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Technical and Environmental Improvements of LNG Carrier's Propulsion Machinery by Using Jatropha Bio - diesel fuel

E. H. Hegazy[1], M. A. Mosaad[2], A.A. Tawfik[3], M. A. Hassan [4], M. Abbas[5]

ABSTRACT

Modern dual fuel engine is looking for a new alternative fuel because of the higher prices and lower production of diesel. Jatropha biodiesel is a promising substitute as an alternative fuel has gained significant attention due to the predicted shortness of conventional fuel and environmental concerns. The operating test values obtained from the Jatropha oil is closely matched with the values of conventional diesel and can be used in the existing diesel engine without any modification. This paper studies the most promising propulsion alternatives of future Liquefied Natural Gas (LNG) carriers running by dual fuel engine used boil off natural gas (BOG) from LNG carriers cargo tanks vessels as main fuel and Jatropha biodiesel oil as pilot fuel. An experimental investigation was performed to investigate the influence of dual-fuel combustion on the performance and exhaust emissions of a direct injection (DI) diesel engine fueled with Jatropha oil and natural gas (NG). A single cylinder diesel engine was set up and arranged to measure and study the engine performance at all running parameters. The test rig aims to simulate the operating conditions of BOG in propulsion machinery of LNG carriers. The present investigation studies the engine performance and emissions experimentally at different loads to obtain the best load condition by using natural gas and Jatropha biodiesel oil. The obtained results showed that the thermal and volumetric efficiency of diesel engine is higher than Jatropha biodiesel engine. The specific fuel consumption, exhaust gas temperature, HC, CO2 and NO were comparatively higher in Jatropha biodiesel. While an appreciable increase in cylinder pressure and CO emission when using diesel. It was observed that the combustion characteristics of the Jatropha biodiesel followed closely with that of the base line diesel. This means that, Jatropha biodiesel can be used instead of diesel fuel oil with safe engine operation.

Keywords: Dual fuel diesel engine; Natural gas; Jatropha oil, Dual fuel combustion emissions

Abbreviations

CR : Compression ratio,
DI : Direct injection,
BSFC : Brake specific fuel consumption,
NOx : Nitrogen oxides
JME : Jatropha methyl esters
CO : Carbon monoxide,
HC : Hydrocarbons,
BOG : Boil off gas
LNG : Liquefied natural gas

1. INTRODUCTION

Jatropha Curcas Linnaeus (or physic) nut is a shrub that originates from Central and South. America and belongs to the euphorbia family. In this study it will be referred to as Jatropha. It is cultivated in Central and South America, South-East Asia, India and Africa. It is a drought resistant plant that can grow in arid and semi-arid areas in the tropics. The shrub or small tree can grow up to 6 meters height and the roots can go 7 meters deep to reach ground water

1, 2, 3, Professor Of Naval Arch. And Marine Eng. Dept., Faculty Of Eng., Port Said Univ.
4, Professor Of Marine Eng. Dept, Faculty Of Eng. And Tech., Aast.
5, Lecturer In College Of Marine Transport, Aast.

From the seeds, oil can be gained. Depending on the variety, peeled or dehusked seeds contain 43-59% oil [1]. In practice 1 kg of seeds gives 200 to 300 ml of oil, depending on the quality of the pressing process. The rest of the weight of the seeds is left as press cake [2].
One of the drawbacks of using pure vegetable oil in an engine is that vegetable oil has a higher viscosity than diesel; this can result in clogging of filters and nozzles especially in cold climates [3].
Jatropha oil can be esterifies like any other vegetable oil to give the oil characteristics that are favorable for operation in a diesel engine [4]. In this study Jatropha Methyl Esters (JME) oil is used. This is done because the stationary engine that is used is capable of operating on pure vegetable oil even in cold climates.
The present research is aimed to exploring technical feasibility of Jatropha biodiesel in direct injection compression ignition engine without any substantial hardware modifications. In this work JME, as shown in Table 1, was investigated for its performance as a diesel engine fuel. The running effect of Jatropha biodiesel was study to predict the performance and exhaust emissions of diesel engine. Performance parameters like indicated power and specific fuel consumption were determined. Exhaust emissions like CO2, CO and HC have been evaluated. For comparison purposes experiments were also carried out on 100% Jatropha biodiesel and diesel fuel With Natural gas.

