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INFLUENCE OF INJECTION TIMING AND COMPRESSION RATIO ON PERFORMANCE, EMISSION AND COMBUSTION CHARACTERISTICS OF JATROPHA METHYL ESTER OPERATED DI DIESEL ENGINE*

R. SENTHIL** AND R. SILAMBARASAN

Dept. of Mechanical Engineering, University College of Engineering Villupuram, Tamilnadu, India
Email: drrs1970@gmail.com

Abstract– Diesel fuel has a limited resource and concerns over environmental pollution are leading to the use of 'bio-origin fuels' as they are renewable and environmentally benign. Jatropha methyl ester, an esterified biofuel, has an excellent cetane number and a reasonable calorific value. It closely resembles the behaviour of diesel. However, being a fuel of different origin, the standard design limits of a diesel engine is not suitable for Jatropha methyl ester. Therefore, in this work, operational parameters are studied to find out the optimum performance of Jatropha methyl ester run diesel engine. The parameters varied are the compression ratio (CR) and injection timing (IT) along with load in a diesel engine. This work targets finding the effects of the engine operating parameters on the performance of the engine with regard to specific fuel consumption (SFC) and brake thermal efficiency (BTHE) with Jatropha methyl ester(J20) as fuel. Further exhaust emissions of the engine for the above conditions are also studied.Thus J20 can be effectively used in a diesel engine without any modification. At compression ratio of 19.5 along with injection timing of 30^{o}bTDC (before top dead centre) will give better performance and lower emission which is very close to diesel. Comparison of performance and emission was done for different values of compression ratio along with injection timing to find the best possible combination for operating engine with J20. It is found that the combined increase of compression ratio and injection timing increases the BTE and reduces SFC while having lower emissions. Diesel (20%) saved, will greatly meet the demand of fuel in railways.

Keywords– Jatropha methyl esters, Transesterification, injection timing, compression ratio, performance, emission, combustion

1. INTRODUCTION

The use of vegetable oils has reduced the levels of particulate matter, HC, and CO compared with the diesel combustion. Various vegetable oils both edible and non-edible can be considered as alternative sources for diesel engines. In most of the developed countries sunflower, peanut, palm, and several other feed stocks are used as alternative sources which are edible in the Indian context. Therefore in developing countries like India, it is desirable to produce biodiesel from non-edible oils which can be extensively grown in wastelands of the country. It has been reported that non-edible oils available in India are Jatropha, Pongamia, rubber seed, Rape seed, cotton seed, Nerium, etc. However the major disadvantage of vegetable oils is their high viscosity which leads to poor atomization and in turn poor combustion, ring sticking, injector coking, injector deposits, injector pump failure and lubricating oil dilution by crank case polymerization. Converting vegetable oils into simple esters is an effective way to overcome all the problems associated with the vegetable oils. For a diesel engine, fuel injection timing and compression ratio are the major parameters that affects combustion and exhaust emissions. Manufacturers and engine

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**Corresponding author

application engineers usually want to know the performance of a Compression ignition engine for various proportions of blends, for various compression ratios and at different injection timings.

As per the US Department of Energy [1], the world's oil supply will reach its maximum production and midpoint of depletion sometime around the year 2020. Future projections indicate that the only feasible option is the production of synthetic fuels derived from non-petroleum sources [2]. For substituting the petroleum fuels used in internal combustion engines, fuels of bio-origin provide a feasible solution to the twin crisis of 'fossil fuel depletion' and 'environmental degradation'. For diesel engines, a significant research effort has been directed towards using vegetable oils and their derivatives as fuels. Non-edible vegetable oils in their natural form called straight vegetable oils (SVO), methyl or ethyl esters known as treated vegetable oils, and esterified vegetable oils referred to as bio-diesel fall in the category of bio fuels. Bio-diesel is considered a promising alternative fuel to be used in diesel engines, boilers and other combustion equipment. They are bio-degradable, can be mixed with diesel in any ratio and are free from sulphur. Although bio-diesel has many advantages over diesel fuel, there are several problems that need to be addressed such as its lower calorific value, higher flash point, higher viscosity, poor cold flow properties, poor oxidative stability and sometimes its comparatively higher emission of nitrogen oxides [3]. Bio-diesel obtained from some feed stocks might produce slightly more oxides of nitrogen (1–6%), which is an ozone depressor, than that of fossil origin fuels but can be managed with the utilisation of blended fuel of bio-diesel and high speed diesel fuel [4]. It is found that the lower concentrations of bio-diesel blends improve the thermal efficiency. Reduction in emission and brake specific fuel consumption is also observed while using B10(10% biodiesel blended with diesel) [5]. The operating parameters must be optimised in light of the specific fuel properties. Effect of injection parameters [6–12]; spray [13], injection timing and compression ratio [14–18] have been studied in detail at many places. Most of the research studies concluded that in the existing design of engine and parameters at which engines are operating, a 20% blend of bio-diesel with diesel works well [5]. Many researchers indicated the need of research in the areas of engine modifications to suit higher blends without drop in performance so that the renewability advantages along with emission reduction can be harnessed to a greater extent. Effect of variations in these parameters has been studied taking one or more parameters at a time [19]. These studies were carried out in different types of engines (stationary/mobile; single cylinder/ multi cylinder; constant speed/variable speed) with bio-diesel prepared from different oil origin. To sum up the results of these studies, a cumulative study taking some or all the parameters at a time in one type of engine is still missing. To fill this gap, the study was done with the objective of finding the optimum engine operating parameters viz. compression ratio and injection timing, for better performance of diesel blended with bio-diesel (B20) obtained from Jatropha oil. The viscosities of B10 and B20 are closer to diesel. Moreover, only the oxidation stability of B10 and B20 meet the European specifications (EN 590) of 20h. It was observed that brake specific fuel consumption (BSFC) increases as the percentage of biodiesel increases. B20 showed a reduction in hydrocarbon (HC) and Carbon monoxide (CO) emission. Hence, B20 can be used in diesel engines without any major modifications.

