Theoretical and Experimental Performance and Emission Analysis of Vegetable Oil Ester Operated CI Engine for Various Injection Timing

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Abstract—A theoretical model was developed to evaluate the performance characteristics and combustion parameters of vegetable oil esters like Jatropha, Mahua and Neem for the various injection timing and they are compared to diesel fuel. The predicted results of these fuels were compared with experimental result and also compared with the result of diesel fuel for the validity of the theoretical model. From the predicted results, it was found that the heat release and work done were reduced by about 4% for Jatropha, 6% for Mahua and 8% for Neem oil esters when compared to diesel. However, slight increase was observed for specific fuel consumption. The experimental work was carried out in a single cylinder computerized diesel engine test rig, which develops 5.2 kW at 1500 rpm and loaded by eddy current dynamometer at various injection timing. The harmful pollutants such as HC, CO, NO_X and smoke were reduced in the vegetable oil esters than those of diesel fuel. The predicted values of performance and emission for all fuel more or less equal to the experimental values. From the investigation it was concluded that the performance of vegetable oil esters such as Jatropha, Mahua, and Neem are much better. Thus the developed model was highly capable for simulation work with bio-diesel as a suitable alternative fuel for diesel

Index Terms— I.CEngine, InjectionTiming, performance , Emission,Vegetable oil, Transesterification and brake power

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1 INTRODUCTION

The tremendous growth of vehicular population and industrialization of the world has led to a steep rise in the demand for petroleum and diesel products. In this investigation to evaluate the engine performance using three vegetable oil esters like jatropha, mahua and neem. They are compared with diesel in C.I engine. An experimental investigation is carried out to assess the performance parameters and emission levels of the three different vegetable oil esters in single cylinder, four strokes constant speed computerized engine test rig. The performance characteristics and emission levels such as NO_x, CO, HC, and Smoke are measured using eddy current dynamometer, Crypton computerized emission analyzer instruments (Model-EN2-390) and Bosch Smoke meter. From the experimental and theoretical results, it is concluded that in terms of performance characteristics and emission levels, vegetable oil esters can be regarded as the best alternative fuel instead of diesel fuel.

1.1Bio-diesel

Due to shortage of petroleum diesel fuel and its increasing cost an alternate source of fuel is very much needed. It has been found that vegetable oils hold special promise in this regard, since they can be produced from the plants grown in rural areas. Vegetable oil from crops such as soybean, peanut, sunflower, jatropha, mahua, neem, rape, coconut, karanja, cotton, mustard, linseed and castor have been tried in many parts of the world, where there is a lack of petroleum reserves as fuels for compression ignition engines. The botanical name of the jatropha shrub is jatropha curcas. Typically it is a drought resistant shrub growing up to about 4-5 feet tall, yielding 4 tons seed per acre under good irrigation. The madhuca indica oil is obtained from well-dried kernels. The madhuca indica tree is a drought resistant and deciduous. Although the long chain hydrocarbon structure, vegetable oils have better igni-

tion characteristics; it causes serious problems such as carbon deposits buildup, poor durability, high density, high viscosity, lower calorific value, more molecular weight and poor combustion. These problems consequently lead to poor thermal efficiency, while using vegetable oil in the diesel engine. However these problems can be rectified by transesterification process.

1.2 Trasesterification

Transesterification is an important and most commonly used method to reduce the viscosity of vegetable oils. In this process triglyceride reacts with three molecules of alcohol in the presence of a catalyst producing a mixture of fatty acids, alkyl ester and glycerol. The process of removal of all the glycerol and the fatty acids from the vegetable oil in the presence of a catalyst is called esterification. This esterified vegetable oil is called bio-diesel. Bio diesel properties are similar to diesel fuel. It is a renewable, non-toxic, biodegradable, clean burning and eco friendly transportation fuel. After esterification of the vegetable oil its density, viscosity, cetane number, calorific value, atomization and vaporization rate, molecular weight, and fuel spray penetration distance are improved considerably. So these improved properties give a better performance in CI engine. Physical and chemical properties are more improved in esterified vegetable oil because esterified vegetable oil contains more cetane number than diesel fuel. These parameters induce good combustion characteristics in vegetable oil esters. The unburnt hydrocarbon level is decreased in the exhaust. It results in lower generation of hydrocarbon and carbon monoxide in the exhaust than diesel fuel. The vegetable oil esters have lower calorific value fuel than diesel. So, it produced lower cylinder mean temperature and generates lower

nitric oxide formation in the exhaust than the diesel fuel.

Properties of Vegetable oil after Trasesterification

Properties	D	J-E	M-E	N-E
Density at 15^0C kg/m^3	840	850	860	873
Cetane number	50	55	53	51
Kinetic viscosity at 40^0C mm^2/s	2.6	3.0	3.1	3.2
Surface tension at 20^0C N/m	0.023	0.024	0.025	0.026
Calorific Value MJ/kg	43	41	40.0	39.6
Specific heat capacity kg^{0}C	1850	1900	1936	1940
10% Distillation point ^{0}C	220	230	241	244
90% Distillation point ^{0}C	300	315	320	326
Oxygen% Weight	0	10	10	10
Latent heat of vaporization kJ/kg	250	245	240	236
Bulk modulus of elasticity (bar)	16000	16700	16840	17150
Stoichiometric air to fuel ratio	15	13.9	13.5	13.2
Molecular weight	170	190	200	215

