SPARK IGNITION ENGINE PERFORMANCE AND EMISSIONS IN A HIGH COMPRESSION ENGINE USING BIOGAS AND METHANE MIXTURES WITHOUT KNOCK OCCURRENCE

by

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With the purpose to use biogas in an internal combustion engine with high compression ratio and in order to get a high output thermal efficiency, this investigation used a Diesel engine with a maximum output power 8.5 kW, which was converted to spark ignition mode to use it with gaseous fuels. Three fuels were used: simulated biogas, biogas enriched with 25%, and 50% methane by volume. After conversion, the output power of the engine decreased by 17.64% when using only biogas, where 7 kW was the new maximum output power of the engine. The compression ratio was kept at 15.5:1, and knocking did not occur during engine operation. Output thermal efficiency operating the engine in spark ignition mode with biogas enriched with 50% methane was almost the same compared with the engine running in diesel-biogas dual mode at full load and was greater at part loads. The dependence of the diesel pilot was eliminated when biogas was used in the engine converted in spark ignition mode. The optimum condition of experiment for the engine without knocking was using biogas enriched with 50% methane, with twelve degrees of spark timing advance and equivalence ratio of 0.95, larger output powers and higher values of methane concentration lead the engine to knock operation. The presence of carbon dioxide allows operating engines at high compression ratios with normal combustion conditions. Emissions of nitrogen oxides, carbon monoxide, and unburnt methane all in g/kWh decreased when the biogas was enriched with 50% methane.

Key words: *high compression ratio spark ignition engines, knock, biogas enrichment with methane*

Introduction

Biogas is the product of anaerobic digestion of waste, whether occurring spontaneously in landfills or under controlled conditions in digesters. Biogas is viewed as an important energy source in current efforts to reduce the use of fossil fuels and dependency on imported resources [1]. Biogas technology offers a unique set of benefits. It can improve the health of users, is a sustainable source of energy, benefits the environment and provides a way to treat and reuse

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various wastes-human, animal, agricultural, industrial, and municipal. In developing countries the expansion of biogas recovery systems has been based upon small-scale reactors designed for digestion of cattle, pig and poultry excreta. Meanwhile, landfill sites and municipal wastewater treatment plants where anaerobic processes produce biogas which is released into the atmosphere, either before or after combustion. Biogas is often used for cooking, heating, lighting or electricity generation. Widespread dissemination of biogas digesters in developing countries stems from the 1970s and there are now around 4 and 27 million biogas plants in India and China, respectively, [2]. Despite the increased research attention, there has only been a slight increase in actual biogas use in electricity generation due to the problems that this fuel presents, such as its low heating value, low flame speed, high percentages of inert gases, and the presence of sulfur [3-7]. Because of these problems, the use of biogas is easier in a biogas-Diesel dual engine [8]. Using biogas as fuel to produce electricity generation could reduce CO_2 emissions and diminish the consumption of conventional fuels [7].

In a Diesel engine converted to spark ignition (SI) mode, a mixture of gaseous fuel and air is admitted via the engine inlet. Ignition energy is required inside the cylinder, needing an adaptation of the engine by including spark plugs with a synchronization system that allows advancing or retarding the spark timing [9-13]. In several cases, engine conversion to SI mode introduces changes in the compression ratio (CR), which requires modification of the geometry of the piston head. The highest output thermal efficiency in the engine is obtained at full load operation, where the heat losses to the walls are low and the specific fuel consumption is small [14]. The energy contained in the gaseous fuel and air mixture affects the power output of the engine. Thus, the energy density of the mixture determines the power developed in the cylinders. The lower heating value of diesel is close to 44 MJ/kg, whereas that of biogas is close to 23.5 MJ/kg. The lower heating value of natural gas is 50 MJ/kg at standard conditions, which makes biogas and methane mixtures attractive [3, 15, 16].

