Performance and Emissions of a Diesel Engine Fueled with a Biofuel Extracted from Jatropha Seeds

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ABSTRACT— With increasing environmental consciousness and the depletion of fossil petroleum fuels, biofuel is being promoted as a green alternative to diesel fuel. Biofuel lowers pollution emissions without having to make major changes to existing engines. Biodiesel is different in its chemical and physical properties based on the types of material or oils used in its preparation. In this research, biodiesel from Jatropha seeds is used as a fuel to run a four-stroke, single-cylinder diesel engine using various blend ratios (B20, B40, B60, B80, and B100) and different loads (no-load, 25% load, 50% load, 75% load, and full load). Performance parameters were evaluated such as brake specific fuel consumption (BSFC), brake thermal efficiency (η_{th}), volumetric efficiency (η_v), and exhaust gas temperature, while exhaust emissions include specific emissions of HC, CO, CO_2, and NOx. Biodiesel blends (B20%) reduce brake thermal efficiency by 6% and volumetric efficiency by 3% while brake-specific fuel consumption increases by 15%. As the blend increases, the emission of NOx increases, for pure Jatropha biodiesel NOx increases by 30% compared to pure diesel. Moreover, for 100% blend, the emissions of CO_2 decrease by 35% and CO decreases by 55% compared to diesel fuel. As biodiesel proportions in fuel blends increase, HC and smoke emissions decrease. Results show that biodiesel made from non-edible oils, such as Jatropha, might be a viable substitute for diesel fuel.

Keywords— Jatropha; Transesterification; Biodiesel Properties; Biodiesel Blend; Performance; Exhaust Emissions;

Abbreviations

B20%	20% Biodiesel + 80% Diesel Fuel
BSFC	Break Specific Fuel Consumption
C	Ceiba Pentandra
CME	Castor Methyl Ester
EGT	Exhaust Gas Temperature
FAME	Fatty Acid Methyl Ester
J	Jatropha Curcas
JBT	Turpentine Oil and Jatropha Biodiesel
PM	Particulate Matter

Nomenclatures

CO	Carbon Monoxide
CO_2	Carbon Dioxide
C_2H_5OH	Ethanol
CH_3OH	Methanol
HC	Hydrocarbon
KOH	Potassium Hydroxide Catalyst
NaOH	Sodium Hydroxide Catalyst
NOx	Oxides of Nitrogen

Greek symbols

η	Efficiency (-)

Subscripts

th	thermal
v	volumetric

1. INTRODUCTION:

The increase in global energy demand has resulted in an increase in the consumption of fossil fuels. Alternative biofuels are a partial solution to the harmful emissions that result. Jatropha oil cannot be used directly in conventional diesel engines due to its higher density, viscosity, and lower calorific value [1–2]. Diesel fuel has the same amount of energy as safflower, Jatropha, sunflower, cottonseed, rapeseed, and peanut oils. When using these oils directly, diesel engines expertise atomization and vaporization problems. Blending, thermal cracking, emulsification, and transesterification all help to overcome this problem. Vegetable oils can be tested in diesel engines after being mixed with diesel oil [3–4] Biodiesel reduces carbon monoxide (CO), hydrocarbon (HC), and particulate matter (PM) emissions when compared to diesel fuel. Besides from its many advantages, biodiesel can be used in railway locomotives, stationary Agri-gensets, and road transport, such as long-distance trucks, garbage trucks, and school buses [5].

Dubey and Gupta [6], investigated the emissions and performance of a diesel engine powered by Turpentine oil and Jatropha biodiesel (JBT). A dual biofuel blend (JBT 50) improved brake thermal efficiency by 2.17 % as compared to diesel oil. NOx, HC, CO, and smoke emissions reduced by 4.72, 4.56, 42.05, and 29.16%, respectively, when compared to crude diesel. As demonstrated by Madiwale et al. [7] ,biodiesel and ethanol mixtures are more efficient when

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blended together. Ethanol improves engine thermal efficiency at a wide range of engine loads when used as an additive. The performance and emissions of a biodiesel blend containing 20% Jatropha oil and 20% fish oil were compared to diesel fuel by Kathirvelu et al. [8], As a result, the HC, CO, and smoke concentration in these fuels are lower than those in diesel oil. Dharma et al. [9], investigated the performance and exhaust emissions of a diesel engine running on diesel and a biodiesel blend of Jatropha curcas (J) and Ceiba pentandra (C). The use of the biodiesel blend J50 C50 increased nitrogen oxide and carbon monoxide emissions from diesel oil. Singh et al.[10] investigated the performance and exhaust emissions of a diesel engine running on conventional diesel and blends of Jatropha biodiesel. Biodiesel blend B30 has shown about 24% reduction in CO emissions and 16.7% reduction in HC emissions. Also has significantly increased NOx and CO_2 emissions by 2.12% and 13.3% as compared to conventional diesel. Gad et al.[11] studied the performance and emissions of a diesel engine fueled with Jatropha biodiesel. Tests were performed at 75% of the engine load and different engine speeds. The maximum decreases in thermal efficiency and volumetric efficiency for B100 were 33 and 9%, respectively. The highest increase of NO_X emission for B100 was 47% compared to diesel oil. Tongroon et al. [12], described the operation of a diesel engine powered by Jatropha oil biodiesel made with fatty acid methyl ester (FAME). According to the results, adding 10% FAME reduced CO, HC, and smoke emissions. In a study comparing diesel engine performance and emissions when Jatropha methyl esters blends were used, Pramanik et al. [13], observed that the specific fuel consumption and thermal efficiency of the B30 biodiesel blend were equivalent to that of diesel oil, there was an increase in NOx emissions, and decrease in CO_2 emissions as the percentage of biodiesel in the fuel mix increased. Based on research using Jatropha biodiesel blends, the combination of Jatropha methyl ester and diesel oil had an effect on engine performance. Sahoo et al. [14], observed that at 1500 rpm and 50 % biodiesel blend the brake output power was increased . The higher the percentage of biodiesel, the greater is the increase in diesel oil specific fuel consumption. Agarwal [15], investigated diesel engine performance and emissions with various Jatropha oil methyl ester blend ratios. Biodiesel blends reduced brake power by 31.8 %, air-to-fuel ratio by 18%, thermal efficiency by 21.8 %, volumetric efficiency by 10.7 %, and specific fuel consumption by 32.18 %. Specific emissions of O_2, CO, and NOx increased as biodiesel content increased. Huang et al.[16], investigated the use of various biodiesel mixtures derived from Jatropha and diesel oils in diesel engines. The amount of carbon monoxide and nitrogen oxide emissions decreased compared to crude diesel. The study by Paul et al.[17], showed that pure Jatropha biodiesel had higher NOx emissions than diesel oil, while specific fuel consumption and thermal efficiency decreased. According to Panwar et al. [18], lower biodiesel blends can improve break thermal efficiency and reduce fuel consumption, exhaust gas temperature increased as biodiesel concentration increased, and NOx emissions are similar to diesel at low loads but higher at full loads. An experiment performed by Singh et al. [19], investigated the effects of Jatropha biodiesel blends on diesel engine performance and emissions. Based on the results, Jatropha methyl ester and its blends had lower brake thermal efficiency than those of diesel engine, while the break specific fuel consumption was higher. It was found that Jatropha biodiesel fuel had lower rates of HC, CO and CO_2. Jatropha biodiesel and its blend emit more NOX than diesel engines

