

Numerical Investigation of Emissions of NO_x and CO using Biofuel in Compression Ignition Engine

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Abstract – Biofuel has been discovered as alternative fuel due to potential of being environmentally friendly. Experimental analysis is limited by high cost, time, and evaluation of engine parameters. This study has numerically investigated potentials of biofuel blends as alternative to diesel fuel in terms of reduction in emissions of NO_x, and CO using MATLAB interface. Test engine was chosen and various biodiesels were identified. Developed models were used to obtain various emission characteristics in respect to each of the biodiesels at various blends. Three biodiesels (Jatropha, Waste cooking oil (WCO) and Algae) were explored and investigated with relative increase in NO_x emission and ranges from Waste Cooking Oil, Jatropha and Algae as 0.0013, 0.0014 to 0.0015 in increasing order respectively at B20 blends of each of the biodiesels. Respective CO emission reduction in Waste Cooking Oil, Jatropha and Algae with values, 0.0064, 0.0091 and 0.0085 in order for B20 blends of each of the biodiesel. For all the emission gases with better properties using biodiesel, significant decrease occurred at lower blends. The study revealed that blend of B20 Waste Cooking Oil biodiesel is environmentally suitable as biofuel for combustion in diesel engine.

Keywords – Biodiesel, Jatropha, Waste Cooking Oil, Algae, NO_x Emission, CO Emission.

I. INTRODUCTION

Energy is essential resource for necessary human existence and development. In 21st century, availability of sustainable energy sources is one of the global crisis confronting all and sundry. Fossil fuels are of great importance because they combust to produce significant amounts of energy to power machines and machinery and also to produce electricity. Many aspects of everyday life rely on fuels, in particular the transport of goods and people. Main energy requirements are met with resources from fossil fuels such as petrol oil, coal and natural gas. Many industries use diesel engine and equipment for production process. In transportation sector, private vehicles, buses, trucks, and ships also consume significant amounts of diesel and gasoline. This leads to strong dependence of everyday life on fossil fuels.

Akinwonmi et-al, 2023 highlighted that efforts has been directed towards reducing the use fossil fuel and coal over the last few decades, because of the emission of NO_x and other hazardous gases and that the fossil fuels and coal also contributes to the continuous rise in the global crude oil prices by many forms of unrest in major producing countries which is a major factor of economic importance.

However, the growth of the population is not covered by domestic crude oil production (Daning *et al.*, 2012). Fossil oils are fuels which come from ancient animals and microorganisms. Fossil fuel formation requires millions of years. Thus, fossil oils belong to non-renewable energy sources. An increase of the oil price often leads to economic recessions, as well as global and international conflicts. In addition, the emission produced by the combustion of fossil fuels also contributes to the air pollution and global warming. Most countries also experience more and more international pressure on global warming issues. Hence, renewable and clean

alternative fuels have received increasing attention for current and future utilization. Appropriate blend of LNG with R134a gives a suitable replacement of R134a in VCR (Vapour Compression Refrigeration) systems eventually revealed reduced environmental pollution without significantly affecting the optimum operational efficiency and Coefficient of Performance of VCR with considerably low Global Warming Potential of between 32 and 1300 (Adeaga, 2023).

Biodiesel as one promising alternative to fossil fuel for diesel engines has become increasingly important due to environmental consequences of petroleum-fueled diesel engines and the decreasing petroleum resources. Biodiesel can replace fossil fuel as “clean energy source”. It protects the environment by reducing CO₂, SO₂, CO and HC (Daning *et-al.*, 2012).

