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Effect of Combustion Chamber Design on a DI Diesel Engine Fuelled with Jatropha Methyl Esters Blends with Diesel

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Abstract

In this paper a single cylinder constant speed air-cooled four-stroke direct injection diesel engine of 4.4 kW is selected for the experimental investigations to evaluate the performance and emission characteristics fuelled with Jatropha oil (JTME), and its blends (20%, 40%, 60%, 80% and 100%). The performance parameters are analyzed include Brake thermal efficiency whereas exhaust emissions include oxides of nitrogen, HC, Smoke and CO. The results of the experiment in each case were compared with baseline data of diesel fuel. It concluded that lower blend of biodiesel 20% JTME act as best alternative fuel among all tested fuel at full load condition.

The experimental investigations on the effect of 20% Jatropha methyl esters (JTME) with diesel on performance, combustion and emission characteristics of diesel engine with different combustion chamber geometries (Spherical, toroidal and Re-entrant).Brake thermal efficiency for toroidal combustion chamber was found higher than that of other two combustion chambers. Smoke density, carbon monoxide and hydrocarbons was observed slightly lower for toroidal combustion chamber compared to the other two but those are lower when compared with standard diesel (SCC). However, nitrogen oxides were slightly lower for toroidal combustion chamber compared to the other two but it is higher when compared with standard diesel (SCC).

Keywords: Combustion Chamber Design ; Performance ;Combustion ; Emission; Jatropha Methyl Esters; Diesel Engine .

Nomenclature

aTDC	After top dead center
BP	Brake power in kW
bTDC	Before top dead center
DI	Direct injection
HC	Hydrocarbon emissions in ppm
IC	Internal combustion engine
ID	Ignition delay in °CA
JTME	Jatropha oil methyl ester
NOx	Nitrogen oxide emissions in ppm
p_{max}	Peak pressure in bar
ppm	Parts Per Million
Q	Heat release rate in J/°CA
RCC	Re-entrant combustion chamber

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SCC	Spherical combustion chamber
TCC	Torroidal combustion chamber
Greek symbols	
$\eta_{b.th}$	Brake thermal efficiency
θ	Crank angle

1.0 Introduction

It is a plant growing almost throughout India. The oil content is approximately 40%. Jatropha curcas is a plant belonging to the family of Euphorbiaceae occurring almost throughout India. It is found in India, in a semi wild condition near villages. Jatropha plant can grow rapidly almost anywhere even on gravelly, sandy and saline soils. It has hardly any special requirement with regard to climate and soil. It can even grow in the crevices of rocks. Its water requirement is extremely low. It yields within 4 to 5 years and has a long productive period of around 50 years yielding handsome returns annually.

Jatropha oil has a high cetane number, very close to diesel. This makes it an ideal alternative fuel compared to other vegetable oils. The flash point of Jatropha oil is around 160ºC compared to 75ºC for diesel. Due to its higher flash point, Jatropha oil has certain advantages over petroleum crude, like greater safety during storage, handling and transport. However, the higher flash point may create problems in engine starting.

The viscosity of Jatropha oil is less compared to other vegetable oils but is higher than diesel [12]. The higher viscosity of Jatropha oil could pose problems related to flow of oil in the fuel supply pipes and nozzle. The fatty acid composition of Jatropha oil as as hown in table 1.

Table 1: Fatty acids composition of Jatropha oil

Sl.No.	Fatty acids	Composition (%)
1	Palmitic (C16:0)	12-17
2	Stearic (C18:0)	5-9.7
3	Oleic (C18:1)	37-63
4	Linoleic (C18:2)	19-41
5	Arachidic (C20:0)	10.3
6	Myrstic (C14:0)	0.5-1.4

2.0 Combustion Parameters

2.1 Variation of cylinder pressure

Figure.1 shows the variation of cylinder pressure with crank angle for diesel, blends of 20%, 40%, 60, 80% and 100% of JTME. There are three distinct regions:

Region I (From the start of combustion to 4^0 bTDC): The cylinder pressure is higher for biodiesel and its blends compared to diesel. In this region, the cylinder pressure increases with the increase in percentage of methyl ester in the blend. This is due to the lower ignition delay of biodiesel and its blends. The combustion starts earlier and the motion of the piston towards TDC also helps the rise in gas pressure.