The objective of this research is to compare the performance and emissions of Jatropha biodiesel and its blends with pure diesel oil in a stationary diesel engine without any modifications at different blend percentages (20%, 40%, 60%,80%,100%). SFC, η_{th} η_V ,AFR ,and EGT measurements were made to determine the performance of the diesel engine fueled with a biofuel, as well as the emission characteristics (HC, CO, CO_2, NOx, O_2) and smoke opacity.

2. MATERIAL AND METHODS:

2.1 Biodiesel Fuel Preparation:

To prepare biodiesel, Jatropha oil is mixed with pure methanol (CH_3OH) or ethanol (C_2H_5OH) and then added pure sodium hydroxide (NaOH) or potassium hydroxide (KOH) as a catalyst [20]. First, in a reaction flask, methanol and concentrated sulfuric acid (1 percent H_2SO_4 based on oil weight) were applied to the oil. Methanol to oil had a molar ratio of 9:1. (20% based on the weight of the oil). The first stage lasted 2 hours at 80°C with a stirring rate of 400 rpm. The mixture was then allowed to settle for 12 hours before proceeding to the second stage[21]. Second stage, using a molar ratio of 6:1 for oil and methanol, 20% methanol is required (based on the weight of the oil). 200 g of methanol and 8 g of NaOH were mixed in a separate vessel before being poured into a round bottom flask and continuously stirred until the NaOH was completely dissolved in the methanol, yielding Sodium Methoxide. A liter of pretreated oil (oil from the first stage) was heated to 80°C after the catalyst and alcohol mixture were added. For 90 minutes at 80°C, the mixture was stirred at 400 rpm in a magnetic stirrer[22]. After allowing the reaction to settle overnight in a separate funnel, the glycerol layer and the methyl ester layer were separated. To remove the unreacted alcohol, and catalyst from the biodiesel, it was mixed and washed with warm distilled water. The oil is heated to 100 °C to allow any water to evaporate[23].

2.2 Physical and Chemical Properties of Jatropha and diesel fuel:

Calorific values are critical in the selection of an alternative fuel for diesel engines in order to improve engine performance. The calorific value of the fuel determines how much heat is available for engine power. The heating values

of Jatropha and diesel oils are 42.1 and 43.3 MJ/kg, respectively. In diesel engines, Cetane number is a measure of a fuel's combustion quality and is related to fuel volatility and ignition delay time. The Cetane number influences the ignition quality, fuel consumption, and combustion roughness. Furthermore, higher Cetane-rated engines have shorter ignition delays and perform less efficiently. Jatropha and diesel oils have Cetane numbers of 56 and 53 °C, respectively. The temperature of the flash point of a fuel is an important consideration in the safe handling and storage of fuel. Because Jatropha oil has higher flash point than diesel fuel, it is safer to handle and store. Jatropha and diesel oil have flash points of 165 and 85 °C, respectively. The pour point of a fuel oil is the lowest temperature at which the oil flows and it may be defined as the lowest temperature at which the oil can be pumped from storage as the temperature drops. Oil can be pumped without heating the storage tank at this temperature. The pouring points of Jatropha, and diesel oil are -7, and -9 ° C, respectively. Table 1, shows the chemical and physical properties of Jatropha biodiesel and diesel oil. All properties of diesel fuel and Jatropha biodiesel were measured in Misr Petroleum company– Ghamrah Research Center. All properties
are compared with the stander properties in reference [24].

Table 1: Fuel properties (Jatropha biodiesel and conventional diesel fuel.

Property	Unit	Test Method	Jatropha Biodiesel	Diesel	Biodiesel Standers	
					ASTM D6751	EN14214
Density	kg/m³	ASTM D-1298	862.5	825	880	860-900
Viscosity	Cp	ASTM D-455	2	1.68	1.9 - 6	3.5-5
Heating Value	MJ/kg	ASTM D-4529	42.1	43.3	---	>35
Cetan Number	----	ASTM D-13	56	53	> 47	>51
Pour Point	°C	ASTM D-6749	-7	-9	< - 3	<0.0
Flash Point	°C	ASTM D-92	165	85	> 130	>120