2. TRANSESTERIFICATION OF VEGETABLE OILS

Transesterification is the process of using alcohol (e.g. methanol or ethanol) in the presence of catalyst such as sodium hydroxide (NaOH) or potassium hydroxide (KOH), which chemically breaks the molecule of the raw oil into methyl or ethyl esters with glycerol as a by-product. This method also reduces the molecular weight of the oil to 1/3 of its original value, reduces the viscosity, increase the volatility and increases cetane number to levels comparable to diesel fuel. Conversion does not greatly affect the gross heat of combustion. Transesterification is the change of the trivalent glycerine molecules against 3 molecules of monovalent alcohol methanol and each is a monoester. In the most vegetable oils, fatty acid with 16 and 18 carbon atoms predominate.

3. EXPERIMENTAL SETUP

The schematic diagram of the engine test rig used is shown in Fig. 1. The engine is fully equipped with measurements of all operating parameters. In the study, the variable compression ratio engine was run with JME (J20) at different compression ratios (16.5, 17.5, 18.5, 19.5 & 20.5) and injection timings (24, 27, 30 & 33 bTDC) to evaluate the characteristics of performance, emissions and combustion. The results were compared with the diesel fuel results as well as for different combinations of compression ratio and injection timing. The properties of Jatropha Methyl Ester are compared with diesel as shown in Table 1.

Fig. 1. Experimental setup

Table 1. Properties of diesel and JME

PROPERTY	Diesel	Jatropha Methyl
Kinematic viscosity in cst at 40^0 C	3.1	4.66
Calorific value in Kj/kg	43200	36694
Density at 15^0C in kg/mm^3	830	876
Cetane no.	46.4	48
Flash point (^{0}C)	56	85
Fire point (^{0}C)	64	92

a) Specifications of the apparatus:

In the test rig, several instruments/ equipment have been used for the experiment. Brief specifications of the instruments are given below.

i. Diesel engine:

Manufacturer : Kirloskar oil engines limited
Type of Engine : Vertical, 4-Stroke Single cylinder
Model : SV1
Rated Output
As per IS: 11170 : 8 HP (5.9kW)
Speed : 1800 rpm

Compression Ratio : 17.5:1
Bore and stroke : 87.5 x 110 (mm)
Injection pressure : 200 bar

ii. Smoke meter:

Smoke meter is used to determine the smoke density of the engine exhaust. The AVL 437 smoke meter has been designed for simple one man operation either from alongside a vehicle for either free acceleration or steady state test procedures. Control is made through a compact and rugged handset with a digital L.C.D. display. Any out of range parameters are automatically flagged to the operator. The brief specification of the smoke meter is given below:

Type : AVL 437 smoke meter
Make : AVL India Pvt. Ltd
Measuring range : 0 to 100 HSU

iii. Exhaust gas analyzer:

Manufacturer : SMS Autoline Equipment private limited
Type : Crypton 290 five gas analyser
Range : HC – 0 to 30000ppm
CO – 0 to 4000ppm
NO_X–0 to 5000ppm
CO_2– 0 to 20%

b) Uncertainty analysis

The errors and uncertainties in the experiments can arise normally from selection of the instruments, condition, calibration, environment, observation, reading and test planning. The uncertainty analysis is necessary to show the accuracy of the experiments. The various parameters like total fuel consumption, brake power; specific fuel consumption and brake thermal efficiency were calculated using the percentage uncertainties of various instruments [20].

Uncertainty analysis involves systematic procedures for calculating error estimate for experimental data. When estimating errors for engine an experiment is usually assumed that data is gathered under fixed conditions. Measurement errors arise from various sources, but they can be broadly classified as bias errors and precision errors. Bias errors remain constant during a set of measurements. They are often estimated from calibration procedures or past experience.

To quantify errors in experimental works, some calculations and estimations have to be applied on sensors, devices and machines that have been used to measure the experimental parameters. The experiment performed as needed to express measurement uncertainty as

$$x' = x \pm u_x \ \ (P\%)$$

where, x', x, u_x and P% are true value, tested value, uncertainty of the measurement and confidence respectively. Total uncertainty of each component or portion of the experiment is determined by finding error due to equipment (basis) and due to environment (precision). Wherever possible, uncertainty of each component or portion of experiment is found to determine where uncertainty must be minimised. Uncertainty is propagated in post-processing phase, to quantities that are non-linear functions of a measurement or functions of multiple measurements with uncertainties based upon the functional relationship.

Both bias and precision errors are present in an experiment. The precision is measured whereas the bias error is usually determined from equipment vendor specification. The total error is the vector sum of these errors and it is to be noted that errors in estimating each error affect the value of the total error.

$$u_x = (B_X^2 + P_X^2)^{1/2}$$

where, B_X and P_X are bias and precision error respectively.

In case of several measurements of the same quantity like engine load, the uncertainty is estimated using statistical measures of speed. Several measurements of the same quantity are: x_1,x_2,x_3,x_4---x_n. Average load of the dynamometer is calculated as

$$\text{Average} = (x_1, x2, x3, x4, ---, x_n)/n$$

Now, there are two ways to describe the scatter in these measurements. The mean deviation from the mean is the sum of the absolute values of the differences between each measurement and the average, divided by the number of measurements:

$$\text{Mean deviation from mean} = \frac{\sum_{i=1}^{n}(Xi - average)}{n}$$

The standard deviation from the mean is the square root of the sum of the squares of the differences between each measurement and the average, divided by one less than the number of measurements:

$$\text{Standard deviation from the mean} = \frac{\sqrt{\sum_{i=1}^{n}(Xi - average)2}}{1-n}$$

Either the mean deviation from the mean, or the standard deviation from the mean, give a reasonable description of the scatter of data around its mean value.

