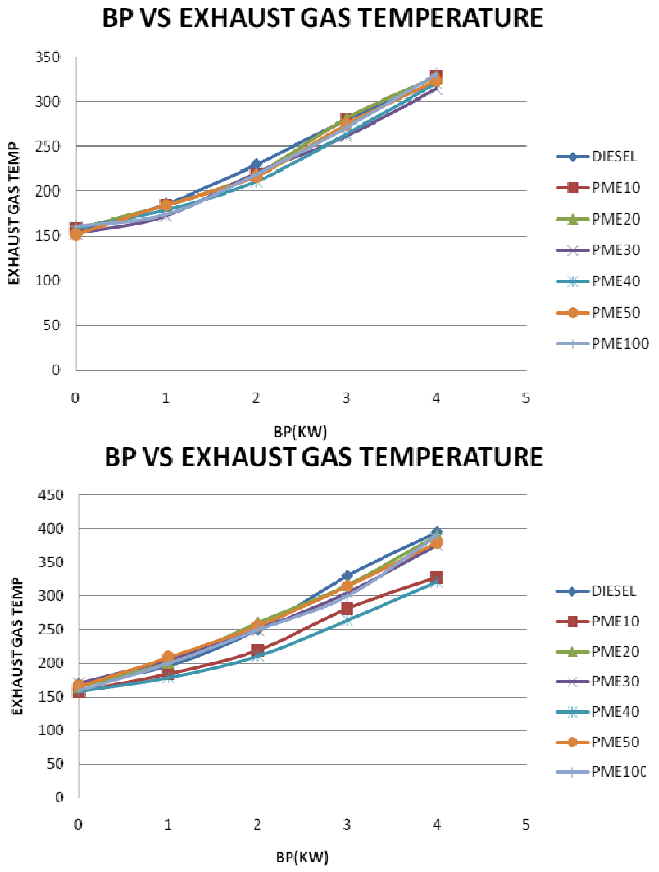


Fig. 6 Variation of exhaust gas temperature for PME, its Blends and diesel for CR 17.5 and 18.5

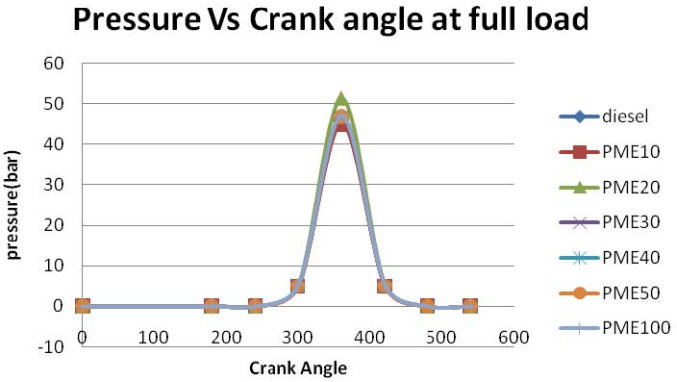


Fig. 7 Variation of combustion pressure for PME, its Blends and diesel

X. Engine Emission Parameters

With problem like global warming, ozone layer depletion and photochemical smog in addition to widespread air pollution, automotive emission are placed under the microscope and every possible method is attempted to reduce emission. Following Engine Emission parameters are evaluated for PME and its blends with diesel.

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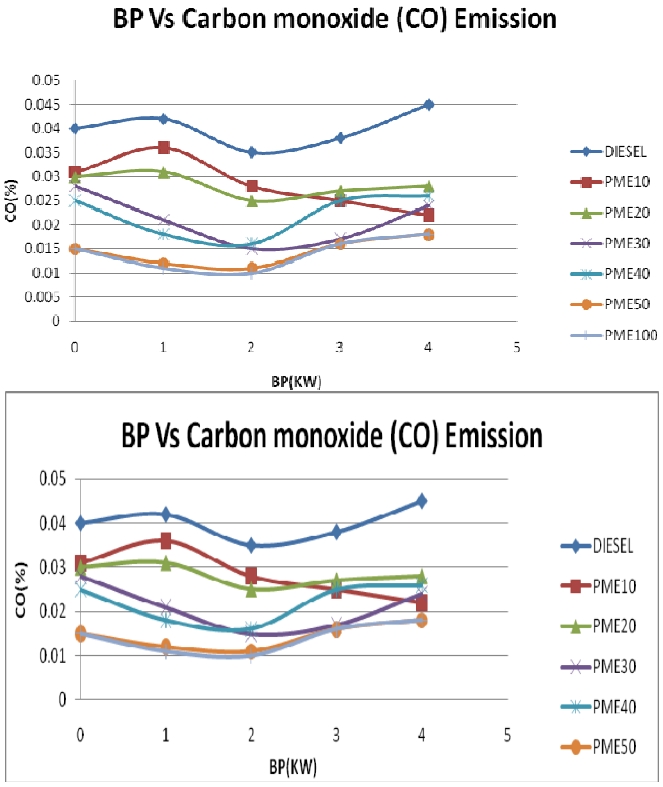