J-E Jatropha oil Ester (Jatropha Curcas - Botanical name) M-E Mahua oil Ester (Madhuca Indica oil -Botanical name) N-E Neem oil Ester
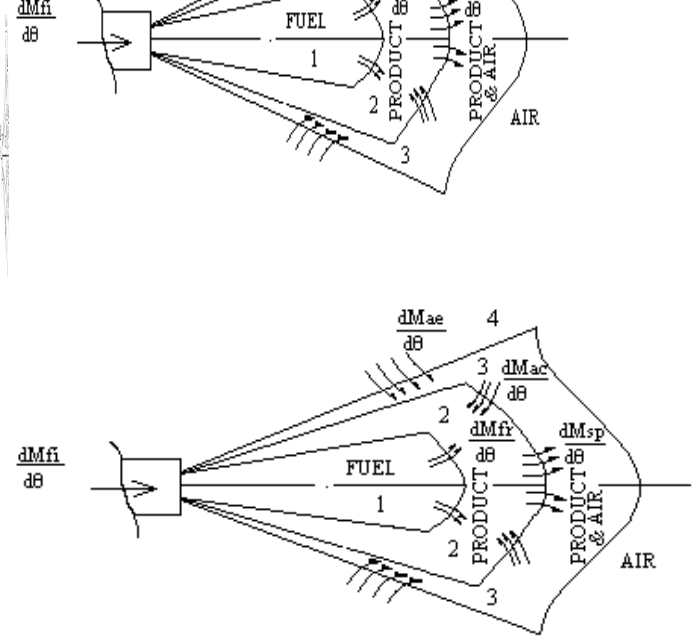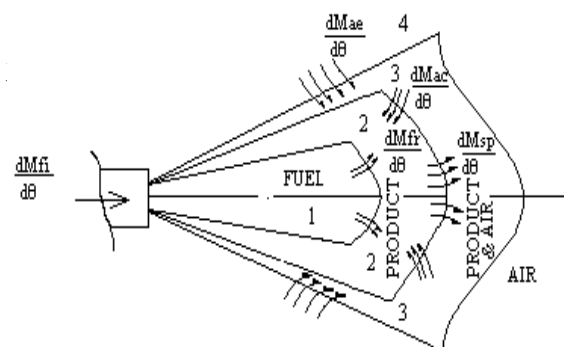
2. Theoretical consideration Discription of the four

Figure 1.schematic representation of combustion zones

1-Fuel injection rate
2-Fuel reaction rate
3-Air Entrainment Rate
4-Air consumption rate for stoichio burning
5-stoichiometric product movement rate

2.1.1Fuel Zone

OF these four zones proposed, the fuel zone is the central core of the spray, which is assumed to contain only fuel, and as shown in zone 1 of the fig (1). The size of the zone or the rate of change of mass of this zone depends upon the rate of injection of the fuel and the rate of reaction of the fuel or the heat release rate both expressed in the same units. The fuel injection rate is supplied as data in the program. The fuel reaction rate is calculated based on the overall conditions existing in the spray. At any step the quantity of fuel burnt or the burning rate is calculated using Whitehouse and Way combustion model. This model is basically in two parts; a preparation model and a reaction or burning model. The preparation model is a correlation between preparation rate, surface area of the fuel droplets all over the spray and the mean partial pressure of oxygen inside the spray. The combustion period in this model is postulated to consist of two periods. They are premixed and diffusion periods. These two periods were rather arbitrarily separated and not integrated in to the combustion calculation at one time, but for some years this difficulty has been met with by using an Arrhenius type equation for reaction rate of the prepared but unburnt fuel. The temperature of the fuel zone is assumed to be the same as that of the average temperature of the spray or the burning zone temperature of the two-zone model. The schematic location of the zones, as shown in fig (1) is not intended to be a true geometric representation, because it is thought that the fuel/air mixture is even more heterogeneous than the model implies, in that apart from the general large scale distribution there may well be small scale variations corresponding to what originated as fuel droplets. Such small scales variations are lumped into these appropriate concentration categories for calculation purposes. This general distribution of small-scale fuel concentrations is a purpose for taking the fuel zone to be at the mean burning zone temperature.

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2.2 Stoichiometric burning zone

This stoichiometric burning zone is the zone 2 of the figure (1). The schematic location of this zone is between fuel zone and the air plus product zone. Combustion is assumed to take place under stoichiometric condition in this zone in accordance with observations that combustion rates in diffusion flames are greater in regions where stoichiometric mixture exists. This zone is the zone of the highest temperature in the combustion chamber. The fuel for this zone comes from fuel zone or zone 1 and the air required for stoichiometric combustion of this fuel comes from the zone 3 or the air plus product zone.

2.3 Air/Product zone

This air plus product zone is the zone 3 in figure (1), which is schematically located between the stoichiometric burning zone and the space outside the spray. The size or the rate of change of mass of this burning zone is mainly controlled by three factors. They are the entrainment rate into the spray, the air consumption rate for stoichiometric combustion of the reacted fuel in the burning zone and finally, the product-mixing rate from the stoichiometric burning zone. If the mixing rate of

product in this zone is zero, the composition of this zone will be the same as that of the trapped charge. Hence, the composition and mainly the quantity of nitric oxide in this zone is directly controlled by the mixing factor which controls the rate of mixing of the stoichiometric product from the burning zone. No reaction is assumed to take place in this zone, which is justified by the fact that the temperature of this zone during the major part of the expansion process is very low in comparison to the stoichiometric burning zone for the practical range of product mixing rate. The product-mixing rate is assumed to be proportional to the amount of product present in the stoichiometric burning zone. This product mixing rate parameter is the only extra empirical factor referred to in the four-zone model, over and above the empirical constants present in the two-zone model. The product-mixing factor is assumed as 0.01 after making elaborate optimization study referred to in Baluswamy. N.

2.4 Air or un burnt zone

This zone is shown as zone 4 in figure (1). The space outside the spray comprises the air zone. The composition of the zone remains constant and is the same as the trapped charge composition. The rate of change of mass of this zone is wholly controlled by the entrainment rate of the air charge into the spray.