Table 1 JME and Diesel fuel properties [5]

	JME	Diesel
Density at15 ^{o}C (kg/l)	0.866	0.820
API gravity	31.78	41.06–31.14
Kinematic viscosity at 40^{o}C (cSt)	19.2	1.6–7.0
Flash point (^{o}C) (closed cup)	61	55
Calorific value (MJ/kg)	47.38	44.3
Pour point (^{o}C)	4.4	1.7
Inorganic acids	Nil	Nil
Cetane no.	63.5	Min. 55
Diesel index	47.12	Min. 48
Water content (% by volume)	0.5	0.1
Specific gravity (29^{0} C)	0.944	0.792
Viscosity at 40^{0} C	36.92	2.86
Carbon residue (% by mass)	0.5	0.1
Ash content (% by mass)	0.002	0.01
Carbon content (% by weight)	86.98	87.45
Hydrogen content (% by weight)	12.99	11.3

1.1 Present Study

The present work is an effort to evaluate the feasibility of popular alternative fuels in the form of JME as a total replacement for diesel oil in LNG carriers. An experimental investigation was performed to investigate the influence of dual-fuel combustion on the performance and exhaust emissions of a DI diesel engine fueled with natural gas. A single cylinder diesel engine was set up and arranged to measure and study the engine performance parameters at all running conditions as described in the paper E. H. Hegazy et al [6]. The test rig aims to simulate the operating conditions of BOG in propulsion machinery of LNG carriers. The engine was operated at a constant speed of 1200 rpm and at four different loads: low (1.5 kw) and high (6 kw), which were about 20% and 85% respectively of the rated torque output of the engine at 1200 rpm and 7 kw maximum. The present investigation studies the percentage of CO, CO2, and HC in the exhaust outlet from the engine by using exhaust gas analyzer to find out the engine performance and emissions experimentally at different loads to obtain the best load condition by using natural gas and diesel fuel.

2. Experimental Test Rig

In the present study, a one cylinder air cooled diesel engine is used and coupled with an electrical dynamometer. The usage of the electrical dynamometer is to load the engine at different modes. Figures 1and 2 show the experimental test rig. Table 2 1indicates the specifications of the experimental setup of the single cylinder diesel engine used in the study.

A complete natural gas unit of a spark ignition (SI) engine was used to feed the engine with natural. It is controlled by a pressure regulator and auxiliary valve.

Different measurement apparatus were used to measure the various group of engine performance parameters such as engine emissions (CO, CO2 & HC), fuel consumption and air to fuel ratio (A/F). The test engine is provided with a surge tank to absorb the pressure pulsation at the intake.

Fig 1: Overall view of experimental setup.

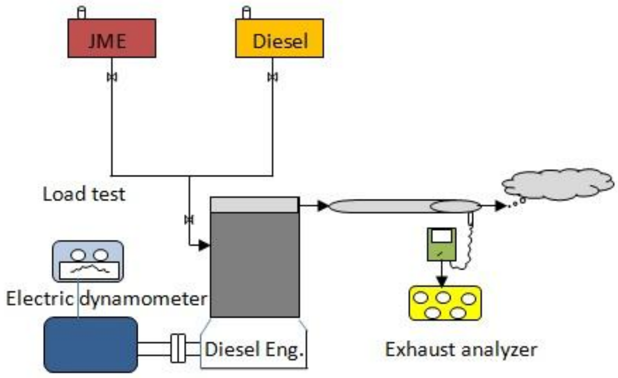

Fig 2: Schematic diagram of the test rig.

Table 2: Diesel Engine and Dynamometer Specifications

Items	Specification
Number of cylinders	One – cylinder
Cooling system	Air cooler
cycles	Four stroke
Bore	102 mm
Stroke	110 mm
Max. power	7.5 kw
Cubic capacity	896 cc
Tank capacity	7.5 liter
Compression ratio	16.5
Dynamometer power	7 kw
Dynamometer Voltage	220 v
Dynamometer Frequency	50 hz
Dynamometer RPM	1500 rpm

2.1 The electrical dynamometer

An electrical dynamometer is coupled to the test engine, as shown in figure 1, to measure and control the engine load at different operating modes. The dynamometer consists of two main parts; the first part is electrical generator and the second one is loading unit. The loading unit, as shown in Fig 3, consists of four rows, each row consists of 3 lamps. Each lamp is 500-w. That means that the load of each row is 1.5 kw and therefore, the overall load of the loading unit 6 Kw.