To get the best of the alternative fuel (biodiesel), researchers has done lots of analysis especially experimental analysis which incorporate a large set up housing a dynamometer, sampler, smoke gas analyzers and other components. The experimental analysis is said to have its limitations as biodiesels could not be compared to conventional fuels simultaneously. That is, apart from the process being expensive and tedious it is very much time consuming. Also, limited number of biodiesel samples can only be analyzed with accurate results not guaranteed. The area of emission of biodiesel is the most sought-after knowledge talking about biodiesel. As a result, a lot of researchers have made emission characteristics their focal point while investigating the several benefits of using biodiesel as against diesel fuel in diesel engine. Some of their findings and submissions are highlighted;

NO_x emission increases with load on engine for all tested biofuels. In this study, simulated biofuels revealed that emissions from NO_x are lower to that of fossil fuel, due to calorific value, viscosity of biodiesel fuel, etc. (Rajak and Verma, 2018). Furthermore, NO_x emission for most biodiesel is relatively higher than conventional diesel (Nirmala *et al.*, 2019). The research results related to the emission of NO_x vary to a certain degree. For different engines and different testing conditions, biodiesel of various origin and different biodiesel/diesel mixtures, NO_x emission can be lower for around 20% but also higher for around 26% (Nikolić *et al.*, 2018).

A study by Nabi *et-al.*, showed that biodiesel play important role in overall emission from diesel engines. According to the study, CO emissions with biodiesel were found to be 11-51% lower relative to the reference diesel. This study based on numerical analysis of biodiesel fuel using a mathematical model which simulates a one-cylinder internal combustion diesel engine with various biodiesel samples (Jatropha, Waste cooking oil and algae) with varying blends and as such various emission characteristics of the engine was analyzed.

II. SCOPE OF INVESTIGATION

This investigation focused on the numerical analysis of various emission effects (CO and NO_x) of compression ignition engines (diesel engine) when fueled with biodiesel (both pure and blends). Dual-Zone thermodynamic model was developed with MATLAB programme to investigate performance and emission level of engine when biofuel is Jatropha, Waste cooking oil and Algae at different blends. However, the model where used to evaluate pressures, temperatures and other similar properties for a predetermined crank angle step. Equations of heat-transfer rate and ignition-delay were infused into the numerical model and combustion model respectively using empirical equations. The computation of heat transfer rate were accomplished taking into consideration premixed and diffusive combustion. Table 1 below shows the technical specification of the test e-

-ngine.

Table 1. Specifications of test.

Engine Parameter	Specification
Make of Engine	Legion Brothers
Type of Engine	Single-cylinder, 4-stroke, Direct CI engine
Bore of cylinder	80 mm
Stroke of piston	110 mm
Connecting Rod	235 mm
Compression Ratio	17.5
Maximum Power output	3.7 kW
Dynamometer	Eddy current
Mode of cooling	Liquid Water
Type of Nozzle	Multiple holes
Number of Nozzle holes	3
Orientation	Vertical
Valve Orientation	Overhead

Various biodiesels which were investigated are; Waste cooking Oil (WCO), Jatropha and Algae biodiesels and at various blends with diesel fuel. For WCO, WC100, WC10 and WC50 (representing pure WCO, 10% blend with diesel fuel and 50% blend with diesel fuel respectively). Similarly, J₁₀₀, J₁₀ and J₅₀ for Jatropha biodiesel and A100, A10 and A50 for that of Algae biodiesel. Tables 2-4 displays the standard fuel properties of waste cooking oil, Jatropha and algae biodiesels.

Table 2. Properties of Waste cooking oil as biodiesel and diesel blends.

Fuel	Density	Viscosity at 40°C	Flash Point	Fire Point	Cetane No
Diesel	0.835	1.385	42°C	68°C	45-50
B ₂₀	0.84	1.407	43°C	65°C	52
B ₄₀	0.845	1.44	53°C	76°C	48
B ₆₀	0.85	1.467	82°C	104°C	46
B ₈₀	0.87	1.556	84°C	124°C	44
B ₁₀₀	0.89	2.72	186°C	208°C	41
Biodiesel	-	1.9-6.0	>130	-	41 mm
Standard					
(ASTM D					
6751-02)					
DIN EN	0.86-0.90	3.5-5.0	>120	-	51 mm
14214					

Table 3. Properties of Jatropha Oil and blends.

