Biogas-Straight Vegetable Oil Dual Fuel Engine for Power Generation in Remote Rural Locations

K. Narayana Reddy and A. Ramesh

***Abstract*—Decentralised power generation in remote rural locations can be done using locally available biomass based renewable energy sources. A single cylinder stationary diesel engine was run in the neat vegetable oil (Jatropha oil) – biogas dual fuel mode to study the potential of this method of operation and also device simple methods to enhance the performance. Increasing the rate of pumping of the injected Jatropha oil and retarding its timing of injection resulted in considerable improvements in thermal efficiency and reduction in emissions. The smoke level at 60% of biogas energy dropped from 2.25 to 0.9 BSU, 2.15 to 0.6 BSU and 1.4 to 0.5 BSU with 7mm, 8mm and 9mm diameter injection pump plungers respectively. Further improvements in the combustion rate of the inducted biogas were possible with enhanced swirl level of the inducted air. At 60% biogas share the thermal efficiency went up by 8%, HC emissions dropped by 26% and smoke level dropped by 17% with enhanced swirl. On the whole, energy shares of biogas as high as 80% could be achieved with good engine performance.**

***Index Terms*—Alternative engine fuels, biogas engines, dual fuel engines, vegetable oil as fuel.**

I. Introduction

Diesel engines have been used for decades for commercial transportation, industrial power generation, locomotives, marine systems and for agricultural applications. Considering this it is essential to explore methods to use alternative fuels in diesel engines. In a vast country like India with remotely located villages, generating the power requirements using locally available resources that are renewable is a good option. Methyl esters of vegetable oils that are generally known as biodiesels (once they meet standards) have properties close to diesel. In the case of rural applications it may be difficult to produce biodiesel using available resources hence it will be wise to evaluate the use of raw or straight vegetable oils for diesel engine applications in rural applications. Most of the work on straight vegetable oils has indicated higher smoke levels, lower thermal efficiencies and lower power outputs when used in conventional diesel engines [1]-[4]. This is mainly because of their significantly higher viscosity and carbon residue than diesel. Lower NOx levels have also been reported with straight vegetable oils [5], [6]. Improving fuel atomization in a diesel engine by altering injection parameters can enhance the performance with non edible vegetable oils. Vegetable oils have a calorific value and cetane number comparable to diesel and their flash point is very high. However, they also have other disadvantages like poor volatility and high carbon residue. Properties of different non-edible vegetable oils are presented in Table I.

TABLE I: Properties of Jatropha Oil [7]

Property	Diesel	Jatropha oil
Density (kg/m^3)	840	918.6
Calorific Value (kJ/kg)	42490	39774
Viscosity (cst)	4.59	49.93
Cetane Number	45-55	40-45
Flash Point (℃)	50	240
Carbon Residue (%)	0.1	0.64

Biogas which mainly contains methane and carbon dioxide along with small proportions of other gases and water vapor can be produced easily through anerobic digestion of waste plant and animal matter. Typical properties of biogas are given in Table II. Biogas has a high self-ignition temperature and hence can be used in the dual fuel mode [8]-[10]. Dual fuel engines are modified diesel engines wherein a gaseous fuel like biogas, natural gas, producer gas or hydrogen is inducted along with air and compressed in the normal way. Then near top dead centre (TDC) a small amount of diesel is injected so that itself ignites and becomes the ignition source for the inducted gaseous fuel air mixture. Biogas diesel dual fuel operation is known to reduce NOx emissions and also particulate emissions significantly. [11]. Thecarbondioxide in biogas also helps suppress knocking [12]. Experiments conducted earlier by the authors indicated that biogas can be inducted in a dual fuel engine while straight vegetable oil can be injected for ignition instead of diesel. This means that an engine can be run by completely renewable fuels that can be sourced from the villages itself. This work focuses on this method of operation.