Fig. 8 Variation of carbon monoxide for PME, its Blends and diesel for CR 17.5 and 18.5

A. Carbon Monoxide

The variation of carbon monoxide with respect to brake power for both fuels and its blends is as shown in Fig. 8. Carbon monoxide emissions are almost same up to moderate load for PME blends and diesel. CO is predominantly formed due to the lack of oxygen. Since PME is an oxygenated fuel, it leads to better combustion of fuel resulting in the decrease in CO emission. The CO emission of the blend B20 is less than the standard diesel and it is found to be higher for compression ratio 18.5. The other blends B30, B40 and B50 have slightly lesser CO emission for compression ratio 18.5. The percentage of CO increases due to rising temperature in the combustion chamber, physical and chemical properties of the fuel, air–fuel ratio, shortage of oxygen at high speed, and lesser amount of time available for complete combustion.

B. Hydrocarbons

The variation of hydrocarbons with respect to load for both fuels and its blends is as shown in Fig. 9. HC emissions reduced drastically, but the higher HC emissions are observed for the blend at low load conditions. At low load conditions the quantity of fuel injected is lower resulting in a leaner mixture and lower gas temperature results in incomplete combustion leading to higher HC emissions. It shows that the hydrocarbon emission of various blends is higher at higher compression ratios. In this research, it shows that the increase in compression ratio increases the HC emission for Blend B40. The other blends produce lesser hydrocarbon emissions at higher compression ratio than the standard diesel. Due to the longer ignition delay, the accumulation of fuel in the combustion chamber may cause the higher hydro carbon emission.

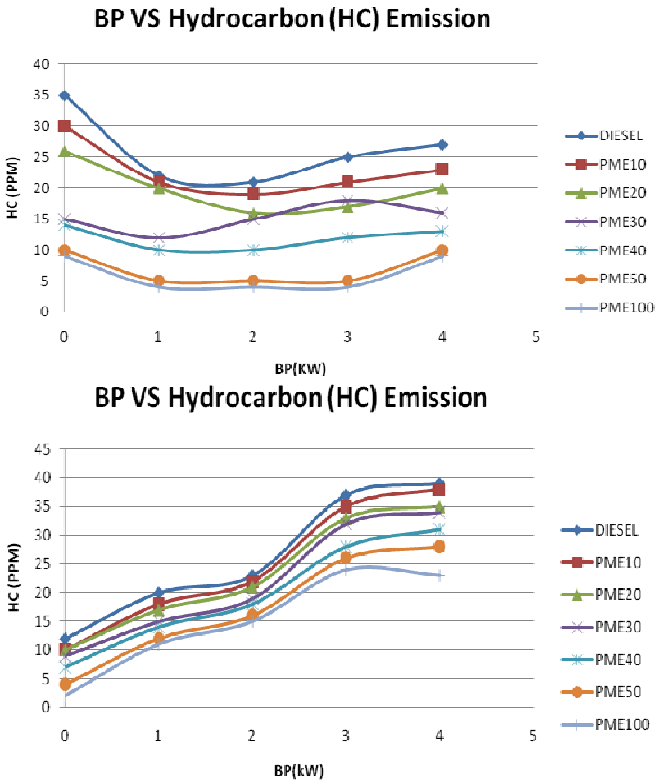


Fig. 9 Variation of Hydrocarbons for PME, its blends and diesel for CR 17.5 and 18.5

C. Nitrogen Oxide

The variation of nitrogen oxide with respect to load for both fuels and its blends is as shown in Fig. 10. NOx gradually increases with the increase in percentage of PME in the fuel. The NOx increase for PME may be associated with the oxygen content of PME, since the oxygen present in the fuel may provide additional oxygen for NOx formation. The formation of NOx emissions are governed mainly by the magnitude of peak cylinder temperature and the crank angle at which it occurs. As observed from the figure, the NOx emission for diesel and other blends increase with the increase of compression ratio. From the figure, it is obvious that for compression ratio 18.5, NOx emission from the PME blend B40 is higher than that of diesel.