3. EXPEMENTAL TEST RIG:

The test engine is a DEUTZ model F1L511, single cylinder, four stroke, air cooled, and direct injection, with the engine specifications shown in table 2. Figure 1, shows the experimental setup, which includes all of the instruments required to measure the various engine parameters. The engine brake power was measured using a 4.5 kW AC generator directly connected to the engine. In this case, an externally controlled variable load bank is used. The loads are (1, 2, 3, and 4) kW from 0 to full load by step 1. Jatropha biodiesel blends were used to run the engine to determine their specific fuel consumption, thermal efficiency, exhausted temperature and air-to-fuel ratio, as well as exhaust emissions parameters like CO, CO_2 HC, and NO_X. The accuracy and range of a gas analyzer for emission are shown in table 3. The engine obtains air through an air box with an orifice that measures air consumption. A pressure differential detector is used to determine the pressure difference between the two orifice sides. The amount of fuel consumed was calculated using a glass burette and a stopwatch. The rated speed of the engine is 1500 rpm. A series of electric lamps was utilized to consume the generator's output power (every two lamps of 500 Watt were connected in series). The engine was warmed up before taking any readings. As soon as the engine reached a stable state, the experiments began. The engine was initially powered by diesel fuel. After that, the engine was then fueled with a biodiesel blend. Readings were taken as the load varied from no load to full load. Figure 2, shows the experiment steps to determine performance and emissions of diesel engine.

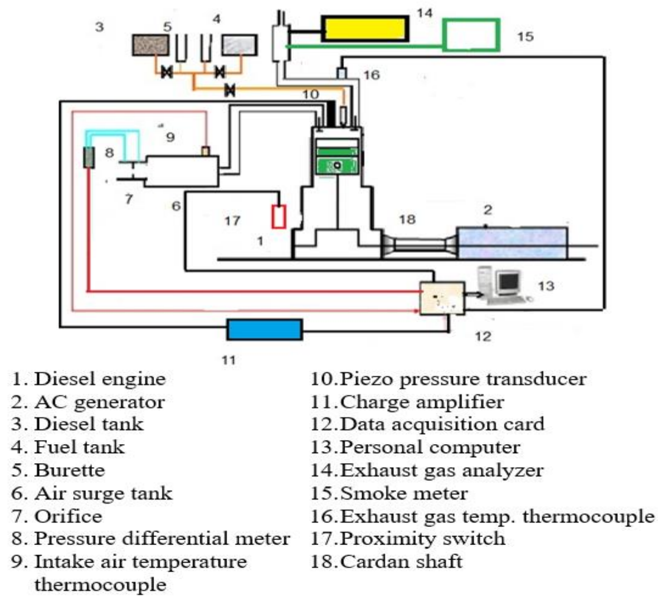

Figure ١: Schematic diagram of the experimental setup.

Table 2: Diesel engine specifications

No.	Engine parameters	Specification
1	Engine model	DUETZ F1L511
2	Number of cylinders:	1
3	Bore	100 mm
4	Stroke	105 mm
5	Displacement	824 cc
6	Rated power	5.775/7.7 kW/hp
7	Rated speed	1500 rpm
8	Maximum torque	44/900 RPM N.m
9	Injection point BTDC	24°C.A
10	Type of Injection:	Direct injection
11	Type of cooling	Air cooling
12	Starting up	Electrical
13	Injection pressure	175 bar

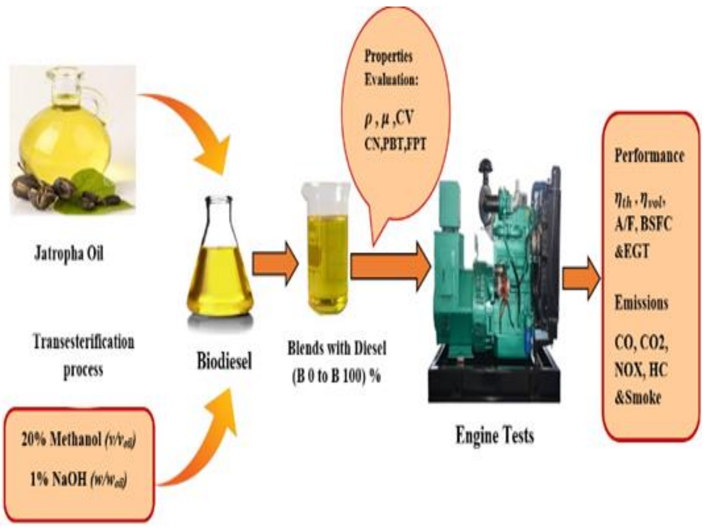

Figure 2: The experiment steps to determine performance and emissions of diesel engine.

Table 3: Accuracy and range of gas analyzer for emission

No.	Exhaust Gas	Range	Resolution
1	Carbon Monoxide (CO)	0-20%	0.01%
2	Carbon Dioxide (CO2)	0-10%	0.1%
3	Oxygen (O2)	0-22%	0.01%
4	Oxides of Nitrogen (NOx)	0-4000 ppm	1 ppm
5	Hydrocarbon (HC)	0-2000 ppm	1 ppm
6	Smoke Opacity	0-99%	0.1

4. COMARISON OF EXPERMENTAL RESULTS WITH OTHER REFRENCES:

The Comparison of the current study and a number of other studies[25–28] using Jatropha biodiesel as fuel are shown in Figure 3. The brake thermal efficiency and carbon monoxide emissions are important characteristics that should be compared. All references used a blend of 20% as biodiesel fuel and used a diesel engine with variable load (no load, quarter load, half load, three-quarter load, and full load). Compared to the current work, figure 3-a shows a relationship between brake thermal efficiency and engine load for different references. All references show an increase in thermal efficiency with engine load, with the peak occurring around 75% load. This study found that Jatropha biodiesel has the same trend and has a higher efficiency than other references; this could be due to its properties, Jatropha biodiesel has lower viscosity and density and a higher heating value. Figure 3-b shows the relationship between carbon monoxide emissions and engine load for different references and the current study. For all fuels, CO emissions decreased at approximately 75% load and then increased at full load operation. A higher engine load or speed results in higher temperatures of the reacting gases inside the cylinder, resulting in lower CO-specific emissions and higher CO_2-specific emissions in the engine. Current study has the same trend with other references

