Effect of air-fuel ratio on the performance and emission characteristic of a biogas fuelled spark ignition engine

S. K. Hotta\textsuperscript{a}, N. Sahoo\textsuperscript{b}, K. Mohanty\textsuperscript{a} and Menelik Walle Mekonen\textsuperscript{b}

\textsuperscript{a}Centre for Energy, IIT Guwahati, Guwahati, India
\textsuperscript{b}Dept. of Mechanical Engineering, IIT Guwahati, Guwahati, India

ABSTRACT
In the current investigation, a single cylinder, 4-stroke, water cooled, manifold injection, variable compression ratio (VCR), spark ignition (SI) engine of rated power of 4.5 kW was operated with raw biogas at three different compression ratios (CRs) ranging from 8 to 10. The fuel was tested in the VCR engine under various operating equivalence ratios (ER) to find out the leaner misfire and the richer knock limit as well. The comparative performance and emission characteristics of the engine with different compression ratios are reported in the paper. It has been found that, the lean misfire limit is considerably extended with increasing compression ratio of the engine. The lean limit indicated by misfire is at an ER of 0.673 with a CR of 10 as against 0.764 with that of a CR of 8. The peak brake power output and brake thermal efficiencies at CR 10, CR 09 and CR 08 were observed at ERs of 1.04, 0.967 and 1.03, respectively. The minimum value of HC and the maximum value of NO\textsubscript{x} concentration coincide with the position of the best thermal efficiency.

Keywords: Raw biogas, SI engine, Equivalence ratio, Engine performance

1. INTRODUCTION
The environmental concerns and the uncertainties associated with future availability of fossil fuel are driving the interest of utilizing renewable fossil fuels\textsuperscript{1}. Liquid fuels like alcohols, gaseous fuels such as natural gas, Liquefied Petroleum Gas (LPG), hydrogen, biogas, and producer gas are promising alternative fuels\textsuperscript{1,2}. Very low level of pollutant emission has been reported in spark ignition (SI) engines when gaseous fuels are effectively utilized\textsuperscript{2}. Gaseous fuels are most attractive because of their wide ignition limit, capability to form homogeneous mixture, higher hydrocarbon ratio and relatively higher auto ignition temperature\textsuperscript{3,4,5}. Even very lean mixtures of these fuels can be burned in air\textsuperscript{6,7}.

Biogas is one such a renewable fuel and attractive source of energy from a greenhouse standpoint. Their abundant availability is due to presence of non-fossil carbon resources like cattle dung, kitchen/agricultural waste and other biomasses\textsuperscript{8}. It is produced through anaerobic
digestion of organic matters and consists of approximately 50-70% of methane (CH₄), 25-50% of carbon dioxide (CO₂), 1-5% of hydrogen (H₂), 0.3-3% of nitrogen (N₂) and traces of other impurities, notably hydrogen sulfide (H₂S). Since biogas has a higher anti-knock index, it can sustain high compression ratio (CR), which enhances the thermal efficiency of the engine.¹⁰

In order to achieve low-emission and the best fuel conversion efficiency in spark ignition (SI) mode, it is highly recommended to maintain the accurate CR, air fuel ratio and ignition timing, when the engine is fueled with biogas¹¹,¹². In order to meet current low-emission technology standards, precisely control of air-fuel ratio is desired. One of the option for generating efficiency is the use of biogas-hydrogen blend¹³, which is the function of excess air ratio (EAR). The peak generating efficiency has been reported at EAR of 1.2. This research investigation was conducted to find out the range of limiting air-fuel ratios both in leaner and richer state of the air-fuel mixture for locally available biogas. Further, the effect of air-fuel ratio on the performance and emission characteristics have been investigated for a biogas fueled SI engine.

2. EXPERIMENTAL SETUP AND PROCEDURE

2.1 Test Fuel and Properties
The fuel selected for the evaluation of the engine performance is the raw biogas produced by anaerobic digestion of cow dung and lignocellulos biomass produced in a Dinabandhu modeled biogas digester of 3m³ capacity. The biogas is collected from the site in a neoprene coated rubber fabric balloon of size 2m³ and the same is being transported to the laboratory for testing. The composition of the used fuel is analyzed each time before testing by using Thermo Fisher Scientific make gas chromatograph (GC). It has been observed that there is no substantial change in the composition of the produced biogas unless until there is drastic change in environmental operating parameters or change in feed materials to the biogas digester. However, there is a little variation in the composition of the biogas with time according to the activities of the anaerobic bacteria. Table 1, describes some of the important properties of biogas used in the experiment.