Fuel	Viscosity at 40°C	Flash Point	Specific Gravity	Calorific Value (Kjkg ⁻¹)
B ₁₀	4.1	74	0.865	43647
B ₂₀	4.2	79	0.868	43093
B ₃₀	4.3	86	0.872	42207
B ₄₀	4.5	92	0.876	41542
B ₅₀	4.6	98	0.882	40877
Diesel	4.0	70	0.853	44755

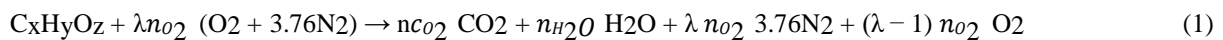
Table 4. Properties of Algae oil and blends as biodiesel.

Fuel	Density (g/cm ³)	Viscosity at 40°C	Flash Point mm °C	Calorific Cetane Value	No
ASTM	D-4052	D-445	D-92	D-240	D-613
Diesel	0.81	4.7	169	42,040	52
Algae-Oil	0.83	5.0	181	42,861	50
B-100	0.83	5.3	182	43,202	51
B-75	0.83	5.6	179	45,760	51
B-50	0.83	5.6	180	45,894	53
B-20	0.83	4.7	179	43,665	54

III. FORMULATION OF MATHEMATICAL MODEL

The first law of thermodynamics and equation of state were explored to formulate model which considered two-zones within combustion chamber. During the start of compression, the mole of different species are considered to coexist with oxygen, nitrogen from intake and carbon dioxide, water (gaseous), nitrogen and oxygen from the residual gases.

The reaction equilibrium is altered by combustion or ignition improvers in the following ways:



It is to be noted that some proportions of oxygen gas required by chemical reactions are bound within molecules of fuel. This makes moles of different reactants for biodiesels in each blend to be expressed as;

$$TMSR_1 = 1 + 4.773\lambda(i_1 + \frac{j_1}{4} - \frac{k_1}{2}) \quad (2)$$

While total amount in molecules of reactant of pure diesel in the blend is expressed by:

$$TMSR_2 = 1 + 4.773\lambda(i_2 + \frac{j_2}{4}) \quad (3)$$

The total amount in moles of species in reactants is expressed by:

$$TMSR = TMSR_1 + TMSR_2 \quad (4)$$

The total amount in molecules of products for biodiesel in the blend is expressed as:

$$\text{TMSP}_1 = i_1 + \frac{i_1}{2} + \left(3.773\lambda \left(i_1 + \frac{i_1}{4} - \frac{k_1}{2}\right)\right) + (\lambda - 1) \left(i_1 + \frac{i_1}{4}\right) - \lambda \frac{k_1}{2} \quad (5)$$

The total amount in molecules of products for pure diesel in the blend is given by:

$$\text{TMSP}_2 = i_2 + \frac{i_2}{2} + \left(3.773\lambda \left(i_2 + \frac{i_2}{4}\right)\right) + (\lambda - 1) \left(i_2 + \frac{i_2}{4}\right) \quad (6)$$

Therefore, the addition of equation (5) and (6) gives the total amount in molecules of products thus;

$$\text{TTMSP} = \text{TMSP}_1 + \text{TMSP}_2 \quad (7)$$

Recalling the first law of thermodynamics for close system we have;

$$\delta Q - \delta W = \delta E \quad (8)$$

where, δQ is heat-loss by conduction at walls of combustion chamber, δW is network done and δE is change in internal energy.

State equation applied to unburned zone consists of pure air and determined by.

$$pV = MR_{\text{mol}}T \quad (9)$$

where M is number of kg-mol. and V is volume of cylinder. Then, volume V_{ca} at each crank-angle δ , is calculated by;

$$V_{\text{ca}} = V_{\text{cyl}} + \left(\frac{\pi d^2}{4}\right) r \left(1 + \frac{1}{\alpha} - \cos\varphi - \sqrt{\frac{1}{\alpha^2} \sin^2\varphi}\right) \quad (10)$$

