ISSN: 2350-0328

International Journal of AdvancedResearch in Science, Engineering and Technology

Vol. 6, Issue 2, February 2019

Diesel Engine Performance Test Using Jatropha Curcas Oil Base Fuel Under a Constant Engine Load Condition

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ABSTRACT: Biodiesel was produced in a modified batch reactor from Jatropha curcas oil using two stage esterification and transesterification processes. The produced biodiesel was washed and dried for use in an unmodified single stroke diesel engine. Quantities of the fossil diesel were measured 10 liters each into four different containers and blended with various percentages of the produced biodiesel as follows; 10% (B_{10}), 20% (B_{20}), 50% (B_{50}) and 70% (B_{70}). The unblended fossil diesel (B_0) and unblended biodiesel 100% (B_{100}) were placed in the fifth and sixth containers respectively as controls. An engine load (L_E) was imposed on the engine through the in-built governor; which was measured as a percentage of the known maximum adjustment. The dynamometer loads were imposed on the engine through the adjustable dynamometer belt and its values were read-off as forces (F_1 and F_2) from the weighing spring attached to the canvas belt. The result of the experiment show that fossil diesel (B_0) had the lowest fuel use value of 1.326kg/h closely followed by B_{20} with a value of 1.485 kg/h while B_{100} had highest value of 1.561kg/h. A maximum torque value of 19.34kW was noted at B_{10} and a minimum value of 12.15kW at $B_{70.}$ The brake power generated improved with blending of biodiesel with fossil diesel. B_{10} generated brake power 4.361 kW higher than the fossil diesel which had a brake power value of 3.98 kW. Brake mean effective pressure of the engine reveals that there was a slight increase from B_0 (245569.90KN/m^2) to B_{10} (268957.50KN/m^2), a steady decrease in value was noted up to B_{70} before moving up again to 184177.40KN/m^2 for B_{100}. Fossil diesel generated a brake specific fuel consumption value of 0.356655kg/kWh which gradually increased with the blends, B_{70} had the highest value of 0.575077kg/kWh. The investigation on engine thermal efficiency against fuel blends reveal a gradual increase from B_0 (961E-05 %) to a peak of (1.00E-4 %) at B_{20}. The results of equivalent power show a minimum value of 15.27493kW for B_0 with a peak value of 17.91891 kW at B_{10}.

KEYWORDS: Jatropha curcas, Biodiesel, Batch-Reactor, Esterification, Trans-esterification.

I. INTRODUCTION

The foreseeable diminishing of fossil fuel reserves and the harmful environmental consequence of exhaust gases from petroleum diesel has attracted much attention during the past few years. According to a publication by Austrian bio-fuels institute**,** the past 10 years saw a dramatic increase in global warming potential from less than 20% to more than 25% while the acidic pollution constitute 75% of the total emissions of this pollution type [1] Researchers have shown that when petroleum is burn especially for transportation industries, some environmentally un-friendly gases are released. [2]. These include – carbon monoxide (C0), carbon-dioxide, ($C0_2$), Nitrogen oxide ($N0_x$), Sulphur-Oxides (SO_x), Volatile Organic Compounds (VOC_s) also known as reactive hydrocarbons, hydroxyl ions (OH) and particulate matter (PM). Ozone(O_3) which is a major component of smog in the presence of sunlight. The carbon-dioxide released from these fossil fuels and other green house gases allow the incoming solar radiation (energy) to enter the earth's atmosphere but, reduce the amount of energy that can re-radiate back into space, this increases the greenhouse gases concentration thus causing reduction in the outgoing infrared radiation thereby causing global warming [3]. Over the years fossil fuel exploration and production has rendered fertile and potential farm land unproductive thus, contributing indirectly to the high pricing of agricultural products [4]. This is as a result of hydrocarbon contamination of the soil through spillage and improper oil waste disposals. Diesel is said to have about 90 times as much sulphur as petrol (premium motor spirit) and heavier hydrocarbon (aromatics) which do not burn easily and are emitted as particulates into the atmosphere. The particulates are regarded as human carcinogens or cancer causing agents [5]. Many industrial workers are probable victims of cancer because of emissions. Carbon emission from fossil fuel combustion and land use changes are considered the main factors causing global warming [6] and [7].