Region II (4^0 bTDC to 10^0 aTDC): In this region the cylinder pressure is lower for all the blends of methyl esters compared to diesel. This is mainly because of the lower heat release of methyl esters and its blends due to their lower calorific values. Since the specific heat capacity of exhaust gas of methyl ester operated engines is high compared to diesel, it absorbs more heat energy thereby reducing the high temperature and pressure of the gas in the cylinder.

Region III (after 10^0 aTDC): The methyl ester and its blends show slightly higher pressure in cylinder due to the late combustion of higher fatty acid components in methyl ester. For instance, the rise in pressure at 20^0 aTDC is 2.0%, 3.2%, 5.0%, 6.7% and 7.7% for 20%JTME, 40%JTME, 60%JTME, 80%JTME and JTME compared to diesel.

It is observed that the crank angle at which peak pressure occurs shifts away from TDC slightly. For example, the peak pressure at rated power (4.4kW) occurs at 6^0CA aTDC for diesel and 20%JTME, 7^0CA aTDC for 40%JTME and 8^0CA aTDC for 60%JTME, 80%JTME and JTME.

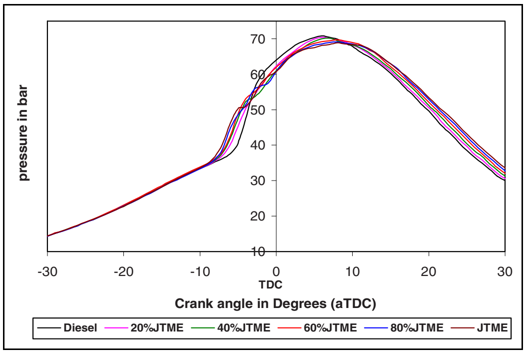

Fig. 1: Pressure – crank angle diagram for JTME and its blends

2.2 Heat Release Rate (Q)

The comparison of heat release rate variations for methyl esters and their blends with diesel at rated power is shown in Figure.2. It can be seen that the heat release rate curves of the diesel, methyl esters and their blends show similar patterns. The peak heat release rates of methyl esters and their blends are lower than that of diesel. The ignition delay for 20%, 40%, 60%, 80%, 100% JTME and diesel, at rated power are 14.65^0, 14.42^0, 14.2^0, 13.8^0, 13.6^0 and 15^0 respectively. It is seen that the delay period at rated power for the blends of JTME decreases with increase in their percentage in the blend. Therefore the peak heat release rate decreases and occurs earlier for methyl esters and their blends as the percentage of methyl ester in the blend increases compared to diesel. For instance from Figure 6.7, the peak heat release rates of diesel, 20% JTME, 40% JTME, 60%JTME, 80%JTME and JTME are 71.46 J/^{0}CA, 68.5 J/^{0}CA, 67.1 J/^{0}CA, 65.5 J/^{0}CA, 62.6 J/^{0}CA and 59.6 J/^{0}CA respectively and they occur at crank angles of 6^0, 6^0, 7^0, 8^0, 8^0 and 9^0 before TDC. It is also seen that all the methyl esters and their blends show higher heat release rates than diesel in last phase of combustion due to the late burning of higher fatty acid components of methyl esters.