Where, V_{cyl} = clearance volume of the cylinder and α = ratio of crank-radius to piston-rod length Applying First law of thermodynamics to unburned zone we have:

$$\delta E = \delta Q - p\delta V - h_a dm_a \quad (11)$$

The burning zone will not only receives mass from the air-zone, but also enthalpy from the fuel which is ready to be burned in time step. So, the first law of thermodynamics for the burning zone becomes:

$$\delta E = \delta Q - p\delta V + h_a dm_a + h_f dm_f \quad (12)$$

Total internal energy was determined by:

$$E(T) = M \sum [x_i e_i(T)] \quad (13)$$

$$e_i(T) = h_g(T) - RT \quad (14)$$

Where, $h_g(T)$ is specific enthalpy of gas species. First law of thermodynamics for combustion in time step 'dt' is:

$$\delta Q - \delta W = E(T_2) - E(T_1) - \delta m_f Q_{\text{vs}} \quad (15)$$

$$E(T_2) = E(T_1) - \delta Q + \delta W + \delta m_f Q_{\text{vs}} \quad (16)$$

If $f(E)$ is greater than accuracy required, the new value of T_2 is calculated thus:

$$T_2 = T_2' - \frac{f(E)}{m C_v T_2'} \quad (17)$$

Unburned fuel temperature is evaluated by the equation,

$$T_u = T_{SOC} \left(\frac{P}{P_{SOC}} \right)^{\frac{\gamma-1}{\gamma}} \quad (18)$$

Rate of Fuel Injection

With the exit of nozzle being constant during injection process, the mass of fuel for crank-angle is;

$$m_{fi} = C_d \frac{\pi}{4} d_n^2 \left(\frac{2\Delta P_n}{\rho_f} \right)^{0.5} \quad (19)$$

Rate of Heat Transfer

It is difficult to compute rate at which heat is transferred to cylinder because of complexities of flow patterns within combustion chamber. Therefore rate of heat transferred is computed by accounting for convection and radiation mode, hence the application of Annad's correlation to estimate the rate of heat transferred from walls of the cylinder to combustion chamber as in eq. (20);

$$\frac{dQ}{dt} = A_n K \frac{Re^b}{D} (T_g - T_w) + C_h (T_g^4 - T_w^4) \quad (20)$$

Where, A_n , b , are constants in convective heat transfer, and C_h , is estimated via mass of fuel injected. First term in (20) represents Prandtl's number of the gases in cylinder as a constant at 0.7. It is to be noted that a major parameter influencing the rate of convective heat transferred is Reynold's number. Second term is radiation term which assumes a grey body radiation. The values of constants are estimated using line of best fit from experimental and predicted values. T_g denotes gas mean-temperature and T_w taken as 450 k denotes cylinder wall temperature.

Formulation of Combustion Model

Process of combustion is chemical reaction of fuel and oxygen to produce considerable quantity of heat. Modeling combustion process helps to describe and predict the rate of release of heat and emissions. However, two-zone combustion model which separate combustion chamber to pre-mixed zone and diffusive zone was explored. Estimation procedures hinges on integrating the first law of thermodynamics with state-equation assuming perfect gas for each zone separately. This method assumed that mixing-rate is dependent on the total surface area of fuel spray formed by droplets. Furthermore, fuel injected into burning zone mixes with entrained air in diffusion process. The rate of burning of fuel is determined by Arrhenius-type expression. However, the pre-mixed rate of combustion is limited by the rate of reaction.

Preparation rate equation is given by:

$$P = K (m_i)^x (m_u)^y (P_{O_2})^m \quad (21)$$

Constraints values x , y and m of combustion process are set to 0.33, 0.67, and 0.4 respectively. Partial pressure of oxygen (P_{O_2}) of (21) allows effects of available oxygen on mixing. Therefore,

$$m_i = n\rho \left(\frac{1}{6} \right) \pi D_0^3 \quad (22)$$

$$m_{ij} = n\rho \left(\frac{1}{6} \right) \pi D^3 \quad (23)$$

Where r is density of droplet of fuel, D_0 is initial droplet diameter, D is diameter of fuel droplets at instant, Number of droplets of fuel is denoted by 'n' while the overall surface area at any instant is ' $n\pi D^2$ '.