ISSN: 2350-0328
International Journal of AdvancedResearch in Science, Engineering and Technology
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These natural and environmental problems surrounding the use of fossil fuel have made it imperative to source for an alternative; and a remedy to these problems. Rudolf Diesel in 1912 experimented on the use of plant oil as alternative to fossil fuel [8]. His reason being that he wanted to make his engine more attractive to farmers and to have a constant source of fuel. Periodic petroleum shortages spurred research into vegetable oil based fuel popularly known as Bio-Biodiesel. Environmental benefits from running biodiesel rather than mineral diesel in the same engine include a 99% reduction of sulphur oxide emissions, a reduction in green house gas emissions of a least 3.2kg of Co_2 per kilogram of biodiesel, a 39% reduction in particulate matter and a high level of biodegradability [1].

A. BIODIESEL CHEMISTRY AND PRODUCTION.

Biodiesel is a renewable, clean burning, biodegradable fuel [9]. It's chemical structure is that of fatty acid alky/ester. There are various forms of bio-fuel and most of them are made from fats or vegetable oils which contain glycerin thus, they are called triglycerides. Bio-fuel is basically solar energy stored in bio-materials in the form a dense chemical [4]. In order words, it is the energy derived from biological carbon fixation. In green plants, this process is achieved through the process of "photosynthesis". It is this plant food stored in form of chemical energy in plants that is available as bio-mass. Jatropha curcas is a flowering plant of the family of *eupheribiaceae* . The plant and it's seeds are non-edible to humans and other animals. This is due to it's toxicity which is as a result of the presence of cursine, deterpine, saponin, tannins, lectinphytate and phorboster. It is a drought resistant oil bearing plant. It grows as a shrub or wild tree when unattended. The tree produces seeds containing 30 to 50 percent oil which is easily convertible into bio-diesel [10]. Biodiesel is typically made by chemically reacting lipids (vegetable oil) with an alcohol producing fatty acid esters. It is commonly produced by transesterification of the vegetable oil or feedstock.

B. LITERATURE SURVEY

Over the years, series of test have been carried out using biodiesel to run engines at different levels of blending. Some of these experiments were carried out on long term basis of about 10 hours straight engine run or short term running of the engine for about 2 hours. Most of these analyses were targeted on parameters like, power, noise level, emission (pollution levels), lubrication, braking ability, operating torque of the engine etc. According to a short term evaluation carried out by Quick (1989) [11] using vegetable oil on a direct injection engine, a maximum power decrease of 3 to 4% was observed. On the other hand, a fuel consumption increase of 5 to 10% was recorded, while the brake thermal efficiency was reduced by 1 to 4%. Prolonged use of crude vegetable oils in the direct-injected diesel engines developed problems leading to deterioration in performance because of their high viscosities. Schumacher, et al (1992) [12]; Recce and Peterson (1993) [13], and Marshall (1993) [14] observed reductions in power ranging from 1 to 7%. Using a 1991 cummins 5.9 litre Direct Injection Turbo-charged engine, Schumacher observed a 3% power increase [15].

II. MATERIALS

The materials used for this experiment were selected mainly based on their physical/mechanical and chemical properties as it affects the stability and reaction with the other parts for smooth running of the selected 16Hp, S1100 model engine. Other material used in the course of the experiment include; Locally fabricated prony brake dynamometer, Tachometer, Stop watch, Burettes, Valves for liquid flow control, Transparent host for monitoring fuel flow and Plastic funnel for directing the fuel flow.

Plate 1. Locally Fabricated Prony Brake Dynamometer.

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International Journal of AdvancedResearch in Science, Engineering and Technology
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III. METHOD USED.

Considering the maximum rated power and speed of the engine (11.63kW and 2200rpm), the engine tests were conducted at; Full load test condition (11.63kW), 75% (¾) load condition (8.72kW), 50% (½) load condition (5.82kW), and 25% (¼) load condition (2.91kW).