TABLE II: Typical Properties and Composition of Biogas [13]-[15]

Composition (%)	Methane (CH_4) = 50-70 Carbon dioxide (CO_2) = 25-50 Hydrogen (H_2) = 1-5 Nitrogen (N_2) = 0.3-3
Auto ignition temperature (℃)	650
Calorific value (MJ/kg)	17
Density at 1 atm and 15℃ (kg/m^3)	1.2
Flame speed at atmospheric pressure and temperature for φ=1 (m/s)	0.25
Flammability limits (vol% in air)	7.5-14
Stoichiometeric air-fuel ratio (kg of air/kg of fuel)	5.7
Research octane number	130

II. Details of the Work

The aim of this work was to use biogas in a constant speed stationary diesel engine in the dual fuel mode with straight

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vegetable oil namely, Jatropha oil being used for ignition instead of diesel. Methods like optimizing the injection timing, enhancing the injection rate and air swirl level have been adopted to improve combustion. Biogas which is the main fuel is itself slow burning due to the high levels of CO_2 present in it and hence swirl enhancement has also been adopted to raise the combustion rate. Performance, emissions and combustion parameters have been presented, compared and discussed with these modifications.

III. Experimental Setup and Experiments

A Single cylinder 4-Stroke air-cooled direct injection diesel engine developing 4.4 kW at 1500 rpm with a displacement volume of was 669 cc and compression ratio of 17.5:1was used. Fuel flow rate was measured on the mass basis. A positive displacement gas flow meter measured the biogas flow rate.

The amounts of methane, carbon dioxide and carbon monoxide in biogas were measured with help of a NDIR (Non Dispersive Infrared) analyzer. Through a positive displacement gas flow meter the biogas was allowed into the intake manifold of the engine. The intake swirl level was enhanced during some of the experiments by properly placing a mask on the inlet valve. A flame ionization detector (FID) was used for analyzing the unburned hydrocarbon emissions. Smoke level was measured using a standard BOSCH smoke measuring system. Nitric oxide emission was measured with a chemiluminescence analyzer. In cylinder pressure was measured with a water-cooled Kistler make flush mounted piezoelectric transducer. Data from 100 consecutive cycles were recorded and processed with specially developed software to obtain combustion parameters. The heat release rate was also computed using this in house developed software. The engine was initially run in the neat Jatropha oil mode and then biogas was allowed in steps. When biogas was allowed at constant BMEP (load) the amount of injected Jatropha oil was automatically reduced by the governor to hold the speed constant. Thus experiments were conducted at various biogas energy ratios which is defined below.

Biogas energy ratio (%) =
(Energy from biogas)/Total Energy Supplied)*100

The maximum energy ratio or biogas flow rate corresponded to the condition where the engine started to misfire because of the small amount of Jatropha oil that was used. Experiments were done to determine the influence of injection rate, injection timing and biogas energy ratio. Three different fuel injection pump plunger diameters namely 7, 8 and 9 mm diameters were tried. Injection rate is proportional to the area of the plunger. With each plunger three static injection timings were evaluated to determine the best based on thermal efficiency. Finally the best configuration was further improved by enhancing the air swirl level to enhance mixing. This was done by using a masked intake valve.

IV. Results and Discussion

The results of experiments conducted in the biogas Jatropha oil dual fuel mode are presented and discussed below. First the injection timing of Jatropha oil was optimized with fuel injection pump plungers of different diameters. Then the best combination was selected for enhancing the swirl level.

- Effect of injection timing in the Jatropha oil-biogas in dual fuel mode with the 7mm diameter plunger.

Though these experiments were done using the three plungers (7, 8 and 9 mm diameter) the results with the plunger diameter of 7 mm alone at 100% load are presented in detail as the trends in the other cases were similar (Figs 1-4).Three different injection timings namely indicated in the figures with the 7mm diameter plunger were chosen. It is seen (Fig. 1) that the brake thermal efficiency reduces as the amount of inducted bio gas (Biogas energy) increases because of the presence of a large amount of CO_2 in the biogas, which tends to suppress the flame speed. At very high biogas energy ratios the thermal efficiency drops significantly because the amount of Jatropha oil injected in very small and this leads to a weak ignition source and even misfiring. The injection timing of 35 °bTDC which is more advanced than the one that is the best for neat Jatropha oil operation is the best in the dual fuel mode based on brake thermal efficiency due to the increase in the ignition delay.