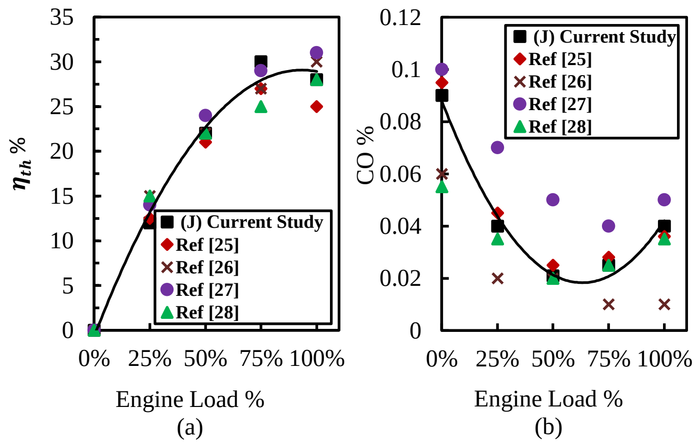

Figure 3: Comparison of experimental results with other references, at (B20%).

5. MESUARING TEQNIQUE:

The engine parameters measured in the present study were the brake power, exhaust gas temperature, fuel consumption, and air consumption, which were recorded at variable engine loads and a constant speed of 1500 rpm in order to evaluate the engine performances such as thermal efficiency, volumetric efficiency, and air to fuel ratio. Also, emissions such as CO, CO_2 HC, and NO_X were measured when the engine was fueled with different blends of Jatropha biodiesel and conventional diesel. All equations that used is taken from reference[29].

5.1 Engine Break Power:

Brake power is the power available at the crankshaft. In the case of the internal combustion engine, it is output power.

$$BP = \frac{V \times I}{\eta} \qquad (1)$$

Where :

BP is engine break power,

V = volt,

I = current, and

η = Efficiency of Generator (typically 85%)

5.2 Mass Flow Rate Of Fuel

Mass flow rate of fuel is the mass of fuel which passes per unit of time.

$$\dot{m}_f = \frac{\forall \times \rho}{t} \quad (2)$$

Where:

$\dot{m}_f$ = mass flow rate of fuel,

$\forall$= a fixed volume of 50 mm L filled with biodiesel,

t = time (sec)

ρ = biodiesel fuel density; 862.5Kg/m^3.

5.3 Mass Flow Rate of Air:

Mass flow rate of air is the mass of air which passes per unit of time.

$$\dot{m}_a = C_d\, A_{orifice} \sqrt{2\, g \Delta\, H\, \rho_a \rho_w} \quad (3)$$

Where:

$\dot{m}_a$= mass flow rate of air,

C_d= orifice discharge coefficient (C_d = 0.6063),

$A_{orifice}$ =orifice cross sectional area in m^2,($d_{orifice}$ = 0.016m)

ρ_a= intake air density in kg/m^3,(=1.17 kg/m^3)

ρ_W= water density in kg/m^3,(=1000 kg/m^3)

5.4 Break Specific Fuel Consumption

Brake specific fuel consumption is the ratio of a mass flow rate of the fuel supplied to the engine to the brake power obtained at a crankshaft and it indicates how efficiently the fuel is used to produce brake power.

$$SFC = \frac{\dot{m}_f}{BP} \quad (4)$$

Where:

SFC = specific fuel consumption,

$\dot{m}_f$ = mass flow rate of fuel in kg/h,

BP = breake power in kW

5.5 Break Thermal Efficiency:

The brake thermal efficiency (BTHE) is defined as the ratio of the brake power developed by the engine to the heat input, the heat energy supplied by the fuel.

$$\eta_{th} = \frac{3600\, BP}{HV \times \dot{m}_f} \quad (5)$$

Where:

η_{th}= thermal efficiency and

HV= heating value for fuel in(kJ/kg).

5.6 Volumetric Efficiency:

The volumetric efficiency of an engine is calculated by dividing the amount of air injected into a cylinder by the mass that occupies the displacement volume at the intake manifold density.

$$\eta_v = \frac{\dot{m}_a}{\rho_a \times V_s \times (\frac{N \times n}{2 \times 60})} \quad (6)$$

$$V_s = \frac{\pi}{4} \times B^2 \times L \quad (7)$$

where :

η_v= volumetric efficiency

Vs =Swept volume in cubic meter

B = Cylinder bore (B = 105 mm),

L = Piston stroke (L = 110 mm),

n = Number of cylinders (n = 1),

N = Engine speed in rpm

5.7 Air-Fuel Ratio:

Air–fuel ratio (AFR) is the mass ratio of air to a solid, liquid, or gaseous fuel present in a combustion process

$$\frac{A}{F} = \frac{\dot{m}_a}{\dot{m}_f} \quad (8)$$

Where:

$\frac{A}{F}$ = Air–Fuel ratio

6. RESULTS AND DISCUSSION:

6.1 Engine Performance Characteristics

6.1.1 Brake-Specific Fuel Consumption (BSFC):

Figure4, shows the brake-specific fuel consumption (BSFC) of various blends as a function of engine brake power. As engine brake power increases, the BSFC values of the Jatropha biodiesel decrease significantly. The higher the brake power, the greater is the influence on fuel consumption[30]. The consumption of Jatropha biodiesel is much higher than that of conventional diesel. This is clearly illustrated by the BSFC's significant increment as the blends are increased. Blend (B100), for example, has a higher BSFC than blend (B20), which is consistent with the result of reference[13] . When compared to conventional diesel, the viscosity and density of blend (B100) increase while the heating value decreases. Such properties may result in greater fuel discharge for a given displacement of the injection pump plunger. As a result, when compared to the performance of conventional diesel, blend (B20) is preferred over other blends.

