An Experimental Investigation on Engine Performance and Emission of a Diesel-piloted Biogas Engine

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Abstract In this work, a Direct Injection Compression Ignition (DICI) engine was modified into a dual fuel engine that utilizes biogas as the primary fuel and diesel as pilot fuel. The main aim of this research was to find an alternative fuel for the diesel engines and to reduce harmful exhaust emissions from these engines while maintaining high thermal efficiency. The performance and emission characteristics of the dual fuel engine was studied. The engine was tested at four different diesel substitution levels, i.e. Biogas 0%-100% Diesel, Biogas 50%-50% Diesel, Biogas 70%-30% Diesel and Biogas 80%-20% Diesel. The effect of diesel substitution with biogas on exhaust gas temperature and performance parameters of brake specific fuel consumption, brake power and brake thermal efficiency were studied. The effect of using biogas on emissions of CO, HC and CO₂ was also investigated. The performance and emission characteristics of the dual fuel engine were compared with those of the conventional diesel engine. The results show that use of biogas leads to an increase in specific fuel consumption and a decrease in brake thermal efficiency. This is because of the lower calorific value of biogas compared to diesel. The results also show that use of biogas causes a decrease in exhaust gas temperature and can therefore be used as a way of reducing NOₓ emission in DICI engines. On the other hand, HC emission increases with increase in biogas.

Keywords Biogas, DICI Engine, Dual fuel engine, Engine performance, Exhaust emission

1. Introduction

The use of compression ignition (CI) engines as power source for automobiles is common in many parts of the world due to their high thermal efficiency, excellent fuel economy and low regulated emissions of unburned hydrocarbon (HC), carbon monoxide (CO) and carbon dioxide (CO₂) compared to those of spark ignition (SI) engines [1]. From an environmental point of view, however, diesel engines generally exhaust a larger amount of particulate matter (PM) and nitrogen oxide (NOₓ) pollutants emissions than those of gasoline engines [2]–[4].

In addition to challenges of emissions from CI engines due to use of fossil fuels, the price of the fossil fuels like diesel is constantly increasing because of the limitations of deposit and increase in demand resulting from industrialization [5]. The regulations for PM and NOₓ emissions from diesel engines are also becoming more stringent hence the need to find alternative fuels for the CI engines, which would reduce harmful exhaust emissions while maintaining high thermal efficiency [6], [7].

Biogas, which is produced by anaerobic fermentation of organic material that can be extracted from varied sources like animal wastes, vegetable wastes, wastes from households, wastes from the food and fodder industry and waste from productive livestock husbandry is abundantly available as fuel for CI engines and is regarded as an alternative clean energy resource in view of its environmental friendly nature [8]–[11]. It is, therefore, a suitable renewable source of energy for CI engines. Biogas is one of the most important renewable fuels, composed mainly of methane (30-70%, by vol.) and carbon dioxide (20-40%, by vol.) and is a promising alternative fuel for internal combustion engines since it is renewable and environmental friendly [12]. Methane (CH₄) is the main component of biogas and it exhibits greater resistance to the knock phenomenon due to its higher octane rating and auto-ignition temperature, making it appropriate for engines with high compression ratios [13]. In addition, the carbon content of methane is also relatively low compared
to that of conventional diesel fuel, resulting in a significant decrease in pollutant exhaust emissions.

Utilizing biogas in engines, when compared to fossil fuels avoids any additional greenhouse gas emissions. Due to organic nature of the components of biogas, burning it in a gas engine for power generation emits the same amount of CO₂ into the atmosphere as was originally absorbed in the process of photosynthesis in the natural CO₂ cycle [14].

Although there is research work that has been done on combustion and emission characteristics of bio-fuels and petroleum fuels with the dual-fuel concept [15]–[21], it is necessary to investigate in more detail the performance characteristics of the biogas-diesel dual-fuel engine as well as finding ways of reducing the exhaust emissions.

2. Experimental Procedure and Equipment

2.1. The Experimental Setup

The engine typically used for this study is a single cylinder four stroke Direct Injection (DI) diesel engine. It is a water cooled, naturally aspirated constant speed compression ignition engine whose major specifications are shown in Table 1. The engine was coupled to a hydraulic dynamometer through which load was applied by increasing the water supply to rotor blades via a centrifugal pump. The engine was tested at 0, 25, 50, 75 and 100 percent brake load conditions. The engine had capability to run either on pure diesel or dual fuel mode. The engine was modified to run on biogas by introducing the gas in the intake manifold pipe through a mixing device. The biogas flow rate was kept fixed for a given speed and load, and a variable pilot injection controlled by a manual regulator mounted on the engine base and in contact with the engine stop lever, fixed at the injection pump. The volume of gas consumed by the engine was calculated by subtracting the initial from the final meter reading. Mass flow rate was calculated using volume of gas consumed, density of the gas and the time duration. The experimental apparatus consisted of the test engine, the dynamometer and control systems, the exhaust emission analyzer, temperature measurement system and the fuel gas supply system as shown in Figure 1.

