SIMULATION STUDIES OF A CI ENGINE FOR BETTER PERFORMANCE AND EMISSION USING DIESEL-DIESEL BIODIESEL BLENDS

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ABSTRACT

The present work describes a theoretical investigation concerning the performance of a four strokes compression ignition engine, which is powered by alternative fuels in the form of diesel and diesel biodiesel blends. The developed simulation model used to estimate the cylinder pressure, heat release rate, brake thermal efficiency, brake specific fuel consumption and engine out emissions. The simulation model includes Honerberg's equation heat transfer model, Zero dimensional combustion model for the prediction of combustion parameters and NO formation during combustion process using Zeldovich mechanism. Experiments were performed in a single cylinder DI diesel engine fuelled with a blend of pungam methyl ester for the proportion of 10%, 20% and 30% by volume with diesel fuel for validation of simulated results. It was observed that there is a good agreement between simulated and experimental results which reveals the fact that the simulation model developed predicts the performance and emission characteristics of any biodiesel and diesel fuel and engine specifications given as input.

Keywords: CI engine, Combustion, Emission, pungan methyl ester, Performance and zero dimensional combustion zone model.

Abbrevations

BTHE	:	brake thermal efficiency
BSFC	:	brake specific fuel consumption
HRR	:	heat release rate
PME10	:	blend of biodiesel and diesel having 10% biodiesel and remaining diesel
PME20	:	blend of biodiesel and diesel having 20% biodiesel and remaining diesel
PME30	:	blend of biodiesel and diesel having 30% biodiesel and remaining diesel

I. INTRODUCTION

There is intense research efforts are being carried out to identify the potential biodiesel to develop less polluting and more efficient engines.The numerical simulation was preferred to study the combustion behavior and emission of diesel engine because the experimental investigations were time consuming and costly affair. A computer based mathematical model have been proposed in the present work to study and understand the combustion and performance.The various models proposed are one zone and multi zone models.

In the present work the analysis of progressive combustion using zero-dimensional single-zone model is used and assumed the entire combustion chamber is considered as single zone.

In the final step using simplified assumptions, the engine heat transfer can be taken into account by means of empirical equations. Gas exchange processes are introduced in order to analyze the engine intake and exhaust. This analysis is called Actual Cycle Simulation (ACS). In this analysis the intake and exhaust processes using fluid dynamic equations, heat transfer and friction are included by means of empirical equations.By means of this step-by-step simulation, the values of pressure, volume and temperature at salient points in the cycle are calculated. Work output as well as thermal efficiency is evaluated. Many researchers have investigated the effects of oxygenated compounds on performance and emissions in a diesel engine [1] Ganesan.V. is the first who introduces the computer simulation is the process of formulating a model of a physical system representing actual processes and analyzing the same. Usually, the model is a mathematical one representing the actual processes

through a set of algebraic, differential or integral equations and the analysis is made using a computer. It considerable reduces the time-consuming tests by narrowing down the variables that must be studied.It helps in optimizing the engine design for a particular application, thereby reducing cost and time.

Benson [2] is the first person who introduces computers to simulate the flow through engine gas exchange system. John B. Heywood [3] analyzed Heat release rate based on first law of thermodynamics and state equation. He proposed to use more sophisticated models for the gas properties before, during and after combustion with accurate heat transfer model. Many researchers have investigated the effects of biodiesel on performance and emissions in a diesel engine [4-5]. Hountalas.D.T and Schwarz.V [6] used a quasi steady gas exchange model combined with a two zone combustion model to calculate performance and emission and optimize the engine operating parameters [7]. Bolem Siva Nageswara Rao[1] & V Ganesan the investigation has been carried out using the software STAR-CD. The effect of piston bowl shapes (bath tub, shallow and flat), compression ratios (11.2, 12 and 12.5), different intake port geometries (helical, tangential and straight) and engine speeds (1500, 2000, 2500 and 3000 rpm) on the turbulence kinetic energy and swirl ratio has been studied. From the analysis of the flow field, shallow bowl seems to be preferable from the point of swirl ratio at lower compression ratio, but at higher compression ratio flat bowl shape is preferable [8]. Joginder Singh Kaliravna 1, Mangesh Nimbalkar 1, Narendra Kumar Jain 1 & V. Ganesan Through statistical tools it is easy to interpret that what were the values of certain parameters when customers rated that particular engine as good, average or bad. Some of these tools used here are standard deviation (SD), coefficient of variance (COV) and lowest normalized value (LVN). The standard deviation (SD) and lowest normalized value (LNV) of indicative mean effective pressure (IMEP) are determined through the analysis of in-cylinder pressure data of the engine.