The methane number (MN) is a resistance to knock quantification methodology for gaseous fuels, was introduced in 1972 by Leiker and associates [17]. As octane number uses a mixture of isooctane and n-heptane as the reference fuel, the reference fuel for MN method is a mixture of methane (CH_4) and hydrogen (H_2). The MN of biogas is larger than the natural gas, allowing biogas to be used in SI engines with larger CR than when natural gas is used, resulting in larger thermal efficiencies; however, NO_x emissions are also increased [18]. These engines also produce relatively less output power than engines with a liquid fuel due to the lower energy density. In addition, the fact that biogas has a low-flame speed means that combustion must begin sooner, resulting in pressure increments from the combustion toward the piston occurring during the end of compression stroke, thereby diminishing net work of the engine [10, 19]. Increasing the percentage of CH_4 in the composition of biogas in a SI engine notably improves the performance of the engine. Increasing the CH_4 proportion also diminishes the emissions of unburnt hydrocarbons, extends the limits of inflammability of the mixture and increases the burning velocity, resulting in a superior reactivity of the mixture with both a larger output power and thermal efficiency. The coefficient of variation of pressure is also reduced [3]. Changes in the heating value of the gas, CR of the engine and the reactant/product mole ratio result in varying the following: output power, pressure peaks, temperature peaks, and the quality of the combustion [20]. Biogas with CO_2 values greater than 40% result in a strong irregular operation of the engine. Biogas with a CO_2 value of 30% has notably improved engine performance relative to that of 40% biogas [3, 21-23]. Biogas purification reduces the CO_2 percentage and increase CH_4 concentration, increasing the lower heating value [24].

Tutak and Jamrozik [25] presented the results of gasification process of dried sewage sludge and use of generator gas as a fuel for dual fuel turbocharged compression ignition engine in dual fuel operation the high value of indicated efficiency was achieved equal to 35%, so better than the efficiency of 30% attainable when being fed with 100% liquid fuel.

Several studies have investigated biogas in SI engines. Porpatham *et al*. [26] published some papers using biogas (60/40, 70/30, and 80/20% CH_4/CO_2 by volume) in a modified single cylinder engine (4.4 kW rated power), they decreased CR from 17:1 to 13:1 and used a speed engine of 1500 rpm. Biogas plus H_2 with a CR of 13:1. Effect of swirl and the influence of the CR on the performance of a converted Diesel engine to SI with CR between 9.3:1 and 15:1. In all the cases, various equivalence ratios were used. At all operating conditions, the engine was free of knock and achieved the highest levels of effective thermal efficiency with biogas consisting of 80%CH_4 + 20%CO_2 with a CR of 13:1. With biogas, it has been found that the higher CR (15:1) provided the greater brake thermal efficiency. The spark timing for the best torque must be retarded as the compression ratio is increased [3, 5, 13]. Midkiff *et al*. evaluated natural gas (NG) in comparison with three types of simulated biogas (60%NG-40%CO_2, 75%NG-25%CO_2, and 55%NG-35%CO_2-10%N_2) in a SI engine with a CR of 11:1. Leaner mixtures, retarded spark timing and diluent addition (CO_2, N_2) yielded reduced NO_x emissions. The presence of a diluent CO_2 or N_2 in a gaseous fuel reduces the volumetric heating value of the fuel and reduces the brake power output [23]. Carrera *et al*. [27] numerically studied the combustion process of a biogas SI engine with a CR of 13:1; this study numerically evaluated the way in which the CR and operating parameters, such as the engine speed, excess air, spark timing, and CO_2 content of biogas affected the evolution in combustion process. Hyang and Crookes *et al*. [7] presented results from tests with variable CR (8:1 to 15:1) of a 7 kW single-cylinder SI engine operating on simulated biogas formed from different mixtures of domestic NG and CO_2. The fraction of CO_2 in the simulated biogas was changed from 23.1 to approximately 40% by volume. The authors concluded that the primary influence of CO_2 in biogas fuel on engine operation was to reduce the NO_x emissions. To increase the CR, bring to higher brake mean effective pressure and brake thermal efficiency. Lee *et al.* [6] investigated the generating efficiency and NO_x emissions of a gas engine generator with a low-pressure loop exhaust gas re-circulation system that was fueled by model biogas; tests were performed at a constant output electric power of 15 kW and an engine speed of 1,800 rpm (the CR was not reported). The test results showed that both NO_x emissions and generating efficiency generally decreased as the exhaust gas re-circulation (EGR) rate increased. Chandra *et al.* [28] presented the performance results of a 5.9 kW stationary Diesel engine that had been converted for SI mode and was run on compressed NG, methane enriched biogas (95%CH_4 + 5%CO_2) and biogas (65%CH_4 + 35%CO_2) produced from biomethanation of jatropha. The performance of the engine at a CR of 12.7 was evaluated at 30º, 35º, and 40º ignition advance for top dead center. In comparison to diesel as original fuel, the power deteriorations of the engine were observed to be 31.8%, 35.6%, and 46.3% for compressed NG, methane enriched biogas, and raw biogas, respectively, due to its conversion from compression ignition to SI mode. This investigation used a Diesel engine adapted to SI mode to eliminate diesel fuel dependence, the engine has a compression ratio of 15.5:1, operating with biogas and additions of CH_4 with an equivalence ratio of 0.95 ± 0.01. Experimental work was conducted in Medellin city, Colombia, located 1,500 meters above sea level, tests were performed at an atmospheric pressure of 852 mbar and 25 ºC. The following assumptions are considered: The high content of CO_2 in the biogas allows the engine to operate at a high CR without knocking occurrence, thus is possible to mitigate the negative effect of the low energy density of biogas on the output thermal