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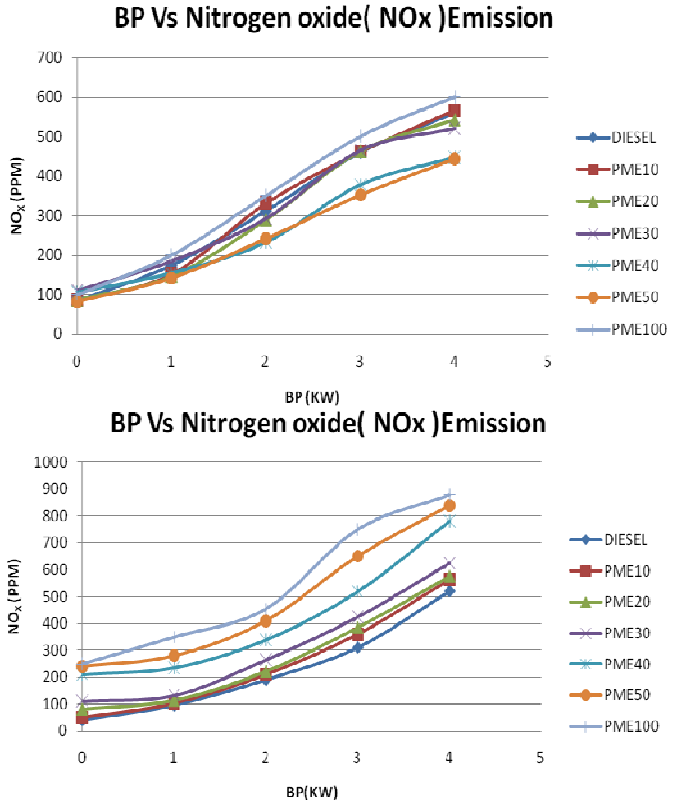


Fig. 10 Variation of nitrogen oxide for PME, its blends and diesel for CR 17.5 and 18.5

XI. Conclusion

The performance, emission and combustion characteristics of a variable compression ratio engine fueled with Pongamia biodiesel and diesel blends have been investigated and compared with that of standard diesel. The experimental results confirm that the BTE, SFC, BP, exhaust gas temperature and mechanical efficiency of variable compression ratio engine, are a function of bio diesel blend, load and compression ratio. For the similar operating conditions, engine performance reduced with increase in biodiesel percentage in the blend. However by increasing the compression ratio the engine performance varied and it becomes comparable with that of standard diesel. The performance, emission and combustion characteristics of a single cylinder direct injection CI engine fuelled with PME and its blends have been analyzed and compared to the base line diesel fuel. The following conclusions are drawn from this investigation

- The specific fuel consumption increases with increase in percentage of PME in the blends due to the lower calorific value of PME.
- Methyl ester of karanja oil results in a slightly increased thermal efficiency as compared to that of diesel at higher compression ratio.
- The exhaust gas temperature decreases at higher compression ratio. The reason is the lower calorific value of blended fuel as compared to that of standard diesel and lower temperature at the end of compression. The exhaust gas temperature for the blends is higher compared to that of standard diesel at lower compression ratios.
- The brake thermal efficiency of the blend B20 is slightly higher than that of standard diesel at higher compression ratios. The specific fuel consumption of blend B20 is lower than that of all other blends and diesel. This may be due to better combustion, and increase in the energy content of the blend.
- The tests on engine running with different fuels (biodiesel and diesel) have resulted in almost overlapped P-V diagrams. The engine running with biodiesel has produced slightly higher in-cylinder pressure and peak heat release rate than the engine running with normal diesel.
- The hydrocarbon emission of various blends is higher at higher compression ratios. The increase in compression ratio increases the HC emission for blend B40. CO emission is low at higher loads for methyl ester of karanja oil when compared with diesel. NOx emission is slightly increased with methyl ester of karanj oil compared to diesel.
- PME satisfies the important fuel properties as per ASTM specification of biodiesel and improves the performance, combustion and emission characteristics of engine significantly.

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