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Modeling and testing of a micro-cogeneration Stirling engine under diverse conditions of the working fluid

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Abstract

Micro-cogeneration Stirling engines are promising for residential applications. This work focuses on the experimental and numerical analyses of a commercial unit generating 8 kW of hot water and 1 kW of electricity burning natural gas. Measurements are coupled to a detailed model based on a modification of Urieli and Berchowitz's work. The results indicate that the thermal efficiency is influenced by the water loop inlet temperature, varying from 90% at 30°C to 84% at 70°C (HHV-based). The measured and simulated powers of the engine are in the 900-964 W range depending on the water temperature and differ by less than 4%. Net electric efficiency of the engine is 15% and of the whole cogeneration unit above 9% (HHV-based). Helium instead of Nitrogen as working fluid is expected to increase the performance.

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Keywords: Stirling engine; numerical modeling; experimental testing; micro-cogeneration; micro-CHP.

1. Introduction

Micro-cogeneration Stirling engines are promising for residential applications because of high total efficiencies, favorable ratios of thermal to electrical power and lower emissions than internal combustion engines. This work focuses on 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. As part of an ongoing study [1], a second campaign employing new instrumentation is carried out at the Laboratory of Micro-Cogeneration [2]. The measures are used to tune an in-house numerical model that is employed to evaluate several parameters that are not easily accessible by direct measurement and to estimate the performance variation by changing the working fluid pressure and type.

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2. Experimental campaign

In the campaign, the initial pressure of the working fluid (Nitrogen) is reduced from the nominal 20 to 8 bar by 4 bar-steps; the inlet temperature of the water loop is set at 30, 50 or 70°C, while its mass flow rate at the nominal 0.194 kg/s. The experimental setup (Figure 1) provides measurements taken externally and internally to the engine. Regarding natural gas and air, the setup allows acquiring temperature and pressure, while for natural gas also mass flow rate and molar composition with a gas chromatograph. Regarding the flue gas the measures include: temperature, emissions (CO, NO, NO_2, SO_2) with an electrochemical analyzer, and molar composition with the same gas chromatograph. For the water loop: temperature, pressure, differential pressure and mass flow rate. The other external instruments are an electric power analyzer and a load cell to weight the condensate water from the flue gas. 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. A thermocouple is placed on one piston crown to detect the highest temperature, namely the heater temperature. Thermoresistances on the water loop allow determining heat exchanges in the four main stages (refer to Figure 1). Thermocouples are placed on the engine walls to evaluate thermal losses. Measures are recorded at steady state conditions, which are verified against the engine electrical power generation and the water loop inlet temperature, which must remain within ±0.3°C of the set point. The length of the acquisition is related to the oscillation of the entire system and equal to 1-2 hours. Data are filtered statistically and used to evaluate mass and energy balances as well as electrical and thermal efficiencies. Considering the measurement uncertainties, the expanded uncertainty on the electrical efficiency is about ±0.3% and on the thermal about ±2.8%.

3. Numerical model

The model is a modification of Urieli and Berchowitz's work [3], which divides the engine into five cells: compression, cooling, regeneration, heating, and expansion. The solution algorithm comprises:

- splitting one engine cycle into short steps (e.g. 10° of shaft rotation);
- evaluating the mass and the thermodynamic properties inside each cell and for each step;
- computing the gas flow between adjacent cells and heat/work exchanged through walls and pistons;
- integrating the cited head/work exchanged over one shaft revolution to evaluate the performances.

The processes inside the compression and expansion cells are assumed adiabatic, while the temperatures inside the cooling and heating cells are assumed uniform within each step. With respect to the original, the present model includes several novel subroutines to predict the following parameters:

- thermal behavior of the metal screen regenerator under oscillating flow conditions;
- irreversibility of exchanged heat within the cells of heater and cooler;
- concentrated and distributed pressure losses within the cells of heater, cooler and regenerator;
- heat losses due to conduction from the heater to the cooler, through the iron walls.

The regenerator is the most critical component due to its strong influence on the performances. The correlations by Gedeon and Wood [4] are adopted for the thermal performance and the pressure drop inside the regenerator. The non-ideality of the exchanged heat inside the heater and the cooler yields a decrease of gas temperature inside the heater, along with an increase of gas temperature inside the cooler, while the losses generated by the pressure drops, called pumping losses, have a direct effect in terms of reducing the cycle useful work. All input information, including geometric parameters, operational data, mechanical losses within the wobble joke and the bearings, electrical losses due to the non-ideal generator and auxiliary power are taken from a previous study [1]. Pressure drop inside the regenerator is estimated to be 8.2 kPa, compared to 2.4 and 6.2 kPa for cooler and heater, respectively. Average gas-to-wall temperature difference is evaluate to be 40.3 and 22.5°C in the heater and cooler, respectively.