2.5 Theory of spray formation and combustion heat transfer

The theory used to simulate the combustion process in the combustion chamber of a diesel engine mainly involves the jet formation, calculation of jet penetration and spray volume, estimation of fuel burning rate and finally the heat transfer between the cylinder contents and the surroundings.

2.6 Fuel jet penetration

The development of the fuel spray or jet in the diesel combustion chamber is based upon the theory of a steady flow semi-infinite free jet and finally modified by the use of empirical factors of transient (real) jets. Assuming that fuel jet penetration model is developed by modifying the transient (real) jet equation. The fuel jet penetration is X_{max}.

$$XU_{max} = 7.414\sqrt{K} \quad \text{--------------------} \quad (1)$$

X - Jet penetration. (m)

U_{max} – Centre line Velocity (m/sec)

K - Kinematics momentum flux (m^4/sec^2)

$$XdX = 7.414\sqrt{K}\,dt \quad \text{----------------------} \quad (2)$$

$$\mathbf{X_{max}} = \mathbf{0.685 \times 2.420} \left(\left[\frac{\Delta p}{\rho_a} \right]^{0.5} dn.t \right)^{0.5} \quad \text{------} \quad (3)$$

2.7 Volume of Fuel Spray

The volume of fuel before impingement consists of conical part of the half cone angle θ and the bell shaped part. The volume of the conical part of the spray is calculated analytically and for the bell shaped part of the jet, numerical integration is used. The volume flow rate at any section along the axis of the spray can be computed

by integrating the product of velocity and area

$$\text{Volume flow rate} = 2\pi r_o u_{max} \int_0^r \frac{r}{r_o} \frac{u}{u_{max}} dr \text{ ------- (4)}$$

$$\frac{r}{r_0} = \varepsilon \text{ -------------------- (5)}$$

$$r_0 = X \tan\theta \text{ ---------------- (6)}$$

$$U_{max} = 7.414 \frac{\sqrt{k}}{x} \text{ From the free jet theory ------- (7)}$$

$$\text{Volume flow rate} = 5.9902 \tan^2\theta \sqrt{K} X . KF \text{ ------ (8)}$$

2.8 Combustion and heat release

In this model the combustion period is assumed to consist of two periods. They are pre-mixed period and diffusion period. The combustion of the reacted fuel is assumed to be ideal. It produces only H_2O and CO_2 for the purpose of calculating the cylinder pressure and the energy level of the spray as a whole. Whitehouse model incorporating the rate of preparation of the fuel, surface area of fuel droplets and partial pressure of oxygen in the cylinder are used for this work. The preparation rate equation is

$$P = K' M_i^{1-x} M_u^{x} PO_2^{L} \text{ -------------------- (9)}$$

$$P = K' M_i^{1/3} M_u^{2/3} PO_2^{0.4} \quad (Kg/^oCA) \text{ ----- (10)}$$

2.9 Heat transfer

$$M_i = n \times \rho \frac{1}{6} \pi D_0^3 \text{ ----------------- (11)}$$

$$M_u = n \times \rho \frac{1}{6} \pi D^3 \text{ ----------------- (12)}$$

Total surface area at any instant $= n \times \pi \times D^2$

Arrhenius type equation is used for reaction rate of the prepared but unburnt fuel, which is as follows,

$$\text{R=K'' } (PO_2/N \sqrt{t}) 3e^{-ACT/T} \int(P-R)dx \quad Kg/8CA \text{ ---(13)}$$

The heat transfer rate is calculated by using Annand's formula (1963). This formula seems more fundamental than the available alternatives. The equation considers net heat transfer as the summation of both radiative and convective heat transfer.

$$dQ_{Loss!} dt = (S_c.H(T-TW) + S_{r}.C(T4-TW^4)) \text{ ---(14)}$$

2.10 Gas properties

The gas properties to be used in the heat transfer formula (The heat transfer calculation uses only four constituents of the ideal combustion process namely N_2, $O2$, $H2O$ and CO_2) and the internal energy of the mixture is calculated from the following expressions.

Total number of moles of the mixture $N = \Sigma N_i$ where i suffix refers to the ith constituent in the mixture.

Molar fraction of constituent $X_i = \frac{N_i}{N}$

Molar weight of the mixture $M = M N_i X_i$ (Kg/Kg.mol)

Dynamic viscosity of the mixture $\mu = Z (T_m)^{0.645}$

$$U = h_0 + \overline{R}(h_1.T + h_2 T^2 + h_3 T^3 + h_4 T^4 + h_5 T^5 \text{ -(15)}$$

2.11 Method of estimating the final cylinder pressure

The final cylinder pressure equation can be written as

$$P_2 = p_1 + p_1(T-T_{b1}/T_{b1})(V_{b1}/V_{u1}+V_{b1}) (V_1/V2)^{Cp/Cv}$$

---------- (16)

$P_2 =$ First estimated value of the final pressure

$P_1 =$ Pressure after entrainment

$T-t_{b1} =$ Temperature rise due to constant volume heat addition

$V_1, V_2 =$ Cylinder volume at beginning and end of the step.

2.12 Equation of conservation of mass

The mass conservation equations satisfy the constraint for the rate of change of mass in the chamber as a whole.

$$dM_{cly}/dQ = dM_1/dQ + dM_2/dQ + dM_3/dQ + dM_4/dQ \text{ -----(17)}$$

dM_1/dQ, dM_2/dQ, dM_3/dQ, dM_4/dQ

(The rate of change of mass in zone 1, 2, 3 and 4).