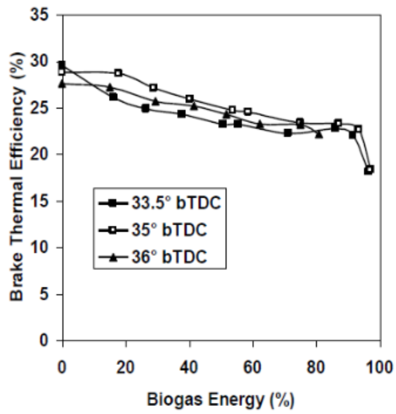

Fig. 1. Brake thermal efficiency with the 7mm plunger.

Unburned hydrocarbon emissions are higher in the dual fuel mode as seen in Fig. 2. The HC is mainly contributed by the Biogas. As the biogas energy level goes up the HC concentration goes up and then there is a fall probably due to improved combustion of the biogas as its concentration goes up. At very high biogas energy levels the HC goes up due to poor combustion (amount of Jatropha oil is low). At full load, in the case of Jatropha oil the NO level (Fig. 3) increases.

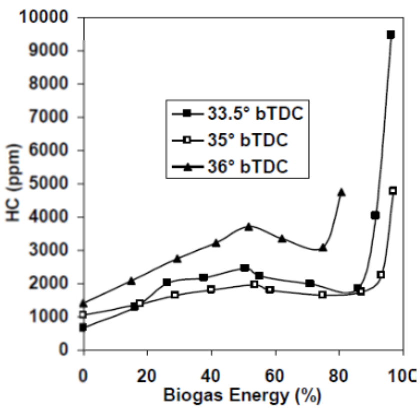

Fig. 2. HC emission with the 7mm plunger.

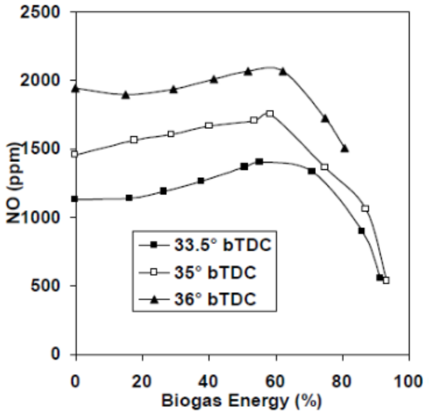

Fig. 3. NO emission with the 7mm plunger.

With a rise in the biogas energy input because of improved combustion rate as seen later in the heat release curve NO goes up. The NO concentration gets increased when the injection timing is advanced because of the increase in the ignition delay and rise in the initial heat release rate.

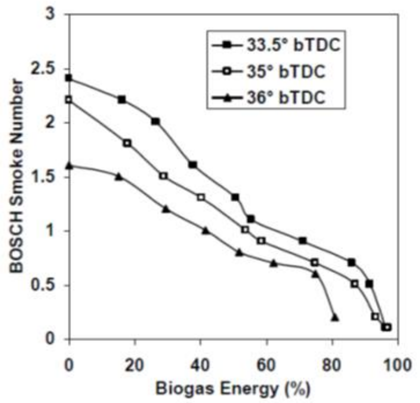

Fig. 4. Smoke emission with the 7mm plunger.

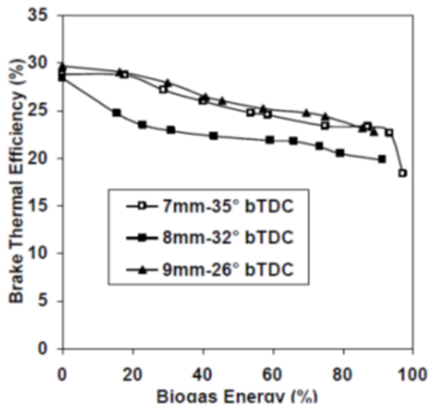

Fig. 5. Brake thermal efficiency with different plungers.