To achieve a good engine working temperature, the engine was run at high idle speed using fossil diesel for about 30mins. Some defined loads were imposed on the engine through the governor which controls the speed. For example, the maximum load was selected by adjusting the speed to 2200rpm. (The speed was measured using a tachometer). The load from the dynamometer was imposed on the engine by adjusting tensions (F_1 and F_2). The engine speed dropped but was not raised back to the initial speed. The speed was not adjusted back because we needed to study the effects of the imposed engine load on the performance of the considered fuel options. After about 30minutes running, the engine sound and vibration stabilized and the following readings were taken: tension F_1 and F_2 on the spring balance of the dynamometer, fuel consumption by the engine with time (t), the speed N was measured with a tachometer. The load F_1 and F_2 were further increased gradually by adjusting the belt screws, and further values of F_1 , F_2 , N, and t were noted. The procedure continued up to six times when the engine begins to stall.

The following mathematical equations were used in analyzing the information obtained in the course of this short term engine performance test.

A. Brake Power (B_P)

This is the engine generated power that is available to the shaft

$$Bp = \frac{2\pi NT}{60,000}\ kW \qquad 1$$

If the angular velocity "ω" is known, then the brake power (Bp) can be calculated using the equation;

$$Bp = \frac{\omega T}{60,000}\ kW \qquad 2$$

where,
$\omega = 2\pi N$
T = Torque
N = Rotational speed of shaft in (rpm)

B. Brake Mean Effective Pressure (Bmep)

$$\text{Bmep} = \frac{2\pi NT}{L\times A\times N}\ \text{bar or kW/m}^2 \qquad 3$$

Bmep and torque since it is independent of engine size and speed.

C. Specific Fuel Consumption (SFC)

This is generally defined by the equation;

$$\text{SFC} = \frac{Mf}{w} \qquad 4$$

Where
M_f = Mass rate of fuel flow into the engine
W = engine power

The two main aspects of specific fuel consumption include;

i. Indicated Specific fuel consumption (isfc) given by;

$$\text{Isfc} = \frac{Mass of fuel flow/hour}{Indicated power}\ \text{Kg/kW-h} \qquad 5$$

ii. Brake specific fuel consumption (bsfc) – this is a measure of the engine's ability to convert fuel energy into mechanical work.

$$\text{bsfc} = \frac{Mass of fuel flow/hour}{Brake Power}\ \text{kg/kW-h} \qquad 6$$

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International Journal of AdvancedResearch in Science, Engineering and Technology
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bsfc decreases as engine speed increases, it reaches minimum and then increases at high speeds. Fuel consumption increases at high speeds because of greater friction losses. At low engine speeds, the longer time per cycle allows more heat loss and fuel consumption goes up. Brake specific fuel consumption, generally decreases with increase in engine size, being best and lowest for very large engines.

D. Mechanical Efficiency (M_{eff})

$$M_{eff} = \frac{BrakePower}{IndicatedPower} = \frac{Bp}{IP} = \frac{bmep}{imep} = \frac{isfc}{bsfc} \quad 7$$

FUEL BLEND	SPEED Rpm	LOAD (F1-F2)g N	FUEL CONSPT RATE Ml	FUEL USED PER HOUR kg/h	TORQUE (F1-F2)g.r N-M	BRAKE POWER kW	BRAKE M.EFF.PR KN/m^2	BRAKE SP.FC kg/kWh	THERMAL EFFICENCY %	FUEL EQUV. POWER kW

E. Brake Thermal Efficiency (B_{etf})

$$B_{etf} = \frac{BrakePower}{HeatSupplied} \quad 8$$

F. Fuel Equivalent Power (Fep)

$$Fep = \frac{Hf \times Mf}{3600} \text{ kw} \quad 9$$

Where;
Fep = Fuel equivalent Power (kW)
H_f = Heat value of fuel KJ/Kg
M_f = Rate of fuel consumption. Kg/h.

IV. DATA PRESENTATION

Tables 1-4 presents the average calculated and measured results for the experiments carried out. The mathematical formulas/equations used in calculating the parameters are as stated in equation 1-9 above. However, some items such as: engine speed, dynamometer load(L_D) and fuel consumption were obtained from direct measurements. The engine speed was measured from the flywheel with a digital tachometer. The engine load (L_E) was imposed on the engine through the in-built governor; which was measured as a percentage of the known maximum adjustment. The dynamometer loads were imposed on the engine through the adjustable dynamometer belt and its values were read-off as forces (F_1 and F_2) from the weighing spring attached to the canvas belt. The test fuel was introduced into the engine through two burettes (50ml each) fixed on a board and connected by flexible hosts. The initial and finial fuel level after the experiment were noted.