![Diagram](image.png)

**Fig. 1. Photograph of the experimental setup**

2.2. Test Fuel

The pilot fuel injected into the combustion chamber which acts as the source of ignition was conventional diesel fuel. Diesel fuel was used as the reference fuel to compare engine performance on single and dual fuel modes. Biogas is produced from anaerobic biodegradation of organic solid waste in the absence of oxygen and the presence of anaerobic micro-organisms. The process is carried out in a biogas reactor within a specified time period. Biogas generated from water hyacinth was collected in a flexible gas bag (3 m³) for test in the laboratory (engine room). Properties of the gas are shown in Table 2.

![Diagram](image.png)

**Table 1. Engine Specifications**

<table>
<thead>
<tr>
<th>Specification</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Number of cylinders</td>
<td>1</td>
</tr>
<tr>
<td>Number of strokes</td>
<td>4</td>
</tr>
<tr>
<td>Number of holes in injector</td>
<td>3</td>
</tr>
<tr>
<td>Rated power (kW)</td>
<td>7.5</td>
</tr>
<tr>
<td>Speed (rpm)</td>
<td>1500</td>
</tr>
<tr>
<td>Compression ratio</td>
<td>16.5:1</td>
</tr>
<tr>
<td>Fuel consumption (g/kWh)</td>
<td>251</td>
</tr>
<tr>
<td>Injection pressure (bar)</td>
<td>200</td>
</tr>
<tr>
<td>Injection timing (°bdc)</td>
<td>23</td>
</tr>
<tr>
<td>Cylinder capacity (cc)</td>
<td>950</td>
</tr>
<tr>
<td>Combustion system</td>
<td>Direct injection</td>
</tr>
</tbody>
</table>

**Table 2. Fuel Properties of Biogas**

<table>
<thead>
<tr>
<th>Quantity</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Methane (%)</td>
<td>49-53</td>
</tr>
<tr>
<td>Carbon dioxide (%)</td>
<td>21-29</td>
</tr>
<tr>
<td>Nitrogen (%)</td>
<td>17-19</td>
</tr>
<tr>
<td>Density of biogas (kg/m³)</td>
<td>1.22</td>
</tr>
<tr>
<td>LHV (kJ/kg)</td>
<td>19,100</td>
</tr>
<tr>
<td>Octane number</td>
<td>130</td>
</tr>
<tr>
<td>Auto-ignition temperature (°C)</td>
<td>632-813</td>
</tr>
</tbody>
</table>
2.3. Experimental Procedure

The engine load was controlled using the hydraulic dynamometer while the exhaust gas was analyzed by Horiba MEXA-544GF emission analyzer. The dynamometer reading (load), engine speed, fuel consumption and exhaust gas temperature were recorded during experiments. Exhaust gases were analyzed on line by the emission analyzer in which HC, CO and CO2 were measured. A portable Laser type digital tachometer - RS 445-9557 was used to measure the engine speed at all operating conditions. Thermocouples were fixed at the exhaust manifold and at engine coolant inlet and outlet to the engine to measure the temperature of exhaust gas, temperature of cooling water at entry to the engine and temperature of cooling water at exit from the engine respectively.

The diesel fuel was pressurized by the high pressure injector system and the flow rate measured by the fuel flow meter. Biogas was introduced during the intake process by the gas mixing device which was installed in the intake pipe. The mixing device (chamber) consisted of a circular flow pipe of length 200mm and internal diameter 94mm, closed on both ends, with an inlet for air and for biogas each and an outlet for the mixture on the side. Air from the filter entered the mixing device tangentially through a pipe of diameter 50mm while biogas entered axially through a copper tube of internal diameter 10mm. The copper tube had three holes of diameter 3mm on the side, at 120° to each other, from which the gas exited as it mixed with air. The fuel-air mixture then flowed tangentially from the mixing chamber into the engine. The design of the mixing device allowed more time for the air and fuel gas to mix as the air flowed in a whirl inside the chamber.

The fuel gas flow rate was regulated by a control valve while the consumption was measured using a gas flow meter. The pilot liquid fuel consumption was measured by a calibrated glass tube (burette) by noting the time required for the consumption of 10ml of fuel. The engine was started and allowed to run at no load for about 5 minutes to reach the steady state at the start of test. After this the engine was loaded in steps and corresponding data for each load was noted. The torque reading by the dynamometer pointer was recorded and used to calculate brake power, BSFC and BTE.