2.2 Experimental Test Rig
The experimental setup shown in Fig. 1, consists of a single cylinder, four stroke, and water-cooled, manifold injection variable compression ratio (VCR) engine of rated power 4.5 kW at 1800 rpm. The engine is coupled with a water-cooled eddy current dynamometer for loading the crankshaft with the help of electromagnetic force. The research engine setup is equipped
with a tilting block VCR attachment developed by Larsen\textsuperscript{14}. This technique is completely manual where the block can be tilted on one side keeping opposite end hinged. The variation on geometric clearance volume achieves the required compression ratio theoretically. The compression ratio on the test bench ranges from 6 to 10. The fuel injection nozzle attached with the engine has three holes of 0.3 mm diameter with a spray angle of 120°. The load on the engine is varied manually by controlling the knob of the potentiometer connected to the eddy current dynamometer. The load applied on the dynamometer is measured by pre calibrated strain gauge type load cell with a digital load indicator located on the engine panel. Rotameters are attached in the panel box to measure the constant flow rate of cooling water to the engine jacket and exhaust gas calorimeter. A diaphragm type piezo pressure sensor with built in amplifier with low noise cable is mounted on the engine head to measure the cylinder pressure during combustion. The optical crank angle encoder attached to the engine delivers a signal for each degree rotation of the crankshaft. The pressure and crank angle signals are then interfaced to the computer through piezo power unit to observe the pressure and crank angle signals as well as to measure the speed of the engine.

The standalone panel box attached to the engine is consists of an air box, fuel tank, manometer, fuel measuring burette, dynamometer loading unit and the electronic panel along with the a NI USB-6210 data accusation system (DAS). The air flow measurement is performed by using air flow transmitter attached to the air box. The biogas flow rate is measured by using a specially designed float type variable area rotameter. The fuel injection time, fuel injection angle, ignition angle can be programmed with open electronic control unit (ECU) at each operating point based on RPM and throttle position. Air temperature, coolant temp, throttle position and trigger sensor are connected to open ECU which control ignition coil, fuel injector, fuel pump and idle air. The engine is connected to the Labview based software “Enginesoft” to record and analyze the data stored via a NI USB 6210 data logger. A “PE3” software package is used to program the open ECU. This ECU can be reconfigured according to the fuel quality. The spark timing (ignition angle) of the engine can be varied using the “PE3” to obtain the maximum brake torque timing (MBT). The engine is equipped with electronically control capacitive discharge ignition (CDI) system. The LAB VIEW based “PE3” package is used to control the ignition timing based on the input received from trigger sensor and crank angle encoder. The emission analysis is carried out by using AVL DIAGAS 444N five-gas analyzer.
Table 1. Specific properties of the biogas.

<table>
<thead>
<tr>
<th>Properties</th>
<th>Biogas</th>
</tr>
</thead>
<tbody>
<tr>
<td>Molecular formula</td>
<td>CH₄- 55.6%²</td>
</tr>
<tr>
<td></td>
<td>CO₂- 42.3%²</td>
</tr>
<tr>
<td></td>
<td>N₂- 2.1%²</td>
</tr>
<tr>
<td>Density at 15°C (kg/m³)</td>
<td>1.11⁷</td>
</tr>
<tr>
<td>Lower heating value (MJ/kg)</td>
<td>17⁷</td>
</tr>
<tr>
<td>Heat of vaporization (MJ/kg)</td>
<td>0.5</td>
</tr>
<tr>
<td>Stoichiometric A/F ratio</td>
<td>5.67⁶</td>
</tr>
<tr>
<td>Research octane number</td>
<td>110</td>
</tr>
<tr>
<td>Auto ignition temperature (°C)</td>
<td>650</td>
</tr>
<tr>
<td>Flame Speed (cm/s)</td>
<td>25¹</td>
</tr>
</tbody>
</table>

* Calculated ² Experimental value

Figure 1 Schematic diagram of experimental engine test rig.