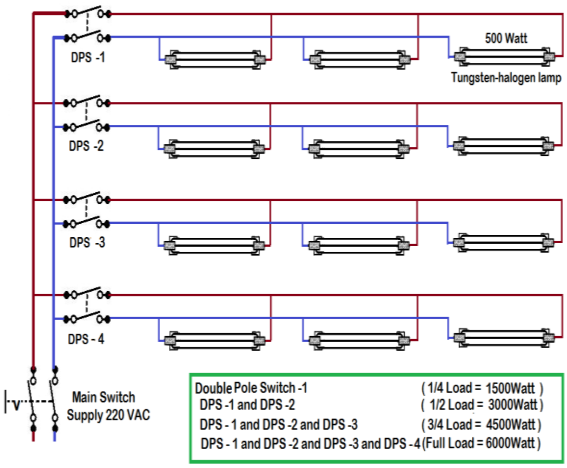

Fig 3: Schematic diagram of the load unit

2.2 Gas kit

The natural gas kit consists of the cylinder, pressure regulator solenoid valve, manual valve, pressure gauge and the filling power valve. The cylinder is pressurized by 200 bar natural gas. The gas kit includes (on/off) feeding valve and pressure gauge with feed line as shown in Figure 1. The function of the feeding valve is to control the amount of natural gas supported to the feed line and the function of pressure gauge is to give an indication to the amount of available natural gas in the tank and the feed line is then connected to the pressure regulator.

2.3 Pressure regulator

The pressure regulator, as shown in Fig 1, has a great role in the natural gas kit structure as it reduces pump pressure from 200 bar to atmospheric pressure. The pressure regulator has a solenoid valve to manage the feeding of the natural gas to the engine. A 12 volt battery used to start the operation of solenoid valve to open and close the feed line of natural gas to the engine. In additional, a power valve is used to manually control the amount of natural gas supplied to the engine.

2.4 Instrumentation and Measurements

The engine is equipped for measurements of engine parameters such as, the mass flow rate of intake air and the natural gas used in some operation modes were measured by using two different calibrated orifice meter-manometer arrangements. The exhaust gases are analyzed by using the calibrated exhaust gas analyzer type IMR 3000 to detect carbon monoxide, carbon dioxide, unburned hydrocarbon and exhaust temperature. The gases are picked up from the exhaust pipe of the engine as shown in figure 1 to measure the exhaust pressure. The rate of fuel consumption was measured by using a calibrated burette and a stopwatch (± 0.1 second) for an adequate volume of fuel (in cubic centimeter) consumed by the engine. An optical tachometer used to measure the engine shaft speed. The rotational speed of the engine shaft used in this study is 1200 rpm at different loads

2-5 Error analysis and experimental uncertainty

From the analysis of the experimental uncertainty, it was found that the uncertainty in the determination of the diesel fuel consumption was less than 1%, whereas the constant rotational speed of the engine measured and monitored within 1 rpm deviation, nominally at 1200 rpm. The uncertainty associated with the measurement of the engine load was 1.2%. The maximum uncertainty in the pressure values was estimated to be 2.2%. The prediction of the exhaust gas emissions depend on the recorded values of the gas analyzer. The overall uncertainty of gas emission values in ppm was estimated to be 3% at the lowest values of flow velocity.

3- Results and Discussion

Engine performance and emissions were studied experimentally at different operating loads. Results of the measurements for pure diesel operation and JME operation are tested

Firstly engine was operated using pure diesel at different loads and then was operated using Jatropha biodiesel at the same loads. The effect of dual fuel performance is predicted theoretically and then verified with experiments. To evaluate the accuracy of the test results, a comparison is held between them and the literature. The following paragraphs discuss the results and experimental output and compare the results in Brake Specific Fuel Consumption (BSFC), thermal efficiency and volumetric efficiency.

3.1Mass flow rate of air

It can be observed from Figure 4 that, as the applied load, and hence the generated power, increases the mass flow rate decreases. This applies to all of the fuel types used in the study. This can be attributed to not using throttle valve in the engine. Therefore, there is no throttling to the inlet air. That is why the amount of air decreased. Furthermore, it can be seen that the mass flow rate in pure diesel fuel operation is higher than that in Jatropha biodiesel operation. This is because of the delay period of combustion between Jatropha and diesel fuel. In the case of running in dual fuel, it is clear that a decreased occurred in mass flow rate of air about 9.97% in diesel and 9.21% in Jatropha biodiesel and that is regarding to natural gas taken place the air in the suction stroke entered to the cylinder.