Tested load - mean deviation < true load < tested load + mean deviation

Tested load - mean deviation < true load < tested load + standard deviation

for parameters that have been evaluated depending on two or more independent parameters, propagation of uncertainty is carried out using

$$Uy/y = \sqrt{(\frac{u_{x1}}{x1})^2 + (u_{x2}/x2)^2 + .,,,,. + (u_{xn}/xn)^2}$$

where U_y and y are uncertainty and the testing value of the evaluated parameter x_1,x_2,x_3,x_4,---$x_n$respectively.

The uncertainty analysis carried out in this Appendix is based on the lines suggested by Kline and McClintock. It should be noted that the uncertainty analysis presented here considers only the errors that relate to the measurement made during testing. Δ is used here to symbolize the error in the quantity.

Uncertainty calculations in Thermal Performance Parameters

Total percentage uncertainty

= Square root of [(Uncertainty of TFC) ^{2} + (Uncertainty of brake power) ^{2} + (Uncertainty of specific fuel consumption) ^{2}+ (Uncertainty of brake thermal efficiency) ^{2} + (Uncertainty of HC)^2 + (Uncertainty of NOx)^2

Uncertainity in Brake Power

BP=2πNT/60*1000, T = W*R

Uncertainity in BMEP

The BMEP is calculated by using the formula

$$BMEP = \frac{(BP(kW)X\,60)}{LXAX\left(\frac{N}{n}\right)XNoofcylindersX\,100}$$

Uncertainity in BTHE

The BTHE is calculated by using the formula

$$BTHE = \frac{BPX\ 3600\ X\ 100}{fuelflowin\frac{kg}{hr}XCalorificValue}$$

Uncertainity in BSFC
The BSFC is calculated by using the formula

$$BSFC = \frac{fuelflowin\frac{kg}{hr}}{BP}$$

Uncertainty in Emission Constituents such as HC, CO, CO_2 and NO_x (Resolution / Range)

c) Testing procedure

Engine was started and warmed up at low idle, long enough to establish the recommended oil pressure, and was checked for any fuel or oil leakage. The engine was run on no-load condition and speed was adjusted to 1800 rpm by adjusting fuel injection pump. Engine was run to gain uniform speed, after which it was gradually loaded. Experiments were conducted at different torque levels (0, 8, 16, 24 and 32 Nm). The engine was run for 10 minutes and data were collected during the last 3 minutes. For 20% biodiesel, performance tests were carried out at five different compression ratios and four different injection timings.

The exhaust gas is passed through a four gas analyzer for measuring the emission of carbon monoxide, carbon dioxide, unburnt hydrocarbon and oxides of nitrogen present in exhaust gases. A smoke meter is used for the measurement of smoke capacity.

Experiments were carried out at steady state for different engine loads at constant speed of 1800rpm. The engine was allowed to run for a few minutes until the exhaust gas temperature, the cooling water temperature, the lubricating oil temperature and the emission have attained steady-state condition and data were recorded subsequently. All the gas concentrations were continuously measured for 10 min and the average results were presented. The experimental uncertainties are shown in Table 2. The steady-state test was repeated thrice. The results of the three tests were found to agree with each other within the experimental data that lie outside the probability of normal variations will incorrectly offset the mean value and inflate the random error estimates. As the error value is too low when compared with the actual values,the error values are not included in the actual result. The equipment is often calibrated and it was kept in error free condition.

Table 2. Experimental uncertainties

Parameters	Systematic Errors (±)
Speed	1 ± rpm
Load	± 0.1 N
Time	± 0.1 s
Brake power	± 0.15 kW
Temperature	± 1°
Pressure	± 1 bar
NOX	± 10 PPM
CO	± 0.03%
CO2	± 0.03%
HC	± 12 PPM
Smoke	± 1 HSU

4. RESULTS AND DISCUSSION

Test engine was run with Jatropha Methyl Ester (J20) and the timing for the consumption of 10cc fuel was calculated.

a) Effect of injection timing on Performance, emission and combustion characteristics:

i. *Specific fuel consumption (SFC)*

Figure 2 shows that the SFC increases by 4.5% for J20 blend at 24º bTDC, decreases by 16.25% for J20 blend at 30º bTDC and increases by 7.5% for J20 blend at 33º bTDC when compared to J20 blend at 27º bTDC. At 30ºbTDC of IT gives the lowest SFC compared to all other ITs. This may be due to higher viscosity and lower calorific value leading to better combustion.

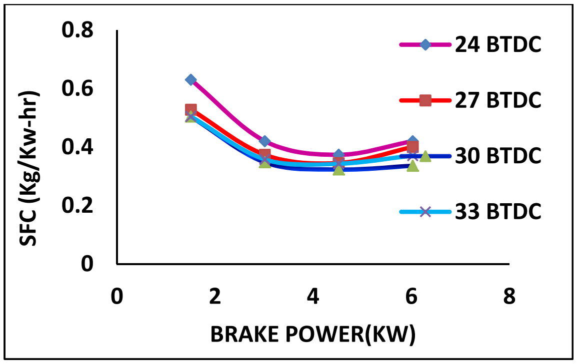

Fig. 2. Brake power Vs SFC for different blends

ii. *Brake thermal efficiency (BTE)*

Figure 3 shows that brake thermal efficiency decreases by 1.48% for J20 blend at 24º bTDC, increases by 2.04% for J20 blend at 30º bTDC and increases by 0.53% for J20 blend at 33º bTDC when compared to J20 blend at 27º bTDC. At 30ºbTDC, J20 gives highest BTE than all other ITs. This may be due to a combination of heating value and mass flow rate indicates inputs to the engine, which in case of J20, are more compared to neat diesel.