2.3 Experimental Procedure

The test is conducted by running the engine solely with raw biogas as a fuel. Biogas balloon stored in atmospheric pressure is connected to a compressor where the pressure is initially built up to 3 atm in the biogas storage and then it is connected to a mass flow controller through a controlling valve, where the pressure is set to 1.3 bar. Biogas fuel line leaving the mass flow controller is then connected to biogas fuel metering unit followed by a diaphragm pump driven by the suction stroke of the engine. Biogas leaving the diaphragm pump is diluted while allowing it to pass through a T-type mixing chamber in the suction stroke. Hence, it was difficult to start the engine with cold raw biogas. The VCR SI engine is initially started at CR 10 with petrol by turning on the ignition switch with some load and throttle opening to achieve the steady state condition. After achieving the steady state condition, the engine was operated at wide open throttle (WOT) condition (90% of the throttle opening).
When the water jacket temperature approaches 60º C, the ignition advance of the engine was reconfigured to 45º crank angle (CA) before top dead center (TDC) through PE3 software. Then, the fuel (biogas) line valve is opened for mixing with air before entering to the inlet manifold of the engine. At the same moment the petrol injector is turned off and the engine was allowed to run on raw biogas. After achieving the steady state condition the “Enginesoft” was configured for the particular CR (10), speed (1650 rpm), fuel calorific value (1700 kJ/kg), fuel density (1.2 kg/M³) and the number of cycles (10) for which data has to be recorded. After configuring the software the data has been logged on for time interval of 60s in the software itself. While maintain the speed of 1650 rpm, the readings of corresponding load applied, fuel consumption, temperature of water at the inlet/outlet of engine jacket, temperature of water at the inlet and outlet of the calorimeter, temperature of the exhaust gas at the inlet and out let of the calorimeter, and volume flow of water to engine and calorimeter were recorded manually. At the same time, the exhaust gas composition was checked to quantify the corresponding emission of CO, CO₂, O₂, NOx and Unburnt Hydro Carbon (UHC) by inserting the probe of AVL flue gas analyzer at the tail end of the exhaust pipe. For enabling the SI engine to operate on different air-fuel ratios, the mass flow rate of the biogas supplied to the engine was varied through the control valve with constant fuel delivery to the engine. With variation in mass flow rate of the fuel, the engine speed varies. But, by controlling the load applied by the eddy current dynamometer, the designated speed (1650 rpm) of the experiment is regained and corresponding data are again recorded for the particular air-fuel ratio. Similarly, for each operating air-fuel ratios the data are recorded by maintain the designated speed of the engine at CR 10. Once the data logging for the operating CR of the engine was completed, the CR of the engine was set to CR 09 and CR 08, respectively by manually operating the compression ratio adjustment lever on the engine. The above mentioned steps are again repeated to log the data for the operating air-fuel ratios of the engine.

3. RESULTS AND DISCUSSIONS

3.1 Performance Analysis

The variation of the brake power (BP) with the equivalence ratio of a biogas fuelled SI engine operated at WOT condition and at different CR is plotted in Fig. 2. As shown, at all CRs the BP follows an increasing trend along with the progressive advance of the equivalence ratio (ER) and deviates from the trend at a point very close to the stoichiometric mixture (Φ=1). The range of ERs covered the lean misfire limit on one side to the knock limit on the rich
side. The leanest point shown in this plot corresponds to the condition just before the onset of misfire beyond which it was difficult to run the engine smoothly. It was also observed that, on rich side there is a drop in BP at all operating CRs of the engine. This may be due to the enriched mixture supplied to the engine which onset knocking during combustion. It was also observed that, irrespective of the ERs the BP output of the engine increases with increasing CRs. This is because of the enhanced thermal efficiency of the engine at higher CRs. It was noticed that, the lean misfire limit considerably extended with increasing CRs of the engine. The lean limit indicated by misfire is at an ER of 0.673 with a CR of 10 as against 0.764 with that of a CR of 8. However, the leanest point of misfire at CRs of 10 and 9 are almost coinciding and appears at an equivalence ratio of 0.67 and 0.66, respectively. On the other hand because of incomplete combustion the engine was unable to run with richer mixture beyond an ER of 1.1. Thus, an increase in CR extends the lean limit of operation because of the higher gas temperature and lesser dilution by the residual exhaust gas. The peak power outputs with CRs of 10, 9 and 8 are, 2.99, 2.64 and 2.18 kW at ERs of 1.04, 0.96 and 1.03, respectively.