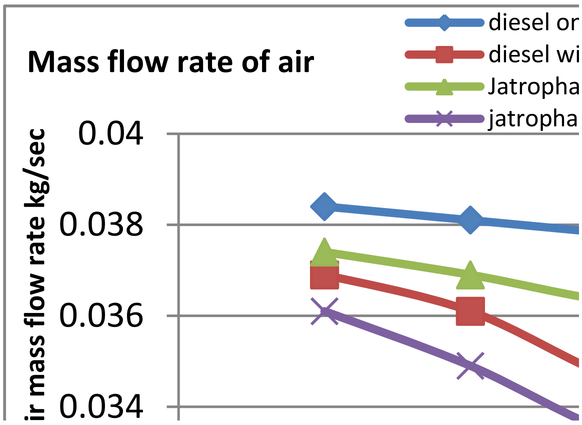

Fig 4: Variation of air mass flow rate using different fuels.

3.2 Fuel Consumption

It can be observed from Figure 5 that the fuel consumption increases as the load increases for all of the types of the fuel used in the study. However, when running in Jatropha biodiesel the fuel consumption, at specific load, is about 11% higher than that when running in pure diesel .This can be attributed to the lower calorific value of Jatropha compared to diesel [7].

Furthermore, it can be seen that, in the case running in dual fuel, at specific load, the fuel consumption of Jatropha biodiesel with natural gas is about 9% higher than that when running in diesel and natural gas. This, also, can be attributed to the lower calorific value of Jatropha biodiesel less than in compared to diesel.

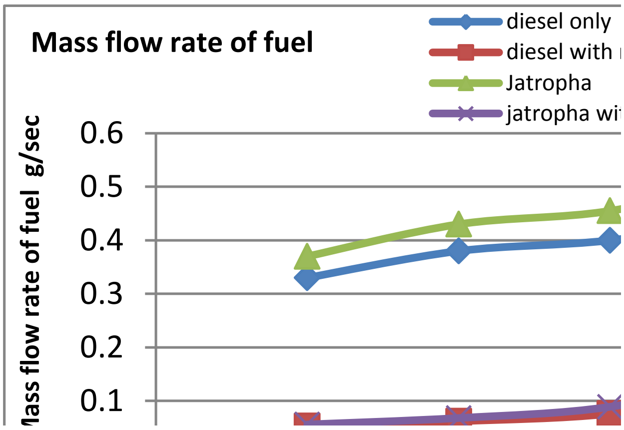

Fig 5: Variation of fuel consumption using different fuels.

3.3 Brake-specific fuel consumption (BSFC)

Brake specific fuel consumption, BSFC, is a key parameter when comparing the performance of different engines. This is because increasing the BSFC means the power cost increases and the running cost increases, as well.

The results presented in figure 6 show the significant increase of BSFC in part loads. However, as the load increased, the BSFC decreased. Figure 6, also, declares that when the engine running in Jatropha biodiesel, the BSFC is about 6.5% higher than when running in diesel. Furthermore, when the engine running in Jatropha biodiesel with natural gas. the BSFC is about 7.4% higher than when running in diesel with natural gas [8] .The total BSFC is affected by the presence of gaseous fuel at part load under dual fuel operation [9].In comparison study with a research paper dated in 2012 by Roussos G and et al, it is found that the trend of BSFC in dual fuel engine almost same with difference about 6.4% in part load and 4.7% in full load and that give a good impression about the results [10].

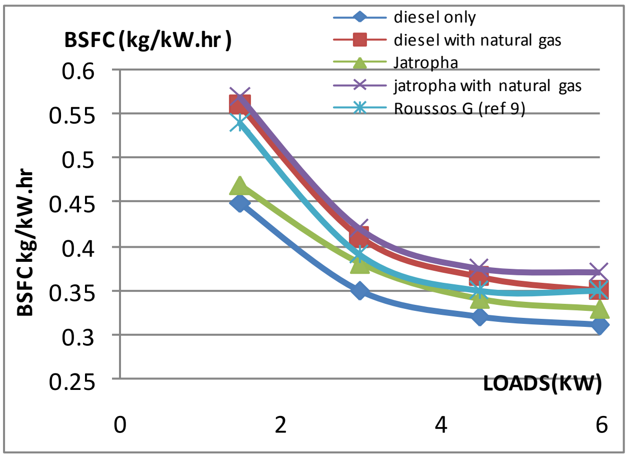

Fig 6: Variation of BSFC using different fuels.