It has been observed that with second camshaft value of SD and LNV of IMEP is improved significantly which in turn shows improved combustion stability at idling in both fuel modes [9-10]. Based on the model description a computer program is developed using C program for the diesel cycle simulation to predict the cylinder pressure for the complete cycle and performance parameters can be effectively evaluated. A CFD studies on combustion and emission of diesel engine using Fluent is required to evaluate the emission data. The present work aims to develop a single zone zero dimensional model for DI diesel engines running with biodiesel to predict the engine performance and emission parameters. An experimental work is also conducted in a 4-stroke single cylinder compression ignition engine for comparison with simulated results. It is revealed from the results that the simulation model developed for biodiesel is in very good agreement with experimental results.

II. MATHEMATICAL TREATMENT

Conservation and state equations

This model is based on the first law of thermodynamics is applied. For a closed system, the first law of thermodynamics states that

$$dQ - dW = dE \qquad ...[1]$$

Where dQ is the heat loss to the chamber walls, dW is the network energy and dE is the change in internal energy.

The state equation is applied for unburned zone consists of pure air.

$$pV = M R_{mol} T \qquad ...[2]$$

where M is the number of kg-mols and V is the Cylinder volume. The volume at each crank angle V_{ca} is calculated using the relation

$$V(THETA)=vdisp*[cr/(cr-1)-(1-\cos\theta)/2+1/s-1/2*\sqrt{[(2l/s)\wedge 2-\sin^2\theta]} \qquad ...[3]$$

A. Fuel injection rate

The fuel injection rate is calculated using the relation,

$$m_{fi} = C_d\ (\pi/4)\ d_n^2\ (2\Delta P_n/\rho_f)^{0.5} \qquad ...[4]$$

Where ΔP_n is the pressure difference across the injector hole.

B. Heat transfer process

Heat transfer coefficient: The heat transfer coefficient between cylinder gases and wall is calculated by

Hohenberg's correlation given as:

H1=pow(69626.179688*0.001/1000000,–0.06);..[5]

H2=PATM*(VPHT+1.4); ...[6]

H3=pow(H2,0.8);

=pow(TEP,-0.4); ...[8]

HTC=0.13*H1*H3*H4; ...[9]

C. Combustion model

Heat release rate is calculated using wiebe heat release model (zero dimensional combustion model) is based on the exponential rate of the chemical reactions. In this model, it is assumed that all the fuel is injected is injected before the end of ignition delay period itself.

ROHR (dQc/dθ) = a*(m+1)*(Qav/COMBDUR) ∧m*[(θ-SOC)/COMBDUR] ∧ m ...[10]

exp [-a(θ-SOC)/COMBDUR] ∧ (m + 1)] ...[11]

where: a=6.908; m=2; Qav=QP2;

(Energy Release At Constant Pressure)

QP2= (hrp1 + N1*282800)*sf2 ...[12]

D. Ignition delay

Ignition delay time can be calculated as the difference between the time at which combustion starts and the time at which injection starts. Ignition delay is a complicated function of mixture temperature, pressure and equivalence ratio and fuel properties. The correlation proposed by wolfer's relation is used in the model.

Ignition delay = [0.44*exp (4650/AT2)] / (AP2 ∧ 1.19)

E. Combustion duration

The duration of combustion varies according to the empirical expression given by

Combustion duration = 40+5*(RPM/600-1) +166*(ycc/y-1.1) ∧ 2

F. NO_x Formation

To calculate the formation of nitric oxide inside each zone, a chemical equilibrium scheme is used to calculate the concentration of various components under equilibrium conditions.