![Figure 2](image)

**Figure 2** Variation of brake power with equivalence ratio at WOT condition

Figure 3 shows the effect of the equivalence ratio on the brake thermal efficiency (BTE) of a biogas fueled SI engine operated at WOT condition. It was observed that, as the air-fuel ratio changed from the lean misfire limit to the rich side (moving left to right in the figure) for the raw biogas (56% CH\(_4\) and 42% of CO\(_2\)), the BTE follows an increasing trend and deviates at a point very nearer to the stoichiometric condition. The peak brake thermal efficiencies of the engine at CR of 10, 9 and 8, were found to be 23.4%, 19.5% and 17.9% at ERs of 1.04, 0.96
and 1.03, respectively. Mixture richer or leaner than this point will cause incomplete combustion or slower the burning rate and hence lead to a drop in thermal efficiency. It was also observed that with increasing compression ratio there is a significant increase in peak BTE. The peak BTE rises by 30%, when the CR was increased from 8 to 10.

![Figure 3](image)

**Figure 3** Variation of brake thermal efficiency with equivalence ratio at WOT condition

### 3.2 Emission Analysis

The effect of air-fuel ratio on the hydrocarbon (HC) concentration in the exhaust emission of a biogas fueled SI engine operated at 1650 rpm at WOT and at different CRs is shown in Fig. 4. It was observed that with increasing CR the unburnt hydrocarbon emission in the exhaust emission of the biogas fueled SI engine is decreasing throughout the range of supplied air-fuel ratios to the engine. By introducing homogenous air-fuel mixture to the combustion chamber causes better combustion at higher operating CRs of the engine. The minimum value of HC concentration in the exhaust emission at CR 10 is 28 ppm which was reduced by 32% compared to HC emission at CR 8.

It was also observed that, the unburnt HC emission follows a decreasing trend as the air-fuel ratio changed from lean burning limit to the rich knock limit. The unburnt HC emissions at CR 10, 9 and 8, were found to vary in the range of 96-28 ppm, 75-35 ppm and 87-37 ppm, respectively. The leaner air-fuel mixture with insufficient oxygen promotes the incomplete combustion of fuel as a result misfire produces the unburnt HC. Since the volumetric efficiency has an increasing trend with increasing CR, the HC emission at higher CR is reduced. The HC emission reaches the minimum value coinciding with the position of the
best thermal efficiency. Mixture deviating from this point will have insufficient oxygen or lower combustion temperature, which will produce higher unburnt hydrocarbon. On the leaner side of the mixture, the HC emissions at higher CRs are quite appreciable and reduced by 9% and 32% at CRs 9 and 10, respectively.

![Figure 4](image)

**Figure 4** Variation of hydrocarbon emission with equivalence ratio at WOT condition

![Figure 5](image)

**Figure 5** Variation of nitric oxide emission with equivalence ratio at WOT condition

Figure 5 shows the nitric oxide emission levels of biogas fueled SI engine operated at 1650 rpm with WOT operating condition and at different compression ratios. As observed, the nitric oxide emission levels shoots up with increasing equivalence ratio of the supplied air-fuel mixture to the engine. The nitric oxide emission level increases for higher CRs of the engine. This is due to increasing in peak cylinder temperature at higher CRs of the engine.
The nitric oxide emissions reach a peak value and it also coincident with the position of the best thermal efficiency. This because of the higher combustion temperature produced due to the better combustion of air-fuel mixture. Any deviation from this point will have insufficient oxygen or lower combustion temperature, which will reduce the production of nitric oxide. The peak values of the nitric oxide concentration in the exhaust emission of the SI engine were found to be 937, 635 and 625 ppm with CR 10:1, 9:1 and 8:1 at ERs of 1.04, 0.96 and 1.03, respectively. The CO levels shown in Fig. 6 are low, as most of the experiments were conducted in the leaner than stoichiometric ratio range. It was noticed that, as the air-fuel ratio changed from the lean misfire limit to the rich knock limit, the CO concentration in the exhaust emission of the SI engine shoots up. This is because of incomplete combustion in the richer region of the mixture. The CO concentration in the exhaust emission varies from 0.009-0.091%, 0.007-0.050% and 0-0.050% at CRs of 10, 9 and 8, respectively. Higher CRs increases the CO emission and is quite noticeable in the rich region of the air-fuel mixture. This may be due to the poor mixing of air and fuel, local rich regions and incomplete combustion. The CO emissions throughout the operating range are falling in between 0-0.091% and are well in the acceptable range.