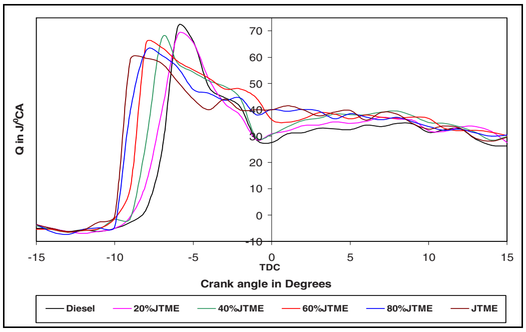

Fig.2: Comparison of rate of heat release for JTME/diesel blends at rated load

2.3 Ignition Delay (ID)

Ignition delay of fuel is one of the important parameters in determining the knocking characteristics of diesel engines. Ignition delay depends upon many factors such as compression ratio, the inlet pressure, injection parameters and the properties of the fuel. Higher the cetane number (CN), the shorter is the ignition delay, and vice versa. The ignition delay of various blends of methyl esters at different loads is compared with diesel and is shown in Figure.3. It can been seen that the ignition delay of methyl esters and its blends is significantly lower than that of diesel and decreases with increase in the percentage of methyl ester in the blend and neat esters record the lowest ignition delay when compared to their blends and diesel. For example, the decrease in delay at full load (4.4kW) is 1.9%, 3.5%, 5.1% 7.5% and 9.2% for 20%JTME, 40%JTME, 60%JTME, 80%JTME and JTME respectively compared to diesel. As the temperature of air in the cylinder is fairly high at the time of injection, esters undergo chemical reactions and polymerization, which result in injection characteristics that are different from those of diesel. In spite of the higher viscosities of esters, lighter compounds (volatile matter) are produced through cracking of higher fatty acids of esters. These lighter compounds in turn produce larger dispersion and shorter ignition delay. A decrease in ignition delay means a smaller amount of fuel accumulation prior to ignition which results in lower heat release rate.

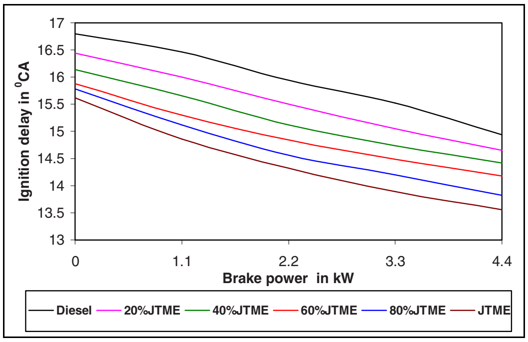

Fig. 3: Comparison of ignition delay for JTME/diesel blends

2.4 Peak pressure (p_{max})

The peak pressure is found to increase with the percentage of methyl ester in few cases and decrease is noted in majority of cases. It is observed that the peak pressure increases with increase in brake load for all the test fuels. This is due to the fact that more fuel is burned at higher loads. It is also seen that the methyl ester and its blends record lower peak pressures compared to diesel at any load. From the figure.4, as the percentage of the methyl ester in the blend increases, the peak pressure decreases. The decrease in peak pressure at rated power (4.4kW) is 0.5%, 0.8%, 1.5%, 2.3% and 2.6% for 20%JTME, 40%JTME, 60%JTME, 80%JTME and JTME respectively compared to diesel. This is due to the lower calorific value of the blends of methyl ester and poor atomization. Also the higher specific heat of the exhaust gas absorbs more heat thereby reducing the high temperatures and peak pressures in the

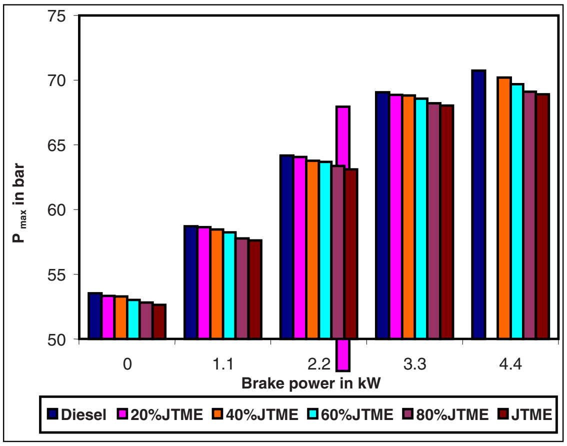

Fig.4: Comparison of peak pressure for JTME/diesel blends

3.0 Performance and Emission Characteristics

Engine performance characteristics are the major criteria that govern the suitability of a fuel. This study is concerned with the evaluation of brake thermal efficiency (BTE) of the methyl ester-diesel blends. With problems like ozone layer depletion and photochemical smog in addition to widespread air pollution, emissions from internal combustion engines are placed under the microscope and every possible method is attempted to reduce them. Hence in this study the emissions of hydrocarbon (HC), carbon monoxide (CO), smoke density and oxides of nitrogen (NOx) of methyl esters and their diesel blends are compared with diesel.