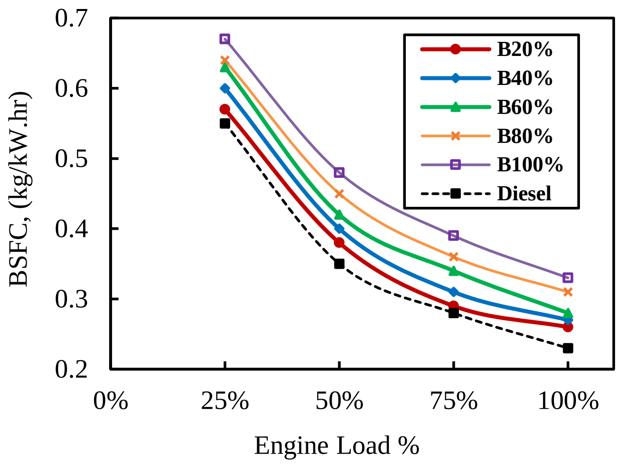

Figure 1: Effect of engine brake power on break specific fuel

6.1.2 Break Thermal Efficiency (η_{th}):

For various blends, Figure 5, illustrates the brake thermal efficiency as a function of engine brake power. The thermal efficiency of Jatropha increases with engine load, with maximum efficiency at 75% loading, which is consistent with references [25]. The thermal efficiency obviously decreases as the blend increases, which may be due to the lower calorific and volatility values, as well as the higher viscosity, resulting in poor atomization and vaporization [30]. Although the thermal efficiency of the blends was significantly higher than that of biodiesel alone, the efficiencies of the blends and biodiesels were generally lower than that of diesel fuel. The maximum thermal efficiency obtained using Jatropha biodiesel is approximately 30% using a blend of 20%, compared to pure diesel, which has a maximum efficiency of approximately 31 %.

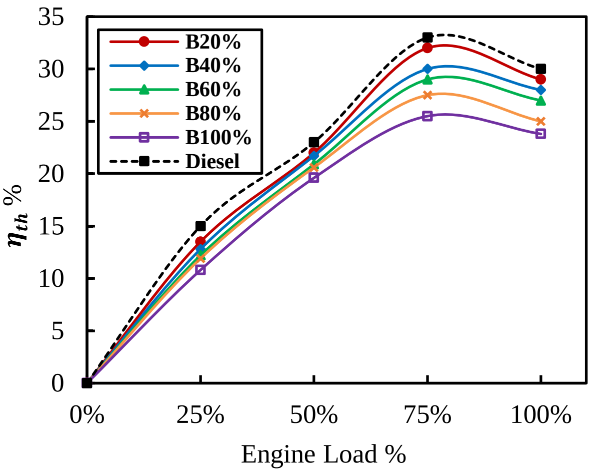

Figure 5: Effect of engine brake power on thermal efficiency.

6.1.3 Air-Fuel Ratio (AFR):

Figure 6, represents the relationship between the air-fuel ratio (AFR) and engine load for different biodiesel blends. As the engine load increases, the air-fuel ratio of Jatropha biodiesel decreases. Under constant engine speed and unchanged air induction for combustion, the amount of fuel injected per cycle increases as engine load increases, potentially decreasing the air-fuel ratio. Also, increased the biodiesel blends Consequent the AFR is gradually decreased. It may be because the fuel blends contained more oxygen, resulting in lower levels of air consumption. [25].

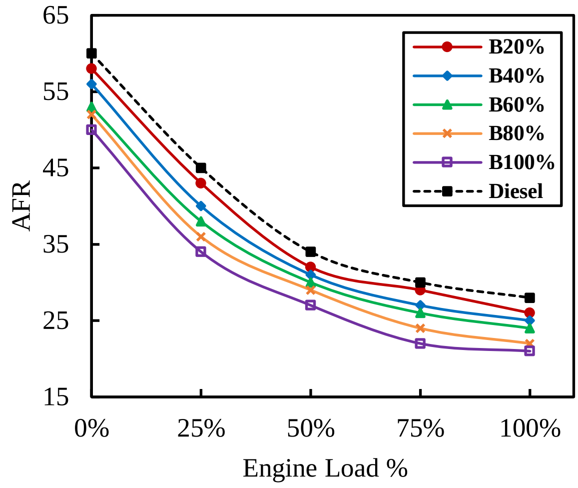

Figure 6: Effect of engine brake power on air to fuel ratio.

6.1.4 Volumetric Efficiency (η_v):

The volumetric efficiency of an engine is calculated by dividing the amount of air injected into a cylinder by the mass that occupies the displacement volume at the intake manifold density. Figure 7, shows the relationship between volumetric efficiency and engine loads for different biodiesel blends. As the load increased, the volumetric efficiency decreases. To overcome engine load, fuel consumption increased at high loads, but the same amount of air was consumed (constant speed) at different loads, resulting in higher gas temperatures. This decreases volumetric efficiency[31]. Hence, it can be stated that when compared to diesel fuel, biodiesel and its blends have lower volumetric efficiency, which may be due to the intake air being preheated as a result of the high temperature of residual gases (from exhaust stroke), which reduces volumetric efficiency. [32]. As shown in the previous paragraph, increasing the blend ratio reduces volumetric efficiency. The volumetric efficiency of pure Jatropha biodiesel drop is 7 at full load (lower than diesel fuel).

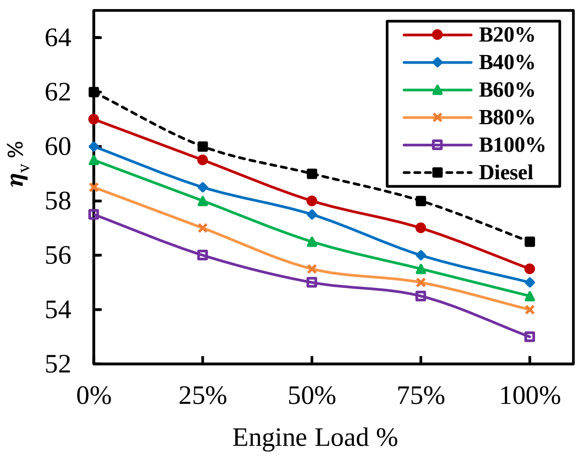

Figure 7: Effect of engine brake power on volumetric efficiency.