Table 1. Engine technical characteristics

Commercial designation	Lister Petter TR2, 4 strokes, 2 cylinders
Cubic capacity	1.550 cm^3
Bore × stroke	98 × 101 mm
Compression ratio	15,5:1
Maximum output power in diesel mode	8.5 kW
Maximum output powr in SI model	7 kW

efficiency, output power and cyclic dispersion. and the increase of CH_4 percentage in biogas in certain values at high CR allows the engine to operate without knocking, with low cyclic dispersion and increased energy density.

Experimental methodology

Experimental set-up

A Diesel engine in SI mode was coupled to an asynchronous generator of electricity, which allowed the engine behavior to be evaluated with several load levels. The most important technical characteristics of the engine/generator combination are listed in tab. 1.

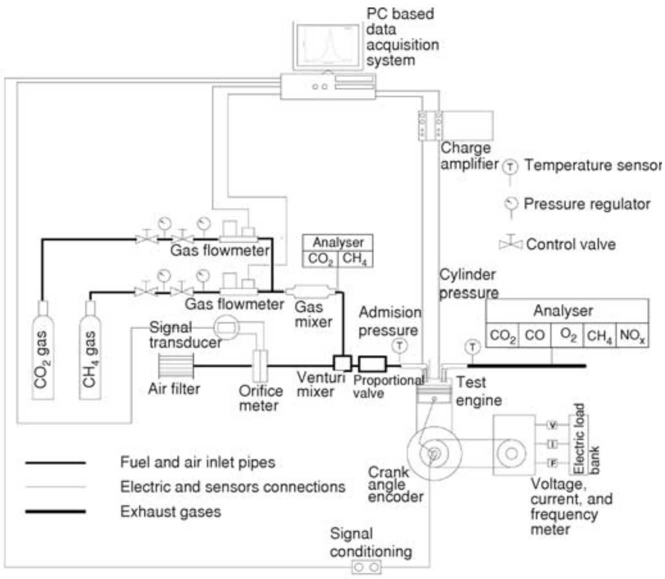

Figure 1. Experimental diagram

In this study, an orifice hole plate differential pressure meter was used to monitor the air flow. Two rotameters were used to measure the volumetric flow and therefore, could measure the consumption of CH_4 and CO_2 and control the different desired biogas mixtures. A gas analyzer, which operates based on the principle of non-dispersive infrared absorption was used to measure the CO, CH_4, and CO_2 emissions. The O_2 measurements were performed with an electrochemical cell. The pressure in-cylinder was obtained by a piezoelectric pressure transducer Kistler 6125B located in the cylinder head of the engine. Measurements of the crankshaft angle were performed with a crank angle (CA) encoder Kistler model 2614A. The charge inlet pressure was measure with a piezoresistive absolute pressure sensor Kistler type 4005B V200S for SCP amplifier Kistler type 4665 (with connecting cable type 4763B). The main elements used in the experiment are shown in fig. 1.

Engine conversion

Several modifications were made and parts were added that allowed the correct operation of the engine in SI mode: high-capacity spark plugs, ignition coil and distributor, air-biogas mixer, proportional valve, and spark timing regulator.

Testing procedure

– Before the conversion, the Diesel engine effective output power was 8.5 kW in compression ignition mode.

- After the conversion, the engine was operated with NG to evaluate the maximum output power that can be achieved in SI mode. This maximum output power was 7.5 kW, however this caused anomalous noisy operation and knocking occurrence, as indicated by the sound of the engine and in the variation of the pressure vs angle curve. The effective output power of the engine in SI mode was established to be 7 kW using biogas, without knocking, which was a decrease of 17.6% of the effective output power.
- The engine was operated with simulated biogas with a composition of 60%CH_4-40%CO_2 for four load levels: 43% (3 kW), 57% (4 kW), 86% (6 kW), and 100% (7 kW). The engine speed was kept constant at 1800 rpm in all the experiments, corresponding to 60 Hz, this is the frequency used in Colombia.
- Then, the fuel was changed to 75% biogas + 25% CH_4 by volume (equivalent to 70%CH_4-30%CO_2), and the engine was operated at the same four loads.
- Finally, 50% biogas + 50% CH_4 (equivalent to 80%CH_4-20%CO_2) was used. In all the cases, the output variables were the output thermal efficiency, equivalence ratio, emissions of CO_2, CO, CH_4, and NO_x, exhaust gas temperature, pressure *vs*. angle curves, and volumetric efficiency. All the experiments were performed three times. These values were averaged to make comparisons, all of the operating conditions evaluated in the engine were free of knocking occurrence. Table 2 shows the main fuel properties, these values were calculated using software developed by GASURE research group, and tab. 3 shows the general procedure for testing.