Table 1 Constant Engine Load (L_E) Test (at a Maximum engine Load condition) For Different Fuel Types

B0	2159.429	117.72	0.430416	1.326371	17.658	3.983425	245569.9	0.356655	9.61E-05	15.27493
B10	2157.714	128.9314	0.51198	1.501618	19.33971	4.361068	268957.5	0.3689	0.000108	17.91891
B20	2178.429	121.9243	0.479802	1.485466	18.28864	4.168691	254340.3	0.355561	1.00E-04	17.16363
B50	2172.286	88.29	0.502268	1.558639	13.2435	3.004084	184177.4	0.540085	7.30E-05	17.81205
B70	2177.286	81.00257	0.469771	1.515292	12.15039	2.764711	168975.5	0.575077	6.83E-05	17.02925
B100	2170.286	88.29	0.475466	1.560699	13.2435	3.000826	184177.4	0.534565	7.60E-05	16.34044

Table 2 CONSTANT ENGINE LOAD TESTS (T)
75% (3/4) Load Test
Average Calculated Values

FUEL TYPE	SPEED Rpm	LOAD (F1-F2)g N	FUEL CONSPT RATE ml	FUEL USED PER HOUR kg/h	TORQUE (F1-F2)g.r N-M	BRAKE POWER kW	BRAKE M.EFF.PR KN/m^2	BRAKE SP.FC kg/kWh	THERMAL EFFICENCY %	FUEL EQUV. POWER kW
B0	1872.833	117.72	0.374191	1.153108	17.658	3.452553	245569.9	0.392967	8.33E-05	13.27957
B10	1876.667	112.815	0.419897	1.304535	16.92225	3.312557	235337.8	0.468719	8.17E-05	14.69607
B20	1847.167	125.895	0.411335	1.273493	18.88425	3.625088	262623.4	0.405362	8.72E-05	14.71441
B50	1854.333	122.625	0.36477	1.131954	18.39375	3.552535	255802	0.356948	8.64E-05	12.93592
B70	1846	114.45	0.412391	1.330208	17.1675	3.304724	238748.5	0.442762	8.17E-05	14.94923
B100	2169.5	101.37	0.507083	1.589826	15.2055	3.449188	211463	0.467257	8.74E-05	17.42702

Average Calculated Values
Table 3 CONSTANT LOAD CONDITION ENGINE TESTS (T)

FUEL TYPE	SPEED Rpm	LOAD (F1-F2)g N	FUEL CONSPT RATE Ml	FUEL USED PER HOUR kg/h	TORQUE (F1-F2)g.r N-M	BRAKE POWER kW	BRAKE M.EFF.PR KN/m^2	BRAKE SP.FC kg/kWh	THERMAL EFFICENCY %	FUEL EQUV. POWER kW
B0	1405.833	191.295	0.414596	1.27762	28.69425	4.178777	399051.1	0.314675	8.91	14.713
B10	1436.333	168.405	0.41968	1.303861	25.26075	3.768062	351301.4	0.363564	9.29	14.689
B20	1455.833	176.58	0.279051	0.863941	26.487	4.020213	368354.9	0.234877	9.664966	9.982
B50	1468.5	122.625	0.276641	0.858471	18.39375	2.81691	255802	0.33205	6.847027	9.811
B70	1471.5	120.99	0.292112	0.942236	18.1485	2.782778	252391.3	0.379336	6.878232	10.589
B100	1458.833	127.53	0.266776	0.836408	19.1295	2.900558	266034.1	0.342139	7.350305	9.168

ISSN: 2350-0328
International Journal of AdvancedResearch in Science, Engineering and Technology
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Table 4 CONSTANT LOAD CONDITION ENGINE TESTS (T)
T4, 25% (1/4) Load Test
Average Calculated Values