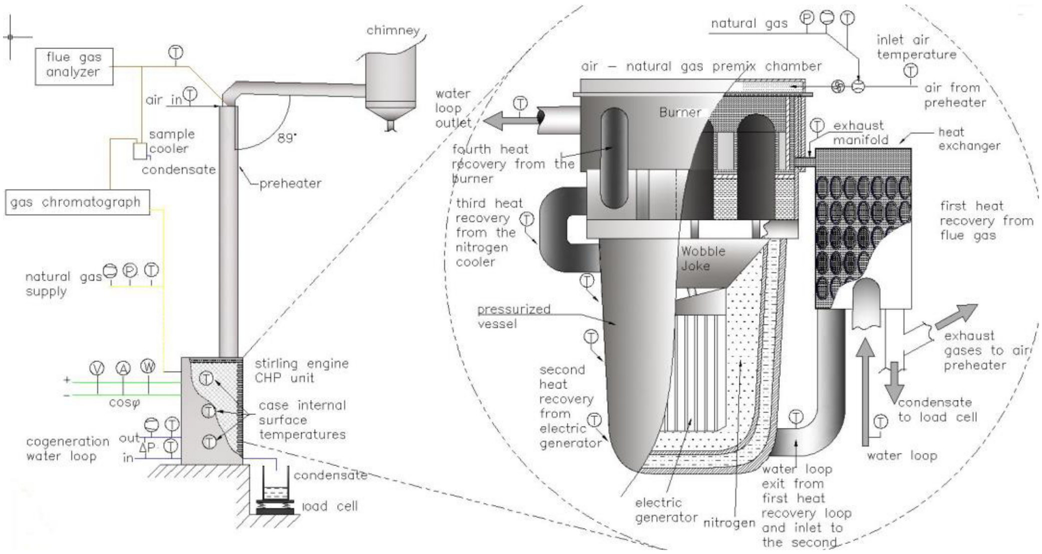

Figure 1. Schematic of the experimental setup showing the measurements acquired internally and externally to the engine.

4. Results and conclusions

Table 1 shows the experimental and simulated results for Nitrogen initial pressure at 20 bar and for water at 0.194 kg/s and at 30, 50 and 70°C. The measured net electrical efficiency of the cogeneration (CHP) unit is found to exceed 9% (based on the fuel HHV), while the measured and simulated net electrical efficiencies of the engine differ by no more than 4% and are both around 15% (based on the heat input to the engine). The measured thermal efficiency is influenced strongly by the water loop inlet temperature and varies from about 90% at 30°C to 84% at 70°C. Figure 2 illustrates the experimental and simulated power and efficiency for Nitrogen initial pressure from 8 to 20 bar and also for helium as working fluid. The values are in good agreement, showing the same decreasing trend as the pressure decreases, but differ by a constant offset, which requires a further investigation. Moreover, the figure indicates that adopting Helium as working fluid allows better performances at higher initial pressure.

Currently the experimental campaign is focusing on increasing the Nitrogen initial pressure from 20 to 24 bar by 2 bar-step and also on varying simultaneously the initial pressure and the water inlet temperature away from 20 bar and 50°C, while maintaining the same water flow rate of 0.194 kg/s.

References

[1] G. Valenti, P. Silva, N. Fergnani, G. Di Marcoberardino, S. Campanari, E. Macchi "Experimental and numerical study of a micro-cogeneration Stirling engine for residential applications", Energy Procedia, 45, pp.1235-44, 2014. doi:10.1016/j.egypro.2014.01.129

[2] S. Campanari, G. Valenti, E. Macchi, G. Lozza, N. Ravidà, N. Lazzari "Development of a microcogeneration laboratory and testing of a natural gas CHP unit based on PEM fuel cells", Applied Thermal Engineering, In Press, 2014. doi:10.1016/j.applthermaleng.2013.10.067

[3] Israel Urieli, David M. Berchowitz "Stirling Cycle Engine Analysis", Adam Hilger Ltd, Bristol, 1984.