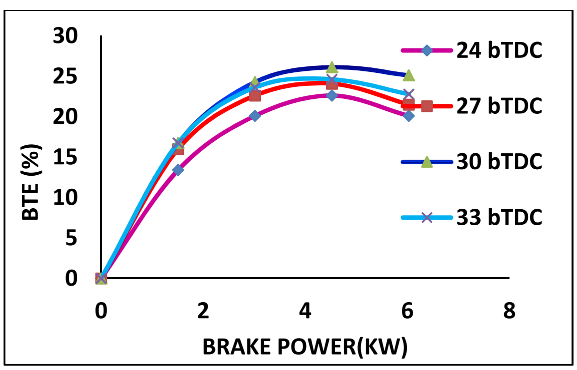

Fig. 3. Brake power Vs BTE for different blends

iii. *Carbon monoxide (CO)*

Figure 4 shows that CO emission increases by 0.72 % for J20 blend at 24º bTDC, decreases by 0.011% for J20 blend at 30º bTDC and 0.006% for J20 blend at 33º bTDC when compared to J20 blend at 27º bTDC. At 30ºbTDC gives lowest CO emission than all other ITs. This may be due to oxygen concentration and cetane number. Since JME fuel contains oxygen in fuel and it acts as a lesser combustion promoter inside the cylinder.

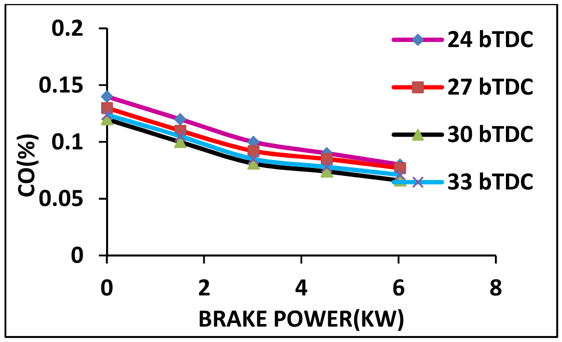

Fig. 4. Brake power Vs CO for different blends

iv. Carbon di-oxide (CO_2)

Figure 5 shows that the CO_2 emission decreases by 0.06% for N20 blend at 24º bTDC, increases by 0.14% for N20 blend at 30º bTDC and 0.03% for N20 blend at 33º bTDC when compared to N20 blend at 27º bTDC.At 30º bTDC gives the highest CO2 emission than all other ITs. This may be due to better combustion taking place.

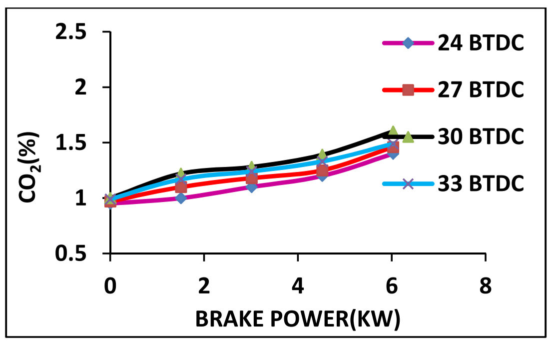

Fig. 5. Brake power Vs CO_2 for different blends

v. Oxides of nitrogen (NO_x)

Figure 6 shows that NOx emission decreases by 8.1% for J20 blend at 24º bTDC, increases by 14%. for J20 blend at 30º bTDC and 8.5% for J20 blend at 33º bTDC when compared to J20 blend at 27º bTDC At 30ºbTDC gives the highest NO_x emission than all other ITs. This may be due to the presence of oxygen in biodiesel, which leads to complete combustion of biodiesel than diesel. As a result, maximum temperature inside cylinder is more in case of biodiesel than diesel.

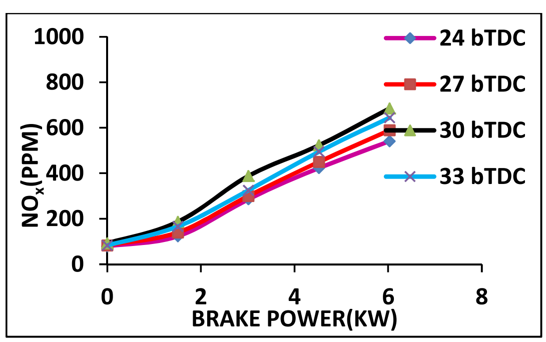

Fig. 6. Brake power VsNO_x for different blends

vi. Hydrocarbon (HC)

Figure 7 shows that HC emission increases by 9.3% for J20 blend at 24º bTDC, decreases by 5.8% for J20 blend at 30º bTDC and 1.4% for J20 blend at 33º bTDC when compared to J20 blend at 27º bTDC. This may be due to viscosity and surface tension that affects penetration rate, droplet size of fuel, which in turn affects mixing of fuel and air. Cetane number of fuel also plays a vital role in ignition process.

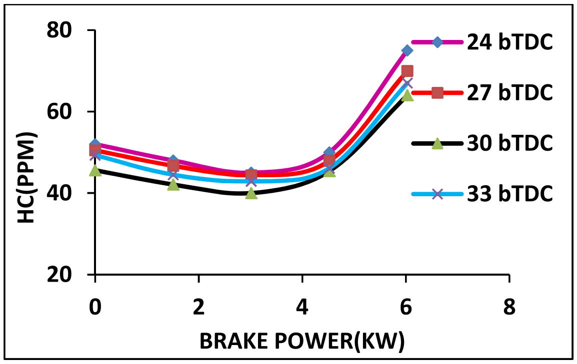

Fig. 7. Brake power Vs HC for different blends

vii. Cylinder pressure

From Fig. 8 we see that the peak pressure decreases by 4.5% for J20 blend at 24º bTDC, increases by 5.7% for J20 blend at 30º bTDC and increases by 1.4% for J20 blend at 33º bTDC when compared to J20 blend at 27º bTDC .