6.1.5 Exhaust Gas Temperature (EGT):

Figure 8, shows the exhaust gas temperature (EGT) variation with load for various biodiesel blends and diesel fuel. The temperature of the exhaust gas rises as the engine load increases. This is consistent with the references [33] .where more fuel is required in the engine to produce the extra power required to handle the extra load as the percentage of biodiesel in the blend increases, so the temperature of the exhaust gas increases. Because biodiesel blends have a higher viscosity, there is poor fuel atomization and vaporization, resulting in late burning of injected fuel and, as a result, higher EGT. Also Because biodiesel fuel and its blends have a lower heating value, more fuel is injected[34].The highest value of exhaust gas temperature was observed with the B 100% percent, which was 350 °C, while the corresponding value with diesel was only 275 °C. The maximum exhaust gas temperature for Jatropha biodiesel with B20 was found to be 290°C, which is very close to the maximum value obtained with pure diesel.

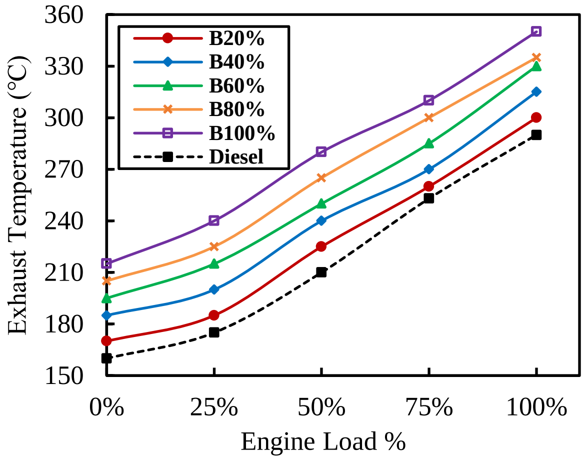

Figure 8: Effect of engine brake power on exhausted temperature.

6.2 Engine Emissions Characteristics

6.2.1 Oxides of Nitrogen Emissions (NOx):

Figure 9, shows the Oxides of Nitrogen NOx emissions from engines running on various diesel-biodiesel blends, pure diesel, and pure biodiesel. Because of the increased amount of fuel injected, NOx emissions rise as engine load increases. Combustion temperature, ignition delay, and oxygen content all have a direct effect on NOx emissions. This effect could be explained by the fact that the amount of fuel injected and the temperature of the cylinder combustion are directly proportional [28]. Obviously, NOx emissions rise as the biodiesel blend percentage rises. Because biodiesel contains more oxygen than conventional diesel, it provides better combustion, which raises combustion temperature[31] .For B100 at full load, the maximum increase in NOx specific emissions for Jatropha biodiesel is 30% (over diesel fuel). Biodiesel blends, 20 percent is considered preferable when compared to other blends that promise to reduce emissions.

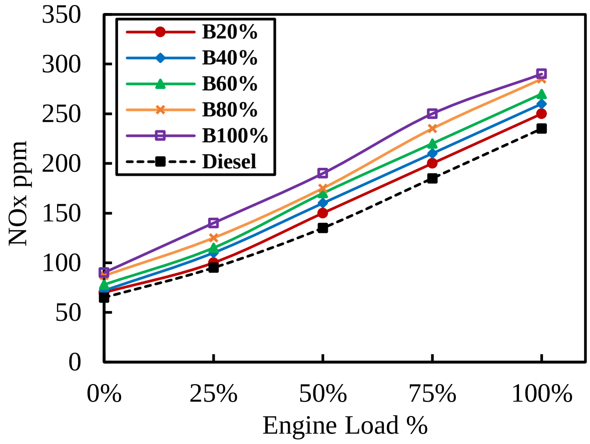

Figure 9: Effect of engine brake power on oxides of nitrogen emissions.

6.2.2 Unburned Hydrocarbon Emissions (HC):

Figure 10, shows the relation between hydrocarbon emission and engine load for various biodiesel and diesel blends. The HC emission increases as the engine load increases because more fuel is injected at higher loads, which explains this result [34]. The HC emission values for biodiesel blends showed similar trends as diesel fuel, but they were lower. Biodiesel and its blends have a lower carbon to hydrogen ratio than conventional diesel, which is due to the presence of oxygen in their molecular structure. Biodiesels with a higher Cetane number may have less combustion delay and ignition lag, resulting in a lower total amount of hydrocarbons emitted [5]. Using pure Jatropha biodiesel instead of diesel fuel reduces HC emissions from diesel engines by 50 percent.

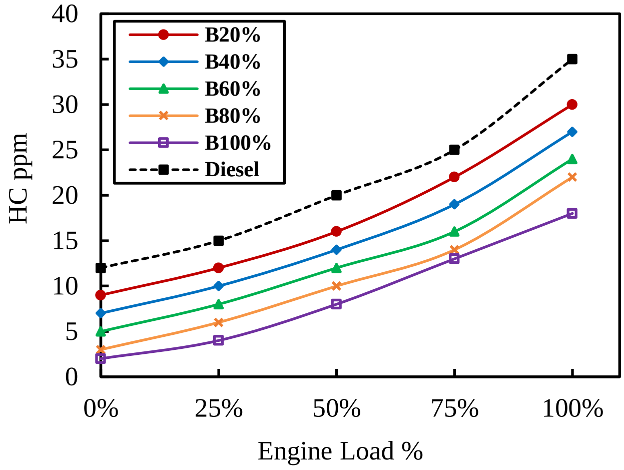

Figure 10: Effect of engine brake power on unburned hydrocarbon emissions.

6.2.3 Carbon Monoxide Emissions (CO):

Figure 11, shows carbon monoxide (CO) emissions at various engine loads for various blends. For all blends, the general trend is that CO gradually decreased up to about 75% load and then increased at full load. Because diesel engines run on lean mixtures, CO emissions are generally low. The specific emissions of carbon monoxide from Jatropha biodiesel decrease as engine load increases. The oxidation of CO into CO_2 occurs at a slower rate in the presence of a lower engine load. This is primarily due to the lower gas temperature in the cylinder. The higher the engine speed or load, the higher the temperature of the reacting gases inside the cylinder, resulting in lower CO specific emissions and higher CO_2 specific emissions in the engine[25]. Specific carbon monoxide emissions decrease as the percentage of biodiesel fuel in the blend increase. This could be attributed to the presence of oxygen in the molecular structure of Jatropha biodiesel, which aids in combustion [28].