3.4 Cylinder indicated pressure

As shown in Figure 7, the results show a slight decrease in cylinder indicated pressure by 6.4% when using Jatropha biodiesel as compared to that when using pure diesel. This can be attributed to the higher viscosity and volatility of Jatropha biodiesel resulting in poor atomization through the fuel injector and thus lower cylinder pressure [11]. Also as shown, the maximum combustion pressure is strongly affected by the presence of gaseous fuel. As the quantity of gaseous fuel increases, keeping engine load constant, peak cylinder pressure increases significantly while the slope remains almost the same regardless of engine load. For Jatropha dual fuel operation, peak cylinder pressure is lowered by 9.2% compared to the dual diesel operation. It is also shown that the peak pressure in Jatropha dual trend is started slightly small compared with dual diesel and when the load increased the peak pressure jumped to close to diesel dual trend and that regarding to increase of natural gas in full load. During operation in dual fuel, there was no dangers exist for the engine structure.

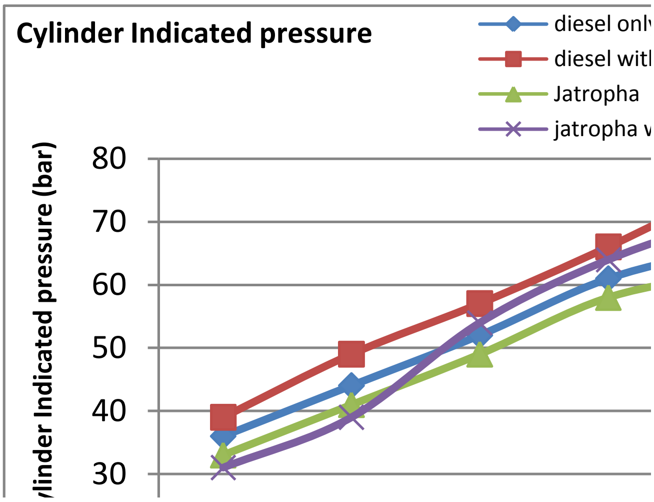

Fig 7: Cylinder indicated pressure using different fuels.

3.5 Exhaust gas temperature

The exhaust gas temperature gives an indication of the amount of waste heat going with exhaust gases. The exhaust gas temperature reflects the status of combustion inside the combustion chamber. The reason for of the rise in the exhaust gas temperature may be due to ignition delay and increased quantity of fuel injected. The exhaust gas temperature of the different fuel combustion, as shown in Figure 8, indicates that the temperature increased by 4.3% when useing Jatropha biodiesel more than when using diesel in different loads and

this may be related to the higher relative density and lower energy of Jatropha, comparing to diesel oil.
On the other hand, the exhaust temperature of dual fuel engine operated using Jatropha biodiesel and natural gas increased by 4.1% more than that when using diesel fuel. This may be because of the late burning of gas mixture in dual fuel mode caused some rise in temperature in the exhaust [12].

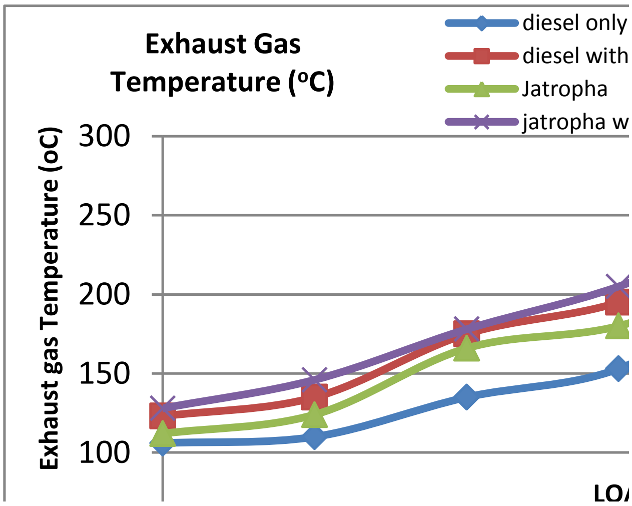

Fig 8 Variation of exhaust using different fuels.