Assuming that velocity of molecules of gas varies as square-root of temperature also density of Oxygen is proportional to partial pressure divided by temperature. Unburned fuel is estimated by;

$$U_{b_f} = \int (P-R) d\alpha \tag{24}$$

Rate of preparation is denoted by P while rate of burning per degree of crank-angle is denoted by R. Period of reaction kinetics delay depend on duration and N is a divisor to determine the rate of reaction per crank-angle. The rate of crank-angle is given as

$$R = \frac{k'}{N\sqrt{T}} e^{\frac{-\alpha}{T}} \int (P - R) d\alpha \tag{25}$$

where, k' is a constant and α is an index and the combustion rate is given by:

$$\begin{aligned} \frac{dm_f}{d\phi} &= P \text{ if } R > P \\ \frac{dm_f}{d\phi} &= R, \text{ if } R < P \end{aligned} \tag{26}$$

The time taken to burn prepared fuel is negligibly small when compared to time taken for preparation during combustion process and it occurs at high temperatures. Therefore, the burning period to release the heat is taken as a measure of preparation rate.

Formation of Nitrogen-Oxide (NO)

Nitrogen-Oxide is formed by process of combustion with influence at high temperature fuel in required quantity of Oxygen. Formation of Nitrogen-Oxide occurs through oxidation of atmospheric Nitrogen. The formation of Nitrogen-Oxide occurs via three principal sources:

- i. Dissociation of molecular air-nitrogen molecular air-nitrogen known as thermal NOx,
- ii. Hydrocarbon fragments attack on the air-nitrogen prompt NOx formed, and
- iii. Nitrogen-Oxide contains components in the fuel.

During combustion process, cylinder pressure rises and burned fuel are compressed to higher temperature thereby rendering formation of Nitrogen-Oxide negligible. Thermal NOx formed in the burnt gases always dominates when in comparison to formation of Nitrogen-Oxide in the flame front.

Initial rate of formation of Nitrogen-Oxide is;

$$\frac{d(NO)}{dt} = \frac{6.10^6}{\sqrt{T}} \exp\left(\frac{-69090}{T}\right) [O_2]_e^{0.5} [N_2]_e \tag{27}$$

Formation of NOx depend on value of temperature and concentration of oxygen in combustion phase.

Equilibrium concentration of Oxygen with Nitrogen gases at burned zone is;

$$[O_2]_e = f_{O_2}(P, T) \quad [N_2]_e = f_{N_2}(P, T) \tag{28}$$

Smoke Formation

The power output in diesel engine is reduced in presence of solid carbon (smoke) at exhaust of combustion chamber. Smokes formation occurs when combustion process is heterogeneous and governed by radical mechanisms. The rate of smoke formation is estimated using semi-empirical model thus;

$$\frac{dm_{sf}}{dt} = A_f m_f P^{0.5} e^{\left(\frac{-E_{sf}}{R_{mol}T}\right)} \quad (29)$$

However, oxidation of soot was predicted, assuming second order reaction between soot, m_b , in oxygen, O:

$$\frac{dm_b}{dt} = A_b m_b \left(\frac{P_{O_2}}{P}\right) P^2 \exp.\left(\frac{-E_b}{R_{mol}T}\right) \quad (30)$$

Where, P_{O_2} is partial pressure of Oxygen. Energies of activation are set to the values of $E_{sf} = 1.25 \times 10^4$ kcal./kmol, $E_b = 1.40 \times 10^4$ kcal./kmol. Estimated smoke value in measured smoke within exhaust gas were explored to estimate the values of A_f and A_b .

Determination of net value of smoke formation is expressed as:

$$\frac{dm_s}{dt} = \frac{dm_{sf}}{dt} - \frac{dm_b}{dt} \quad (31)$$

IV. RESULTS AND DISCUSSION

Nitrogen-Oxide and NO_2 emissions generated during combustion process were due to presence of oxygen and nitrogen at high temperatures. It has been reported that regular fossil fuel has trace amount of nitrogen molecules in their structure but the Nitrogen content within the fuel has little effect on Nitrogen-Oxide emission. NO_x emission depend on combustion cylinder temperature, pressure, air-fuel ratio during combustion process and properties such as viscosity, density and flash point also affect the NO_x emission.