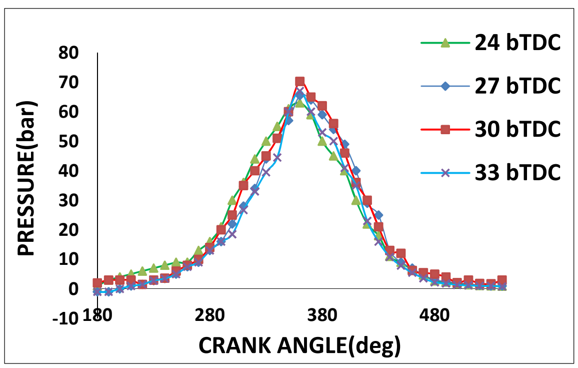

Fig. 8. Crank angle Vs Pressure

viii. Heat release rate

Figure 9 shows that the maximum HRR increases by 6.8% for J20 blend at 24º bTDC, decreases by 2.03% for J20 blend at 30º bTDC and decreases by 0.66% for J20 blend at 33º bTDC when compared to J20 blend at 27º bTDC .

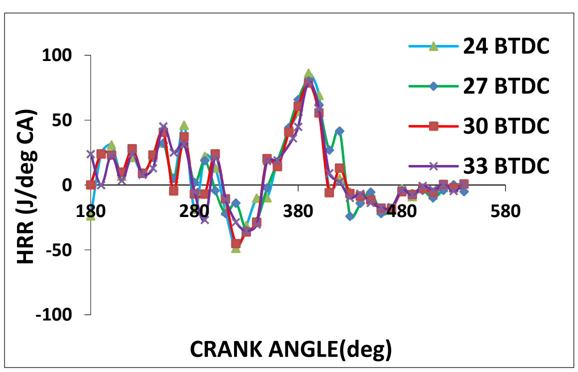

Fig. 9. Crank angle Vs HRR

ix. Cumulative heat release rate

Figure 10 shows that the maximum Cumulative HRR decreases by 13.8% for J20 blend at 24º bTDC, decreases by 20.23% for J20 blend at 30º bTDC and decreases by 19.04% for J20 blend at 33º bTDC when compared to J20 blend at 27º bTDC .

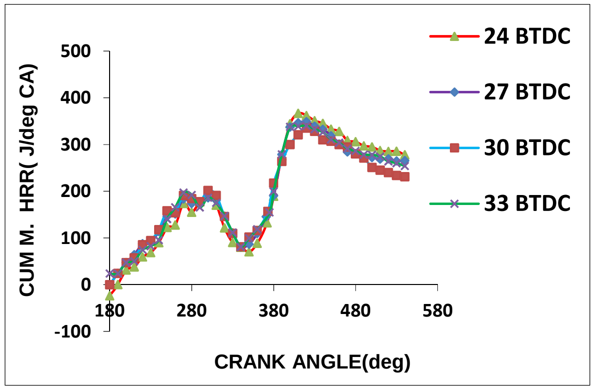

Fig. 10. Crank angle Vs Cumulative HRR

b) Effect of compression ratio on Performance, emission and combustion characteristics:

i. Specific fuel consumption (SFC)

Figure 11 shows that the SFC increases by 9.7% for J20 blend at 16.5:1, decreases by 2.75% for J20 blend at 18.5:1 , 8.25% for J20 blend at 19.5:1 and 5.5% for J20 blend at 20.5:1 when compared to J20 blend at 17.5:1. This is due to the fact that increase in compression ratio and reduction in BSFC due to reduction in dilution of charge by residual gases.This results in better BTE and lower BSFC. However, increase in BSFC is observed with lower compression ratio due to slow combustion pressure because of more charge diameter and lower compression pressure and temperature.

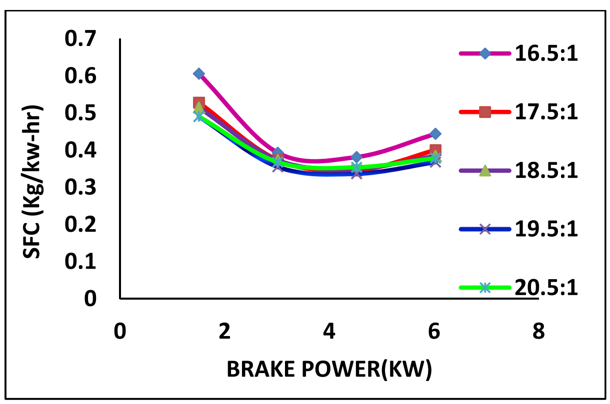

Fig. 11. Brake power Vs SFC for different blends

ii. Brake thermal efficiency (BTE)

Figure 12 shows that the brake thermal efficiency decreases by 1.66% for J20 blend at 16.5:1, increases by 0.33% for J20 blend at 18.5:1 , 1.33% for J20 blend at 19.5:1 and 0.08% for J20 blend at 20.5:1 when compared to J20 blend at 17.5:1. This is due to increase in compression ratio, injection of fuel in higher temperature and pressure compressed air, better air-fuel mixing and faster evaporation leads to complete combustion. Further, reduction in compression ratio resulted in lower BTE due to lower compression pressure and temperature, slow combustion process, and more dilution by residual gas.

