Development of a Biogas Fuel Supply System for an Internal Combustion Engine

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ABSTRACT

A supply system of dual biogas/gasoline is proposed for a small spark ignited internal combustion (SIIC) engine. This work comprehends the design, manufacturing and testing of the fuel supply system, including the devices to maintain the constant engine speed at all loads. For the development of the carburetor—mixing device, the authors have employed a mathematical model to optimise its geometry, with the aim of reducing the pressure losses and maintaining an adequate fuel/air ratio. Moreover, this work shows the experimental results carried out at Institute of Internal Combustion Engines—at Universidad Nacional de Ingeniería (UNI) in Peru, using a 6 kW, one cylinder, stationary engine. With respect to original gasoline engine, with the biogas fired engine, the team was able to achieve a drastic reduction in toxic emissions.

INTRODUCTION

The degree of a country's energy self sufficiency is measured by the energy coverage index (ECI). In Perú, in 2010, it is 80,8%, indicating there is a 19,2% energy deficit, which needs to be imported. Another worrying piece of information is that in Peru, over one million families in rural areas, have no electricity and its benefits. Thus, as an alternative, we advocate the rational exploitation of natural energy resources, higher energy efficiencies in energy utilization and the development of clean energy technologies applied to both energy production and consumption.

Within this strategic scheme, using biomass for energy generation is a viable alternative to address such deficit. It's well understood that solid waste from agriculture, industry and residences can be processed to obtain valuable products such as biogas and fertilizer.

Power generation from biogas using internal combustion engines (ICE) has been a recurring theme in various research projects at Universidad Nacional de Ingeniería (UNI, Peru). Lately, researchers at this university have developed viable ICE alternatives which are appropriate for the remote regions of Peru which often lack electricity. The key for success in these developments has been the adaptation of existing ICE technology to newer fuels, while keeping the engines simple and inexpensive to operate and maintain by the end user. This paper summarizes the development of the fuel feeding system needed to convert a gasoline-fired ICE to biogas. We show both the engineering calculations and the experimental validation carried out during the development of the fuel system. Next, a primer on biogas is given.

BIOGAS

Biogas is a mix of gases resulting from organic matter decomposition, which occurs through bacterial action in anaerobic conditions (absence of oxygen). This process occurs naturally in swamps, landfills and in the digestive tract of many mammals. Industrially, *digesters* obtain the biogas from the organic feedstuff and *scrubbers* are used to remove impurities to achieve a gas with up to 98% methane content.

The main components of biogas [1, 6] are methane (CH_4: 50 to 80%) and carbon dioxide (CO_2: 20 to 25%). The balance makes less than 15% and includes hydrogen sulfide (H_2S), hydrogen (H_2), nitrogen (N_2), oxygen (O_2), argon (Ar), carbon monoxide (CO) and ammonia traces (NH_3). Biogas caloric value (heating value) and other properties depend on the amount of methane, which in turn depends on the raw material, and the biogas production and refining efficiencies. Table 1 shows physical-chemical properties for biogas with different CH_4 contents at standard conditions (101,3 kPa y 273 K) in comparison with gasoline. In this work, biogas is obtained in a 6-meter deep Ascending Flow Anaerobic Reactor (AFAR) with a retention or cycle time of 7 hours. This set up outputs 10 m^3 of biogas per cycle with 80% CH_4.

The hydrogen sulfide (H_2S) in biogas (0,10 to 0,50%) is highly cor-

Table 1. Biogas Properties

Properties	Fuel		
	Gasoline	Biogás (60% CH_4)	Biogás (80% CH_4)
Lower heating value, MJ/m^3	44,0*	21,6	28,8
Density, kg/m^3	720-780	1,21	0,96
Laminar Flame Velocity, m/s	0,35-0,55	0,25	---
Stoichiometric air:fuel ratio, m^3/m^3	---	5,71	7,57
Stoichiometric air:fuel ratio, kg/kg	14,7-15,1	6,1	10,2
Caloric value per mix volume (with λ =1), MJ/m^3	3,74	3,23	3,36
Octane number (research method) [6]	66-97	130**	---

*in MJ/kg. ** with 70% de CH_4.

rosive. For some engines, the maximum allowable content is 1500 ppm or 0,15% by volume [5]. For this reason, scrubbers or wet filters are needed to remove it. Raw biogas from a plant is usually saturated with water (its relative humidity is 100%). So, depending on the piping set-up at the digester outlet, the gas moisture can be reduced through cooling and condensation. Condensate is retained and removed in water traps and biogas is considered dry when it is fed into the engine.