Table 2. Main fuel properties

Fueal properties	100% biogas	75% blogas +25% CH_4	50% blogas +40% CH_4
Low heating value [MJm^{-3}]	20.36	23.75	27.14
Theoretical air volume [m^3airm^{-3}gas]	5.71	6.66	7.62
Low wobbe index [MJm^{-3}]	20.99	25.86	31.40
Flammability low limit [%]	7.2	6.7	6.2
Flammability high limit [%]	18.2	17.0	15.8
Relativity density [kg/kmol$_{fuel}$ per kg/kmol$_{air}$]	0.94	0.84	0.75

Table 3. General procedure for testing

<table>
<tr><th>Factors</th><th>Levels</th><th>Output variables</th></tr>
<tr><td rowspan="4">Load levels</td><td>3 kw</td><td rowspan="10">Effective thermal efficiency
Volumetric efficiency
Specific fuel consumption
Equivalence ratio
Specific emissions of CO_2, CO, CH_4 and NO_x
Exhaust gas temperature
Pressure in cylinder *vs*. CA curves</td></tr>
<tr><td>4 kW</td></tr>
<tr><td>6 kW</td></tr>
<tr><td>7 kW</td></tr>
<tr><td rowspan="3">Spark timing advance</td><td>– 100% biogas, 20 CA deg. bTDC</td></tr>
<tr><td>– 75% biogas + 25%CH_4, 16 CA deg. bTDC</td></tr>
<tr><td>– 25% biogas + 50%CH_4, 12 CA deg. bTDC</td></tr>
<tr><td rowspan="3">Biogas composition</td><td>– 100% biogas, 60%CH_4 + 40%CO_2</td></tr>
<tr><td>–75% biogas + 25%CH_4, 70%CH_4 + 30%CO_2</td></tr>
<tr><td>– 50% biogas + 50%CH_4 80%CH_4 + 20%CO_2</td></tr>
</table>

– All the experiments were performed in the same way, the equivalence ratio was hold constant in 0.95 for each point evaluated in the engine operation.
– The spark timing advance in CA deg. was calibrated according to the next: to 100% biogas 20 deg. bTDC, to 75% biogas + 25%CH_4 16 deg. bTDC, and to 50% biogas + 50% CH_4 12 deg. bTDC.

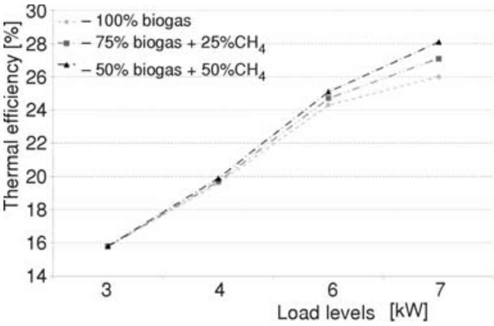

Figure 2. Variation of the otuput thermal efficiency with load levels

Experimental results

Effective parameters of the engine

Figure 2 shows the variation of the output thermal efficiency for the fuels and load levels studied. The highest thermal efficiencies were operating the engine with 50% biogas + 50%CH_4. Additionally, the thermal efficiency increased as the engine load level increased. The increase in thermal effective efficiency with CH_4 addition occurred because the percentage of CO_2 in the mixture decreased, which increased the low heating value and deflagration velocity of the fuel, resulting in a higher reactivity of the mixture. With 50% biogas + 50%CH_4 , the output thermal efficiency increase by 12% at 7 kW operation *vs.* 3 kW. The increase in the thermal efficiency when the engine is working at full load at a constant 1,800 rpm *vs.* a partial load occurred because the improvement in volumetric efficiency for maximum throttle opening. Decreasing load at constant speed, increases the specific fuel consumption due to the bigger magnitude of friction and increases the pumping work [14]. Figure 3 shows the variation of the CO emissions for the fuels and loads tested. The CO emissions in ppm increases as the engine load and methane addition are increased. The CO emissions are caused by incomplete combustion, as the load is increased, there is more fuel to burn. Moreover, when biogas is enriched with methane, there is a higher percentage of CH_4 in the mixture, which must be burned at the same time because the engine speed is constant. Emissions in ppm are helpful to analyze combustion but specific emissions give a point of view related with the process inside the engine to produce a required work, CO emissions in g/kWh are higher to biogas because is required more gas flow to produce the same output power and the output thermal efficiency is lower to 100% biogas operation.