In the present work extended zeldovich mechanism is used for calculation of NO formation. NOx formation is strong dependant on temperature and oxygen concentration. ...[13]

$$d(NO)/dt = 6*10^6/T^{0.5} \exp(-69090/T)[O_2]_e^{0.5}[N_2]_e$$

The equilibrium oxygen and nitrogen concentration in the burned zone are given by

$$[O_2]_e = f_{O2}(P,T) \quad [N_2]_e = f_{N2}(P,T) \quad ...[14]$$

III. EXPERIMENTAL DESCRIPTION

Specification are given in table 1. A Electrical dynamometer was used to apply the load on the engine. A water rheostat with an adjustable depth of immersion electrode was provided to dissipate the power generated. Tests were carried out at various loads starting from no load to full load condition at a constant rated speed of 1500 rpm. At each load, the fuel flow rate various constituents of exhaust gases such as Hydrocarbon (HC), carbon monoxide (CO) and nitrogen oxides (NO_x), were measured with a 5-gas MRU Delta exhaust gas analyzer. The analyzer uses the principle of non-dispersive infrared (NDIR) for the measurement of CO and HC emissions while NO_x measurement was by means of electrochemical sensors.

Table 1. Engine Specifications

Make	Kirloskar
Model	TAF 1
Type	Direct injection, air cooled
Bore × Stroke (mm)	87.5 × 110
Compression ratio	17.5:1
Cubic capacity	0.661 lit
Rated power	4.4 KW
Rated speed	1500 rpm
Start of injection	24° bTDC
Connecting rod length	220 mm
Injector operating Pressure	220 bar

Combustion analysis was carried out by means of an AVL pressure pick-up fitted on the cylinder head and a TDC encoder fixed on the output shaft of the engine. The pressure and the crank angle signals were fed to pentium personal computer. Various combustion

parameters like heat release rate, cumulative heat release rate and peak pressure and its accurance were obtained using data acquisition system.The properties of the fuel at various volumetric proportions as specified in table 2. The engine was first operated with diesel oil to generate the baseline data followed by pungam methyl esters and their blends such as PME10, PME20 and PME30 blends.

Table 2. Properties of the Different Fuel Blends

Details	Diesel	Raw pongamia	PME
Density kg/m^3	821	934	892
CV kj / kg	43000	35648	39149
Flashpoint °C	48	270	156
KV cst	3.52	45.62	5.405
CN	48-56	47	57

Table 3. Operating Parameters Considered in the Present Investigations

% Load	0, 25, 50, 75, 100
Speed (rev/min)	1500
Compression ratio	16:1, 17.5:1, 19:1
Injection Timing ° bTDC	21, 24, 27
Injection Pressure (bar)	200, 220, 240

IV. RESULTS AND DISCUSSION

The simulation model is capable of predicting various performance, combustion and emission parameters based on the any engine and any fuel specifications given as input. The developed simulation model can be useful for both diesel and any biodiesel blends. It serves as a versatile tool for a better understanding of the variables involved and their effect on engine performance and also it helps in optimizing the engine design for a particular application, thereby reducing cost and time. Experimental investigation were carried out at different compression ratio, injection timing and injection pressure and its details are mentioned in table 3. At injector opening pressure of 200 bar and 220 bar and injection timing of 21° bTDC and 24° bTDC and compression ratio of 17.5:1 and 16:1 were tried for PME10, PME20 and PME30 but from the investigation it was found that the performance was very poor. Further the engine were set to run at higher compression ratio of 19:1, advanced injection timing of 27° bTDC and higher injector opening pressure of 240 bar it arrives at the optimum range operating parameters for PME20. It was observed that PME20 it gives better performance for the optimized operating parameters. Finally the theoretical results of cylinder pressure, brake power, thermal efficiency, Specific fuel consumption and harmful pollutants such as nitric oxide, carbon monoxide and hydrocarbon are validated with experimental results. The predicted results of performance and emissions have been in close agreement with experimental results. The developed simulation model appears to be a useful tool for analyzing the diesel engine combustion accurately.