3.1 Brake Thermal Efficiency (BTE)

Brake thermal efficiency is slightly lower for methyl esters and their blends compared to diesel at all loads. For instance, at rated power (4.4kW) the brake thermal efficiency of diesel, 20%JTME, 40%JTME, 60%JTME, 80%JTME and JTME are 33.36%, 32.8% (0.56%), 31.6% (2%), 31.22% (2.14%), 30.87% (2.49%) and 29.37% (3.99%) respectively. The Figures in bracket show the decrease in brake thermal efficiency of blends of JTME compared to diesel. The maximum decrease for various blends of JTME at rated power is only 3.99%) compared to diesel. The engine is operated under constant injection advance and methyl esters and their blends have smaller ignition delay, combustion is initiated much before TDC is reached. This increases compression work and reduces the brake thermal efficiency of the engine. The maximum efficiency is obtained when most of the heat is released close to TDC. The start of heat release occurs much before TDC for methyl esters and their blends. This results in larger deviation from the ideal cycle and hence lower thermal efficiency.

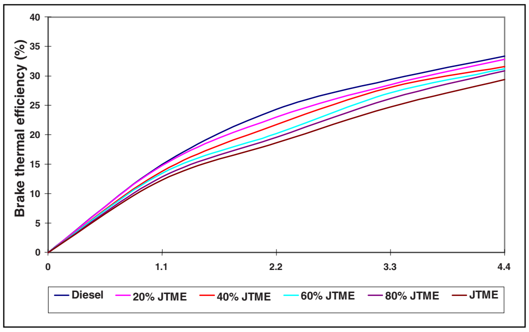

Fig. 5: Comparison of brake thermal efficiency for JTME/ diesel blends

3.2 Carbon Monoxide (CO) Emissions

In general the CO emissions are low for diesel engines as they are operated under lean mixtures. Due to intrinsic oxygen content in the ester, the oxygen availability for oxidation of CO is more in esters and their blends compared to diesel which results in reduced CO emissions. Thus CO emissions which are already low for diesel engines are further reduced by use of methyl ester and its blends. As percentage of methyl ester in the fuel increases the % of CO is continuously reduced. Similar trend is observed for blends of other methyl esters.

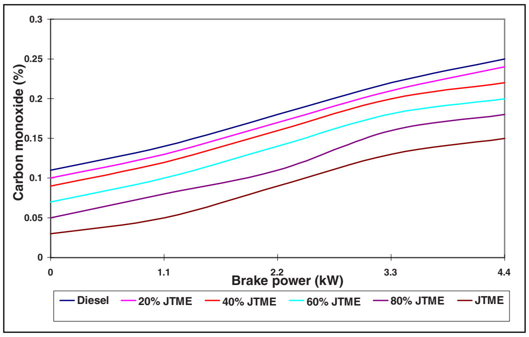

Fig.6: Comparison of CO for JTME/diesel blends

3.3 Hydrocarbon (HC) Emissions

The variation of HC emissions with brake power for various blends of methyl ester is shown in figure.7. It can be seen that there is an increase in HC emissions for all the test fuels as the load increases. This is perhaps due to the presence of fuel rich mixture at higher brake powers. There is significant reduction in HC emissions for the methyl esters and their blends at all loads compared to diesel. Adding methyl ester to diesel increases oxygen content resulting in better combustion, and this results in lower HC emissions. Increasing the percentage of the methyl ester in the fuel drastically reduces HC emissions.