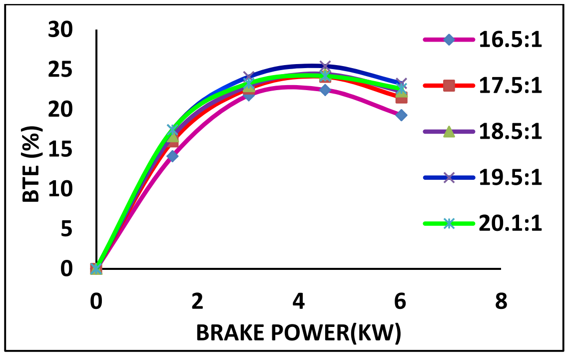

Fig. 12. Brake power Vs BTE for different blends

iii. Carbon monoxide (CO)

Figure 13 shows that the CO emission increases by 0.003% for J20 blend at 16.5:1, decreases by 0.002% for J20 blend at 18.5:1, 0.008% for J20 blend at 19.5:1 and decreases by 0.004% for J20 blend at 20.5:1 when compared to J20 blend at 17.5:1. This may due to better combustion, less dilution of charge by residual gases accelerating the carbon oxidation to form carbondioxide. At lower compression ratio, the carbon monoxide emissions are increased due to more dilution of fresh air with residual gases, lower compression temperature and poor mixing of fuel and air.

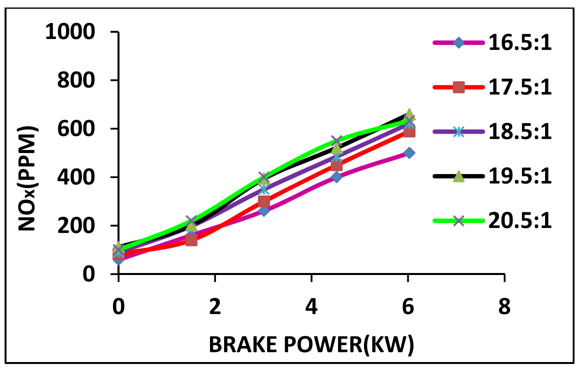

Fig. 13. Brake power Vs NO_x for different blends

iv. Carbon di-oxide (CO_2)

Figure 14 shows that the CO_2 emission decreases by 0.11% for J20 blend at 16.5:1, increases by 0.06% for J20 blend at 18.5:1, 0% for J20 blend at 19.5:1 and decreases by 0.03% for J20 blend at 20.5:1 when compared to J20 blend at 17.5:1. The CO_2 emissions are increased with increase in CR due to better combustion. Whereas at lower CR, carbon dioxide emissions are lower due to slower and incomplete combustion.

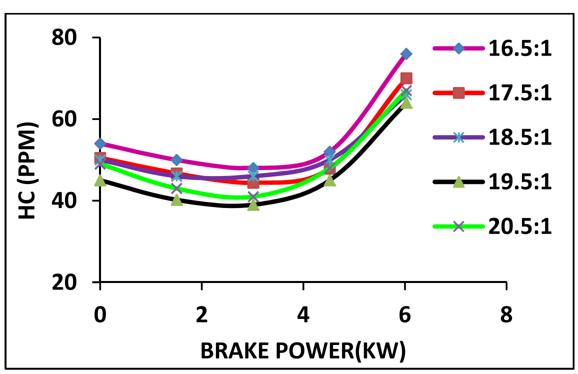

Fig. 14. Brake power Vs HC for different blends

v. Oxides of nitrogen (NO_x)

Figure 15 shows that the NO_x emission decreases by 15.1% for J20 blend at 16.5:1, increases by 5.1% for J20 blend at 18.5:1, 10.7% for J20 blend at 19.5:1 and 6.95% for J20 blend at 20.5:1 when compared to J20 blend at 17.5:1. As compression ratio increases the combustion pressure and temperature increase which accelerates the oxidation of nitrogen to form nitrogen oxides. At lower compression ratio, the combustion takes place during expansion stroke which results in lower combustion temperature and pressure which leads to lower NO_xemission.

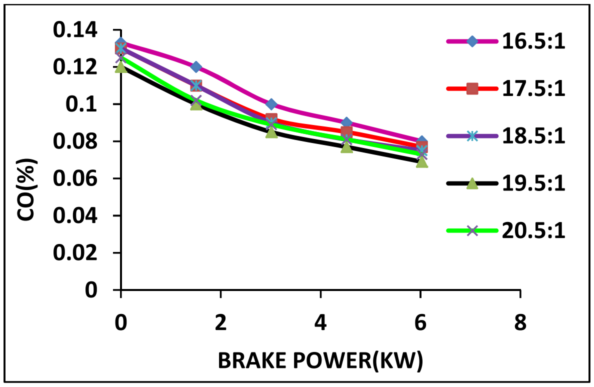

Fig. 15. Brake power Vs CO for different blends

vi. Hydrocarbon (HC)

Figure 16 shows that HC emission increases by 10.52% for J20 blend at 16.5:1, decreases by 2.9% for J20 blend at 18.5:1, 5.8% for J20 blend at 19.5:1 and decreases by 1.47% for J20 blend at 20.5:1 when compared to J20 blend at 17.5:1. This may be due to increase in air temperature at the end of compression stroke, enhancement in combustion temperature and reduction in charge dilution leads to better combustion and reduction in hydrocarbon emissions at high compression ratio. Increase in hydrocarbon emission is observed with reduction in compression ratio which is due to slow combustion process.

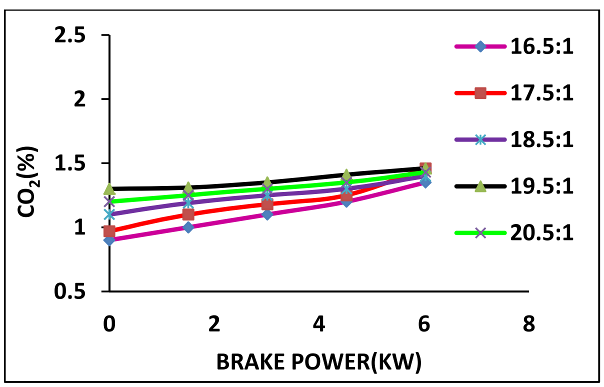

Fig. 16. Brake power VsCO_2 for different blends

vii. Cylinder pressure

Figure 17 shows that the peak pressure decreases by 7.5% for J20 blend at 16.5:1, increases by 2.2% for J20 blend at 18.5:1, 4.4% for J20 blend at 19.5:1 and 1.78% for J20 blend at 20.5:1 when compared to J20 blend at 17.5:1. This may be due to increase in density of air fuel mixture, better mixing of unburnt and burnt charges results in fast and efficient combustion.