Figure 1. shows that the comparison of experimental and predicted cylinder pressure for three different biodiesel blends and diesel fuel with respect

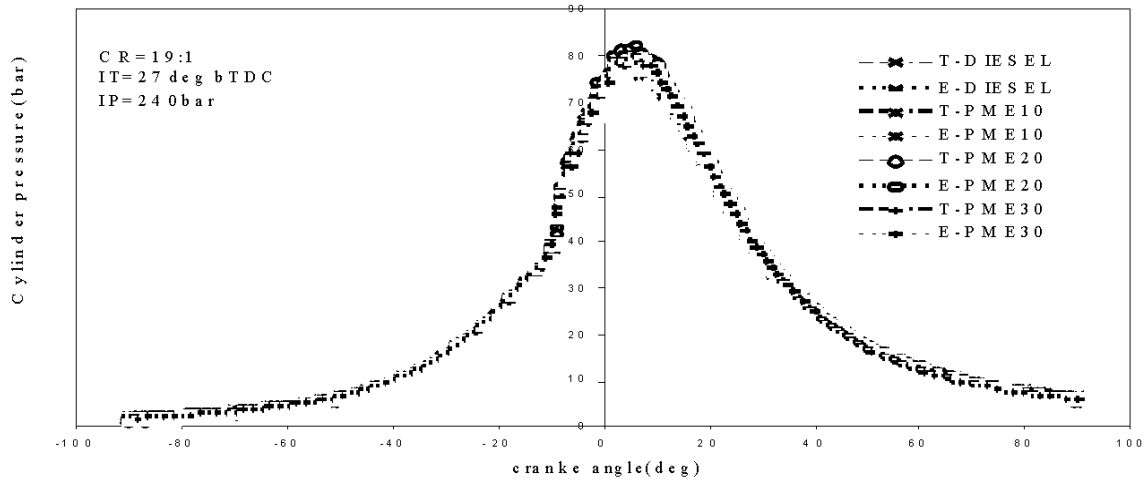

Fig. 1. Comparison of experimental and predicted cylinder pressure for three different biodiesel blends with respect to crank angle

to crank angle for various engine operating conditions. The agreement between the predicted values and experimental values of cylinder pressure are satisfactory.

Figure 2 shows the comparison of experimental and predicted heat release rates for diesel and biodiesel blends. From the results it was observed that the heat release rate increases during premixed combustion and decreases during diffusion combustion. The increase in heat release rate is due to the decrease in delay period. As revealed by the figure the combustion model predicts very well the heat release rates in the engine cylinder.

Figure 3. shows the comparison of experimental and predicted brake thermal efficiency for three different biodiesel blends with respect to brake power.The predicted values of brake thermal efficiency for all biodiesel blends are similar experimental brake thermal efficiency.

Figure 4. shows the brake specific fuel consumption predicted by the simulation model and the experimental one. It is seen that the fuel consumption of biodiesel blends is lower than that of diesel fuel operation which is due to better combustion in the cylinder. Due to the better consumption of the fuel BSFC have moderate improvement. The coincidence between the predicted and measured values is good.

Figure 5. shows the calculated and measured CO (in %) with respect to the loads. CO emissions are greatly dependant on the air fuel ratio relative to stoichiometric proportions. It is seen from the figure that the full load emission of CO for diesel is 0.49% and that of biodiesel blend with diesel is 0.2% predicted by the simulation model. The experimental values are 0.51% for diesel and 0.48% for DGM blends with diesel at full load. Due to better combustion of biodiesel blends with diesel, CO emissions present in the exhaust are reduced.

Figure 6. shows the predicted and measured NOx (in ppm) for various loads. The predicted NOx values are slightly higher than the measured ones. The maximum value of NOx is 1245 ppm reached at full load for the DGM in experiment due to high temperatures during combustion. NOx emission is 1115 ppm at full load with diesel and it is increased to 1238 ppm with the addition of diesel-biodiesel blend at full load predicted by the simulation model. The coincidence of simulation and experimental values are found to be good agreement.