DESIGN APPROACH FOR THE BIOGAS CARBURETOR-MIXER

The biogas carburetor-mixer is the most important fuel-supply system component. In this work, a venturi type mixer was developed by using a math model. Such mixer has two parts: a housing and a venturi, which is replaceable.

The carburetor-mixer is responsible for mixing biogas and air in the right proportion for all engine load levels. The mixer has the same fluid dynamics principle as a conventional carburetor, that is, the air flow enters the engine's combustion chamber, due to the vacuum created by the piston's down stroke, thus defining the admission stage. Air flow velocity increases in the venturi's throat creating a depression (Bernoulli effect), thus making the biogas flow through throat holes and allowing the air to be mixed with the biogas. The throat depression depends on the position of the carburetor's butterfly or throttle valve. Note biogas enters the mixer's air stream radially, mixing with air along the admission duct before entering the engine's combustion chamber (cylinder).

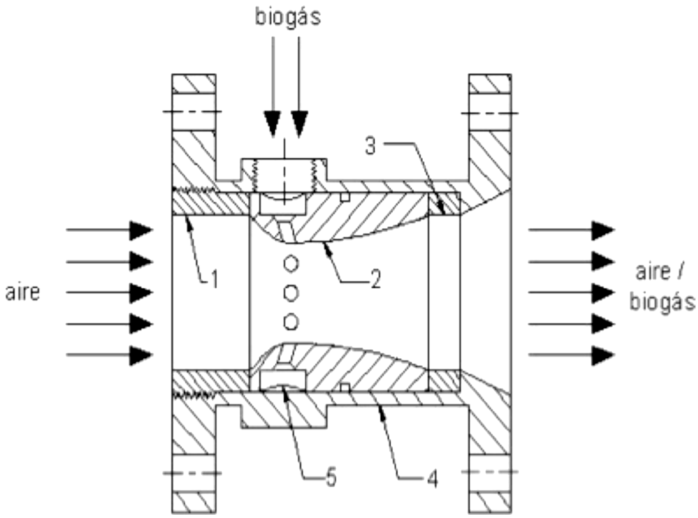

Figure 1. Biogas mixer: (1) Aligner, (2) Nozzle, (3) Spacer, (4) Body, (5) Biogas feeding ring (with radial holes).

Math Modeling

Math modeling for the mixer yielded the main body dimensions, such as: Nozzle inlet and outlet diameters, throat diameter, the nozzle length and optimal nozzle profile. The modeling aim for the nozzle profile was to minimize pressure losses. The math model allowed us to determine the number and diameter of the biogas feed holes for a homogenous mix and with adequate excess air to assure complete combustion. The inclination of the holes' axis with respect to the nozzle axis was also considered, so the holes' discharge coefficient was maximized. The main dimensions obtained for the mixer are shown in Figure 2.

As shown in Figure 2, to model the profile for mixer nozzle, two connected third-degree curves or splines were needed [4], by using polynomials of the form