2.13 Energy equations

Energy equation can be written as,

$$dE_{cyl}/dQ = DQF - DW - DQC - DQR \text{ ------(18)}$$

DQC – DQR   = Heat transfer to the system

DQF =   Energy associated with the external flow to the system

DW   = External work done by the system

3. DESCRIPTION OF THE COMPUTER PROGRAMME

The program can be divided into four parts. The first part is reading the data, which are either permanent (enthalpy coefficients, viscosities, molecular weights) or vary with the engines and the running conditions, (engine geometry, fuel injection data, Annand's coefficients, Whitehouse constants for heat release, trapping conditions). The second part covers the first part of the working cycle, it is from the crank angle when inlet value closes (IVC) to the one at which the fuel injection just begins. This part is essentially similar to the single zone synthesis model. The third part is the two-zone part of the program. Calculation of the two-zone model being with the injection of the fuel into the combustion chamber. The two zones are burning zone and the non-burning zone. The growth of the fuel spray is considered to be developed mainly in two stages, based on the type of equations used for entrainment calculation. The fourth part of the program deals with the subdividing of the spray (or the burning zone of the two zone model) into three sub zones within the boundary conditions of the spray, (like the volume of the spray, heat transfer from the burning zone, internal energy of the spray), as dictated by the two zone calculations. In fact this part of the program does the calculations not after the completion of the two zone calculations up to EVO but in parallel with the two zone calculations at every crank angle from the beginning of injection the fourth part of the computer program deals with the calculation of the sizes of the three sub zones.

4. EXPERIMENTAL WORK

The performance tests were carried out on a single cylinder, four stroke naturally aspirated and water cooled kirloskar computerized diesel engine test rig. Diesel engine was directly coupled to an eddy current dynamometer. The engine and dynamometer were interfaced to a control panel which was connected to a computer. This computerized test rig was used for recording the test parameters such as fuel flow rate,air flow rate, temperature and load for calculating the engine performance, brake thermal efficiency and emissions like CO,HC,NOx and Smoke. After setting the engine load torequired values the following observations and subsequent calculations were made,

1) Time for 60cc of fuel consumption (sec)

2) Exhaust gas temperature

3) Water inlet and outlet temperature.

4) Air flow rate.

5)HC, CO, NOx and smoke are recorded for various

Load Conditions.
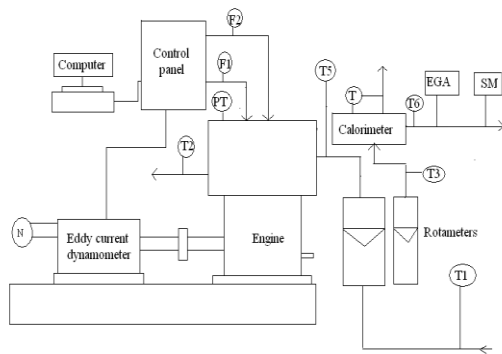
Figure 2. Schematic of experimental set-up

T1, T3- Inlet water temperature
T2- Outlet engine jacket water temperature
T4- Outlet calorimeter water temperature
T5- Exhaust gas temperature before calorimeter
T6- Exhaust gas temperature after calorimeter
N- Rpm decoder

F1- Fuel flow for differential pressure unit
F2- Air intake difference pressure unit
PT- Pressure transducer
EGA- Exhaust gas analyzer (Five gases)
SM- Smoke meter
Engine Specifications

Engine parameters	Specifications
Engine type	Kirloskar, Four stroke
No of cylinder	Single
Bore	87.5 mm
Stroke	110 mm
Cubic capacity	661 cc
Compression ratio	17.5
Rated speed	1500 rpm
Dynamometer	Eddycurrent,Water cooling
Propeller shaft	With universal joints
Fuel injection pressure	200 bar
Rated output	5.2 kW

5. THEORETICAL PERFORMANCE AND EMISSION CURVES

The performance and emission characteristics curves are carried out experimentally like predicting various performance characteristics such thermal efficiency, specific fuel consumption for different vegetable oil esters such as jatropha, mahua and neem. These experimental results of vegetable oil esters are compared with diesel fuel. From the results it is concluded the performance of vegetable oil esters are reduced about 5% to 8% and the harmful pollutants are reduced by 18% when compared to diesel. The results are shown in figure (1) to figure (6).

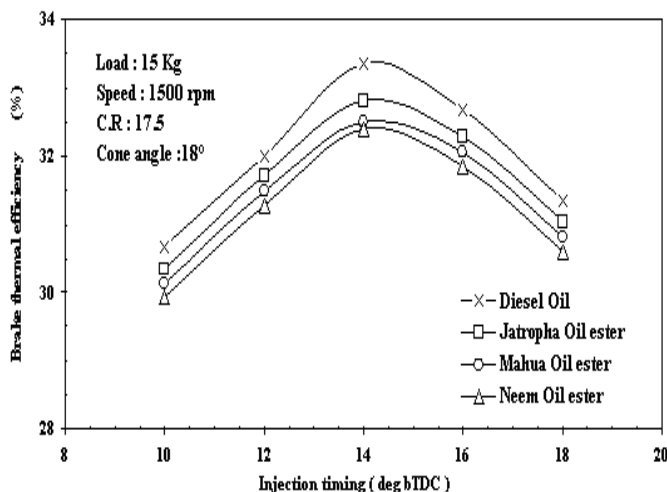### 5.1 Theoretical Brake Thermal Efficiency for various injection Timing

Figure 5.1 COMPARISON OF PREDICTED BRAKE THERMAL EFFICIENCY FOR THREE DIFFERENT BIODIESEL WITH RESPECT TO INJECTION TIMING

The brake thermal efficiency is also predicted with respect to various injection pressure for different vegetable oil esters and compared with diesel fuel. Injection pressure is one of the very important operating variables, which affects the brake thermal efficiency. The brake thermal efficiency is predicted for various injection pressure The

injection pressure tried are 140bar to 240 bar. When the injection pressure 200bar it is found that the brake thermal efficiency is reach maximum. Fuel particles are uniformly mixed with air particles at injection pressure 200bar. when the injection pressure is increased above 200bar the wall-wetting problem is created. This problem leads to a decrease in the performance of the engine. When the cone angle is decreased below 200 bar the fuel penetration and velocity are decreased. It generates more un burnt hydrocarbon. So injection pressure 200 bar is found optimum value for reach maximum efficiency in the engine.

