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Experimental and numerical investigation of a MILD-based Stirling engine fed with landfill gas

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Abstract

Cogeneration systems based on Stirling cycles appear particularly appealing to achieve the goals of improving efficiency and reducing pollutants. Moreover, it is possible to couple such engines with MILD combustion burners. The industrial company Cleanergy provides energy solutions based on a Stirling engine that employs landfill gas. The objective of the present study is to evaluate the performances of such Stirling engine when the gas mixture fed to the engine itself and the operating conditions are changed. For all the combinations analyzed, the engine shows a good behavior in terms of emissions (NO_x in the order of 10 ppm) stability and efficiency. Numerical simulations on the combustion chamber of this engine are also carried on. The main objective is to validate models in presence of different fuel compositions.

Keywords: Type your keywords here, separated by semicolons ;

1. Introduction

Nowadays, in the field of energy production, particular attention must be paid to improving efficiency and reducing pollutants. To this goal, especially for residential applications, micro-cogeneration systems seem very appealing. Several cogeneration technologies are nowadays available. Among those, Stirling engines represent an optimal choice because of their ability to attain high efficiency, fuel flexibility, low emissions, low noise/vibration

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levels and good performance at partial load [1]. Unlike reciprocating internal combustion engines, the heat supply is from external sources, allowing the use of a wide range of energy sources including fossil fuels such as oil or gas, and renewable energy sources like solar or biomass. Since the combustion process takes place outside the engine, it is a well-controlled continuous combustion process, and the products of combustion do not enter the engine. Furthermore, it is possible to couple those engines with MILD combustion [2] burners.

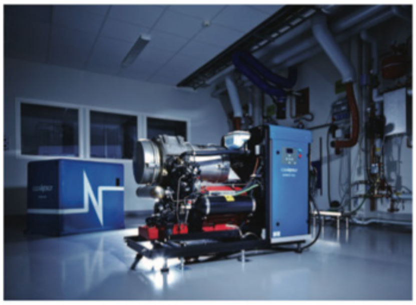
MILD combustion, also known as flameless combustion [3], is a rather new combustion technology that provides high efficiency in fuel consumption with low NO and soot emissions. It requires the reactants to be preheated above their self-ignition temperature and enough inert combustion products to be entrained in the reaction region, in order to dilute the flame. As a result, the temperature field is more uniform than in traditional non-premixed combustion systems, and it does not show high temperature peaks. Hence, NO formation is suppressed as well as soot formation, due to the lean conditions, low temperatures and the large CO_2 concentration in the exhausts. Moreover, MILD combustion is very stable and noiseless [4].

The growing trend today is that combustors should be fuel flexible. These different fuels are typically of Low Caloric Value (LCV), such as biofuels, syngas and landfill mixtures. The increasing interest in flameless combustion is motivated by the large fuel flexibility, representing a promising technology for low-calorific value fuels [5], high-calorific industrial wastes as well as in presence of hydrogen [6]. Recently several studies [7-9] have shown the compatibility of such regime with biogas. The industrial company Cleanergy provides energy solutions based on the Stirling engine. Cleanergy currently focuses on renewable, gaseous mixtures that are relatively difficult to burn since the energy content is small compared to natural gas. One such gas is landfill gas. In a landfill gas extraction, the methane content is decaying with time. More details about this technology can be found in Abou-Taouk et al. [10-11].

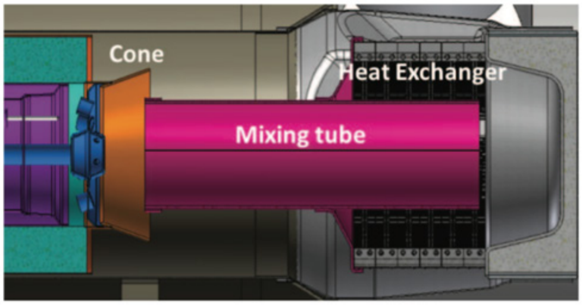
In the present work first the experimentally assessment of the performances of the combustion chamber of the aforementioned Stirling engine will be presented. Then the results of the CFD investigation will be also shown.

2. Experimental set-up

The experimental campaign has been carried on at Chalmers University of Technology in Gothenburg. The object of the tests is an alpha type Stirling engine equipped with a combustion chamber operating in MILD combustion regime. The burner it is called GasBox and it is shown in Figure 1.



