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Experimental and numerical study of a micro-cogeneration Stirling unit under diverse conditions of the working fluid ☆

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HIGHLIGHTS

- A Stirling rated at 1 kWe and 8 kWth is analyzed experimentally and numerically.
- The developed model is an extension of the work by Urieli and Berchowitz.
- The initial pressure of the working fluid (nitrogen) is varied from 9 to 24 bar_g.
- The initial pressure influences strongly the fuel input and the electrical power.
- Major losses: wall heat conduction and non-unitary regenerator effectiveness.

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ABSTRACT

Micro-cogeneration Stirling units are promising for residential applications because of high total efficiencies, favorable ratios of thermal to electrical powers and low CO as well as NO_x emissions. This work focuses on the experimental and the numerical analysis of a commercial unit generating 8 kW of hot water (up to 15 kW with an auxiliary burner) and 1 kW of electricity burning natural gas. In the experimental campaign, the initial pressure of the working fluid is changed in a range from 9 to 24 bar_g – 20 bar_g being the nominal value – while the inlet temperature of the water loop and its mass flow rate are kept at the nominal conditions of, respectively, 50 °C and 0.194 kg/s. The experimental results indicate clearly that the initial pressure of the working fluid – Nitrogen – affects strongly the net electrical power output and efficiency. The best performance for the output and efficiency of 943 W and 9.6% (based on the higher heating value of the burnt natural gas) are achieved at 22 bar_g. On the other hand, the thermal power trend indicates a maximum value of 8420 W at the working pressure of 24 bar_g, which corresponds to a thermal efficiency of 84.7% (again based on higher heating value). Measurements are coupled to a detailed model based on a modification of the work by Urieli and Berchowitz. Thanks to the tuning with the experimental results, the numerical model allows investigating the profiles of the main thermodynamic parameters and heat losses during the cycle, as well as estimating those physical properties that are not directly measurable. The major losses turn to be the wall parasitic heat conduction from heater to cooler and the non-unitary effectiveness of the regenerator.

1. Introduction

In the scenario of micro-cogeneration (or micro Combined Heat and Power, micro-CHP), Stirling units are a promising technology for residential applications because of high total efficiencies, favorable ratios of thermal to electrical powers and lower emissions compared to reciprocating engines and micro gas turbines.[1] The present work covers the experimental and numerical analyses of a natural gas-fired commercial unit capable of generating 8 kW of hot water (up to 15 kW with an auxiliary burner) and 1 kW of electricity. Although featuring in general a low electrical efficiency, these micro-cogeneration units are able to recover heat with a high

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[1] In this work, the term "Stirling engine" refers strictly to the engine itself, while "Stirling unit" to the combination of the engine and the ancillaries used in a cogeneration installation, as for instance burner, fan, cogeneration heat exchanger, air preheating, etc.

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thermal efficiency, thus qualifying positively in terms of primary energy savings. Moreover, the thermal-to-electrical power ratios of 8-to-1 (and 15-to-1 with the additional burner) are well suited for covering simultaneously the thermal and electrical loads of typical residences in continental climates.

Chen and Griffin [1] as well as Cheng and Yu [2] conduct two detailed reviews of the numerical investigations on Stirling engines in 1983 and in 2010, respectively. Among these investigations, the work by Urieli and Berchowitz provides a fundamental contribution in 1977 [3]. Their numerical model, especially the so-called “simple analysis” variant, represents a balanced compromise between simplicity, accuracy and generality for the calculation of the performance of any Stirling engine.

The first study investigating the heat transfer phenomena inside the engine, which influence strongly the engine performance, is likely that by Finkelstein and Organ [4]. The authors develop a specific approach to quantify the regenerator heat exchange and to include the effects of a compressible working fluid in the estimation. Shultz and Schwendig [5] propose a simulation model of the general Stirling cycle. A further development is the work by Kongtragool and Wongwises [6], which studies the effects of dead volumes by manufacturing and testing twin- and four-piston engines.

In later years, Karabulut investigates specifically the beta free piston engines [7], whereas Parlak et al. deal with the gamma engine under almost stationary flow conditions [8], and Andersen et al. develop a one-dimensional model [9]. The first computational fluid-dynamics models are realized probably by Organ [10] and by Mahkamov [11], whose numerical results indicate clearly that the fluid-dynamic losses within the regenerator and the “dead” volumes are the major factors reducing the power output from the engine. Timoumi et al. [12] prove that the actual engine efficiency remains very low compared to the high theoretical value. Several engine losses are identified by Walker [13] and by Wilson et al. [14]: conduction losses in the heat exchangers, dissipation by pressure drop, shuttle and gas spring hysteresis losses. Costea et al. [15] highlight how the irreversibility due to heat transfer and pressure drop in the thermodynamic cycle has a significant importance in predicting the performance of the engine itself.