$$y = a_0 + a_1 z + a_2 z^2 + a_3 z^3 \tag{1}$$

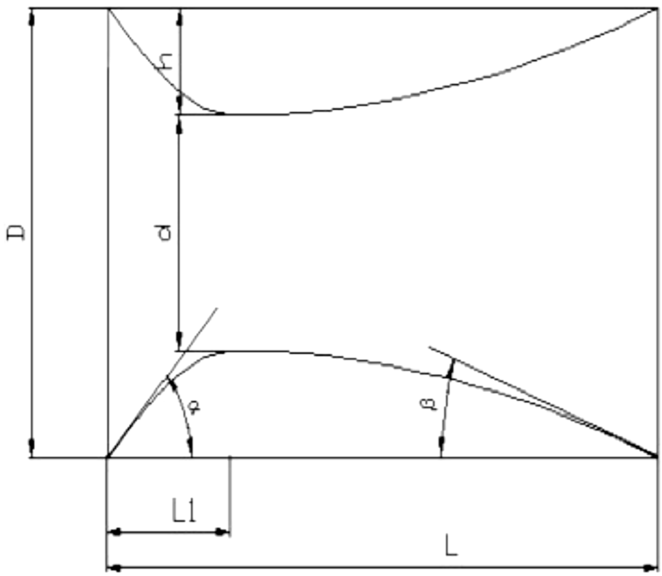

Figure 2. Main nozzle dimensions.

The numeric values for the polynomial coefficients are obtained by using the profile boundary values:

$$h = \frac{D-d}{2}\;; \qquad \frac{L_1}{L} \approx 0{,}20 - 0{,}25; \qquad \frac{L}{h} \approx 6{,}5 - 7{,}5\;;$$

$$\alpha \approx 40 - 50°; \qquad \beta \approx 15\;\; - 25°$$

and the following recommendations [4]:

Once the nozzle profile and dimensions are known, we derive the equations for energy conservation, mass continuity and an adiabatic (reversible isentropic) process for any two points in the nozzle (separated by the differential *dz*). Then, by using Darcy's equation (for the same differential *dz*) and Moody's diagram, we compute the average pressure and velocity values for the nozzle. See results in Figure 3.

Next, prior to estimating the total area needed for the biogas feed holes, we computed the stoichiometric air-fuel ratio for biogas, by using the following equation:

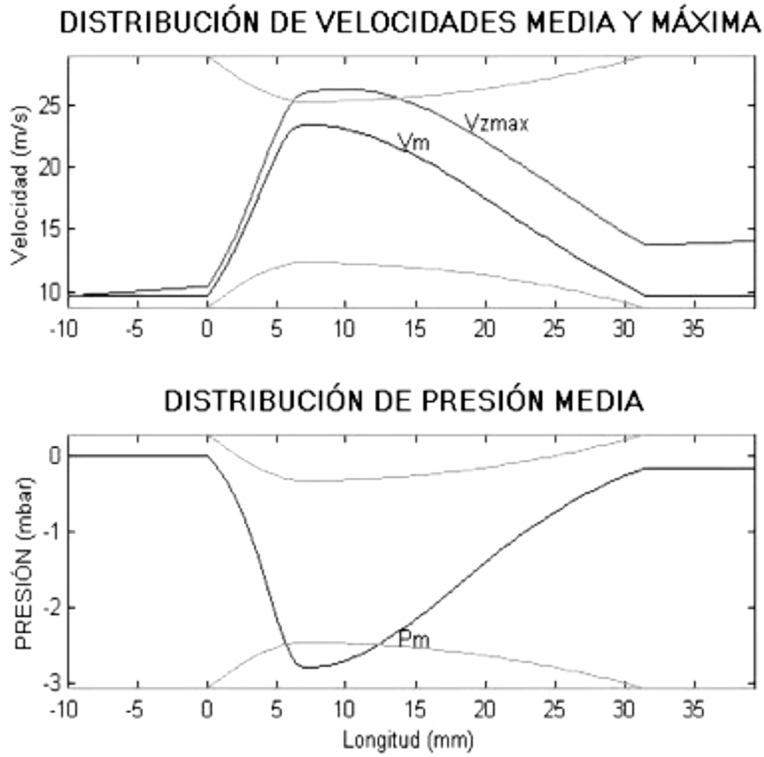

Figure 3. Mean Velocity and Pressure along the length of the mixer nozzle axis.

$$CH_4 + \frac{b}{a}CO_2 + 2(O_2 + 3{,}76N_2) \rightarrow (1 + \frac{b}{a})CO_2 + 2H_2O + 7{,}52N_2$$

Where a and b are molar fractions of CH_4 y CO_2, respectively. Testing for complete combustion found that a = 0,8 y b = 0,2. Then, in this case, the computed stoichiometric air-fuel ratio or $AFR_{stoich} = l_o$ for biogas was found to be 10,2 kg air/kg biogas.