The injected Jatropha oil is the source of smoke which reduces with increase in the biogas energy proportion (Fig. 4). A biogas energy rate of about 80% at an injection timing of 35°bTDC seems to be the most suitable from the point of view of thermal efficiency smoke, HC and NO emissions.

- Effect of plunger diameter at the best injection timings.

Experiments like the ones described in the previous section were done with the three plungers and the best static injection timings were selected as were selected as 35°, 32° and 26° bTDC with 7mm, 8mm and 9mm diameter plungers respectively (Figs. 5-9). These experiments were done at the best injector opening pressure (IOP) of 220 bar. In the case of the 7mm diameter plunger the low injection pressure developed could form a localized pilot spray which could lead to strong ignition source for the combustion of the biogas. In the case of the 9mm diameter plunger the high rate of injection probably leads to accumulation of a large quantity of pilot fuel during the delay period which leads to good combustion of the biogas. At the biogas energy share of 60% the brake thermal efficiency with 7mm, 8mm, 9mm diameter plungers are 24.5%, 21.8% and 25.1% respectively.

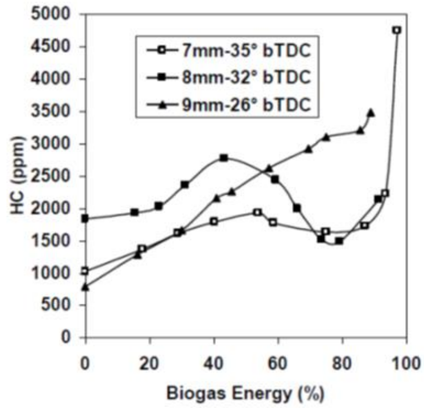

Fig. 6. HC emission with different plungers.

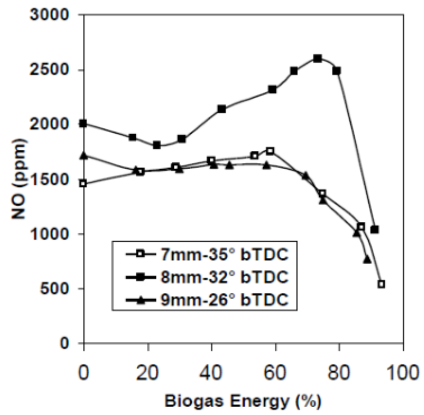

Fig. 7. NO emission with different plungers.

HC emissions are lower with 7mm and 9mm diameter plungers (Fig. 6) where the brake thermal efficiency is better. Though the 9mm diameter plunger leads to high brake thermal efficiency it does not lead to low HC levels at high biogas flow rates. The higher injection rate with 9mm diameter plunger will lead to high injection pressures and then greater dispersion of the pilot fuel. This will lead to poor combustion particularly when the amount of pilot fuel is low i.e. at high biogas flow rates. NO emission is higher with the 8mm diameter plunger as shown in Fig. 7 due to the dominance of the premixed phase combustion. The NO level first rises with increase in the biogas energy rate due to improved combustion and falls when the pilot fuel quantity reduces. At biogas energy rate of about 60% the NO levels are 1746ppm, 2310ppm and 1625ppm with 7mm, 8mm and 9mm diameter plungers respectively. The values of smoke at 60% of biogas energy are 0.9BSU, 0.6BSU and 0.5BSU with 7mm, 8mm and 9mm diameter plungers (Fig. 8). Smoke level mainly results from combustion of the injected pilot fuel. High injection pressures (9mm diameter plunger with high injection rate) lead to lower smoke.

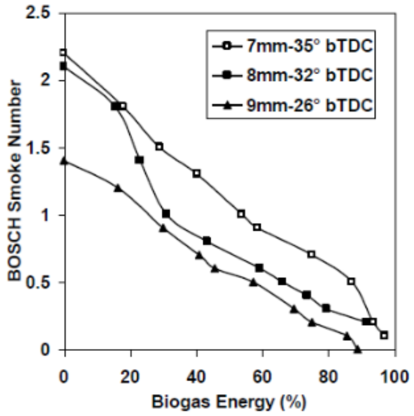

Fig. 8. Smoke emission with the different plungers.