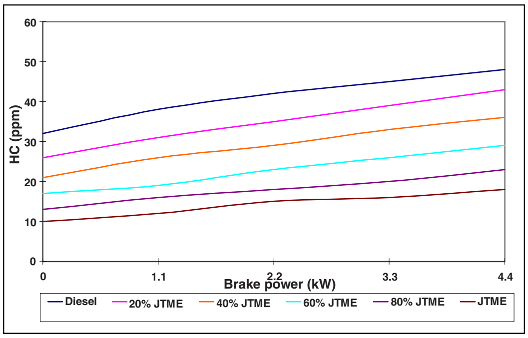

Fig. 7: Comparison of HC for JTME/diesel blends

3.4 Nitrogen Oxide (NO_x) Emissions

The variations of NOx emissions with brake power for methyl esters and their blends are compared with those of diesel in figure 8t. It is observed that the NO_x emissions increase with increase in power for all the test fuels. This is due to increase in the amount of fuel burned with load, which results in increase in combustion temperature. At any brake power increase in the emission of nitrogen oxides (NO_X) with increase in percentage of methyl ester in the fuel is observed. Methyl esters are oxygenated fuels more oxygen available for the formation of NOx compared to diesel. Hence the NO_X emissions increase with increase in percentage of methyl esters in the blend.

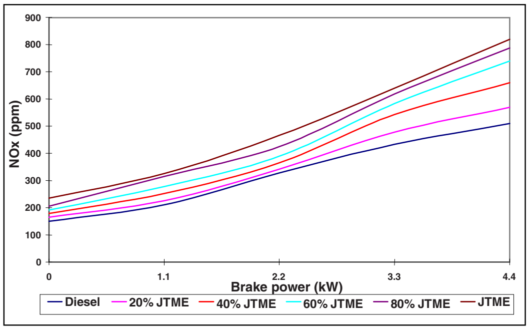

Fig.8: Comparison of nitrogen oxides for JTME/diesel blends

3.5 Smoke Density

The variation of Smoke density at various brake power is shown in figure 9 for the blends of different methyl esters. From this figure, it is clear that smoke density decreases with increase in methyl ester in the fuel blend. The particulate reducing effect of methyl esters can be attributed to its lower aromatic and short-chain paraffin Hydrocarbons and higher oxygen content. Similar trend is observed for other methyl esters.

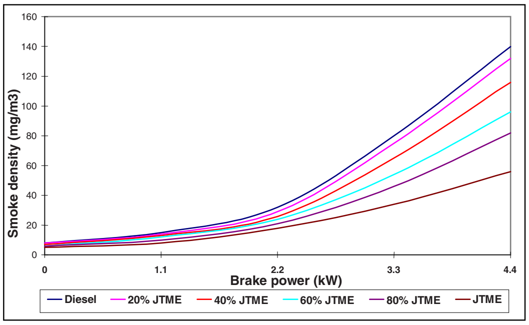

Fig.9: Comparison of Smoke density for JTME/diesel blends

4.0 Effect of Combustion Chamber Design

The experimental investigations on the effect of 20% Jatropha methyl esters (JTME) with diesel on performance, combustion and emission characteristics of diesel engine with different combustion chamber geometries (Spherical, toroidal and Re-entrant).

Fig.10(a) : Spherical combustion chamber chamber

Fig.10(b): Re-entrant combustion chamber

Fig.10(c): Torroidal combustion

The existing spherical combustion chamber was altered to re-entrant and torroidal shapes for better air motion to reduce spray penetration in order to achieve better atomization. These shapes were expected to reduce the formation of carbon deposits after long time use. Figures 11(a), 11(b), &11(c) show the Spherical, Re-entrant and Torroidal combustion chamber shapes respectively.

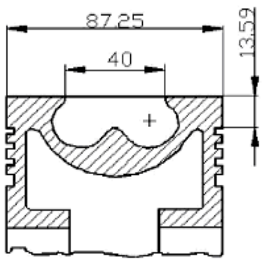

Fig. 11(a): Spherical Combustion Chamber Chamber

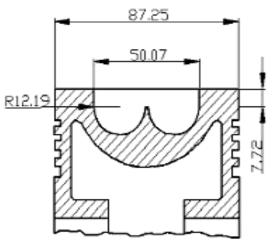

Fig.11(b): Re entrant Combustion Chamber

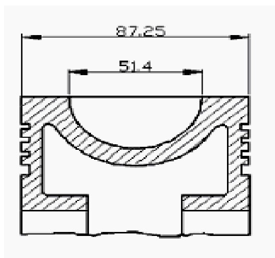

Fig.11(c): Torroidal Combustion

4.1 Combustion Parameters

4.1.1 Variation of Cylinder Pressure

Pressure variation of three types of open combustion chambers follow similar pattern of pressure rise as that of diesel at all loads as shown in figure 12. However when compared to diesel oil, pressure data of 20 % JTME are lower for all types of combustion chambers, may be due to variations of viscosity in fuel.