3.6 Thermal Efficiency

Figure 9 shows thermal efficiency as a function of power output. It shows that the thermal efficiency increases drastically with load conditions. The thermal efficiency for both fuels deviations not more than 1.2%, and the characteristics are almost same. This means that the use of Jatropha biodiesel that is not preheated, instead of diesel, does not deteriorate thermal efficiency.
In dual fuel combustion the diesel portion of the fuel auto ignites first due to compression heat and remaining gas-air mixture burns in progression. The combustion stages are a bit different from typical compression ignition of diesel. The flame propagation speed of natural gas is relatively slower compared to diesel, especially when the mixture is lean [13]. In comparison study with a research paper dated in 2009 by Md. Ehsan and et al [14], it was found that the trend of thermal efficiency in dual fuel engine almost same with difference about 1.8% in part load and 1.2% in full load. This gives a good relation between diesel and Jatropha. However, it was found that for loads varying in the higher range, the engine could be run almost as efficiently as diesel-only operation, without changing the injection timing.

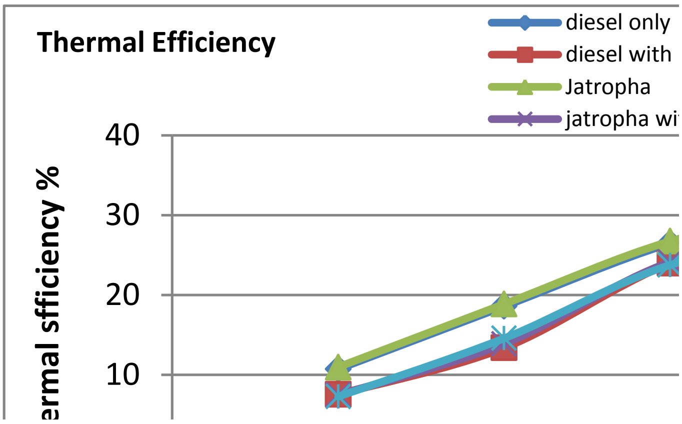

Fig 9: Thermal Efficiency

3.7 Volumetric Efficiency

The volumetric efficiency for the reference measurement results in a value for volume of the intake mixture. The volumetric efficiency should give similar results for each measurement, since it is assumed that the engine takes in a fixed volume each inlet stroke.
Figure 10 shows volumetric efficiency as a function of power output; it shows that the volumetric efficiency decreases drastically as the load increases. It also shows that the engine has slightly higher volumetric efficiency when diesel is used as a fuel, but the maximum volumetric efficiency increased by 1.4% and that means no big changed occurred when used Jatropha as main fuel.
In case of dual fuel, an increase in the natural gas fraction in the total fuel mixture results in a decrease in air inlet. This is because natural gas replaces air in the intake mixture. A decrease in air fraction in the intake mixture results in a decrease in the value of actual volumetric efficiency, [15]. Consequently, volumetric efficiency decreases for both diesel and Jatropha biodiesel. In comparison study with a research paper dated in 2008 E. Kerkhof [16] and et al, it was found that the trend of volumetric efficiency in dual fuel engine almost the same with a difference about 1.9% in part load and 0.9% in full load. This gives a good relation between diesel and Jatropha.