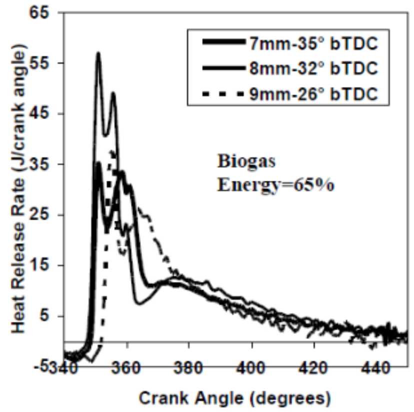

Fig. 9. Heat release rate with different plungers.

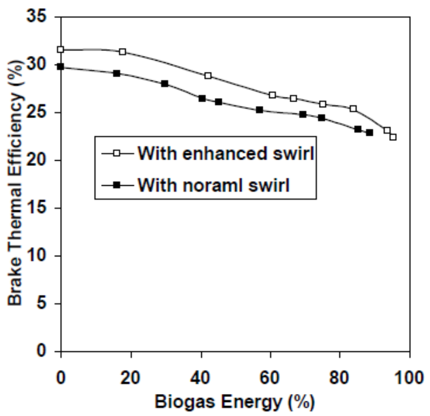

Fig. 10. Brake thermal efficiency with swirl.

Fig. 9 clearly shows that there is a lot of difference in the second phase of combustion with the three types of plungers. The heat release rate in the first phase of combustion is mainly due to the pilot fuel combustion along with a small quantity of the entrained biogas. The second phase is mainly due to the burning of biogas thought to be from multiple ignition centers. The high ignition delay with the 8mm diameter plunger and the high rate of injection lead to a strong first phase which is rather advanced and this one of the reasons for the lower thermal efficiency. Maximum rate of pressure rise was also highest with 8mm diameter plunger followed by 9mm then 7mm diameter plunger. The maximum rate of pressure rise increased with biogas admission indicating higher combustion rates and fell off when the pilot Jatropha oil quantity was reduced. The 9mm diameter plunger was the best based on the brake thermal efficiency. Hence it was taken up for further experimentation.

- Effect of Enhanced Swirl in the Jatropha oil-Biogas Dual Fuel Mode of Operation.

Fig. 10 and Fig. 11 indicate that there is a significant increase in brake thermal efficiency and reduction in HC emission with enhanced swirl. This is because of the enhanced combustion rate of biogas. This resulted in enhanced NO level and reduction in the smoke level due to improved mixing. As the biogas energy was raised (i.e. the pilot fuel quantity falls) the second phase of combustion which is due to burning of the biogas becomes more significant.

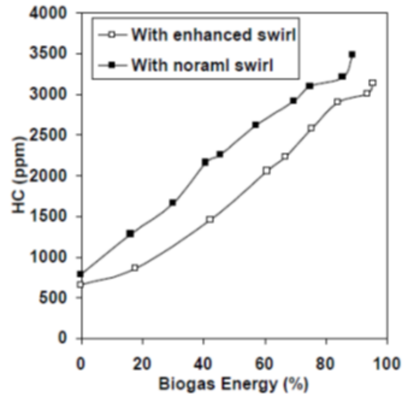

Fig. 11. HC emission reduction with swirl.

V. Conclusions

- Straight vegetable oil – biogas dual fuel mode is viable for small scale decentralized power generation using locally available renewable resources. Up to 80% of the energy input can come from biogas at full load.
- Injection timing, injection rate and swirl level play an important role in enhancing the combustion process. High injection rates achieved with the 9mm diameter plunger coupled with relatively retarded injection timings coupled with enhanced swirl levels can improve the performance and reduce emissions considerably.
- Significant reduction in NO and smoke emissions can be achieved with biogas induction as compared to neat vegetable oil operation due to reduced vegetable oil injection.
- The smoke level at 60% of biogas energy dropped from 2.25 to 0.9 BSU, 2.15 to 0.6 BSU and 1.4 to 0.5 BSU with 7mm, 8mm and 9mm diameter plungers respectively.
- Enhanced swirl improves combustion of the injected vegetable oil and also the combustion rate of biogas. At a biogas energy share of 60% the thermal efficiency went up by 8%, HC emissions dropped by 26%, NO emission went up by 6.8% and smoke level dropped by 17% when enhanced swirl.