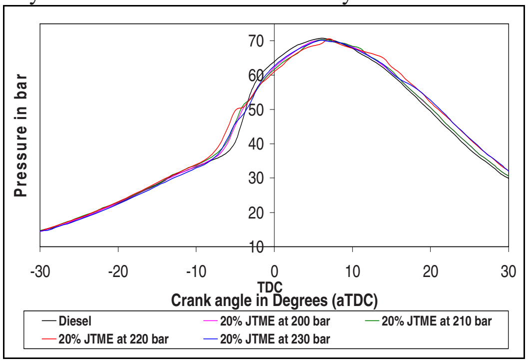

Fig.12: Pressure – crank angle diagram for 20% JTME at different Combustion chambers

4.1.2 Heat Release Rate (Q)

From figure 13 shows that the maximum hat release rate for 20% JTME is 40.21 J/^{0}CA with TCC when compared with the other two combustion chambers SCC (37.884 J/^{0}CA) and RCC (36.343 J/^{0}CA).But for diesel with SCC is 41.079 J/^{0}CA.

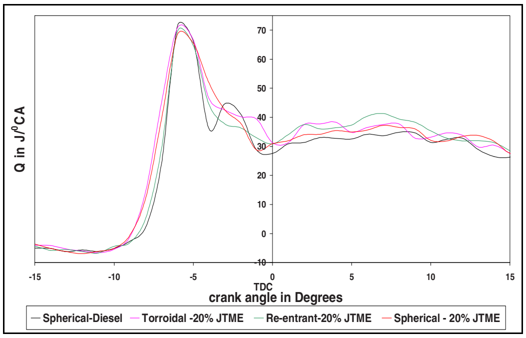

Fig.13: Comparison of rate of heat release for 20% JTME at different Combustion chambers

4.2 Performance and Emission Characteristics

4.2.1 Brake Thermal Efficiency

Figure 14 shows the variation of brake thermal efficiency for 20% JTME for different Combustion chambers. Brake thermal efficiency of 20% JTME is lower compared to that of diesel with standard engine SCC. Since the engine is operated under constant injection timing and combustion is initiated much before TDC is reached. This increases compression work and more hat loss and thus reduces brake thermal efficiency of engine. Brake thermal efficiency of TCC is higher when compared to SCC and RCC at all loads may be due to better mixture formation of 20 % JTME and air, as a result of better air motion in TCC, which leads to better combustion of biodiesel and thus increases brake thermal efficiency.

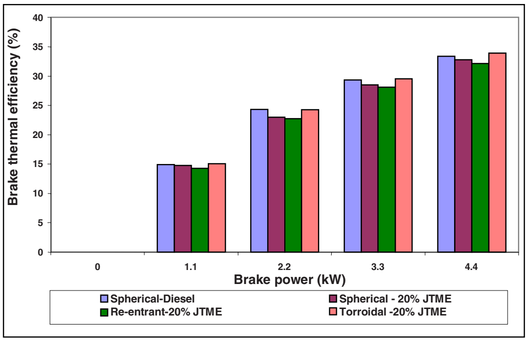

Fig.14: Variation of Brake thermal efficiency for 20% JTME at different Combustion chambers

4.2.2 Carbon Monoxide (CO) Emissions

Figure 15 Variation of CO emission for 20% JTME at different Combustion chambers. At full load, CO emissions of the 20% JTME decreased significantly when compared with those of standard diesel engine (SCC) .CO emissions slightly increases with TCC than SCC and RCC for 20% JTME. Lower air movement in TCC and presence of oxygen in 20% JTME, lead to poor combustion of fuel, resulting in increase in CO emissions.