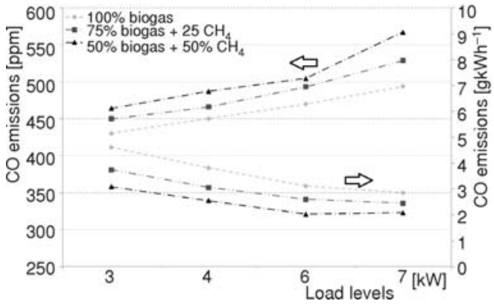

Figure 3. Variation of the CO emissions with load levels

Figure 4 shows the CH_4 emissions in ppm and g/kWh, in both cases the emissions are lower for the more enriched fuel at all load levels and the emissions decrease with increases in load level. This behavior is the opposite of the behavior of output thermal efficiency. Thus, improving engine operating conditions will reduce CH_4 emissions. Figure 5 shows the variation of the NO_x emissions, which shows that increasing the engine output power increases the NO_x levels. Using a higher addition of CH_4 with biogas decreases the NO_x/kWh because the gas flow is

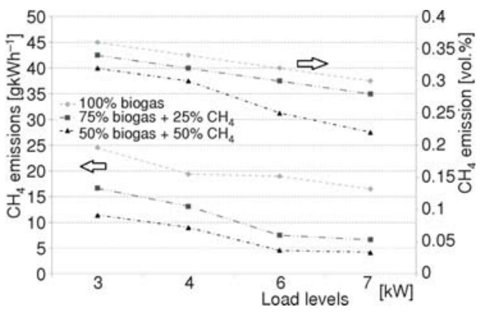

Figure 4. Variation of the CH_4 emissions with load levels

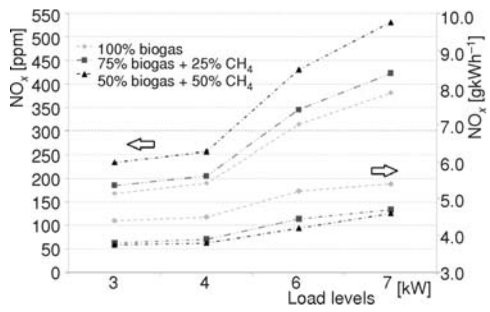

Figure 5. Variation of the NO_x emissions with load levels

lower than when the engine is operating with biogas. The increase of the load results in higher pressures in the cylinder, which results in greater cylinder temperatures, this is the reason because NO_x levels at these conditions increase; the formation of the NO_x are facilitated as the temperature is increased, N is dissociated into NO and NO_2. The NO_x emissions in ppm are lower to biogas because the CO_2 reduce the adiabatic flame temperature, when biogas is enrichment with the CH_4 addition, the adiabatic flame temperature is increased. However, the gas and air flows decrease, and the relation between mass flow and output power decreases too, NO_x/kWh values are lower for 50% biogas + 50%CH_4 followed by 75% biogas + 25%CH_4 and biogas exhibited the highest levels of NO_x for all the loads evaluated. Figure 6 shows the flue temperature. In general, the flue temperature is higher for fuel with a higher percentage of methane because the reduction in CO_2 causes both, the heating value of the biogas and the pressure inside the engine would be increased. To ensure that the initiation of combustion occurred at the appropriate point (just before the piston reached top dead center in the compression cycle), it is required to fix the spark timing advance for each composition. Variation in the spark timing was implemented according to the concept that less-reactive mixtures require a larger advance. Each additional 25%CH_4 enrichment should decrease the spark timing advance by 4 CA deg., the correct selection of the spark timing advance avoids knocking.

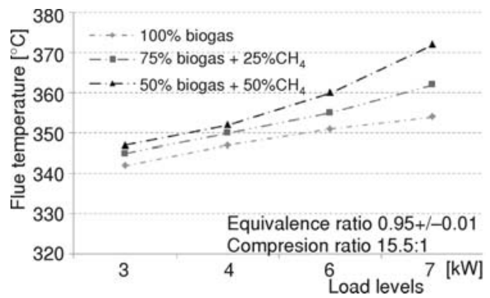

Figure 6. Variation of the flue temperature with load levels

Indicated parameters of the engine

Figure 7 shows the pressure inside the engine cylinder as function of the crankshaft angle. A comparison was made between the three fuels for a fixed output power of 7 kW at 1,800 rpm. A CA of 0° corresponds to top dead center. The figure shows that the higher peak pressure occurs for 50% biogas + 50%CH_4, followed by 75% biogas + 25%CH_4, and biogas. Additionally, the