5.2 Theoretical brake specific fuel consumption for various injection pressures

The brake specific fuel consumption is found for diesel is 0.257 kg/kW-hr; where as for jatropha oil ester it is 0.277 Kg/kwhr; for mahua oil ester it is 0.286 kg/kW-hr; and for neem oil ester it is 0.291 kg/kwhr for optimum engine conditions such as 75% of load, injection pressure of 200bar. Because at optimum engine condition fuels burn completely. Misfiring is not occurred this time. This increases the temperature and the number of moles of the burned gases in the cylinder. This effect increases the pressure to give increased thermal efficiency and decreased the specific fuel consumption. The specific fuel consumption is predicted with respect to injection pressure for various vegetable oil esters and compared with diesel fuel.
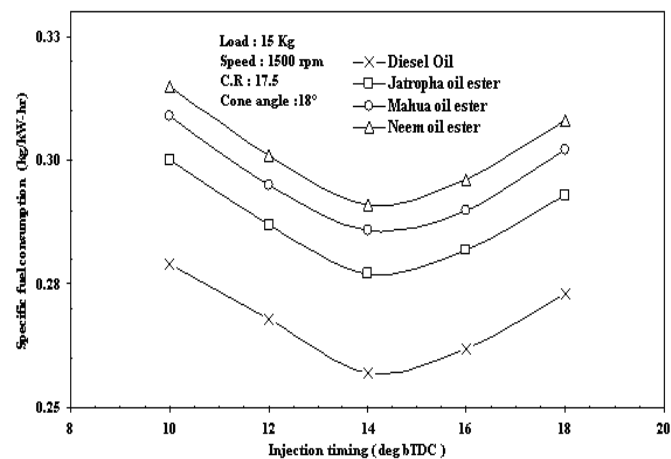

Figure 5.2 COMPARISON OF PREDICTED SPECIFIC FUEL CONSUMPTION FOR THREE DIFFERENT BIODIESEL WITH RESPECT TO INJECTION TIMING

5.3 Theoretical carbon monoxide for various injection pressures

Figure (3) shows the comparison of predicted emission result of carbon monoxide emission with various injection pressures for different vegetable oil esters and compared with diesel fuel. The carbon monoxide for vegetable oil esters are nearly 18 % reduced than diesel. The main difference in ester-based fuel compared to diesel is the oxygen content and cetane number. As the ester based fuel contains some oxygen, which acts as a combustion promoter inside the cylinder result better combustion than diesel fuel. Hence carbon monoxide, which is present in the exhaust due to incomplete combustion, reduces

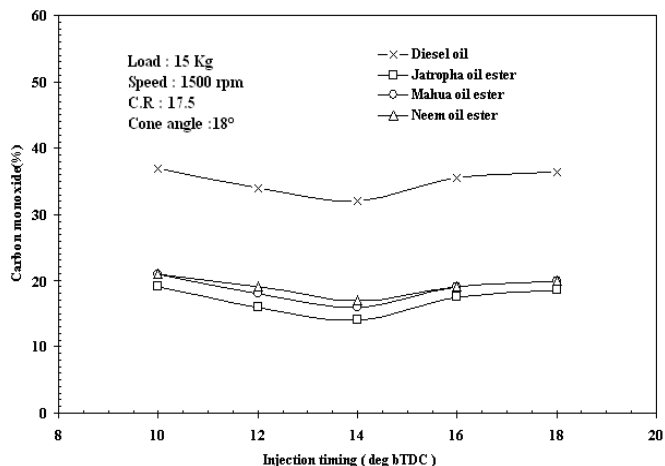drastically. The reduction of carbon monoxide in case of ester is lower when it is compared to diesel.

Figure 5.3 COMPARISION OF PREDICTED CARBON MONOXIDE OF THREE DIFFERENT BIODIESEL WITH RESPECT TO INJECTION TIMING

5.4 Theoretical hydrocarbon for various injection pressure.

Figure (4) shows the comparison of predicted result of hydrocarbon emission with various injection pressure for different vegetable oil esters and compared with diesel fuel. The hydrocarbon emission for diesel is about 119 PPM; while for jatropha oil ester, mahua oil ester and for neem oil ester it is 98PPM, 100ppm, and 101ppm respectively. Cetane number of the fuel plays a vital role in ignition process. As cetane number of ester-based fuel is higher than diesel, it exhibits a shorter delay period and the fuel undergoes better combustion. Here oxygen content of the fuel comes into picture as it enhances the combustion process. Therefore overall result of oxygen content and cetane number of the fuel leads to low CO and HC emission. Thus it is very clear from the graph that esters emit lower Hydrocarbon emission than that of diesel.

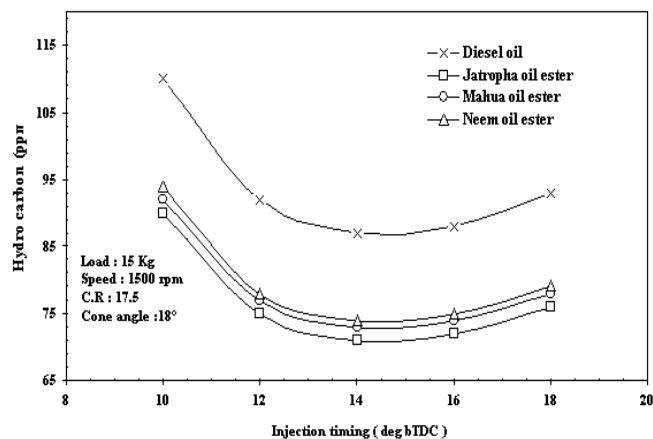## 5.5 Theoretical Nitric oxide for various injection pressures.