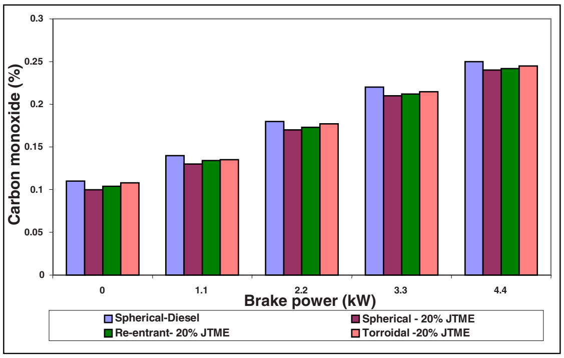

Fig.15: Variation of CO emission for 20% JTME at different Combustion chambers

4.2.3 Hydrocarbon (HC) Emissions

Figure 16 shows the variation of HC emission for 20% JTME at different Combustion chambers. At full load, HC emissions of the 20% JTME decreased significantly when compared with those of standard diesel engine (SCC) .HC emissions slightly increases with TCC than SCC and RCC for 20% JTME.

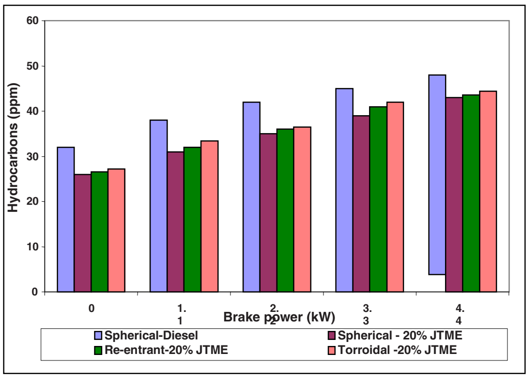

Fig.16: Variation of HC emission for 20% JTME at different Combustion chambers

4.2.4 Nitrogen Oxides (NO_x) Emissions

Figure 17 shows the variation of oxides of nitrogen for 20% JTME at different combustion chambers. NOx emissions are lower for TCC when compared to SCC and RCC for 20 % JTME, but it is higher when compared with standard diesel (SCC) may be due to higher combustion temperatures arising from improved combustion due to better mixture formation and availability of oxygen .

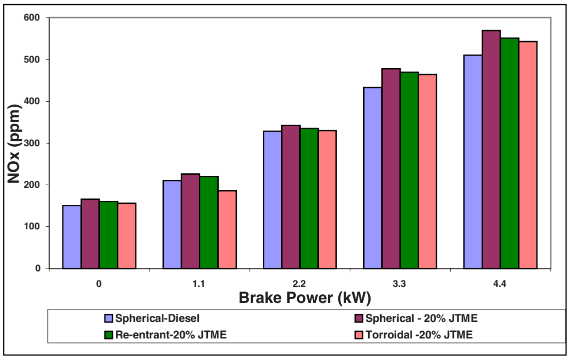

Fig.17: Variation of Oxides of nitrogen for 20% JTME at different Combustion chambers

4.2.5 Smoke Density

Figure 18 shows the variation of smoke density for 20% JTME at different combustion chambers. At all loads, smoke emissions for the 20% JTME decreased significantly than those of standard diesel engine (SCC), may be decrease due to presence of oxygen in biodiesel blend. Oxygenated 20% JTME fuel leads to an improvement in diffusive combustion. Smoke emissions were found lower for TCC than SCC and RCC, may be due to better fuel mixing and presence oxygen.

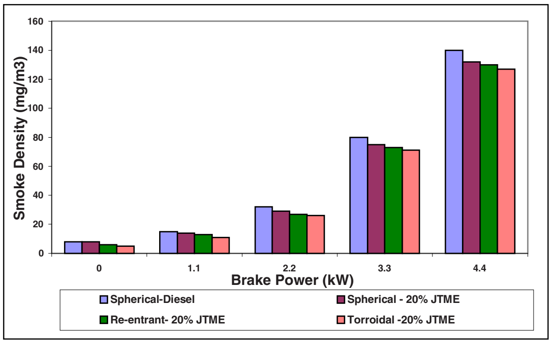

Fig. 18: Variation of smoke density for 20% JTME at different Combustion chambers