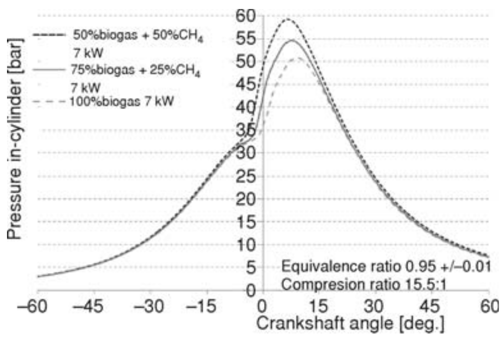

Figure 7. Pressure inside the engine cylinder for the three fuels

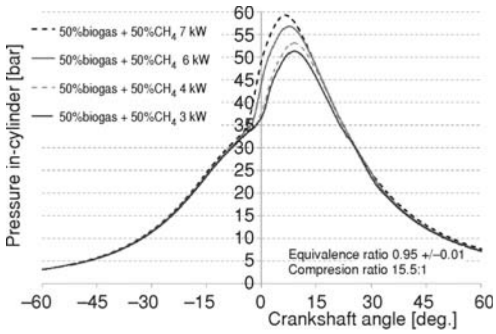

Figure 8. Pressure inside the engine cylinder with load levels

beginning of the pressure increase occurred earlier for biogas enriched with 50% methane than the other two fuels, with the biogas taking the longest time. These characteristics of the three curves are due to a higher low heating value and burning velocity for the more enriched biogas respect to the other two fuels. These higher values are because of the pressure increase begun earlier and achieving a higher value for the more enriched biogas than the other two fuels. For pure biogas, the low heating value and burning velocity are lower than for the enriched fuels, leading to a delayed combustion initiation and a lower peak pressure. For 50% biogas + 50%CH_4, the higher peak pressure leads to the highest temperatures, with a higher burning velocity, which reduces the residence time with less heat loses, all of which cause an increase in the thermal effective efficiency of the engine. Figure 8 shows the pressure inside the engine cylinder for four different loads using 50% biogas + 50%CH_4 as fuel. The highest peak pressure occurred with the maximum engine load, though the initiation of the pressure increase did not vary significantly among all loads; however, the slope of the curve is greater for the maximum load. In this case, as the fuel and its properties are the same, the highest peak pressure is the result of the larger fuel quantity used to achieve the required engine load. This pressure increase corresponds to the higher thermal effective efficiency of the engine at full load, as was previously mentioned. Figure 9 shows the fraction of burnt fuel for the three fuels at 7 kW. In general, the fuel with the highest level of enrichment begins combustion earlier, whereas combustion with biogas begins later. This result is because of the more enriched biogas has a larger CH_4 concentration and a smaller amount of CO_2. The more enriched fuel has a higher burning velocity than the other fuels, causing earlier combustion even the spark timing advance of the other fuels are increased.

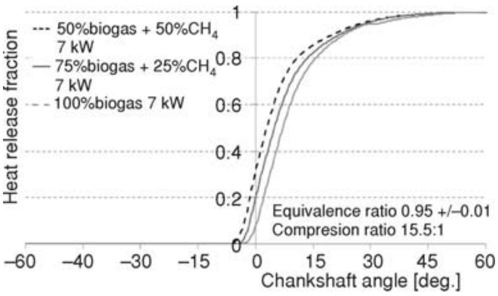

Figure 9. Heat release fraction for the three fuels

Figure 10 shows the variation of the temperature in the engine cylinder for the three fuels with a load of 7 kW. The highest temperatures occurred with the fuel that is most enriched with CH_4. This enrichment diminishes the CO_2 content, which increases the low heating value of the fuel and the flame speed, resulting in a more reactive mixture with a higher adiabatic flame temperature. Figure 11 shows the profiles of the net heat release rate, which take in account the total heat release and the heat lost to the walls for the three fuels with a load of 7 kW. These profiles illustrate that using 50% biogas + 50%CH_4, have both the largest net heat release and the earliest initiation of heat release. The faster and larger