Emissions of NO_x by diesel engine is attributed to flame temperature, oxygen percentage within fuel and time of reaction process. Biodiesel blends of B-20 (Jathropa) gives a favourable increase in NO_x though that of diesel fuel is a little reduced. In figure 1 below the NO_x emissions values at different blends of the various biodiesels (Jathropa, Waste cooking oil and algae) with diesel fuel serving as the reference on the vertical NO_x axis. It was observed that, as the blends increases (i.e. addition of more biodiesels to the diesel fuel) NO_x values increase. This trend is in line with the findings of Muralidharan and Vasudevan (2011).

Table 5. NO_x emissions of each of the biodiesel at different blends.

Fuels	Biodiesel Jatropha (g/L)	WCO (g/L)	Algae (g/L)
Diesel	0.001	0.0010	0.0010
B-10	0.0012	0.0012	0.0012
B-20	0.0014	0.0013	0.0015
B-30	0.0020	0.0019	0.0022
B-40	0.0022	0.0020	0.0025
B-50	0.0030	0.0025	0.0030
B-60	0.0050	0.0035	0.0055
B-70	0.0075	0.0055	0.0070

Fuels	Biodiesel Jatropha (g/L)	WCO (g/L)	Algae (g/L)
B-80	0.0080	0.0060	0.0080
B-90	0.0090	0.0065	0.0090
B-100	0.0120	0.0068	0.0015

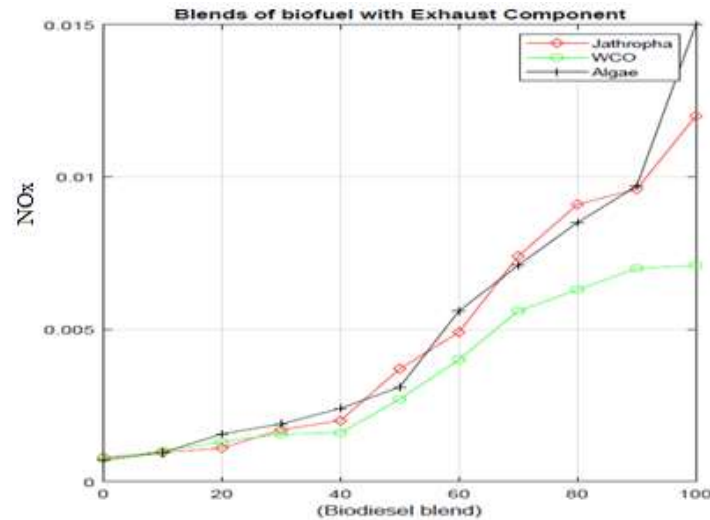


Fig. 1. Plot of NO_x emission against Biodiesels blends.

Table 5 shows the increase in values of NO_x for each of the biodiesels and at their various blends. This implies that biodiesels (Jatropha, Waste cooking oil and Algae) at various blends decreases NO_x emissions, hence suitable in terms of environmental protection.

Since blend 20 of most biodiesels are the highest blend that favors application in diesel engines without having to change or adapt the engine into a form different from its original, it will be made a reference point, to choose the biodiesel with the most suitable characteristics in terms of NO_x emission. From table 4.1, the biodiesel with the least emission NO_x is waste cooking oil biodiesel with a value of 0.0013g/L which make it the most suitable since it is closer to the value of conventional diesel, closely followed by Jatropha which has 0.0014g/L, and algae is the least favourable with a relatively high value of NO_x formation (0.0015mg/L).

CO Emission

Carbon monoxide is produced when combustion reactions are not fully completed, either due to the lack of oxygen or due to inadequate mixing. Low excess Oxygen increases CO formation. The higher the air ratio and the better the mixing, the lower is the CO emission. Biodiesel as a fuel known to be rich in Oxygen have a lower emission of this CO gas, which is clearly shown in figure 2. In figure 2, in the beginning a blend of biodiesel has small effect on the amount of CO emitted, though reduction occurred. Along the line the reduction became more pronounced, and it is within this range of blend 20 to blend 40 that the greatest fall occurred. The rate of this reduction reduced and continued uniformly up until blend 70, and from this point the reduction continued to dwindle. Although at pure biodiesel, it has the best CO emission characteristics.