**3.3 Combustion Analysis**

As a fuel, biogas has an extremely low energy density on the volume basis on account of its higher CO₂ content. The larger quantity of CO₂ content reduces its calorific value, flame velocity and flammability limit. The combustion process is completely dependent on the flame propagation, which ultimately depends on the flame speed (25 cms⁻¹ in biogas) and auto ignition temperature (650°C in biogas). Figure 7 explains the effect of air-fuel ratio on
the average cylinder pressure for a biogas fueled SI engine at WOT condition. The ignition advance, speed and compression ratios are remained constant for the particular study. This particular plot explains the effect of air-fuel ratio on the average cylinder pressure of the engine operated at a CR10 and IA $45^\circ$ before TDC. It was observed that with progressive equivalence ratio (as the fuel-air ratio changed from the lean burning limit to the rich knock limit) the average peak cylinder pressure increases and shift towards the top dead center. However, the peak cylinder pressures at $\Phi= 1.041$ and 1.107 coincide each other and appears at 371$^\circ$CA. The limiting peak cylinder pressures were observed to be 26.81 bar at $\Phi= 1.041$ and 1.107. This is because of the faster flame propagation, which results in improved thermal efficiency at this condition. Because of this pressure rise, the BP and BTE are found maximum at $\Phi = 1.041$ while the engine was operated at CR 10. At lower ERs the average peak cylinder pressures are lower and as expected it increases with the richer configuration of the air-fuel mixture. However, at a particular equivalence ratio the peak pressure seems to be retired due to onset of knock in the engine. The best observable ER at CR 10 was noticed to be 1.041.

![Figure 7](image)

**Figure 7** Effect of ER on the average peak pressure of the cylinder of the biogas fuelled SI engine

The variation of NHRR with progressive advance of ER of biogas fueled SI engine is shown in Fig. 8 with WOT operating condition. The heat release analysis evaluate the data on differential basis and this lead to noise in the computed result, especially at the lower pressure regions where the discretization are a larger proportion of the signal. The negative NHRR implies that there is heat transfer to the cylinder wall and ignition should be close to the minimum NHRR. This is because that the heat transfer rate increases as the temperature and
density of the gas increases during combustion. But once the combustion completes, the heat release rate will start adding negative value to the heat transfer. The minimum NHRR is ill defined in SI engine, because of the lower initial rate of combustion. It has been noticed that with progressive ERs, the NHRR increases and shift towards TDC up to definite point ($\Phi =$ 1.041). This is due to the enhanced flame propagation rate where the conversion efficiency is maximum. Beyond this operating range of ER ($\Phi >$ 1.041), the NHRR starts deviating with decrease in the trend. This is due to onset of knock with the richer air-fuel mixture in the engine. The maximum NHRR with $\Phi =$ 1.041 was found to be 15.49 J/°CA at 375° CA.

![Figure 8](image.png)

**Figure 8** Effect of ER on the neat heat release rate of the biogas fuelled SI engine

**4. CONCLUSIONS**

Base on the experimental investigations on the use of biogas to understand the effect of air-fuel ratio on the performance, combustion and emission characteristics of a SI engine, the following conclusions are drawn:

- The lean misfire limit is considerably extended with increasing CRs of the engine. The misfire is at an ER of 0.673 with a CR of 10 as against 0.764 with CR of 8. On the other hand because of incomplete combustion, the engine was unable to run with richer mixture beyond an ER of 1.1. The brake power output at CR 10, 9 and 8 were observed at $\Phi =$ 1.04, 0.96 and 1.03, respectively.

- The peak brake thermal efficiencies of the engine at CR of 10, 9 and 8 were found to be 23.4%, 19.5% and 17.9% at ERs of 1.04, 0.96 and 1.03, respectively. Mixture richer or leaner than this point will cause incomplete combustion or slower the burning rate and hence lead to a drop in thermal efficiency.
• The HC emission reaches the minimum value coinciding with the position of the best thermal efficiency. The minimum value of HC concentration in the exhaust emission at CR 10 is 28 ppm which was reduced by 32% compared to HC emission at CR 8.

• The nitric oxide emissions reach a peak value and it also coincident with the position of the best thermal efficiency. This because of the higher combustion temperature produced due to the better combustion of air-fuel mixture. The peak values of the nitric oxide concentration in the exhaust emission of the SI engine were found to be 937, 635 and 625 ppm with CR 10, 9 and 8 at ERs of 1.04, 0.96 and 1.03, respectively.

ACKNOWLEDGMENTS
The authors wish to thank the Ministry of New and Renewable Energy (MNRE) for the financial support of this project.

REFERENCES