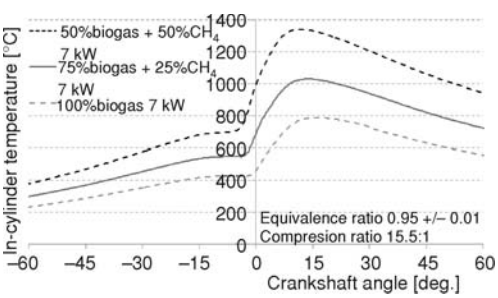

Figure 10. Temperature inside the cylinder for the three fuels

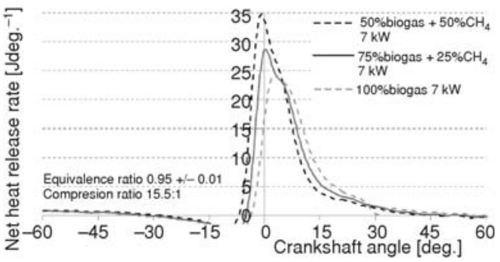

Figure 11. Net heat release for the three fuels

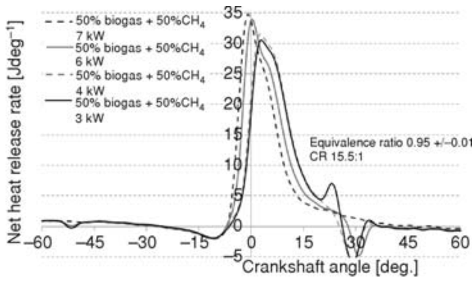

Figure 12. Net that release with load levels

net heat release occurs because the mixture is more reactive with a larger heating value and flame speed, leading to higher temperature and pressure profiles. All of these benefits of higher CH_4 enrichment result in higher levels of output thermal efficiency of the engine. Figure 12 shows the profile of the net heat release for 50% biogas + 50%CH_4 for the four load levels. The highest rate of net heat release occurs at the highest operational load, while the begun of the heat release is the same for the four loads. The larger rate of net heat release values observed with increases in the load is caused by the higher temperatures and pressures inside the cylinder.

Figure 13 shows the burn duration of combustion, burn duration as CA deg. measure of burn progress in cycle, between 0-90% of mass fraction burned, for the four load levels and the three fuels used, because of at high loads the pressure inside the cylinder is upper and thus the burn duration is a few brief regarding to low loads. Besides, for the fuel 50% biogas + + 50%CH_4, the burn duration in all loads is smaller than the others two fuels because the high flame speed. The coefficient of variation (COV) of indicated mean effective pressure (IMEP) seen in fig. 14, shows the four load levels and the three fuels used, than in all the cases the COV of the IMEP is lower than 5%, representing a permanent work. At high loads levels and using the fuel 50% biogas + 50%CH_4 the COV of the IMEP is the lowest in all the tests.

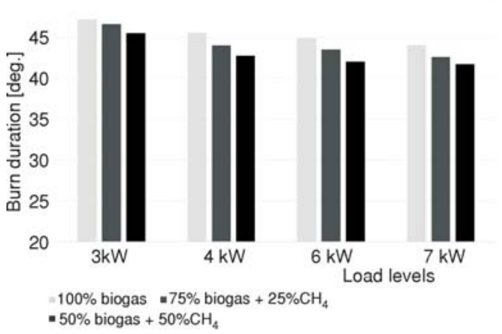

Figure 13. Burn duration (degrees)

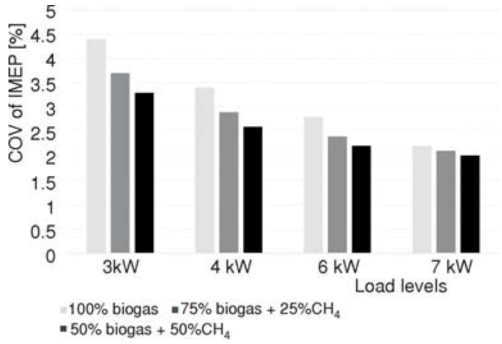

Figure 14. COV of IMEP

Comparison of results for engine in dual and SI mode

This research found similar results of engine performance than those reported by Porpatham *et al*. [3], however, this work achieved lower emissions of CH_4 (approximately 50% reduction), NO_x emissions were decreased significantly, and CO emissions are very low in both cases. Additionally, this research used the original CR of the engine (15.5:1), a higher engine speed (1800 rpm) with a higher output power (7 kW).