Figure 5.4 COMPARISION OF PREDICTED HYDRO CARBON OF THREE DIFFERENT BIODIESEL WITH RESPECT TO INJECTION TIMING

Figure (5) shows the comparison of predicted result of NO_x formation with various injection pressures for different vegetable oil esters and compared with diesel fuel. The NO_x for

diesel is about 985 ppm; while for jatropha oil ester, mahua oil ester and for neem oil ester it is 967ppm, 960ppm, 955ppm respectively. In a direct injection naturally aspirated four-stroke diesel engine NO_x emission is sensitive to oxygen content, adiabatic flame temperature and spray properties. A change in any of those properties may change the NO_x production. Further, more fuel chemistry effects in the flame region could account for a change in NO_x production. NO_x formation is increased with increase in temperature.
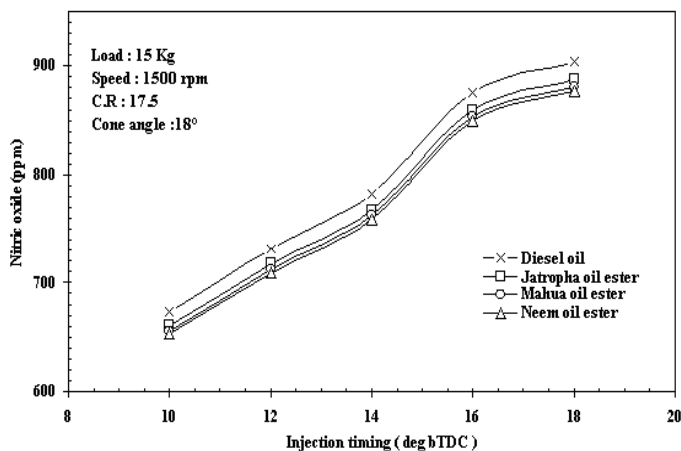
Figure 5.5 COMPARISION OF PREDICTED NITRIC OXIDE OF THREE DIFFERENT BIODIESEL WITH RESPECT TO INJECTION TIMING

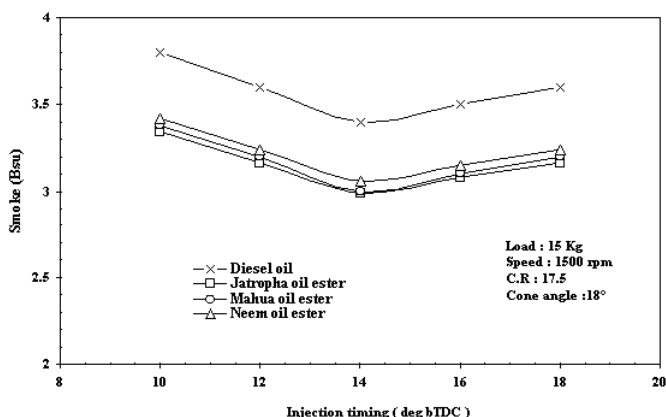## 5.6 Theoretical smoke for various injection pressures

Figure 5.6 COMPARISION OF PREDICTED SMOKE OF THREE DIFFERENT BIODIESEL WITH RESPECT TO INJECTION TIMING

Figure (6) shows the comparison of experimental results of smoke with various engine injection pressure for different vegetable oil esters and compared with diesel fuel. The smoke for diesel is 4 bsu; for jatropha oil ester, it is 3.5bsu; for mahua oil ester it is 3.6bsu; and for neem oil ester it is 3.6bsu. The smoke that formed due to incomplete combustion is much lower for esters compared to diesel. This is because of better combustion of esters. The main difference in ester-based fuel compared to diesel is the oxygen content and cetane number.

As the ester based fuel contain some oxygen, which acts as a combustion promoter inside the cylinder result better combustion than diesel fuel.

6. COMPARISION OF THEORETICAL AND EXPERIMENTAL PERFORMANCE AND EMISSION CURVES

6.1 Theoretical and experimental Brake Thermal Efficiency for various injection Timing

Figure 6.1 exhibits Figure 6.1 exhibits the result of theoretical brake thermal efficiency of all the vegetable oil esters for various injection timings compared in this work. When the injection timing is retarded from 18° to 14° bTDC, brake thermal efficiency is considerably increased. But for injection timing of 12° bTDC and 10° bTDC, it is found that the brake thermal efficiency is decreased. The maximum brake thermal efficiency is predicted for injection timing of 14° bTDC. Probably more complete combustion possible, at the injection timing of 14° bTDC, may be the reason for the increase in brake thermal efficiency. However, advanced and retarded injection timing of 16° and 18° and 12° and 10° may result in the decrease of complete combustion. Hence, at 14° bTDC, maximum brake thermal efficiency is obtained as expected.
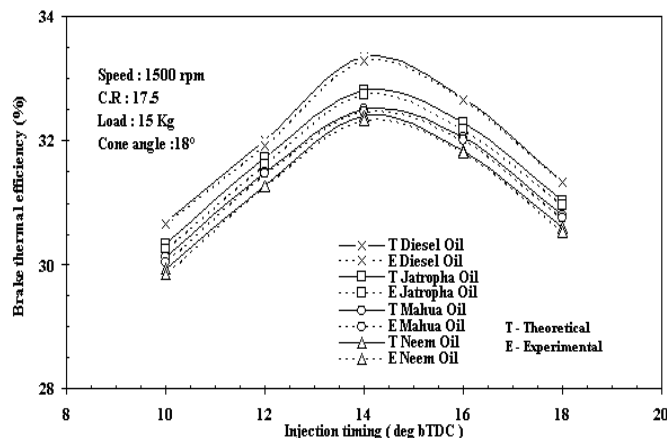