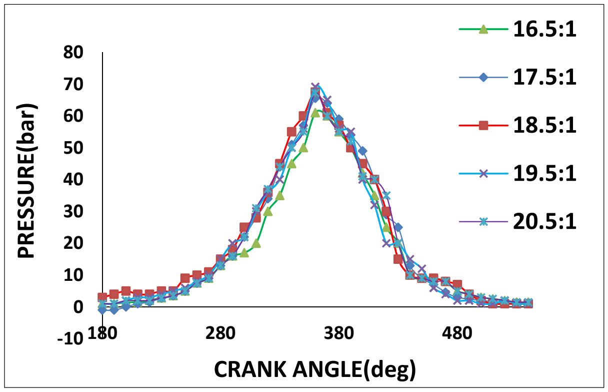

Fig. 17. Crank angle Vs Pressure

viii. Heat release rate

Figure 18 shows that the HRR increases by 7.68% for J20 blend at 16.5:1, increases by 4.65% for J20 blend at 18.5:1, 1.28% for J20 blend at 19.5:1 and 1.08% for J20 blend at 20.5:1 when compared to J20 blend at 17.5:1.

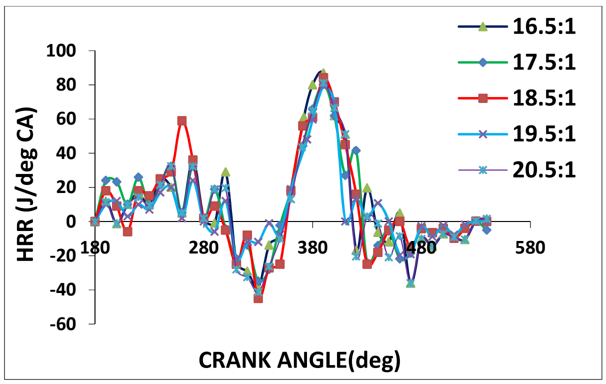

Fig. 18. Crank angle Vs HRR

ix. Cumulative heat release rate

Figure 19 shows that the maximum Cumulative HRR decreases by 12.20% for J20 blend at 16.5:1, decreases by 6.84% for J20 blend at 18.5:1, 15.50% for J20 blend at 19.5:1 and 11.39% for J20 blend at 20.5:1 when compared to J20 blend at 17.5:1. This may be due to injection of a higher quantity of fuel during larger delay period and slow combustion.

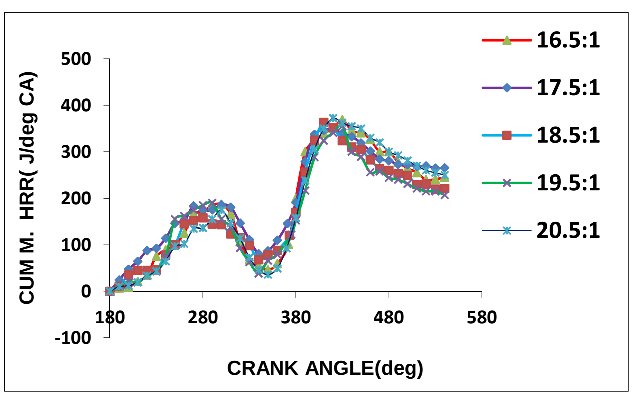

Fig. 19. Crank angle Vs Cumulative HRR

c) Economic analysis of Jatropha biodiesel

Transportation sector has a dominant role in global fuel consumption and greenhouse gas emissions. Due to the drastic increase in greenhouse gas emissions, sustainable development of this sector has raised the concern in many countries. Biodiesel is a renewable energy that has great potential to serve as an alternative fuel to fossil diesel in the compression ignition (CI) engine. A wide variety of biodiesel research on transesterification, performance and emission analysis is currently available worldwide. However, the study on techno-economic and feasibility of biodiesel fuel is limited. Biodiesel and diesel fuel are similar in chemical structure and have similar properties. So they can be burnt in diesel engine, also biodiesel has significant lubricity to the fuel (something that sulphur formally did in diesel fuel, but has since been reduced, hence ultra-low-sulphur-diesel or ULSD), reducing the engine and fuel pump wear and reportedly extending engine life. Adding just 1% biodiesel to ULSD will restore lubricity to the fuel.

- Cost of raw Jatropha oil = $ 0.367/ litre.
- Biodiesel processing cost = $ 0.15/ litre.
- Cost of production = $ 0.517/ litre.
- Less return from crude glycerol = $ 0.05/litre.
- Net cost of production = $ 0.467/ litre.
- Dealer's margin = $ 0.167/ litre.
- Profit = $ 0.05/ litre.
- Sales price of biodiesel = $ 0.533/ litre.

d) Commercial viability

According to estimations, 3.21 million tons of biodiesel would be required from 3.42 million hectares to meet 5-percent blending by Fiscal 2011/12. Considering Jatropha to be a major feedstock for biodiesel with an average seed yield of 2.5 tons/hectare and 30 percent biodiesel recovery rate, 18.6 million hectares would need to be brought under Jatropha cultivation to meet the 20-percent blending target by 2017. The above assessment assumes a steady rise in demand (circa 6.4 percent / annum) for diesel in India. Diesel demand during the 12th five year plan (fiscal year 2012/13 through 2016/17) is likely to grow by 35 percent to 87.4 million tons. Meeting a 5-percent blending target will require an additional 4.1 million hectares under Jatropha.