Figure 7. shows the comparison of HC with respect to the load predicted by the simulation model and the experimental one. Biodiesel blend addition decreases the HC values with respect to increase of load as shown. Hydrocarbon emission is decreased by about 3% predicted by the simulation model. Diesel fuel

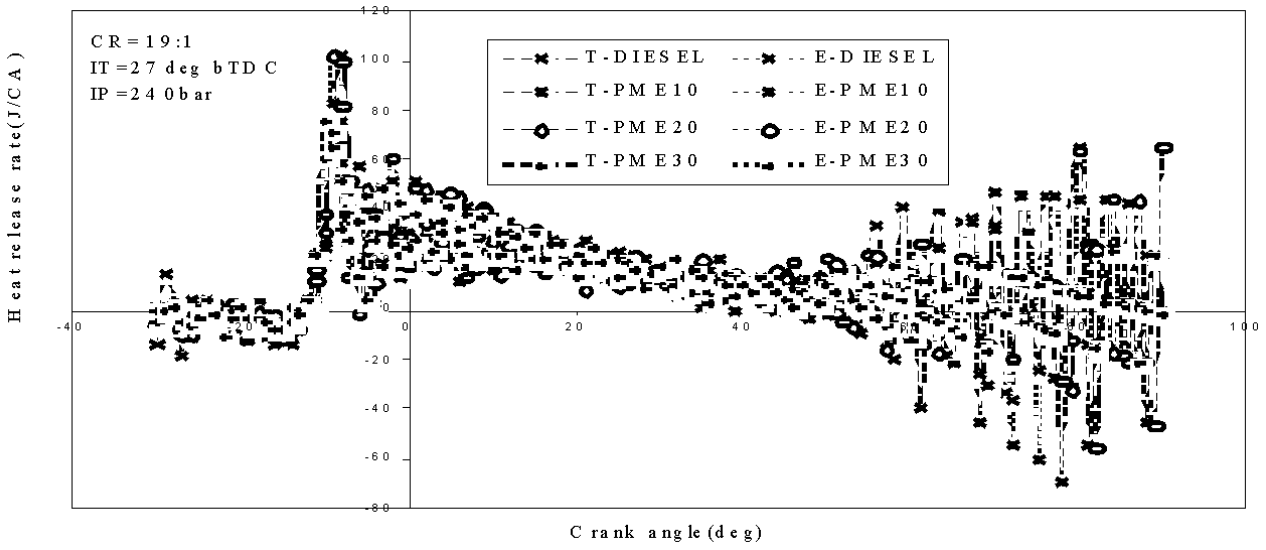

Fig. 2. Comparision of experimental and predicted heat release rate for three different biodiesel blends with respect to crank angle

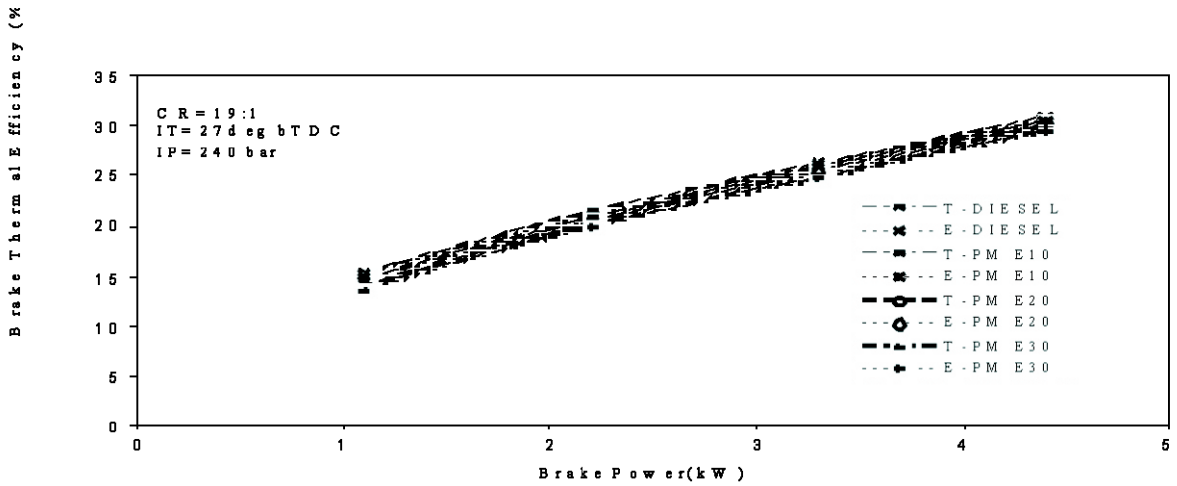

Fig. 3. Comparison of experimental and predicted brake thermal efficiency for three different biodisel blends with respect to brake power