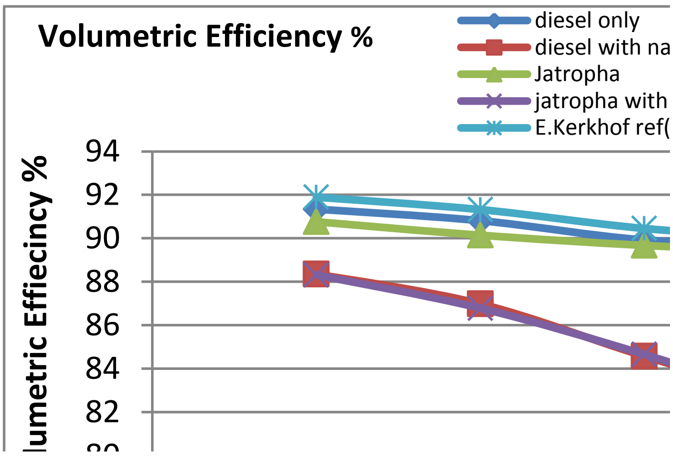

Fig 10: Volumetric Efficiency

3.8 CO emissions

As it is clear from the Figure 11 that the average values of CO emissions for Jatropha biodiesel about 6.4% lower than these for pure diesel.. Several reasons have been reported to explain the general CO decrease when substituting conventional diesel for Jatropha: (1) the additional oxygen content in the Jatropha fuel, which enhances complete combustion of the fuel, thus reducing CO emissions. (2) The increased Jatropha cetane number. The higher the cetane number, the lower the probability of fuel- rich zones formation, usually related to CO emissions. In case of dual fuel engine, it is revealed that, at part load, increasing the amount of gaseous fuel leads to a sharp increase of CO concentration. This is due to the slow combustion rate of gaseous fuel which maintains the charge temperature at low levels resulting to a reduction of the oxidation process of carbon monoxide [17]. At higher engine load, CO emissions increase with increasing natural gas mass ratio and beyond a certain value of gaseous fuel mass ratio they start to decrease as a result of the high gas temperature and faster combustion rate. In general CO emission values under dual fuel operation are considerably higher when compared to normal diesel operation. At ¾ loads, it is clear that the deviation between dual Jatropha and dual diesel increased more than at full load. This is due to the exhaust rate outlet at ¾ load is less than the exhaust rate outlet at full load.

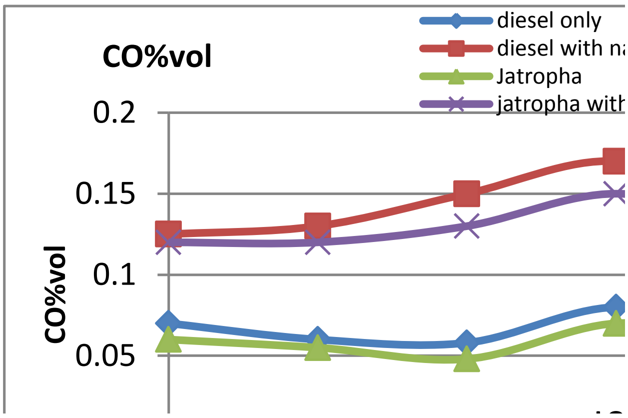

Fig 11: CO emissions using different fuels

3.9 CO2 emissions

The carbon dioxide emission from the diesel engine with different fuel is shown in Figure 12. It can be seen that there was a slight increase in CO2 emissions when using Jatropha biodiesel as compared when using diesel fuel. The Jatropha biodiesel followed the same trend of CO2 emission, which was about 7.8% higher than that in the case of diesel This is due to high carbon content in Jatropha molar construction, $C_{56}H_{101}O_6$, and to lower carbon content in diesel oil $C_{10}H_{20}$. However, when using natural gas, the amount of CO2 decreased in both types of fuel because the amount of pilot fuel decreased but still slightly high in using Jatropha (about 4.6%) because of the carbon content in Jatropha [18].

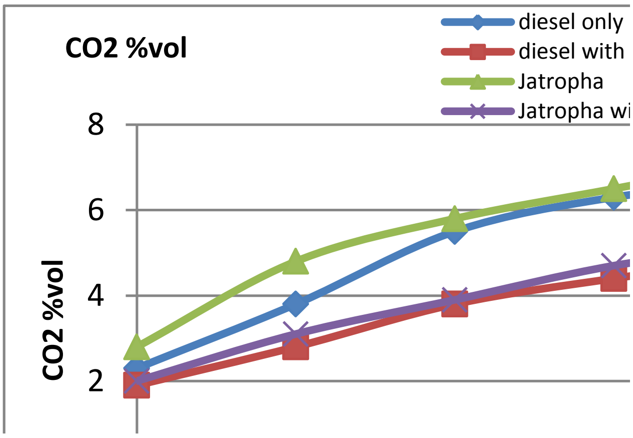

Fig 12: CO2 emissions using different fuels.

3.10 Unburnt hydrocarbon (HC) emissions

The variation of HC emissions with brake load, as shown in figure 13, indicates that the unburnt hydrocarbon (HC) emissions of Jatropha biodiesel fuel was significantly increased as compared with diesel fuel HC emissions. This could be attributed to the higher oxygen content of HC in Jatropha molar construction $C_{56}H_{101}O_6$. When compared to that in diesel $C_{10}H_{20}$. It, also, can be seen that, at half brake load, there is an increase of about 8% of HC emissions in Jatropha more than that in diesel with increasing gaseous fuel mass ratio until a certain limit where they start to decrease to 5.5% in full load. This is due to the increase of burnt gas temperature, which promotes the oxidation of unburned hydrocarbons [19]. In the case of running in dual fuel engine, it is observed that at low load the HC emissions increase as the percentage of gaseous fuel increases. Under dual fuel operation, the filling of the crevice volumes with unburned mixture of air and gaseous fuel during compression and combustion while the cylinder pressure continues to rise, is an important source dominating the formation of HC emissions. This offers an explanation for the considerably increased HC emissions in dual fuel operation by Jatropha; about 15.7% higher than that of dual fuel operation by diesel.