In our case, the maximum biogas supply for maximum power are gotten when the throttle is fully opened and the excess air coefficient is 0,9 (nominal conditions). The excess air coefficient (λ) is the ratio of the actual Air to Fuel Ratio to the stoichiometric Air to Fuel Ratio (AFR):

$$\lambda = \frac{AFR_{actual}}{AFR_{stoich}} = \frac{G_a}{(l_0 \cdot G_c)}$$

Where: G_a y G_c are the actual air and fuel mass flow rates, respectively and l_o is the stoichiometric air-fuel ratio.

Next, by using the well-known formula for gas flow through orifices, we have:

$$\lambda = \frac{1}{l_0} \cdot \frac{A_d \cdot C_d}{A_c \cdot C_c} \cdot \sqrt{\frac{\rho_0}{\rho_c}} \cdot \sqrt{\frac{\Delta P_d}{\Delta P_d + \Delta P_c}}$$

Where: A_d y $A_{c\ a}$ are the areas of the diffuser and the biogas feed holes (total), respectively; C_d y C_c are the discharge coefficients of such areas; $_o$ y ρ_c are the air and fuel densities, respectively. ΔP_d y ΔP_c are the depressions at the venturi throat and the biogas gage pressure, respectively (in mbar).

From the math model results, the max throat depression was obtained ($\Delta P_d = 4{,}5\ mbar$) for the Honda GX-240 engine at nominal operating conditions. Next, by knowing the air flow and the excess air coefficient ($\lambda = 0{,}9$), the required fuel flow can be estimated. So, assuming that both gas and air are at the same ambient temperature (T_0), we have:

and $$\frac{\rho_0}{\rho_c} = \frac{\mu_0 P_0}{\mu_c P_c}$$

$$\mu_c = a\mu_{CH_4} + b\mu_{CO_2}$$

Where: μ_o and μ_c, P_o and P_c are the molecular masses and absolute pressures for air and fuel, respectively.

From these equations, we obtained the total orifice area A_c to supply gas to the engine, as follows:

$$A_c = \frac{1}{\lambda \cdot l_0} \cdot \frac{A_d C_d}{C_c} \cdot \sqrt{\frac{\mu_0}{a \cdot \mu_{CH_4} + b \cdot \mu_{CO_2}}} \cdot \sqrt{\frac{P_0}{P_0 + \Delta P_c}} \cdot \sqrt{\frac{\Delta P_d}{\Delta P_d + \Delta P_c}}$$

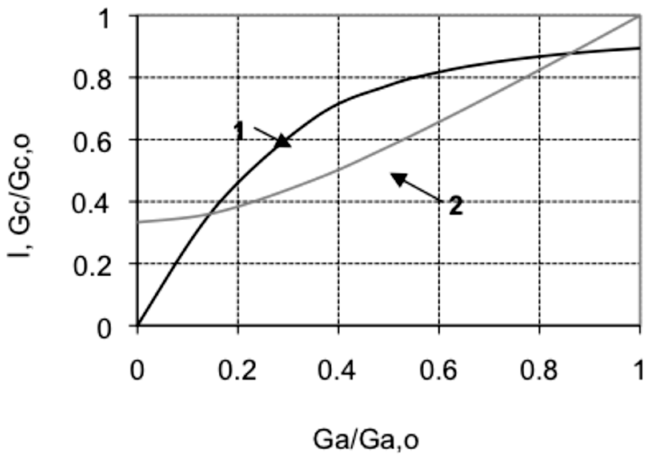

Figure 4. Variation of the excess air coefficient λ (1) and the relative biogas mass flow $G_c/G_{c,0}$ (2) as a function of the relative air mass flow $G_a/G_{a,0}$