Substitution of diesel with biogas in the ratio of Biogas:Diesel equal to 0:100, 50:50, 70:30 and 80:20 was achieved by regulating the flow rate of biogas using a flow control valve on the fuel line at the exit from the gas reservoir while the flow rate of diesel was regulated using the pilot fuel regulator (a bolt and nut mounted at the engine stop lever which could be tightened or loosened to push or release the lever in order to regulate the pilot fuel from pump outlet). Through a number of trials at different diesel and biogas flow rates, diesel substitution was approximated using Equation 2. The flow control valve and pilot fuel regulator were then marked and graduated at the various points corresponding to the required diesel substitution ratios.

2.4. Performance Evaluation of Engine

The engine (specifications given in Table 1) was tested on diesel as well as on dual fuel mode at the engine speed 1500 rpm and five loading conditions (0%, 25%, 50%, 75% and 100%). The flow rate of biogas was kept fixed at each substitution level and controlled by the biogas flow control valve. Each test was conducted with four replications. During each test, the engine load, engine speed and fuel consumption were measured. The observed data were utilized to calculate the engine thermal efficiency, specific fuel consumption and percent diesel substitution. The performance of the engine was evaluated in terms of brake power, brake specific fuel consumption, brake thermal efficiency and percent diesel substitution. The brake thermal efficiency and percent diesel substitution in dual fuel operation were determined as follows:

Brake thermal efficiency of engine on dual fuel mode:

$$\eta_{th} = \frac{\dot{W}_b}{\dot{m}_{f,d} \times CV_d + \dot{m}_{f,g} \times CV_g}$$

(1)

where $\eta_{th}$ = brake thermal efficiency, $\dot{W}_b$ = brake power, $\dot{m}_{f,d}$ = rate of flow of diesel fuel into engine, $\dot{m}_{f,g}$ = rate of flow of biogas fuel into engine, $CV_d$ = calorific value of diesel fuel and $CV_g$ = calorific value of biogas fuel [22].

Diesel Substitution:

$$ds = \frac{D_d - D_{dg}}{D_d} \times 100$$

(2)

where $ds$ = diesel substitution, per cent, $D_d$ = diesel consumption by the engine on single fuel mode, in kg/h, $D_{dg}$ = diesel consumption by the engine on dual fuel mode, in kg/h [22].

3. Results and Discussions

3.1. Brake Specific Fuel Consumption (BSFC)

Figure 2 shows variation of BSFC with engine load. The BSFC for dual fuel operation is the sum of the biogas and the liquid fuel i.e. diesel. It decreases with increase in load for all values of biogas flow rate. At low engine load condition of 25%, the BSFC for dual fuel combustion is much higher than for single fuel combustion. These results obtained indicate the lower rate of combustion of gaseous fuel due to the lower air-fuel ratio in the combustion chamber and a lower combustion temperature. Whereas the differences in the BSFCs between single and dual fuel combustions are much lower at higher engine loads. It was seen that at engine loads over 50%, where a high thermal load was imposed on the engine, the increase in combustion rate of biogas led to a significant improvement in the BSFC with dual-fuel combustions.
3.2. Brake Power

Figure 3 shows the variation in engine power output at different engine loads for both single and dual fuel modes of the engine. From the figure, it can be seen that the power output of the engine is slightly higher during single fuel mode than that of dual fuel mode under all test conditions. This is because of higher heating value of diesel as compared to biogas. The figure also indicates that with increase in engine load, the brake power increases linearly from no load to full load for single fuel mode and linearly from half load for dual fuel operation. The increase in brake power with load is due to the higher fuel consumption rate of the engine with increase in load.

3.3. Brake Thermal Efficiency

Figure 4 shows that the thermal efficiency of dual-fuel combustion was lower as compared to single-fuel mode at all engine loads. It has been reported that the lower thermal efficiency of the dual-fuel combustion is because of the effect of biogas residuals, combustion gas residuals and low combustion temperatures during the combustion process.

3.4. Exhaust Gas Temperature (EGT)

Figure 5 shows the variation of the exhaust gas temperature with engine load for all the fuel modes. The exhaust gas temperature measurements were conducted using a thermocouple mounted at a connection on the exhaust manifold. In this figure, the exhaust gas temperature increases linearly as the engine load is increased due to the increase of total energy input which is due to higher fuel consumption. However, the exhaust gas temperatures were found to be lower for dual-fuel combustion compared to single-fuel modes and these differences increased at higher engine loads. The low exhaust gas temperature can be explained by the decreased charge temperature with induction of biogas in the engine which acts as a heat sink due to the carbon dioxide component. It could also be due to the fact that the flame propagation speed of the pilot fuel in intake charge is reduced and then the fuel in the combustion chamber was not burned completely and therefore the combustion pressure and temperature are reduced[23].