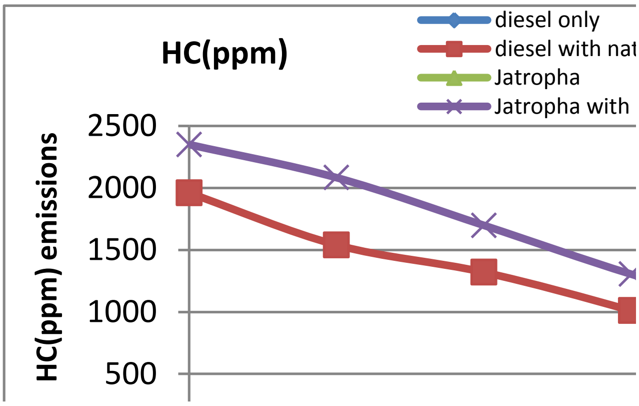

Figure 13: HC emissions using different fuels.

3.11 NO emissions

The results in Figure 14 indicated that NO emissions of Jatropha biodiesel fuel was significantly increased by about 9% as compared to diesel fuel NO emissions. Reasons for this behavior could be: (1)The adiabatic flame temperature; some authors state that it is slightly higher for Jatropha because of its oxygen at denature which help for more complete combustion and so higher temperature and NOx emission. (2) The reduction

in soot formation with Jatropha; radiation from soot produced in the flame zone is a major source of heat transfer away from the flame, and can lower bulk flame temperatures by25–125K, depending on the amount of soot produced at the engine operating conditions. (3) Jatropha typically contains more double bonded molecules than diesel. These double bonded molecules have a slightly higher adiabatic flame temperature, which leads to the increase in NOx production for Jatropha.
In the case of running with dual fuel engine, it was noticed that at high engine load and low mass ratios of natural gas there is a sharp decrease of NO concentration from about 900 ppm in pure diesel to about 200 ppm in dual diesel and from 100 in JME to about 200 ppm in Dual Jatropha JME. A possible explanation for the reduction of NO concentration, observed in most cases is the less intense premixed combustion, the reduction of gas temperature due to increase of the specific heat capacity [20], the slower combustion and finally the reduction of oxygen concentration due to presence of the natural gas mass ratio, which replaces an equal amount of air in the cylinder charge.

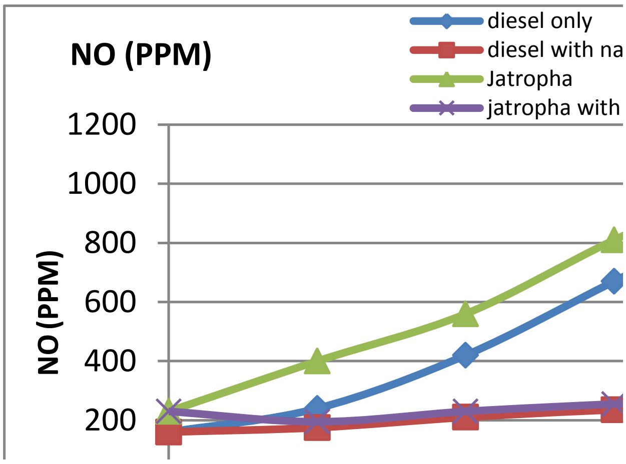

Figure 14 NO emissions using different fuels.

4. Conclusions

The present work is an experimental investigation on the use of Jatropha methyl ester as pilot fuel in dual fuel engine using natural gas as the main fuel.
The results indicate that the temperature increased by 4.3% when used Jatropha biodiesel. Moreover, the indicated cylinder pressure decreased by 6.4% when using Jatropha biodiesel as compared when using diesel. The thermal efficiency for both fuels deviates not more than 1.2% and the characteristics are almost same. Slightly higher volumetric efficiency when diesel is used as a fuel but the maximum increased about 1.4%. NOx emissions of Jatropha biodiesel fuel were significantly increased by about 9% as compared to diesel fuel NOx emissions. the average values of CO emissions for Jatropha biodiesel about 6.4% lower than these for pure diesel. The Jatropha biodiesel followed the same trend of CO2 emission, which was higher than in case of diesel about 7.8%.
The measuring results for the test rig indicate that the combustion characteristics of the Jatropha biodiesel followed closely with that of the base line diesel and this means that, it is possible to use Jatropha biodiesel instead of diesel fuel oil with safe engine operation.

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