Figure 4 shows the variations of λ (curve 1) and the relative fuel mass flow $G_c/G_{c,0}$ (curve 2) as a function of the relative air mass flow $G_a/G_{a,0}$. We have assumed that A_c, A_d, ΔP_c, a, b, C_d y C_c maintain approximately constant values. Note the plot of the excess air coefficient (λ) can be moved to a higher or lower position. That is, the mix can be made richer (lower AFR) or poorer (higher AFR), by adjusting the biogas flow. This can be accomplished, for example, by increasing or reducing the area of the feed holes (A_c). Also note the mix can be made poorer by opening the throttle (butterfly valve), or by increasing the load. For this reason, after some testing, the team decided to gradually increase (one hole at the time) the diameter of biogas feed holes to increase the fuel flow and the engine power output. Such modification was done with the understanding that the specific fuel consumption would increase, but with the aim to increase the engine power output.

Notice the downward trend of the λ curve (1). This shows that by enriching the mix (lower λ) and thus reducing the air flow by closing the throttle with lower loads, we don't have to install a special system to maintain engine idle at very low loads.

Note the nozzle must be fabricated with the smoothest surface finish possible. The body rings are made to mate the rings of the carburetor and air filter, respectively. And the biogas enters the mixer through a needle valve, which regulates the fuel flow.

Figure 5 depicts the layout of the components of the fuel feeding system and speed regulation system.

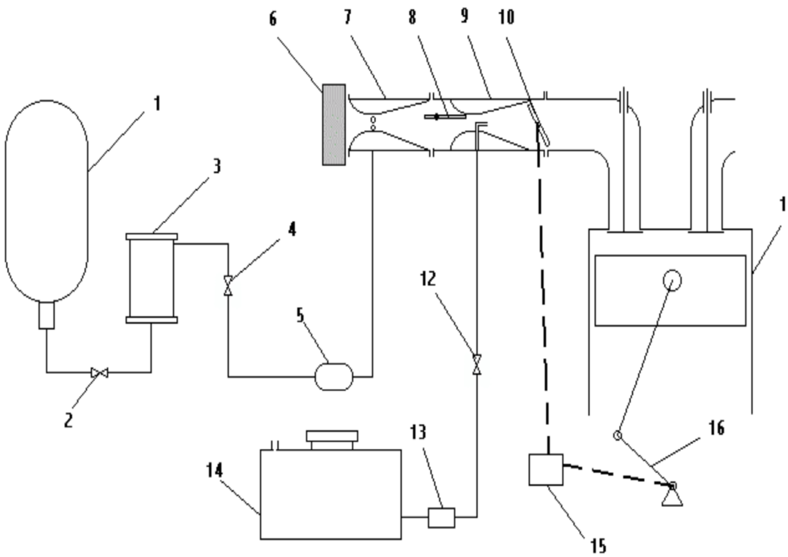

Figure 5. Biogas engine schematic: 1- Biogas storage, 2-Shut off valve, 3- H_2S filter, 4- Gate valve, 5-Adjustment valve, 6-Filter, 7-Mixer, 8-Choke, 9-Gasoline Carburetor, 10-Butterfly Valve, 11-Engine, 12-Shout off valve, 13-Filter,14-Gasoline Tank, 15-Speed control, 16- Crankshaft.

OTHER MODIFICATIONS

Other modifications done to the engine included:

Biogas Filter. To eliminate or reduce the H_2S content in the raw biogas, it's customary to use filters containing lime, iron filings, or other earths such as hematite and gray limonite, which are iron rich and react with H_2S to make iron sulfide. In our case, the filter was installed vertically in the biogas line, about 4 m away from the inlet to the engine. The filter was made of a PVC tube (30 cm diameter by y 100 cm long) filled with iron filings in three compartments.

Idle control. Note the engine does not have any device to maintain or regulate the engine speed al low loads. This is because the biogas mixer developed here allows the engine to run smoothly at low loads.

To the engine. Since biogas has a higher octane rating than any commercial gasoline, the engine can operate at much higher compression ratio. Thus, the head was ground to increase the compression ration from 8:1 to 10,5:1. This resulted an improvement in the engine's specific fuel consumption or efficiency (gr of fuel input/kWh output). Note we made this change only for the experimental study. This change is not convenient for dual fuel engines that need to operate with both biogas and gasoline. With gasoline, knocking and detonation can occur with higher compression ratios, unless more expensive high octane gasoline is used.