Figure 6.1 COMPARISION OF EXPERIMENTAL AND PREDICTED BRAKE THERMAL EFFICIENCY OF THREE DIFFERENT BIODIESEL WITH RESPECT TO INJECTION TIMING

6.2 Theoretical and experimental Specific fuel consumption for various injection Timing

The variation of brake specific fuel consumption for various injection timings for different vegetable oil esters is seen from Figure 6.2. The minimum specific fuel consumption is found for injection timing of 14° bTDC for all fuels. When the injection timing is retarded from 18° bTDC to 14° bTDC, it is found that the brake specific fuel consumption is decreased. But for injection timing of 12° bTDC and 10° bTDC, it is found that the specific fuel consumption is increased. The minimum specific fuel consumption is predicted for injection timing of 14°

bTDC. However, advanced and retarded injection timing of 16° and 18° bTDC and 12°and10° bTDC may result in the occurrence of more energy consumption. Hence, at 14°bTDC, we get minimum specific fuel consumption as expected.
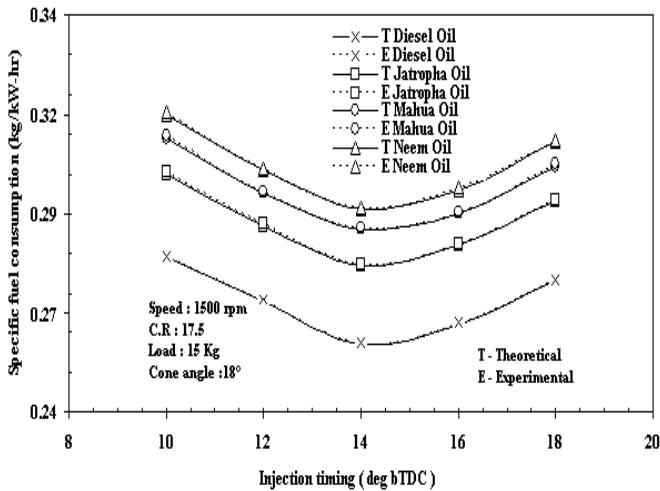

Figure 6.2 COMPARISION OF EXPERIMENTAL AND PREDICTED SPECIFIC FUEL CONSUMPTION OF THREE DIFFERENT BIODIESEL WITH RESPECT TO INJECTION TIMING

6.3 Theoretical and experimental Carbon monoxide for various injection Timing
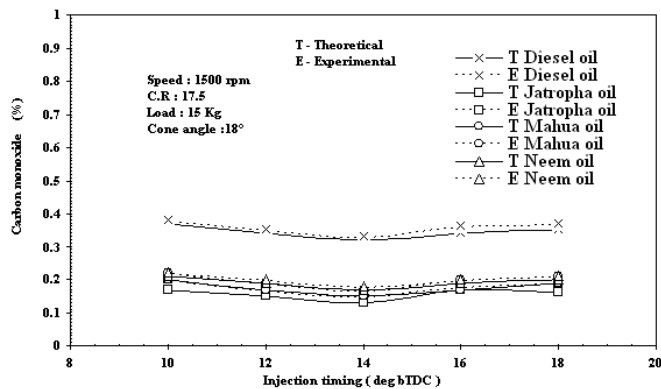

Figure 6.3 COMPARISION OF EXPERIMENTAL AND PREDICTED CARBON MONOXIDE OF THREE DIFFERENT BIODIESEL WITH RESPECT TO INJECTION TIMING WITH DIESEL

Figure 6.3 shows the comparison of predicted result of carbon monoxide emission of the three vegetable oil esters considered for various injection timings. The injection timings tried are 18°, 16°, 14°, 12°and 10° crank angle. The minimum carbon monoxide emission is found for injection timing of 14° bTDC for all fuels. Maximum peak pressure has to occur at 5° aTDC for maximum utilization of pressure. If the injection timing is 14° btdc maximum peak pressure will occur around 5° aTDC. However, advanced and retarded injection timing of 16° and 18°bTDC, 12° and 10°bTDC may result

in the occurrence of incomplete combustion. Hence, at 14°bTDC as expected, we get minimum carbon monoxide and such complete combustion takes place at injection timing of 14° bTDC.

6.4 Theoretical and experimental Hydrocarbon for various injection Timing
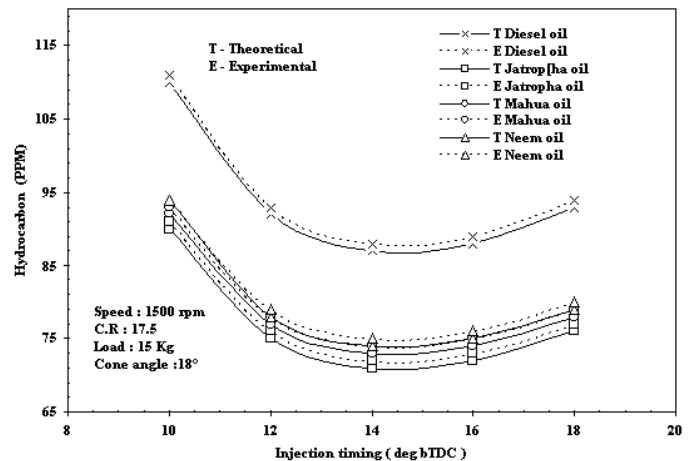

Figure 6.4 COMPARISION OF EXPERIMENTAL AND PREDICTED HYDRO CARBON OF THREE DIFFERENT BIODIESEL WITH RESPECT TO INJECTION TIMING WITH DIESEL

Figure 6.4 shows the comparison of predicted result of hydrocarbon emission for various injection timings for three vegetable oil esters namely jatropha, mahua, and neem. The injection timings tried are 18°, 16°, 14°, 12°and 10° crank angle. The minimum hydrocarbon emission is found for injection timing of 14° bTDC for all fuels. However, advanced and retarded injection timing of 16° and 18°bTDC and 12° and 10°bTDC may result in the occurrence of un burnt hydrocarbon due to pressure fluctuations and incomplete combustion. Hence, at 14°bTDC because of complete combustion minimum hydrocarbon emission is observed.