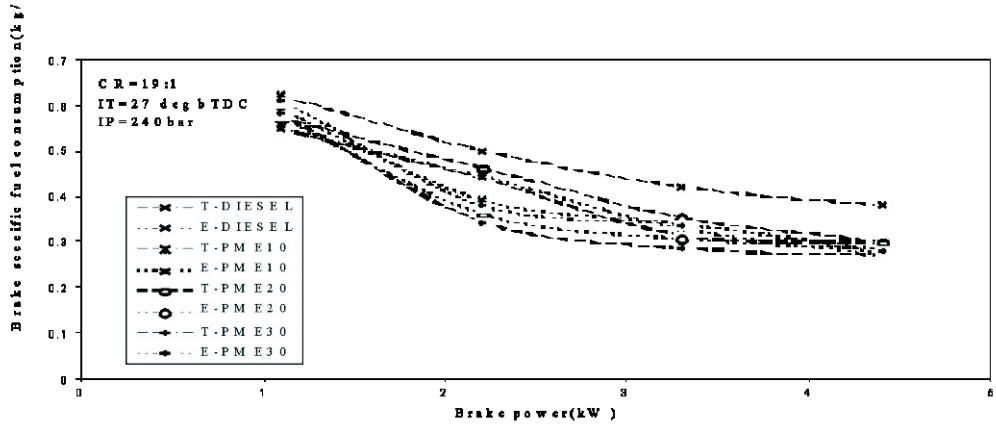

Fig. 4. Comparison of experimental and predicted brake specific fuel consumption for three different biodiesel blends with respect to brake power

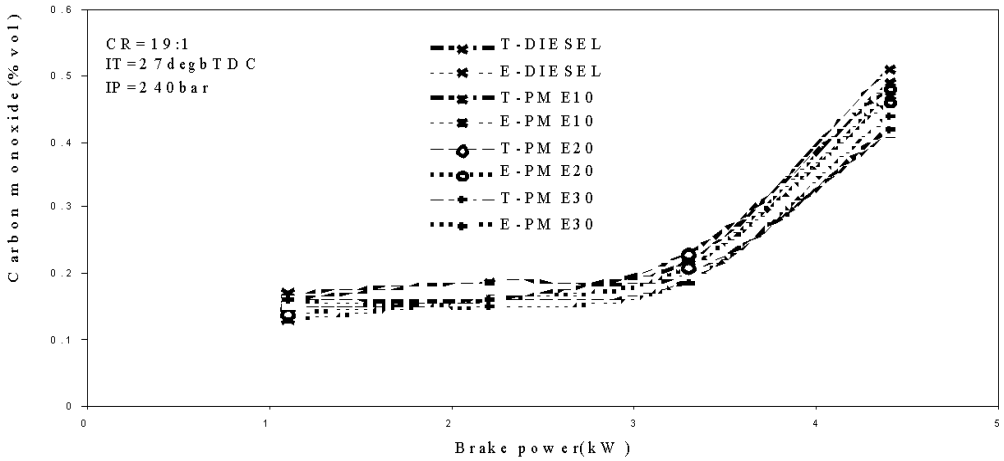

Fig. 5. Comparison of experimental and produced carbon monoxide for three different biodiesel blends with respect to brake power