Decrease in exhaust gas temperature with use of biogas indirectly shows the potential of biogas-diesel dual fuel operation for reduction of NOX emission. This is due to the fact that the most important reason for the formation of NOX in the combustion chamber is the high temperature of about 2000K at the site of combustion. In diesel engines, NOX formation is a highly temperature dependent phenomenon and takes place when the temperature in the combustion chamber exceeds 2000 K [3]. It was found that use of biogas reduces the exhaust gas temperature by up to 26% at minimum engine loading and upto 36.9% at maximum loading.

3.5. Carbon Monoxide Emission

Figure 6 shows the CO emissions at various engine loads. From the figure, the concentration of CO emissions for dual fuel operation were lower than those of diesel fuel operation for all combustion modes. In addition, CO concentration decreases with increase in biogas supply. In comparison between single and dual-fuel combustion, the concentration of CO emissions for the dual-fuel mode
were considerably lower than those of the single-fuel mode under all test conditions.

### 3.6. Unburned Hydrocarbon Emission

Figure 7 shows the HC emissions at different engine loads under different combustion modes. As shown in the figure, the HC emissions for diesel in single fuel combustion were lower than those of dual fuel combustion using biogas and diesel for all test conditions. With the induction of biogas into the engine, the CO₂ content in the mixture increases at the expense of fresh air which in turn reduces the air-fuel ratio and combustion temperature. At higher engine loads the HC emissions were found to be reducing for dual fuel mode whereas the same was found to be slightly increasing for single fuel mode. At part loads, the dual fuelled operation suffers from higher unburnt hydrocarbon emissions also due to overall lean mixture and incomplete combustion because of small quantity of pilot fuel [24]. At light loads the pilot quantity being small so flame cannot propagate fast and far enough to ignite the entire mixture. As the result it causes higher HC emissions but with increase in load the hydrocarbon emission decreases. As load progresses the pilot quantity increases and burns the surrounding fuel-air mixture sufficiently [23], [24].

### 3.7. Carbon Dioxide Emission

Figure 8 shows the concentrations of CO₂ emissions at different engine loads for single and dual fuel combustion modes. In this figure, the highest CO₂ emissions were produced by single fuel combustion, where as biogas-diesel dual fuel combustion produced lower CO₂ emissions. It can be seen that emission of carbon dioxide decreases with increase in biogas supply to the engine. This is as a result of substitution of some of the air with the gaseous fuel leading to decrease in CO₂ which is a product of complete fuel combustion.

### 4. Conclusions

The study was conducted to investigate various effects of dual fuel combustion of diesel and biogas on the performance and exhaust emission characteristics of a dual fuel single cylinder four stroke DI diesel engine under various experimental conditions. The following conclusions were drawn from the analysis:

1) The brake power was found higher for single fuel combustion than for dual fuel combustion at all loads. This is due to the higher calorific value of diesel as compared to biogas.

2) The test results showed very high brake specific fuel consumption of the engine at 25% load and comparatively much lower BSFC at higher engine loads.
loads, for dual fuel combustion. Also, the BSFC in dual fuel mode was higher than that of single fuel mode operation at all loads but it is comparable for both modes at higher engine loads.

3) The brake thermal efficiency of the test engine in dual fuel combustion mode was lower as compared to that in single fuel mode at all engine loads. It was also found that under high load conditions i.e. above 50% engine load, the BTE was much higher in single fuel combustion mode than that of dual fuel combustion mode for all the substitution percentages.

4) The CO emissions of the test engine were found higher for diesel in single fuel combustion mode as compared to dual fuel combustion mode. The results also indicated almost linear increase in the CO emissions with increase in engine load in case of both single and dual combustion modes. On the other hand, the results showed a linear decrease in HC emissions for dual combustion mode and a linear increase in the same in single fuel combustion mode with increase in engine load. This shows an increase in the combustion rate of HC at high loads for dual fuel operation.

5) Exhaust gas temperature increases with increase in engine load for both single and dual fuel mode combustion but it decreases with increase in biogas in the fuel mixture.

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Nomenclature

DIC1 - Direct Injection Compression Ignition
DI - Direct Injection
CI - Compression Ignition
SI - Spark Ignition
BTE - Brake Thermal Efficiency
BSFC - Brake Specific Fuel Consumption
EGT - Exhaust Gas Temperature
PM - Particulate Matter
CO - Carbon monoxide
CO₂ - Carbon dioxide
HC - Hydrocarbons
NOX - Oxides of Nitrogen
LHV - Lower Heating Value

References