Jatropha seemed to follow a trend similar to that of algae, though it maintains uniform decline rate at intervals than algae biodiesel.

Waste cooking oil biodiesel picked its origin alongside others from conventional diesel, it maintained the best CO emission characteristics till blend 70. The decline was a sharp one right from the first instance at blend 10, but as it tends towards achieving its original form especially at the final blends a little increase in CO emission characteristics is observed. The results of Karthikeyan *et- al.*, (2020) were in agreement with this present study.

Since blend 20 of most biodiesels are the highest blend that favors application in diesel engines without having to change or adapt the engine into a form different from its original, it will be made a reference point to choose the biodiesel with the most suitable characteristics in terms of CO emission. From Table 6 below, at this blend Jatropha has the highest CO value of 0.0091 g/L, followed by algae biodiesel which has the value, 0.0085 g/L and then waste cooking oil with 0.0064 g/L CO produced. This indicates that waste cooking oil has the best CO emission characteristics.

Table 6. CO emission as biodiesel at different blends.

Fuel	Jatropha (g/L)	WCO (g/L)	Algae (g/L)
Diesel	0.0100	0.0100	0.01
B-10	0.0099	0.0070	0.0098
B-20	0.0091	0.0064	0.0085
B-30	0.0073	0.0050	0.0071
B-40	0.0049	0.0040	0.0056
B-50	0.0038	0.0028	0.0031
B-60	0.0028	0.0017	0.0024
B-70	0.0018	0.0015	0.0019
B-80	0.0011	0.0013	0.0015
B-100	0.0005	0.0018	0.0008

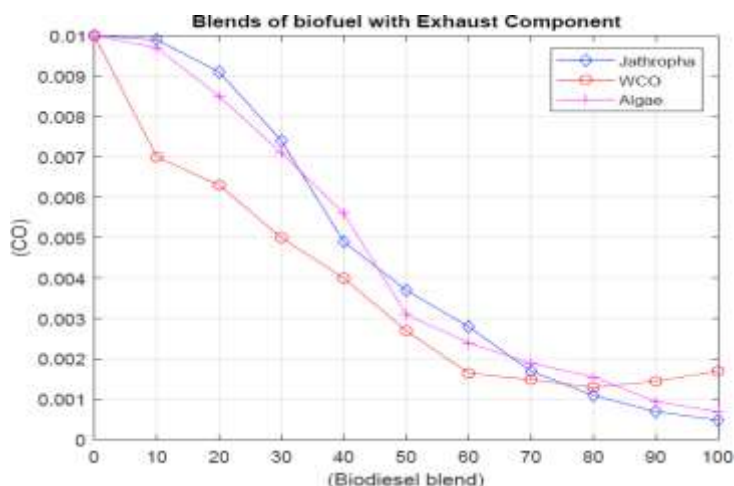


Fig. 2. Plot of CO emission against biodiesels at different blends.

V. CONCLUSION

A 2-dimensional model simulation using MATLAB was explored to simulate the emission characteristics of conventional diesel and biodiesel at different blends. The emission characteristics (NO_x and CO) were evaluated

in g/L at every 10% increase of the blend of various biodiesel. The biodiesels considered in this project work are: Jatropa, Waste Cooking Oil and Algae and the conclusions are summarized below:

- i. Overall, biodiesels are of better emission characteristics because they exhibit significant decrease in CO.
- ii. The higher the blend of the biodiesel, the better the emission characteristics and so they are better at blend 100 i.e., pure form of the biodiesel. But due to the fact that blend 25-100 are not good for diesel engine output, blend 20 is then chosen to be the best blend.
- iii. For all the emission gases with better properties using biodiesel, their most significant fall occurs at the early blends and a few of these gases show no consequential change at later blends (beyond blend 60) Waste cooking oil was discovered as the best of analyzed biodiesel at blend 20. It is therefore the most suitable of the biodiesels analyzed to take the place of conventional diesel in diesel engine.

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