Starting system. We kept the original engine starter. Since biogas heating value per unit volume of biogas and air mix (See Table 1), compared with gasoline is 10 to 20% lower, the engine will always be started with gasoline.

Speed regulator. There was no need to make any changes to the original speed regulator. Notice the regulator is needed to maintain constant voltage and frequency when the engine is coupled to an AC generator.

EXPERIMENTAL RESULTS

To verify the functionality and reliability of the designed fuel system, and to obtain the engine technical characteristics, several tests were carried out in a test bench purposely designed and built. The test engine specs are:

- Brand: Honda, model GX-240
- Engine Type: E.CH., with carburetor, 4 strokes, overhead valves, one cylinder
- Compression ratio: 8:1
- Displacement: 242 cm^3
- Diameter x stroke: 73x58mm
- Max Power: 5,97 kW @ 3600 rpm
- Coupled to an AC generator.

For the experimental conditions with biogas (3600 rpm and variable load) we found a small voltage variation at the generator terminals, from 235 V (6,8% higher than the nominal voltage of 220V) with no load, to the nominal value of 225 V (2,3% higher than nominal) with the engine at full open throttle. This demonstrated a reliable operation at all possible load levels. Note that to produce quality electricity, the allowable voltage fluctuation must be ±5% of the nominal (220V) [9]. And the allowable current variation must be less than 2%.

Figure 6 graphs the fuel consumption as a function of power for two tests conducted. Note the curves for the two tests almost overlap, indicating (a) consistent performance and (b) test repeatability. In the graph, the lowest specific consumption (0,65-0,70 m^3/kW.h) corresponds to the max tested engine output of 4,25 kW. This value falls within the lower consumption range specified by Mitzlaff [6] (0,65-,0 m^3/(kW.h)) for ICE with modified Otto cycle for biogas with 60% de $CH_{4.}$ In our case, the biogas is 80% CH_4. Since the curves are not flat and still show a downward trend, the engine can develop more power by feeding it more fuel and air (possibly by using a larger feed holes in the mixer-carburetor).

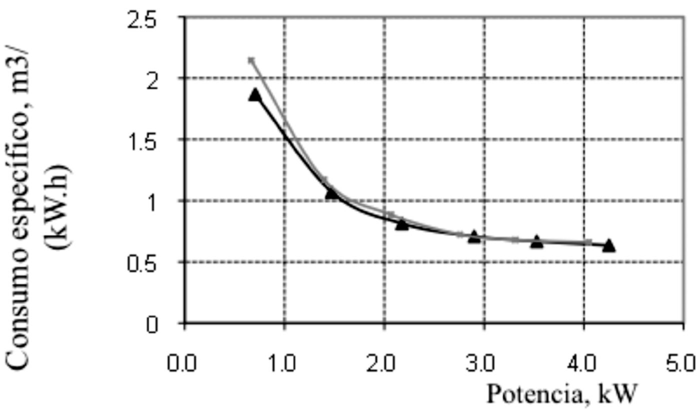

Figure 6. Specific biogas consumption (m^3/(kW.h)) as a function of power output (kW).

Both tests were done with two holes of ϕ 2,2 mm and 8 holes of ϕ 2 mm.

Determining the hourly gas consumption helps understand the engine's economy of operation and it is input to sizing the biogas plants, mixer and ancillary devices. The biogas hourly consumption is 2,6 m^3/h for an output of 4,25 kW (max load). Next, the thermal efficiency for the engine was estimated to be 18,7% at 3,1 kW electrical output. This relatively low efficiency is attributed to smaller engines (<20 kW) and considering the AC generator efficiency is around 75 to 80%.

The excess air coefficient λ was less than 1 throughout the load range; meaning the tested engine always operated with a rich mix. Without loads λ was 0,72 and 0,9 at max load. So, feeding the engine with rich mixes caused it to operate al lower thermal efficiencies. This concern opens the opportunity for further development to improve the present fuel feed system for leaner mixes and better thermal efficiencies.