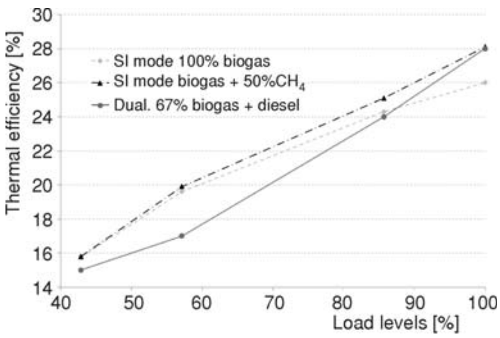

Figure 15. Comparison of output thermal efficiency of the Diesel engine in biogas-disel duel mode *vs.* SI mode with biogas

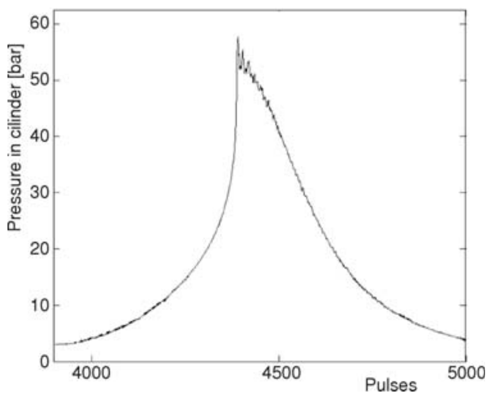

Figure 16. Pressure in cylinder with knock ocurrence

Table 4. Sensitivity and uncertainty analysis

Description	Value	Units
Voltage sensitivity	1.00	V
Current sensitivity	0.05	Amp
Flow sensitivity	0.01	l/min
CO emission sensitivity	1	ppm
CH_4 emission sensitivity	0.1	%
NO_x emission sensitivity	1	ppm
In-energy uncertainty calculation	0.01	kW
Out-energy uncertainty calculation	0.04	kW
Thermal efficiency uncertainty calculation	0.20%	%

A brief comparison is presented between the original Diesel engine operating in diesel-biogas dual mode and the engine after the conversion to SI mode. Figure 15 shows the comparison of the output thermal efficiency, this figure shows that the conversion to SI mode resulted in higher levels of the output thermal efficiency of the engine compared to when was working in diesel-biogas dual mode in part loads. The highest thermal efficiency (close to 28%) occurred at full load operating in both cases, SI mode using 50% biogas + 50%CH_4 and diesel-biogas dual mode. The highest efficiency (28.0%) in diesel-biogas dual mode occurred with a 67% substitution level of diesel in terms of an energy basis at full load. However, diesel injection was not optimized. At part load, SI engine had better output thermal efficiency than the dual mode engine, because of the lower equivalence ratio lead to decrease speed flame and increasing the heat loss to the walls for part load in dual engines. This relatively acceptable level of the output thermal efficiency in SI mode occurred because of the CR of the engine was not modified, CO_2 from biogas was present which acted like a knocking attenuator and the original Diesel engine now is working in Otto cycle with a gaseous fuel. The optimal condition operation for the engine without knocking and the highest output thermal efficiency operating in SI mode was with 50% biogas + + 50%CH_4 using a spark timing of 12 CA° bTDC, at a CR of 15.5:1 and an equivalence ratio of 0.95. Higher values of 80% CH_4 leads the engine to light knocking operation, this could be seen in fig. 16, where was used biogas + + 60%CH_4 and more than the 10% of cycles presented knocking. The curve of pressure inside cylinder versus pulses of CA help to identify when is occurring knocking inside the cylinder. All the tests presented in this job are free of knocking. Table 4 shows the sensitivity and uncertainty analysis.

Conclusions

This investigation used a Diesel engine converted to SI mode operating with biogas, 75%

biogas + 25%CH_4 and 50%biogas + 50%CH_4 at a CR of 15.5:1, an equivalence ratio of 0.95 ± 0.01, and changing the spark timing for each fuel, from which we made the following deductions.

- The optimal condition operation for the engine without knocking occurrence was with 50% biogas + 50% CH_4 using a spark timing of 12 CA° bTDC, at a CR of 15.5:1. The presence of CO_2 diminishes the possibility knocking.
- Engine in SI mode provides upper levels of the output thermal efficiency, operating with 50% biogas + 50%CH_4, with better levels of emissions of CO, CH_4, and NO_x in g/kWh.
- The maximum output power of the engine in SI mode was 7 kW, representing a decrease of 17.6% due to the conversion from 8.5 kW of the original Diesel engine. The conversion of a Diesel engine to SI at lower levels of power using alternative gaseous fuels guarantees conditions of high output thermal efficiency, low emissions and good stability of the combustion.
- The highest thermal efficiency (close to 28%) occurred at full load operating in both cases, SI mode using 50% biogas + 50%CH_4 and diesel biogas dual mode. At part load, SI engine had better output thermal efficiency than the dual mode engine.

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Nomenclature

bTDC – before top dead center, [deg.]
BTE – brake thermal efficiency, [%]
CA – crank angle
CAD – crank angle deg.
CR – compression ratio
SI – spark ignition
ST – spark timing, [CAD]
TDC – top dead center

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