FUEL TYPE	SPEED Rpm	LOAD (F1-F2)g N	FUEL CONSPT RATE Ml	FUEL USED PER HOUR kg/h	TORQUE (F1-F2)g.r N-M	BRAKE POWER kW	BRAKE M.EFF.PR KN/m^2	BRAKE SP.FC kg/kWh	THERMAL EFFICENCY %	FUEL EQUV. POWER kW
B0	1065.667	117.72	0.199149	0.613698	17.658	1.954517	245569.9	0.332648	4.71E+00	7.067556
B10	1058.167	137.34	0.209063	0.649518	20.601	2.253487	286498.2	0.327726	5.556572	7.317066
B20	1058.333	138.975	0.202686	0.627517	20.84625	2.2822	289908.9	0.331589	5.486621	7.250558
B50	1064.5	122.625	0.20963	0.650524	18.39375	2.033208	255802	0.3537	4.942093	7.434161
B70	1037.333	117.72	0.219815	0.709034	17.658	1.901226	245569.9	0.389773	4.699287	7.968313
B100	1054	104.64	0.212772	0.667092	15.696	1.717287	218284.4	0.397959	4.351778	7.312388

V. GRAPHICAL PRESENTATIONS OF EXPERIMENTAL DATA

For easier and in-depth understanding of the experimental results, graphical presentations as shown on Fig.1-7 were used to express and compare the relationship between selected fuel parameters against the various blends.

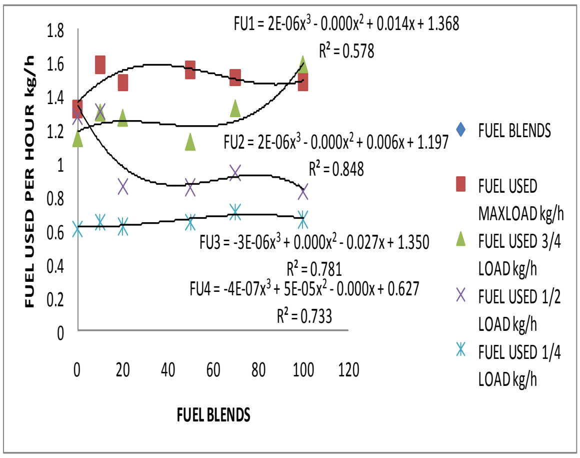

Fig 1 Comparing the Trends of Fuel used per Hour against Fuel Blends

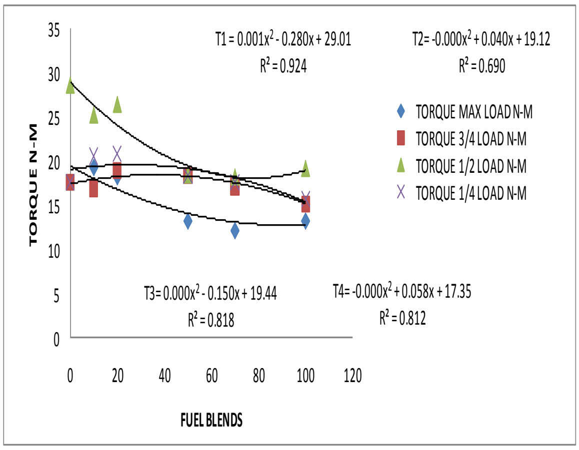

Fig 2 Comparing the Trends of Engine Fuel Torque against Fuel Blends

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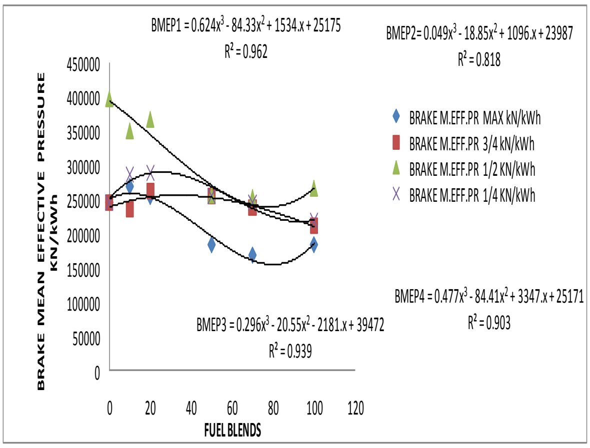