(a) GasBox inside view.



(b) Centerline plane of the combustor.

Fig. 1 - Cleanergy GasBox and combustor.

The engine power output is 7.2 kW electric power and 16 kW heat power at the highest engine pressure (120 bar). The GasBox burner consists of a main burner and a heat exchanger. The gas is injected from one location and there are 6 main injection holes at the entrance of the burner for the air, which is preheated above the auto-ignition temperature of the fuel. The air and the fuel are also diluted by exhaust gas recirculation. The controlling parameter for the combustor is the value of the air-fuel equivalence ratio (λ). λ is kept constant at 1.3, by opening or closing the throttle for the air inlet. For stability reasons, the throttle cannot be open completely. Its opening percentage has to be kept between 2 % and 30 %.

During the experimental campaign presented in the this work three parameters are investigated:

- Engine pressure.
- Fuel composition.
- Air-to-fuel-ratio λ.

As far as the engine pressure is concerned, 3 different set points are considered: 75, 100 and 118 bar. The main variation concerns the fuel composition. The objective is to simulate the composition of Swedish landfill gas, at different moments in time. Indeed, in the landfill gas the CH_4 content decays with time. The different compositions are listed in Table 1. The mixtures are obtained blending pure gases, by means of the control system shown in Figure 2. Furthermore, tests are run for different values of the air-fuel equivalence ratio (λ) for a blend 60% CH_4 – 40 % CO_2 as fuel. Four different λ values are tested, ranging 1.2 to 2. For each experimental run, the emissions of the engines are measured with a Horiba Mexa 7000 tester. It can measure CO, CO_2, NO and UHC.

Table 1. Fuel mixtures composition, vol %.

	CH_4 [%]	CO_2 [%]	O_2 [%]	N_2 [%]
1	24.2	21.6	2.0	52.2
2	50.05	45.0	1.05	3.9
3	39.57	38.66	2.0	19.5
4	26.78	32.15	2.0	39.0
5	52.36	47.32	0	0
6	60.06	39.94	0	0
7	43.53	35.15	2.0	19.5

Fig. 2 - Fuel mixture control system

3. Experimental results

3.1 Effect of the gas mixture and engine pressure

The engine performs well at all the conditions analyzed, despite the high variation on the parameters. This is particularly true for the gas composition. Figure 3 shows the lower heating value for all the seven gas mixtures.

It can be noticed that the lower heating value changes over a wide range of values, going from 6.8 MJ/kg to 17.7 MJ/kg. All the values are well below the lower heating value of the pure methane (50 MJ/kg). As mentioned before, experimental tests are run at three different values of the engine pressure. Figure 4 shows the values of emissions for the different values of the engine pressure.

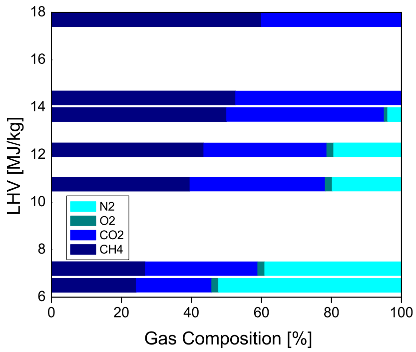


Fig. 3 - Lower heating value for the different gas mixtures.

It is possible to notice that the engine pressure does not have a great influence on the performances of the combustion chamber. At 75 bar, for all the gas mixtures tested, the values of the CO_2 on the outlet are almost constant. It decreases slightly when the value of CO_2 in the fuel mixture decreases. CO values are also pretty much constant around 20 ppm. Same observation can be made for the UHC (Unburnt Hydrocarbons), whose values are in the order of 15 ppm - 35 ppm. Those values for CO and UHC show that the combustion is complete and the performances of the burner are very good. As far as NO_x are concerned, their value increases with the LHV value of the mixture due to the higher temperatures reached when those mixtures are burned. For all the combinations, NO_x values are well below 20 ppm. At 100 bar, one can notice the same behavior for all the species examined. CO_2 values increase when the CO_2 in the fuel mixture increases. The values for CO and UHC are low, 23 ppm for CO and around 12 ppm for UHC, showing a very good performance of the burner. The NO_x emissions are in the order of 12 ppm for all the three gas mixtures. Finally, at 118 bar, it is still possible to notice the same behavior of the emissions although the values are slightly higher than in the two previous cases. CO values as well as UHC values are around 40 ppm. The NO_x emissions are in the order of 20 ppm.