5.0 Conclusions

- Based on Brake thermal efficiency B20 and B40 blends are better than B100 but still inferior to diesel. 20% jatropha methyl ester (JTME) had better brake thermal efficiency (32.8%) value at maximum brake power than compared to diesel (33.36%)
- Properties of 20% of biodiesel are very close to the diesel compared to other % of blends.
- Smoke, HC, CO emissions at different loads were found to be higher for diesel, compared to other fuels.
- Brake thermal efficiency for toroidal combustion chamber (33.92 %) was found higher than that of other two combustion chambers. Smoke density, carbon monoxide and hydrocarbons was observed slightly lower for toroidal combustion chamber compared to the other two but those are lower when compared with standard diesel (SCC). However, nitrogen oxides were slightly lower for toroidal combustion chamber compared to the other two but it is higher when compared with standard diesel (SCC).

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Appendix – A

Specifications of Pressure Transducer

GH12D Miniature Pressure Transducer	
Standard Specifications	
Measuring Range	0...250 bar (3625 psi) 25 MPa
Lifetime	> 108 load changes
Overload	300 bar (4350 psi) 30 MPa
Sensitivity (nominal)	15 pC/bar (1.03 pC/psi) 150 pC/MPa
Linearity	< ±0.3 (0.2)* % FSO
Natural Frequency	115 kHz
Acceleration Sensitivity	< 0.001 bar/g
Shock Resistance	> 2000 g
Operating Temperature Range	up to 400°C (750°F)
Thermal Sensitivity Shift	20...400°C < ±2%, 200...300°C < ±0.5%
Insulation Resistance at 20°C (68°F)	> 10^{13} Ω
Capacitance	7 pF
Mass (without cable)	2.3 grams

Mounting Torque	1.5 Nm
Thermodynamic Specifications Cyclic Temperature Drift	< ±0.5 (0.3)* bar
Load Change Drift Max. Zero-line Gradient	dp/dt 1 mbar/ms
Permanent Zero-line Deviation	7 bar
IMEP-Stability	< 3 %

Appendix B

Specifications of the Engine

Make	:	Kirloskar
Type	:	TAF 1, vertical, air cooled, single cylinder, 4-stroke, compression ignition engine
Bore × Stroke (mm)	:	87.5 × 110
Compression ratio	:	17.5:1
Cubic capacity	:	0.661 lit
Rated power and speed	:	4.4 KW and 1500 rpm
Start of injection	:	23.4º BTDC
Connecting rod length	:	220 mm
Injector opening Pressure	:	200 – 205 bar
Valve timing		
IVO	:	4.5º BTDC
IVC	:	35.5º ABDC
EVO	:	35.5º BBDC
EVC	:	4.5º ATDC

Appendix – C

Specifications of the Charge Amplifier
Input type: unbalanced high insulation app 10 ^14 ohms for connecting piezo electric transducers
Insulation resistance: ≥ 10^14 ohms
Connection: BNC socket on front panel
Overload capability: input protected against electrostatic voltages and charges occurring during operation or handling
Output
Type: unbalanced, output ground separated from protective ground
Connection: 13a to 13c of 64- pin connector (channel 1)
23a to 23c of 64- pin connector (channel 2)
Voltage: 0 ± 10v at load ≥1.5kohm
Current: max± 6.7 mA
Quiescent potential: 0v 0r –8v, selectable with slide switch
Thermal drift: (DRCO, CAL, RANGE 50 pc /v); ≤0.1mv /c

Appendix – D

AVL 617 INDIMETER
ANALOG PART

- Maximum 4 to 8 different voltage input ± 10 V (input resistance 2 × 100 KΩ)
- Input low pass filter fg = 100 KHz
- Simultaneous sample & hold input
- ADC 12 bit ± 1LSB

DIGITAL PART

- Digital signal processor ADSp2101
- RAM 256 KB
- Measurement resolution 0.1, 0.2, 0.5, 1 deg. CA
- ISA or EISA 8MHz I/O timing
- Data transfer rate : 420 – 570 KHz (depend on no. of Channels)
- Power consumption : max 2.1 A
- Temperature range : 0… + 50º C