e) Life cycle assessment

The production and use of Jatropha biodiesel triggers an 82% decrease in non-renewable energy requirement (Net Energy Ratio, NER = 1.85) and a 55% reduction in global warming potential (GWP) compared to the reference fossil-fuel based system. However, there is an increase in acidification (49%) and eutrophication (430%) from the Jatropha system relative to the reference case. Although adding biogas production to the system boosts the energy efficiency of the system (NER = 3.40), the GWP reduction would not increase (51%) due to additional CH4 emissions. For the land use impact, Jatropha improved the structural ecosystem quality when planted on wasteland, but reduces the functional ecosystem quality. Fertilizer application (mainly N) is an important contributor to most negative impact categories. Optimizing fertilization, agronomic practices and genetics are the major system improvement options.

5. CONCLUSION

Injection Timing of 30^{o}bTDC, along with Compression Ratio of 19.5 gives better performance, combustion and lower emissions when compared with standard Injection Timing of 27^{o}bTDC and

Compression Ratio of 17.5. For all tested values, J20 provides best results in terms of BTE, higher heat release rate, and lower emissions of HC, CO and NO_x. Hence J20 can be effectively used as an alternative biodiesel with Injection Timing of 30^{o}bTDC along with Compression Ratio of 19.5 in tested engine. Even so, only 20% of Jatropha methyl ester added to 80% pure diesel will, to a certain extent meet the shortage of availability of pure diesel. Jatropha is available with lower cost when compared to diesel in the present scenario. Hence JME will also be economical for diesel trains.

REFERENCES

1. Al-Baghdadi, M. A. R. S. (2004). Effect of compression ratio. Equivalence ratio and engine speed on the performance and emission characteristics of a spark ignition engine using hydrogen as a fuel. *Renewable Energy*, Vol. 29, pp. 2245–2260.
2. Anand, R., Kannan, G. R., Nagarajan, S. & Velmathi, S. (2010). Performance emission and combustion characteristics of a diesel engine fueled with biodiesel produced from waste cooking oil. *SAE*, 2010-01-0478.
3. Babu, A. K. & Devaradjane, G. (2003). Vegetable oils and their derivatives as fuels for CI engines. *SAE* 2003-01-0767.
4. Balat, M. & Balat, H. (2010). Progress in biodiesel processing. *Appl Ener*, Vol. 87, pp. 1815-1835.
5. Demirbas A. (2009). Progress and recent trends in biodiesel fuels. *Energy Conversion Management*, Vol. 50, pp.14-34.
6. Dennis, Leung, Y. C., Wu, X. & Leung, M. K. H. (2010). A review on biodiesel production using catalyzed transesterification. *Applied Energy*, Vol. 87, pp. 1083-1095.
7. Kim, H. J., Kang, B. S., Kim, M. J., Park, Y. M., Kim, D. K. & Lee, K. Y. (2004). Transesterification of vegetable oil to biodiesel using heterogeneous base catalyst. *Catalysis Today*, Vol. 93, pp. 315-320.
8. HanumanthaRao, Y. V. & Ram S. V. (2009). Experimental investigations on Jatropha biodiesel and additive in diesel engine. *Indian Journal of Science and Technology*, Vol. 2, No. 4.
9. Jindal, S. Nandwana, B. P., Rathore, N. S. & Vashistha, V. (2010). Experimental investigation of the effect of compression ratio and injection pressure in a direct injection diesel engine running on Jatropha methyl ester. *Applied Thermal Engineering*, Vol. 30, pp. 442–448.
10. Jon, H., Van G., Peterson, C. L. & Goering, C. E. (2007). Biodiesel: An Alternative Fuel for Compression Ignition Engines. *American Society of Agricultural and Biological Engineers*, Vol. 31, pp. 1-22.
11. Kegl, B. (2006). Numerical analysis of injection characteristics using biodiesel fuel. *Fuel*, Vol. 85, pp. 2377–2387.
12. Kjarstad, J. & Johnsson, F. (2009). Resources and future supply of oil. *Energy Policy*, Vol. 37, pp. 441–64.
13. Mohanty, C., Jaiswal, A., Meda, V. S., Behera, P. & Murugan, S. (2011). An experimental investigation on the combustion, performance and emissions of a diesel engine using vegetable oil- diesel fuel blends. SAE 2011-01-1187.
14. Parlak, A., Yasar, H. & Sahin, B. (2003). Performance and exhaust emission characteristics of a lower compression ratio LHR Diesel engine. *Energy Conversion and Management*, Vol. 44, pp. 163–175.
15. Raheman, H. & Ghadge, S. V. (2008). Performance of diesel engine with biodiesel at varying compression ratio and ignition timing. *Fuel*, Vol. 87, 12, pp. 2659–2666.
16. Shahi, R. V. (2011). Energy markets and technologies in India, keynote address in global energy dialogue (Hanover, Germany).
17. Watanabe, H., Tahara, T., Tamanouchi M. & Iida, J. (1998). Study of the effects on exhaust emissions in direct injection diesel engines: effect of fuel injection system, distillation properties and cetane number. *JSAE Review*, Vol. 19, pp. 21–26.

18. Wijesinghe, K. & Rukshan, P. D. (2004). The influence of water injection on the thermal efficiency & specific fuel consumption of a ci engine. *Fisita World Automotive Congress*, Barcelona.
19. Ueki, S. & Miura, A. (1999). Effect of difference of high pressure fuel injection systems on exhaust emissions from HDDI diesel engine. *JSAE Review*, Vol. 20, pp. 555– 561.
20. Kline, S. J. & McClintock, F. A. (1953). Describing uncertainties in single sample experiment.