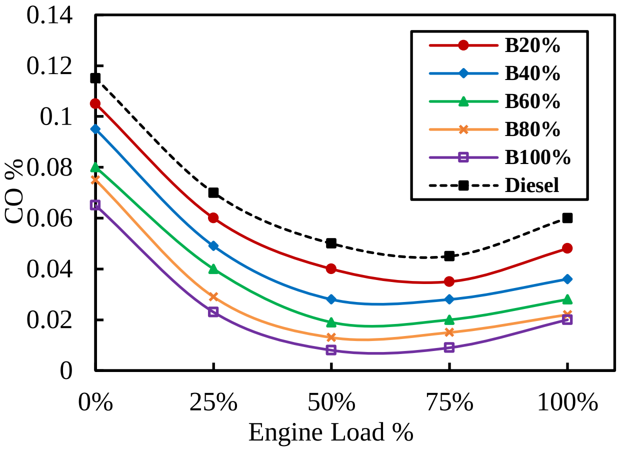

Figure 11: Effect of engine brake power on carbon monoxide emissions.

6.2.4 Carbon Dioxide Emissions (CO_2)

Figure 12, shows the relationship between Carbon Dioxide Emissions CO_2 emissions and engine load for diesel and biodiesel blend fuels. The better the combustion, the higher the CO_2 emissions of an engine. Because higher fuel consumption rates are required for higher engine loads, specific CO_2 emissions increase as engine load increases. As a result, the temperatures of the reacting gas in the cylinders rise. As a result, CO_2 emissions are rising[30]. Specific emissions of carbon dioxide decreased as the percentage of biodiesel fuel in the blends increased. Because biodiesel fuel has a lower carbon to hydrogen ratio than diesel fuel, it is a low-carbon fuel. When biodiesel is burned with air, it produces less CO_2 than when diesel fuel is burned [25]. At full load, pure Jatropha biodiesel emits 35% less CO_2 than diesel.

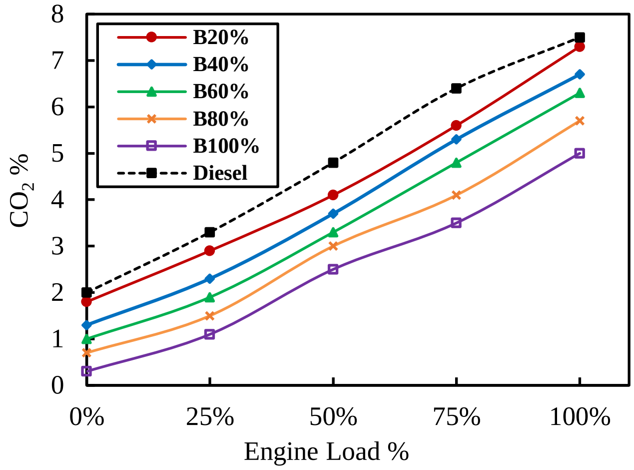

Figure 12: Effect of engine brake power on carbon dioxide emissions.

6.2.5 Smoke Emission:

Figure 13, represents the relationship between smoke emissions and engine load. The results of the tests show that higher engine loads result in higher smoke concentrations due to an increase in fuel injected into the engine[35]. Furthermore, when a higher percentage of biodiesel was used, there was a reduction in smoke opacity. This is due to the higher oxygen content of blends compared to pure diesel, as well as the oxidation of biodiesel carbon residues when burned [36]. When compared to pure diesel, B 100 percent reduced smoke emissions by 30% at full load.

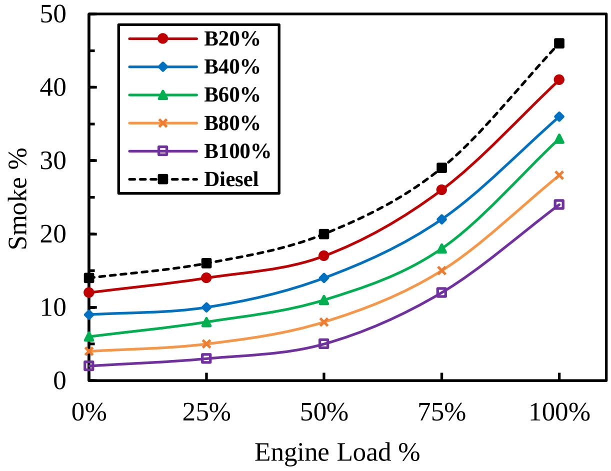

Figure 13: Effect of engine brake power smoke emissions.

7. UNCERTAINTY ANALYSIS:

Measurement errors are resulted from various causes such as, instrument calibrations, the data set finite statistics, and the methods used. In fact, we do not know the exact value of the measured parameters. There are two main types of errors; systematic error and random error. The main difference between systematic and random errors is that, random errors lead to fluctuations of measured value of instrument around the true value. Systematic errors; lead to predictable and consistent deviation from the true value due to a problem related to equipment calibrations[37] .