Regarding the regenerator of the Stirling engine, which affects majorly its performance, Gedeon and Wood provide a fundamental work in 1996 [16]. The authors verify experimentally the heat exchange and the pressure drop inside wowed matrix regenerators under oscillating flow conditions and, consequently, formulate a full set of empirical correlations. Later, Costa et al. confirm these correlations by numerical means [17]. Recently, Cheng and Yang [18] analyze the effects of the geometrical parameters on the shaft work for alpha, beta and gamma engine configurations. The simulation of the reference alpha engine yields a maximum shaft work for a phase angle shift of 80° and a dead to swept volume ratio of 0.5 (these parameters are close to the ones of the engine studied in this work).

If the open literature has plenty of numerical predictions of Stirling engine performance, it appears not to be as rich of experimental verifications of the actual performances. Aliabadi et al. examine the efficiency and the emissions of a residential Stirling engine fueled by diesel and biodiesel. Regarding specifically the electrical efficiency, the authors report it is around 11.5% with respect to heating values determine by ASTM D4809 standard [19]. That engine is the preceding version of the one used in this work and is fed by a liquid fuel instead of natural gas. Lastly, García et al. test the performances of two alpha engines under several working pressures and carry out a comparison among different numerical models [20]. They report a mechanical and an alternator efficiencies falling respectively in the range of 62–80% and 68–90% and depending mainly on the mechanical power output, which in its turn depends upon the mean pressure of the working gas.

As part of an ongoing study [21], a second campaign on the above-mentioned natural gas-fueled commercial Stirling unit is carried out at the Laboratory of Micro-Cogeneration of Politecnico di Milano [22]. The objective is to collect experimental data under diverse initial pressures of the working fluid, taking measurements at the boundary of the engine as well as inside the engine itself. The data are employed to reconstruct mass and energy balances and to assess energy and emission performances. They are also used to tune an in-house numerical model that is utilized ultimately to evaluate several parameters that are not accessible by direct measurement. To the knowledge of the authors, studies that are both numerical and experimental about Stirling engines are not common in the literature and, furthermore, there are no studies about diverse pressures of the working fluid.

The following sections illustrate first the experimental setup and then the numerical model. Next, the results from the experimental campaign as well as of the tuned numerical code are reported. Finally, conclusions are drawn.

2. Experimental campaign

The experimental campaign aims at creating a databank of measurements and at evaluating the performances in terms of electric and thermal efficiencies as well as of emissions upon the change of the initial pressure of the working fluid, Nitrogen (where “initial” indicates prior to starting up the unit). The initial pressure of Nitrogen is varied in discrete steps, 9, 12, 16, 20 (the nominal value), 22 and 24 bar_g. Other main parameters are instead kept constant, as described below. A high purity Nitrogen, of 5.0 quality, is employed in the campaign.

The experimental setup, shown in Fig. 1, provides measurements taken both externally and internally to the Stirling unit. Regarding the fed gases, which are natural gas from the national grid and the ambient air, the setup allows acquiring the inlet temperature and pressure. For the sole natural gas, also mass flow rate as well as molar composition with a micro gas chromatograph are collected. The temperature of air required for the combustion is controlled by a heating, ventilating, and air conditioning system (with a specified set point of 25 °C in this campaign). Regarding the flue gas, the measures include temperature, specific emissions (CO, NO, NO_2, SO_2) with an electrochemical analyzer, and molar composition with the same gas chromatograph adopted for the natural gas. A great attention is paid to the value of O_2 measured by both the analyzer and the gas chromatograph. For the water loop, temperatures, pressures, differential pressure and mass flow rate are acquired. In particular, the inlet flow rate of the water loop is managed by way of a variable-speed pump (set point of 0.194 kg/s); similarly, the inlet temperature is managed by a controlled-valve system (set point of 50 °C). The other external instruments are an electric power analyzer and a load cell to weight the condensate water from the flue gas. Nitrogen initial pressure, water inlet temperature and water mass flow rate define a single operating condition of the campaign.

Internally, on the air side, temperature is measured at the inlet of the engine after the air preheater and, on the flue gas side, at the outlet of the engine before entering the preheater itself. A thermocouple is placed in contact with one piston crown to detect the highest temperature of the cycle, namely the heater wall temperature (which is an important parameter for the numerical model). Thermoresistances on the water loop allow determining heat exchanges in the four main stages (refer to Fig. 1). Thermocouples are also placed on the engine walls. All instruments, external and internal, are outlined in Table 1.