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/m^3	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.

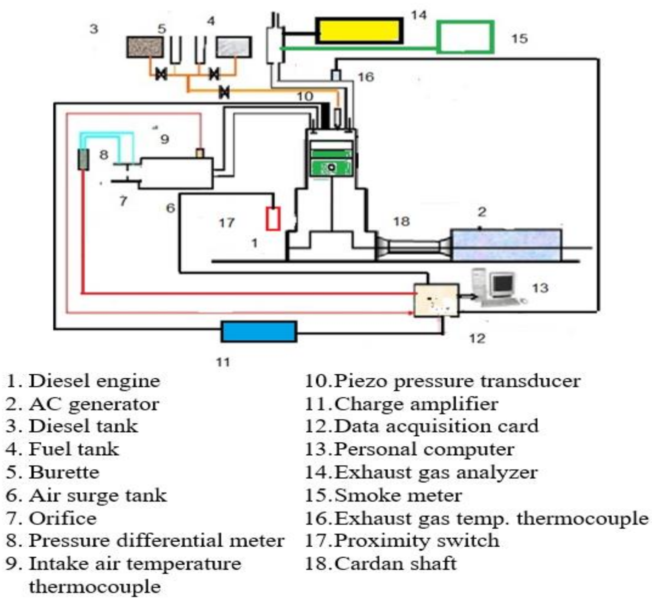

Figure ١: Schematic diagram of the experimental setup.

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

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.

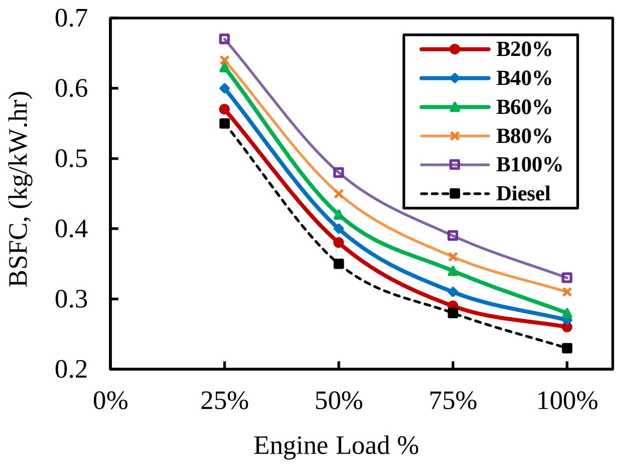

Figure 1: Effect of engine brake power on break specific fuel

6.1.5 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.

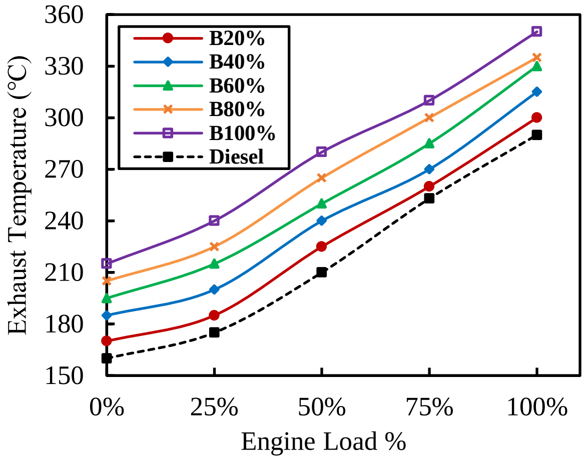

Figure 8: Effect of engine brake power on exhausted temperature.

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.

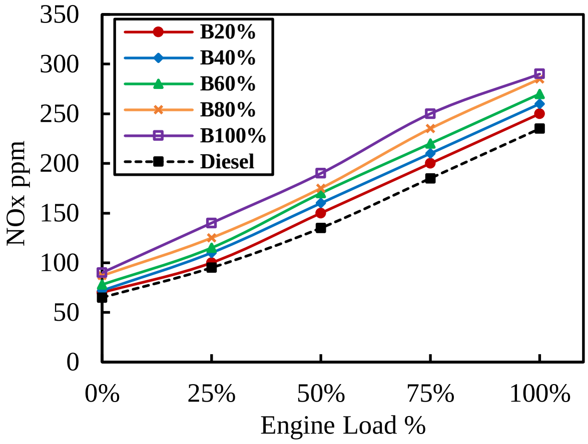

Figure 9: Effect of engine brake power on oxides of nitrogen emissions.

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.

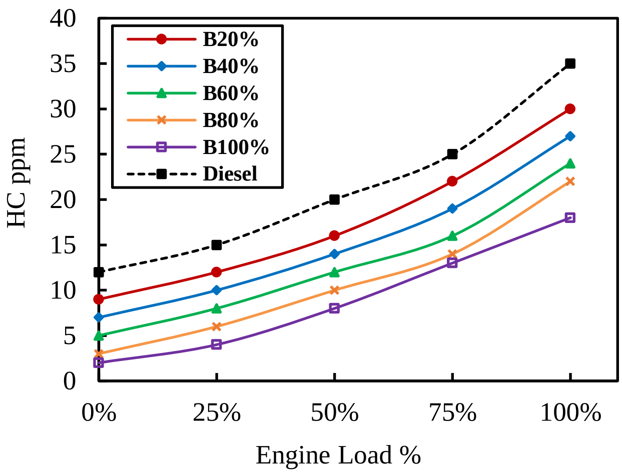

Figure 10: Effect of engine brake power on unburned hydrocarbon emissions.

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].

$$V = M + U(X) \pm; P \;\; Percent \qquad (9)$$

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).

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

7.1 Uncertainty in Measuring Mass Flow Rate of Fuel:

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

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

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

7.2 Uncertainty in Measuring Mass Flow Rate of Air:

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

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

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

7.3 Uncertainty in Measuring The Engine Break Power:

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

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

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

7.4 Uncertainty in Measuring Specific Fuel Consumption:

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

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

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

7.5 Uncertainty in Measuring Thermal Efficiency:

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

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

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

7.6 Uncertainty in Measuring Volumetric Efficiency:

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

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

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

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

7.7 Uncertainty in Measuring Air-Fuel Ratio:

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

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

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%

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Faculty of Energy Engineering - Aswan University - Aswan - Egypt