Uncertainty analysis helps in describing the interval about the measured value within which we suspect that the true value must fall with a stated probability. Uncertainty analysis is the process of identifying, quantifying, and combining the errors [37].

$$V = M + U(X) \pm ; P \ \ Percent \tag{9}$$

Where V is the variable, M is its best value, U(X) is uncertainty, and P is the confidence level. In general, uncertainty analysis can be expressed using the following equation; Eq. (10).

$$U(X) = \sqrt{\left(\frac{\delta X_1}{\delta X} U(X_1)\right)^2 + \left(\frac{\delta X_2}{\delta X} U(X_2)\right)^2 + \cdots + \left(\frac{\delta X_n}{\delta X} U(X_n)\right)^2} \tag{10}$$

7.1 Uncertainty in Measuring Mass Flow Rate of Fuel:

$$\dot{m}_f = \frac{\forall \times \rho}{t} \tag{2}$$

$$U_{\dot{m}_f} = \sqrt{\left(\frac{-\forall \times \rho}{t^2} \times \ u_t\right)^2} = \left(\frac{-\forall \times \rho}{t^2} \times \ u_t\right) \tag{11}$$

At full load condition (BP=4 kW) ,the error in measuring is $\pm 5\%$

7.2 Uncertainty in Measuring Mass Flow Rate of Air:

$$\dot{m}_a = C_d \ A_{orifice} \sqrt{2 \ g \Delta \ H \ \rho_a \rho_w} \tag{3}$$

$$U_{\dot{m}_a} = \sqrt{\left(0.01846 \sqrt{\frac{1}{\Delta \ H}} \times u_{\Delta \ H}\right)^2} = \left(0.01846 \sqrt{\frac{1}{\Delta \ H}} \ u_{\Delta \ H}\right) \tag{12}$$

At full load condition (BP=4 kW) ,the error in measuring is $\pm 0.02\%$.

7.3 Uncertainty in Measuring The Engine Break Power:

$$BP = \frac{V \times I}{\eta} \tag{1}$$

$$U_{BP} = \left(\frac{1}{\eta}\right) \sqrt{(I \ u_V)^2 + (V \ u_I)^2} \tag{13}$$

At full load condition (BP=4 kW) ,the error in measuring is $\pm 2.17\%$

7.4 Uncertainty in Measuring Specific Fuel Consumption:

$$SFC = \frac{\dot{m}_f}{BP} \tag{4}$$

$$U_{SFC} = \forall \times \rho \times \eta \times \sqrt{\left[\left(\frac{-1}{V \times I \times t^2} \times U_t\right)^2 + \left(\frac{-1}{t \times I \times V^2} \times U_V\right)^2 + \left(\frac{-1}{t \times V \times I^2} \times U_I\right)^2\right]} \tag{14}$$

At full load condition (BP=4 kW) ,the error in measuring is $\pm 5.13\%$

7.5 Uncertainty in Measuring Thermal Efficiency:

$$\eta_{th} = \frac{BP}{HV \times \dot{m}_f} \tag{5}$$

$$U_{\eta_{th}} = \frac{1}{\eta \times \rho_f \times HV \times \forall} \times \sqrt{\left[(V \ \times t \times U_I)^2 + (\ I \times t \times U_V)^2 + (V \ \times \ I \times U_t)^2\right]} \tag{15}$$

At full load condition (BP=4 kW) .The error in measuring is $\pm 5.87\%$

7.6 Uncertainty in Measuring Volumetric Efficiency:

$$\eta_v = \frac{\dot{m}_a}{\rho_a \times V_s \times (\frac{N \times n}{2 \times 60})} \tag{6}$$

$$V_s = \frac{\pi}{4} \times B^2 \times L \tag{7}$$

$$U_{\eta_v} = \frac{0.5 \times C_d \ A_{orifice} \sqrt{2 \ g \ \rho_a \rho_w}}{\rho_a \times V_s \times (\frac{N \times n}{2 \times 60})} \sqrt{\left(\frac{1}{\sqrt{\Delta \ H}} \times U_{\Delta \ H}\right)^2} \tag{16}$$

At full load condition (BP=4 kW) .The error in measuring is $\pm 1.87\%$

7.7 Uncertainty in Measuring Air-Fuel Ratio:

$$\frac{A}{F} = \frac{\dot{m}_a}{\dot{m}_f} \tag{8}$$

$$U_{A/F} = \frac{C_d \ A_{orifice} \sqrt{2 \ g \ \rho_a \rho_w}}{\rho_a \times \forall} \sqrt{\left(-0.5t \times \sqrt{\frac{1}{\Delta \ H}} \times u_{\Delta \ H}\right)^2 + \left(\sqrt{\Delta \ H} \ \times U_t\right)^2} \tag{17}$$

At full load condition (BP=4 kW) The error in measuring is $\pm 3.8\%$

7.8 Uncertainty In Measuring Exhaust Emissions:

Resolution and range of gas analyzer for the emission concentrations are shown in (Table 3)

$u_{CO_2} = \pm 1.0\%$

$u_{CO} = \pm 0.2\%$

$u_{NO_x} = \pm 0.2\%$

$u_{HC} = \pm 0.2\%$

$u_{Smoke} = \pm 0.1\%$

8. CONCLOUSTION:

The performance and emission characteristics of a diesel engine using Jatropha biodiesel blended with diesel were evaluated in this study. This research was divided into three sections. The first section discussed the transesterification process for extracting and preparing biodiesel. In the second section, parameters were measured to determine the properties of diesel, biodiesel, and their blends. The engine was tested in the third section to compare the performance and emission characteristics of various fuels. The main conclusions are as follows:

- In terms of engine performance and emissions, diesel and biodiesel fuel were comparable.
- As the biodiesel blends increase, the brake thermal efficiency decreases, and the maximum engine efficiency is 30% and 20% with conventional diesel and pure Jatropha.
- Pure Jatropha has a significantly higher BSFC than diesel, while a 20% blend is very close to diesel fuel consumption.
- The air-to-fuel ratio of pure Jatropha biodiesel is reduced by 25% when compared to diesel.
- NOx emissions rise as the load rises and the biodiesel concentration rises.
- When compared to diesel fuel for pure Jatropha at full load, CO_2 emissions decrease by 35% as the blend increases.
- Increasing biodiesel blends significantly reduce CO, HC, and smoke opacity emissions.
- Uncertainty analysis was calculated to ensure that all measured values were accurate and the results were $\pm$5.13%, $\pm$5.87%, $\pm$1.87% and $\pm$3.8% for SFC, η_{th}, η_v and AFR, respectively.
- According to this study, using 20 percent Jatropha biodiesel blended with diesel in conventional diesel engines is viable.

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