Drastic Reduction in Toxic Emissions

Figures 7 and 8 show the levels of CO (%) and hydrocarbon HC (ppm) as a function of the excess air coefficient. These results are for tests done with gasoline and biogas, as well. Using biogas in an ICE significantly reduces its emissions with respect to gasoline. In our tests with the engine at idle, biogas instead of gasoline reduced CO by about 90% while hydrocarbon emissions were down by 65%. By using an optimized fuel system, with leaner mixes, further emission reductions can be achieved. Again, exploring a system to feed leaner mixtures should be the subject of further research.

CONCLUSIONS AND FURTHER DEVELOPMENTS

A biogas fuel feeding system for a small ICE was developed. The system allows one to start, operate and maintain the ICE in a easy and

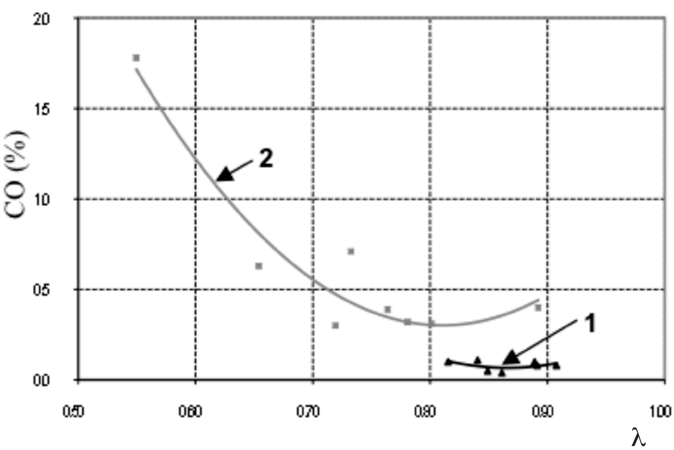

Figure 7. CO (%) in exhaust gases as a function of the excess air coefficient (λ). 1- biogas, 2-gasoline.

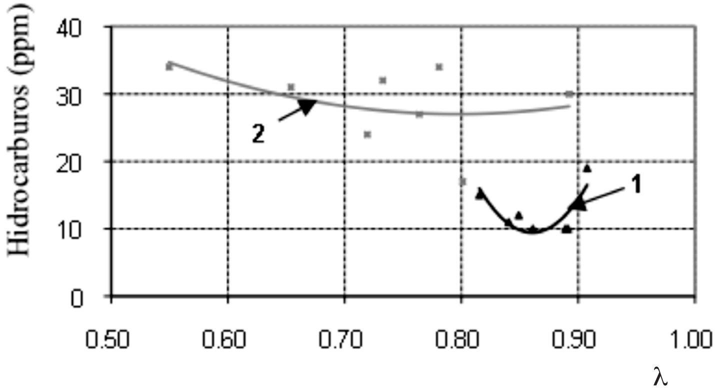

Figure 8. Hydrocarbon emissions (ppm) Vs. excess air coefficient(λ). 1- biogas, 2- gasoline.

trouble-free way. The team used a simple and low cost technology to adapt a Honda GX-240 with biogas. With this development, we have adapted an existing technology so to help rural and marginal urban communities to have access to electricity and power for water pumping, grain milling, wood and metal working, etc. Implementing the developed system here does not require any modification to the engine, existing gasoline carburetor or speed regulator.

However, the max developed power with the biogas fired engine has been reduced to 4,25 kW, which is 29% less than the nominal output for the engine (5,97 kW) when fueled with gasoline. While testing showed the specific biogas consumption was 0,65–0,70 m^3/(kW.h) at full load (4,25kW). The engine operated stable at idle and throughout the operating range, without needing an ad hoc idle control system.

Toxic gas emissions were drastically reduced, thus contributing to the lower carbon footprint and environmental impact of the developed system.

Since the system makes the engine operate rich ($\lambda <1$), and this causes low thermal efficiencies, further developments should include an improved fuel system to achieve leaner biogas/air mixes ($\lambda \geq 1$), with the objective of obtaining higher thermal efficiencies and even lower emissions.

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