The case with the best values is the one at 100 bar, which is the nominal functioning point of the Stirling engine. Nonetheless, the performances of the engine are very good also for the other two points studied, which represent a variation of 25% with respect to the nominal value.

3.1 Effect of the gas mixture and engine pressure

The second part of the experimental campaign was focused on evaluating the effect of the air-to-fuel ratio λ on the performances of the Stirling engine. This parameter was kept constant in the first part of the experimental campaign. The mixture analyzed has a LHV of 17 MJ/kg.

As far the emissions are concerned, the engine performs quite well even for those different values of λ. Figure 5 shows the emissions at different pressures for the different values of λ. Three values of engine pressure are considered: 75, 100 and 120 bar. The engine performs better, in terms of emission, at 120 bar. Indeed, it is possible to notice a peak in the emissions of CO and UHC for λ=1.2, indicating lower combustion efficiency for this point. The values of both CO and UHC decreases almost to 0 ppm when λ is increased. The NO_x emissions, still of the same order of magnitude of all the other tests, decrease from 30 to 5 ppm with increased λ. This means that it could be possible to slightly increase λ to compensate for the higher emissions at higher pressure showed in Figure 4.

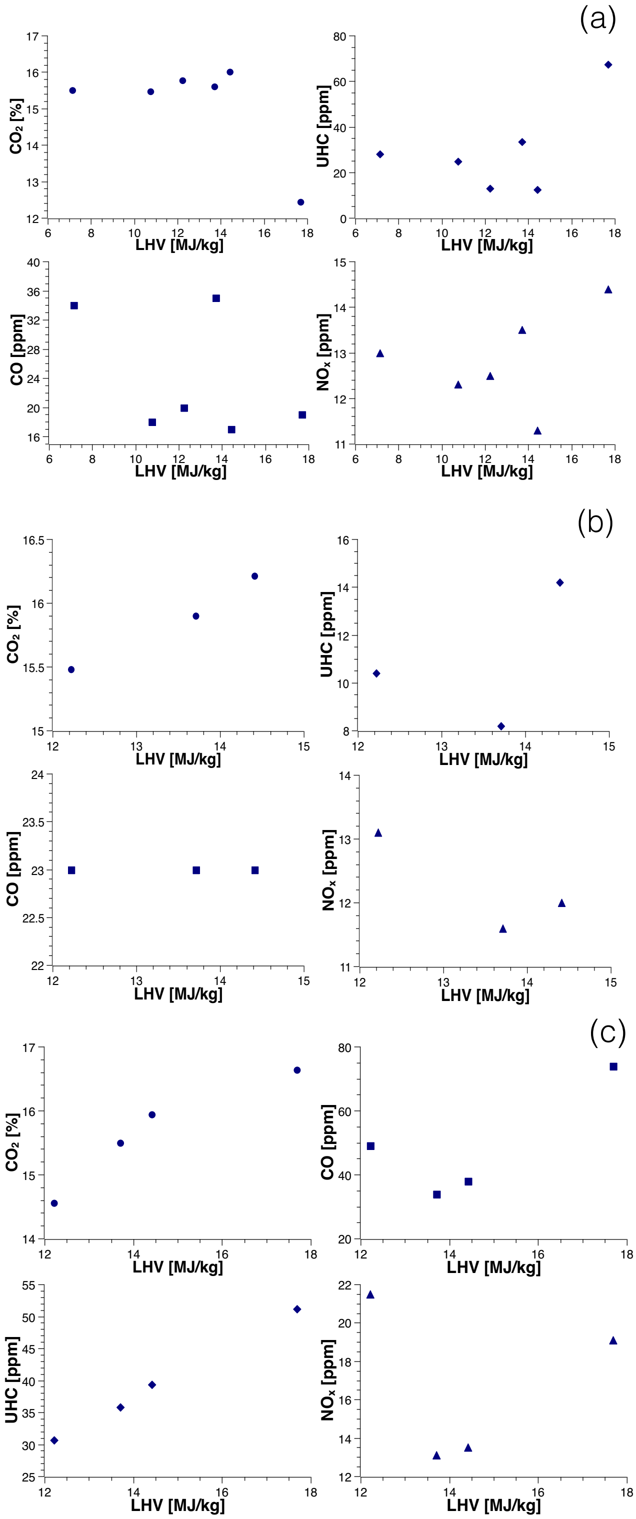


Fig. 4 - Emissions of the engine for different values of the engine pressure.
(a) 75 bar; (b) 100 bar; (c) 118 bar.