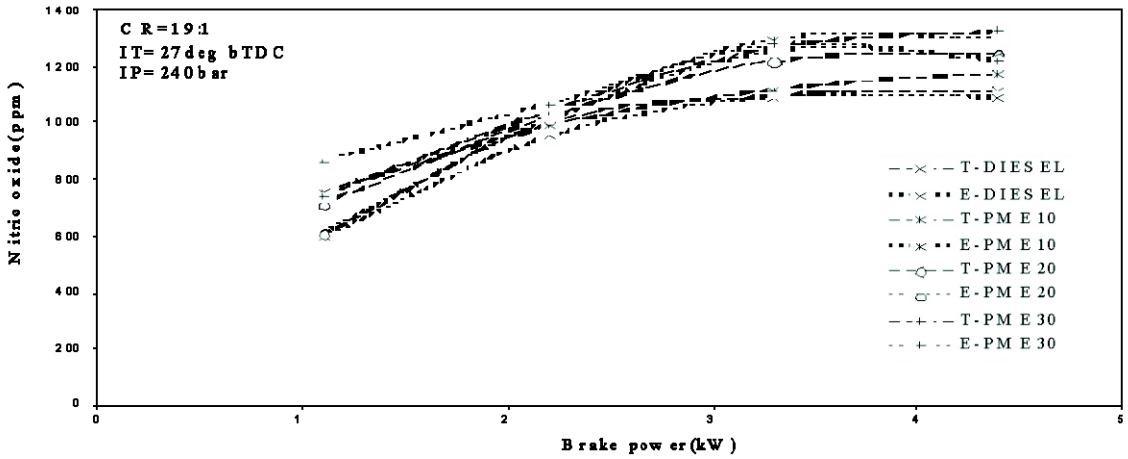

Fig. 7. Comparision of experimental and predicted hydrocarbons for three different biodiesel with respect to brake power

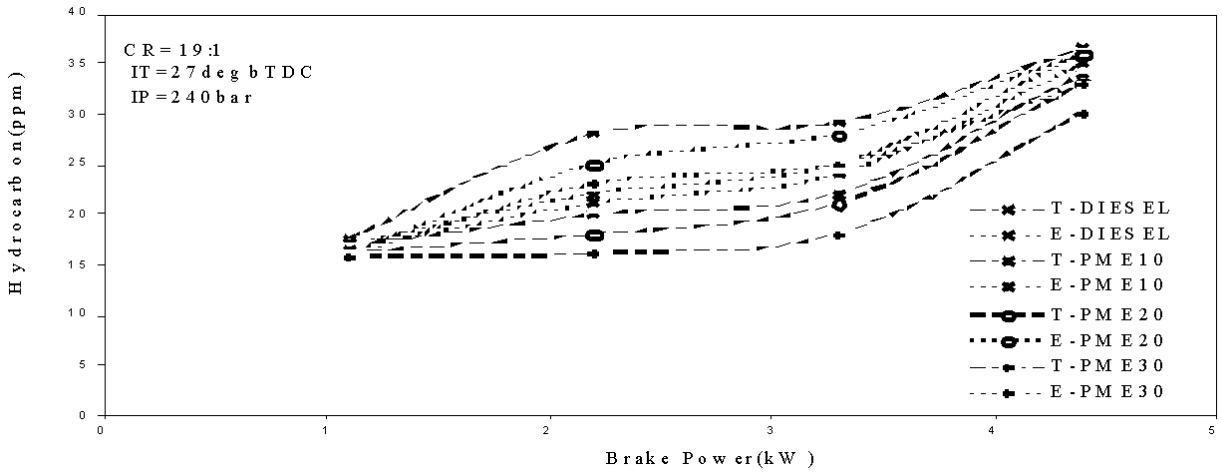

Fig. 6. Comparison of experimental and predicted nitric oxide for three different biodiesel blends with respect to brake power

contains chain and substituted types of aromatic compounds, burns more slowly and produces higher amounts of unburnt hydrocarbons. The HC emissions of biodiesel blends is reduced due to better combustion of the fuel.

V. CONCLUSION

In the present work a single zone zero dimensional model for direct injection diesel engine has been developed to predict the performance characteristics.

The predicted results of performance and emissions have been in close agreement with experimental results. The developed simulation model appears to be a useful tool for analyzing the diesel engine combustion accurately.

Thus the developed computer model can predict the various performance and emission parameters of any vegetable oilesters with minimum inputs such as density, calorific value, chemical formula and engine specification.

Moreover the developed model can be considered as an efficient tool to calculate the effect of engine operating parameters such as compression ratio, injection timing and injection pressure.

Finally it is concluded that the predictions revealed fair agreement with experimental results.Hence the developed simulation C-programming model is validated of any given vegetable oil esters at various operating parameters of diesel engine.

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