References

[1] N. Hemmerieіn and V. Korte, "Performance, exhaust emissions and durability of modern diesel engines running on rapeseed oil," *Society of Automotive Engineers, Paper No.* 910848, USA, 1991.

[2] K. R. Babu, P. Ramesh, and K. V. Reddy, "Utilization of cotton seed oil in a diesel engine," in *Proc. the National Conference on IC Engines And Combustion*, Bangalore, India, 1994.

[3] R. Altin, S. Cetinkaya, and H. S. Yucesu, "The potential of using vegetable oil fuels as fuel for diesel engines," *International Journal of Energy Conversion Management*, vol. 42, pp. 529-538, 2001.

[4] A. S. Huzayyin, A. H. Bawady, M. A. Rady, and Dawood, "Experimental evaluation of diesel engine performance and emission

using blends of jojoba oil and diesel fuel," *International Journal of Energy Conversion Management*, vol. 45, pp. 2093-2112, 2003.

[5] C. A. Silvio, R. Carlos, V. G. Marious, S. R. Leonardodos, and F. Guilherme, "Performance of diesel generator fueled with palm oil," *Journal of Fuels*, vol. 81, pp. 2097-2102, 2002.

[6] A. S. Ramdas, S. Jayaraji, and C. Muraleedharan, "Characterization of and effect of using rubber seed oil as fuel in the compression ignition engines," *International Journal of Renewable Energy*, vol. 30, pp. 795-803, 2005.

[7] M. S. Kumar, A. Ramesh, and B. Nagalingam, "An experimental comparison of methods to use methanol and Jatropha oil in a compression ignition engine," *Biomass and Bioenergy*, vol. 25, pp. 309–318, 2003.

[8] A. Henham and M. K. Makkar, "Combustion of simulated biogas in a dual-fuel diesel engine," *Energy Conversion and Management*, vol. 39, issue 16-18, pp. 2001-2009, 1998.

[9] R. G. Papagiannakis and D. T. Hountalas, "Combustion and exhaust characteristics of a dual fuel compression ignition engine operated with pilot diesel fuel," *Energy Conversion and Management*, vol. 45, pp. 2971-2987, 2007.

[10] M. Feroskhan and I. Saleel, "Investigation of the effects of biogas composition on the performance of a biogas-diesel dual fuel CI engine," *Taylor & Francis*, 2016.

[11] N. N. Mustafi and R. R. Raine, "A study of the emissions of a dual fuel engine operating with alternative gaseous fuels," *SAE paper 2008-01-1394*, 2008.

[12] M. Lounici, K. Loubar, M. Tazerout, M. Balistrou *et al*., "Experimental investigation on the performance and exhaust emission of biogas-diesel dual-fuel combustion in a CI engine," *SAE Technical Paper*, 2014-01-2689, 2014.

[13] E. Porpatham, A. Ramesh, and B. Nagalingam, "Investigation on the effect of concentration of methane in biogas when used as a fuel for a spark ignition engine," *Elsevier, Fuel*, vol. 87, issues 8–9, pp. 1651-1659, 2008.

[14] S. Bari, "Effect of carbon dioxide on the performance of biogas-diesel dual fuel engine," *International Journal of Renewable Energy*, vol. 9, pp. 1007–1010, 1996.

[15] W. Anggono, M. Wardana *et al*., "Effect of inhibitors on biogas laminar burning velocity and flammability limits in spark ignited premix combustion," *International Journal of Engineering and Technology*, vol. 5, pp. 4980–4987, 2013.

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