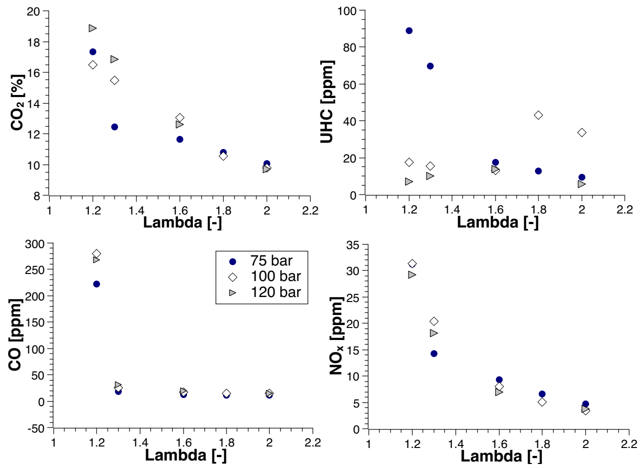


Fig. 5 - Emissions of the engine for different values of λ, at different engine pressure.

4. Numerical modelling

4.1 Numerical model set-up

Numerical simulations were run with the commercial software Ansys 17.0. Due to the combustion chamber symmetry, just 60° of the geometry are considered for the numerical modelling. The grid, chosen after a mesh independency study, contains 925k cells. Favre-averaged Navier-Stokes equations were solved, using the standard *k–ε* turbulence model. Radiation effects were accounted for using the Discrete Ordinate (DO) radiation model with the Weighted Sum of Grey Gases (WSGG) model for the participating media radiation. Turbulence-chemistry interactions were modelled using the Eddy Dissipation Concept (EDC) coupled with the GRI 2.11 oxidation scheme. The In-Situ Adaptive Table (ISAT) was coupled to EDC to reduce the computational costs. An error tolerance of 10^{-5} was selected to obtain table-independent results. As for NO emissions, different formation routes were considered: thermal NO mechanism, Fenimore's Prompt mechanism and intermediate N_2O mechanism. Those pathways were included as a one step global reaction for each path. All the above NO formation kinetic rates are integrated over a probability density function (PDF) for temperature, to account for the effect of temperature fluctuations on the mean reaction rates. The assumed PDF shape is that of a beta function and is evaluated through temperature variance for which a transport equation is solved.

As far as the boundary conditions are concerned, mass flow conditions are specified at both air inlets and fuel inlets whereas the outlet is modelled as a pressure outlet. The heat exchange between the combustion chamber and the working fluid of the Stirling engine (Helium), is calculated by means of a User Defined Function (UDF), which is tuned on the value of the temperature inside the chamber. In other words, it is not possible to extract heat if the temperature inside the chamber is below a certain threshold.

A second-order upwind discretization scheme was used for all equations and the SIMPLE algorithm was employed for pressure-velocity coupling. The simulations were run until the residuals for all the resolved quantities levelled out, resulting in a decrease of at least six orders of magnitude. In addition flow field variables at different locations were monitored to check convergence to the steady state solution.

4.2 Results

Simulations were run to investigate numerically the behavior of the engine, for different values of λ. The comparison between the experimental data and the numerical simulations is shown in Figure 6.