[4] D. Gedeon, J.G. Wood, "Oscillating-Flow Regenerator Test Rig: Hardware and Theory With Derived Correlations for Screens and Felts", NASA Contractor Report 198442. (1996)

Table 1. Experimental and simulated results for Nitrogen initial pressure at 20 bar and water inlet at 0.194 kg/s and 30, 50 and 70°C.

Energy balances of engine and CHP unit	Water inlet 30°C			Water inlet 50°C			Water inlet 70°C		
	Sim.	Exp.	Diff.	Sim.	Exp.	Diff.	Sim.	Exp.	Diff.
Heat input to Stirling engine [W]	6290	6297	0.1%	6227	6188	-0.6%	6152	6158	0.1%
Heat rejected from the cooler [W]	4868	4915	1.0%	4876	4856	-0.4%	4879	4888	0.2%
Gross mechanical power [W]	1406	-	-	1324	-	-	1226	-	-
Heat loss due to piston stroke [W]	172	-	-	165	-	-	157	-	-
Heat loss due to wall conduction [W]	1725	-	-	1654	-	-	1573	-	-
Pumping loss [W]	103	-	-	105	-	-	100	-	-
Heat loss due to non-ideal regenerator [W]	768	-	-	736	-	-	699	-	-
Mechanical frictional loss [W]	225	-	-	212	-	-	196	-	-
Generator energy loss [W]	118	-	-	111	-	-	103	-	-
Auxiliary power absorption [W]	60	-	-	60	-	-	60	-	-
Engine net electrical power output [W]	1003	964	-4.1%	941	930	-1.2%	897	900	3.7%
Heat transferred to the ambient [W]	-	16	-	-	28	-	-	46	-
Engine net electrical efficiency (%)	15.9%	15.3%	-4.2%	15.1%	15.0%	-0.5%	14.1%	14.6%	3.6%
Heat input to CHP unit (HHV) [W]	-	10077	-	-	9573	-	-	9557	-
Thermal output from CHP unit [W]	-	9082	-	-	8033	-	-	8009	-
Thermal efficiency of CHP unit (HHV, %)	-	90.1%	-	-	83.9%	-	-	83.8%	-
Net electric efficiency of CHP unit (HHV, %)	-	9.6%	-	-	9.7%	-	-	9.4%	-
Total efficiency of CHP unit (HHV, %)	-	99.7%	-	-	93.6%	-	-	93.2%	-

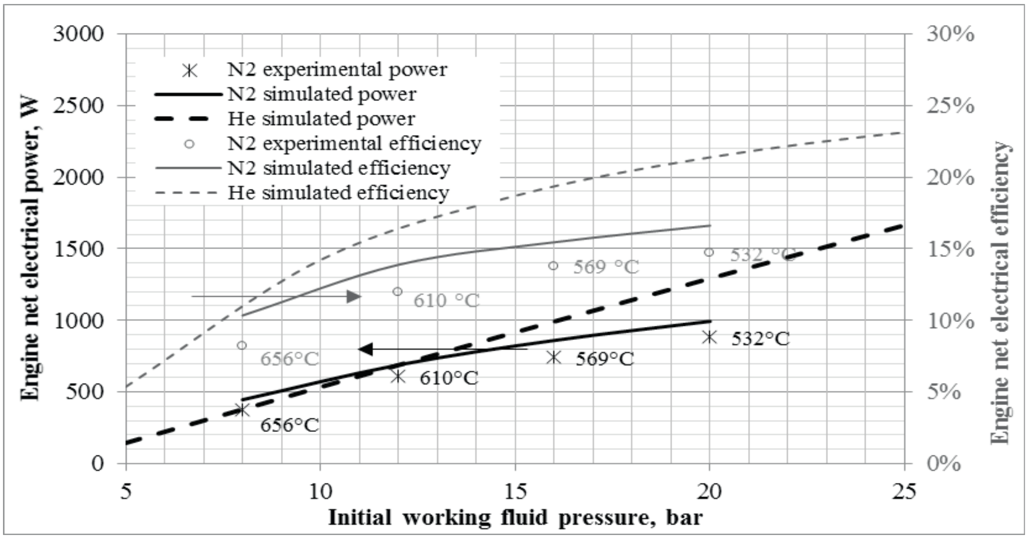

Figure 2. Experimental and simulated engine net electrical power and efficiency as a function of Nitrogen initial pressure from 8 to 20 bar (labels indicate the measured heater temperatures) and simulated results for helium as working fluid from 5 to 25 bar.