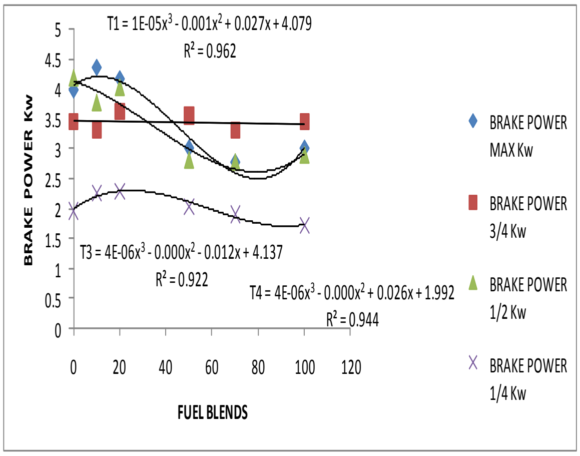
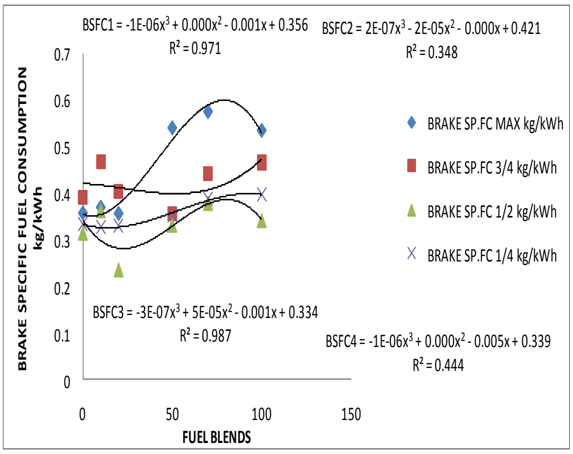


Fig 3 Comparing the Trends of Brake Power against Fuel Blends

Fig 4 Comparing the Trends of Brake Mean Effective Pressure against Fuel Blends



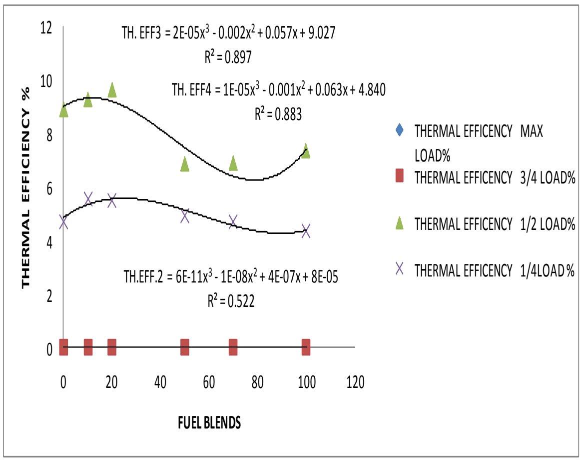


Fig 5 Comparing the Trends of Brake Specific Fuel Consumption against Fuel Blends

Fig 6 Comparing the Trends of Thermal Efficiency against Fuel Blends

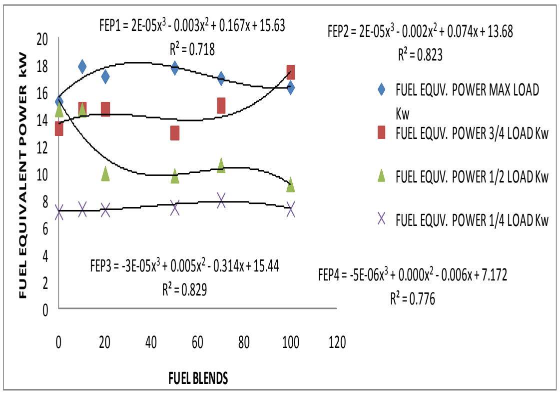


Fig 7 Comparing the Trends of Fuel Equivalent Power against Fuel Blends.