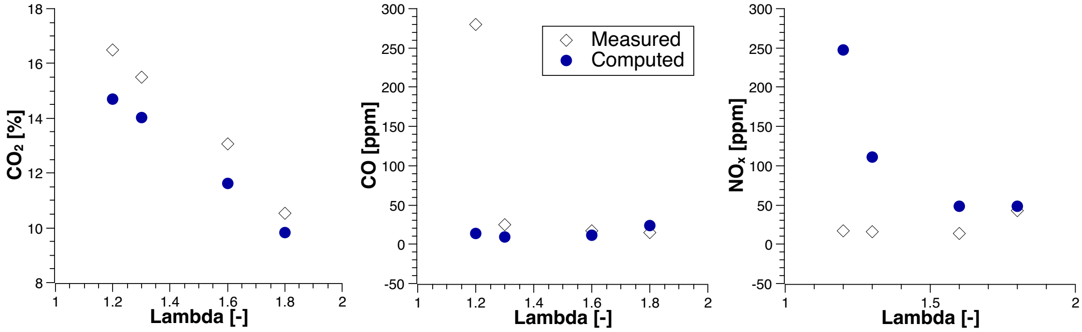


Fig. 6 – Comparison of the emissions of the engine measured and computed, for different values of λ.

The model is able to predict quite well the CO_2 and CO concentrations. The CO_2 is slightly underestimated in the numerical simulations, whereas the CO values are quite close to the experimental data, except for the run at λ=1.2. As far as the NO_x are concerned, there is an overestimation for the first two runs, namely λ=1.2 and λ=1.3. The other two points are predicted better by the model. Figure 7 shows the distribution, on a plane inside the combustion chamber, of temperature, the radical OH mass fraction and the NO production rate. The discrepancy on the NO emissions might be due to an overestimation of the temperature inside the combustion chamber. Indeed, in Figure 7 it is possible to notice that there is a region of the flow characterized by higher temperature. This region is the one in which there is the peak of the radical OH, indicating that it is the main reaction zone, and the peak for the production of NO. Further investigations are ongoing, to better understand the discrepancies which may be due to the heat extraction calculation.

5. Conclusions

In the present study, the performances of the Stirling engine of the industrial company Cleanergy are evaluated. The effect of the gas mixture fed to the engine, the engine pressure and the air-to-fuel ratio λ on the emissions of the engine are studied.

For all the combinations analysed, the engine shows a good behaviour in terms of emissions, stability and efficiency. This Stirling engine is very fuel flexible. The emissions are very low for all the gas mixture analysed, despite the high variation of the lower heating value of those mixtures (ranging from 6.7 to 50 MJ/kg). In particular, the NO_x emissions are always below 20 ppm. The performances of the engine are quite constant also when the pressure is changed. Three values are analysed, with a variation respect to the nominal case (100 bar) of 25%. As far as λ is concerned, the performances of the engine increase when λ increases, leading to lower emissions for λ values around 2 (nominal value λ=1.3).

The numerical model was able to capture quite well the trend of the experimental measurements, although some discrepancies may be noticed.

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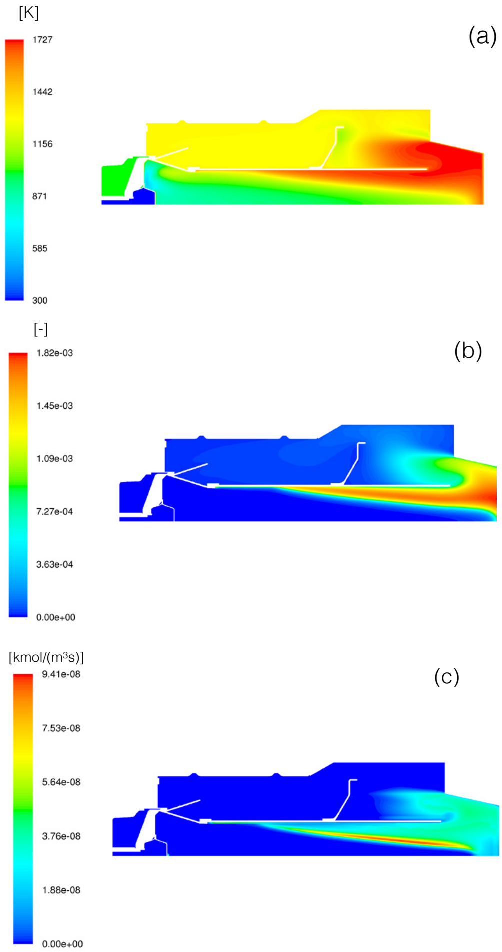

Fig. 7 – Distribution inside the combustion chamber of temperature (a), radical OH mass fraction (b) and NO production rate (c). EDC – GRI 2.11 numerical model.

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