6.5 Theoretical and experimental Nitric oxide for various injection Timing

Figure 6.5 shows the comparison of predicted result of nitric oxide emission for various injection timings for three vegetable oil esters of jatropha, mahua and neem. The injection timings tried are 18°, 16°, 14°, 12° and 10° crank angle. When the injection timing is retarded from 18° btdc to 10° btdc, it is found that the emission of nitric oxide is decreased. Oxides of nitrogen which occur only in the engine exhaust in the form of nitric oxide. Nitrogen and oxygen react at relatively high temperatures. Therefore, high temperatures and availability of oxygen are the two main reasons for the formation of NO_x. When proper amount of oxygen is available, the higher the peak combustion temperature, the more is the NO_x formed. The NO_x

concentration in exhaust is affected by mode of vehicle operating parameters. Air- fuel ratio and the spark advance are the two important factors which significantly affect NO_x emissions. At lean and rich air- fuel mixtures the NO_x concentration is comparatively low. Increasing the ignition advance will result in lower peak combustion temperatures and higher exhaust temperatures. This will result in high NO_x concentration in the exhaust.
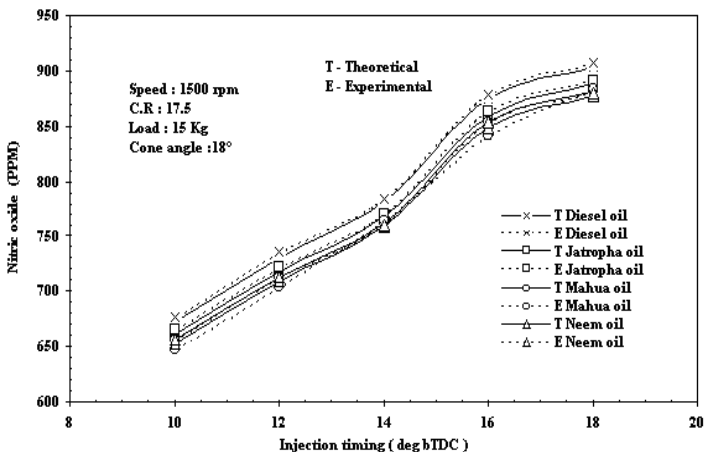

Figure 6.5 COMPARISION OF EXPERIMENTAL AND PREDICTED NITRIC OXIDE OF THREE DIFFERENT BIODIESEL WITH RESPECT TO INJECTION TIMING WITH

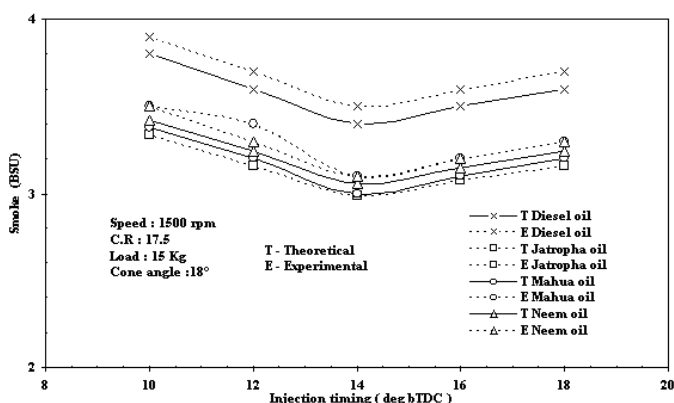## 6.6 Theoretical and experimental Smoke for various injection Timing

Figure 6.6 COMPARISION OF EXPERIMENTAL AND PREDICTED SMOKE OF THREE DIFFERENT BIODIESEL WITH RESPECT TO INJECTION TIMING WITH

Figure 6.6 expresses the comparison of predicted result of smoke emission for various injection timings for the three vegetable oil esters of jatropha, mahua, and neem. The injection timings tried are 18°, 16°, 14°, 12°and 10° crank angle. The minimum smoke formation is found for injection timing of 14° bTDC for all fuels. Increased premixed burning rate may be responsible for the decrease in smoke

level of the injection timing of 14°bTDC. However, advanced and retarded injection timing of 16° and 18° and 12° and 10° may result in the decrease of premixed burning rate. Hence, at 14° bTDC, as expected minimum smoke is observed.

7. CONCULUSION

1. The brake thermal efficiency is reduced about 3 % for jatropha, 4 % for mahua and 5 % for neem oil ester when compared to diesel. It is concluded that the brake thermal efficiency for vegetable oil ester is slightly decreased than diesel.

2.The specific fuel consumption for vegetable oil esters are increased about 8 %, 11 % and 13 % respectively for jatropha, mahua and neem oil ester when compared to diesel. It is concluded that the specificfuel consumption for vegetable oil ester is slightly increased than diesel.

3. The Carbon monoxide is reduced by 18 % for jatropha, 17 % for mahua, and 16 % for neem oil ester when compared to that of diesel. It is concluded that the carbon monoxide for vegetable oil ester is less when compared to diesel fuel.

4. The concentration of Hydro carbon is decreased by 18 % for jatropha oil ester, 16 % for mahua oil ester and 15 % for neem oil ester when compared to diesel fuel.

5.The formation of Nitric oxides is decreased by 1.8% for jatropha oil ester, 2.5% for mahua oil ester, and 3 % for neem oil ester when compared to diesel fuel.

6.The Smoke level is decreased by 12 %for jatropha oil ester, 11 % for mahua oil ester, and 10 % for neem oil ester when compared to diesel fuel.

7.Hence, it is concluded that in terms of performance characteristics, vegetable oil esters serve as a potential substitute for diesel fuel.

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