ISSN: 2350-0328

International Journal of AdvancedResearch in Science, Engineering and Technology

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VI. RESULTS DISCUSSION

The average values of the results obtained from the experiments carried out are as presented in tables 1-4, graphical illustrations with developed model equations were also presented on fig. 1-7 as shown above. The various graphical trends obtained in the course of the analysis were differentiated by the line colors as indicated on the graphs for clearer understanding of the work.

The trend of fuel used per hour (FU) kg/h against blends is shown in Fig 1. The trend of the graph revealed that Fossil diesel had the lowest fuel use value of 1.326kg/h closely followed by B_{20} with a value of 1.485 kg/h while B_{100} had highest value of 1.561kg/h. This could be attributed to the high viscosity of the green fuel. Lowering of the viscosity level of the biodiesel by blending with fossil fuel improved the fuel consumption considerably. This is in agreement with the work of Sahoo and Puhan, (2008) [10] which advocated for the proper blending of biodiesel and fossil fuel for optimal performance. A third order polynomial equation ($FU = 9 \in -07X^3 - 0.000X^2 + 1.324$) was developed for the trend with a determinant coefficient of $R^2 = 0.994$. (x = diesel blend).

The trend of torque against fuel blends (Fig 2) shows that the engine torque decreased from left to the right of the graph. A maximum torque value of 19.34kW was recorded at B_{10} and a minimum value of 12.15kW at $B_{70.}$ The produced fuel exhibited higher torque generation potentials with its blends than pure fossil diesel. This could be as a result of the fact that the engine burns more biodiesel per hour which results in higher power generation and torque value when compared with fossil fuel.

The brake power trend (Fig. 3) show a sharp decline from left to right hand side of the graph. The implication is that engine brake power generation is lower with pure biodiesel. This may be connected with the low injector atomization of green fuels. However, the brake power generated improved with blending of biodiesel with fossil diesel.B_{10} generated brake power (4.361 kW) higher than the fossil diesel (3.98 kW)

Brake mean effective pressure of the engine was investigated as shown in Fig 4. The result revealed that there was a slight increase in Brake mean effective pressure from B_0 (245569.90KN/m^2) to B_{10} (268957.50KN/m^2). Thereafter, a steady decrease in value was noted up to B_{70} before moving up again to 184177.40KN/m^2 for B_{100}. This could be as a result of the minimal blending with fossil fuel which improved the viscosity and flash point of the biodiesel.

The brake specific fuel consumption which is also a measure of the fuel economy of an engine based on the fuel used was equally investigated as shown on Fig 5. Fossil diesel generated 0.356655kg/kWh which gradually increased with the blends. B_{70} had the highest value of 0.575077kg/kWh. This improved performance could be as a result of the controlled blending of the produced fuel which enhanced the fuel flow within the engine and ignition point.

Graph of engine thermal efficiency against fuel blends (Fig 6) reveal that a gradual increase from B_0 (961E-05 %) to a peak of (1.00E-4 %) at B_{20}. Thereafter, a sharp decrease was recorded. The blending of the biodiesel with minimal volume of fossil diesel could be the reason for the performance recorded. It was also observed that the engine thermal efficiency decreased with increase in the blending ratio.

Fig 7 investigated the trend of equivalent power against the fuel blends in a single cylinder diesel engine. The result show a gradual increase from B_0 (15.27493kW) with a peak at about B_{30} thereafter decreased constantly to a value of (16.34044 kW) for B_{100}. The equivalent power is a function of the heat value of the fuel and rate of fuel consumption, the result recorded could be as a result of the minimal blending with fossil fuel which improved the viscosity and heat vale of the produced fuel.

VII. CONCLUSION AND RECOMMENDATION

One of the key factors affecting the performance of diesel engine is the quality of fuel it uses. The produced fuel was tested in an unmodified single cylinder diesel engine to determine its characteristics under constant engine loads. The results of the experiments conducted show that biodiesel requires blending of about 10 – 30 % with fossil diesel for an improved performance in an unmodified diesel engine under constant engine loads. It is therefore recommended given the outcome of this research work that, about 20–30% of biodiesel should be properly blend with about 80-70% of fossil diesel in an unmodified diesel engine for best engine performance.

ISSN: 2350-0328
International Journal of AdvancedResearch in Science, Engineering and Technology
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