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Effect of coupling parameters on the performance of Fluidized Bed Combustor - Stirling Engine for a microCHP System

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Abstract

This work investigates the advantages of placing the hot side heat exchanger of a Stirling engine (SE) immersed in a Fluidized Bed Combustor (FBC). The general objective is to obtain both heat and electric energy using biomass as primary source [1]. This choice is primarily suggested by the more efficient heat transfer between the multiphase fluidized bed medium and the heat exchanger, as compared with immersion in the flue gases [2]. Moreover, the mechanical action of the solid particles reduces fouling of the heat exchange surfaces, a typical problem that arises with biomass combustion. In this paper we explore the possibility of achieving maximum mechanical power, useful to produce electric energy, with the minimum possible amount of fuel. To this aim, key parameters of the heat exchanger are studied in order to profit of the much enhanced heat exchange coefficients attainable with immersion in the fluidized bed. Indeed, the possibility of reducing the global surface area of the heat exchanger can have a positive impact on the efficiency of the SE due to the corresponding reduction of the dead space.

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Keywords: CHP,STIRLING ENGINE, COMBUSTION, BIOMASS, OPTIMIZATION, FLUIDIZED BED.

1. Introduction

This work represents the continuation of a study of CHP system characterized by the presence of a Stirling Engine (SE) having the head, and more specifically the hot side heat exchanger, in direct contact with the sand of a Fluidized Bed Combustor (FBC) [1]. The use of a FBC coupled with a SE offers several advantages. Heat transfer between the multiphase fluidized bed medium and the heat exchanger is much more efficient as compared with immersion in the flue gases. Heat transfer rates between the FB and a body immersed in the fluidized sand bed are typically at least 10 times larger than those attained

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with a gas [2]. This improvement in the heat transfer coefficient can be exploited by reducing the surface area – hence the volume – of the heat exchanger, and/or by maintaining a high heat flux at lower temperature gradients. The latter option may allow an increase of the hot side temperature of the SE, which results in a higher thermodynamic efficiency, and/or a decrease of the temperature of the exhaust gases, which results in reduced heat losses in the fumes. Moreover, FBCs are specially indicated for biomass fuels. Indeed, the mechanical action exerted by the fluidized solid particles substantially reduces the fouling usually caused by impurities in exhaust gases of a biomass combustion process: the physical cleaning action of the sand keeps the exchange coefficients constant over time and reduces the need for maintenance [1]. Another advantage comes from the typical temperatures of biomass combustion in a FBC, about 850 °C, perfectly compatible with those required for optimal operation of the SE. For example, typical temperatures required for the supply of SE EG-1000 employed by Sunpower [3] range from 600 °C (873 K) to 950 °C, with the upper temperature limited by the characteristics of the construction materials. At these temperatures, emissions of nitrogen oxides are low [4].

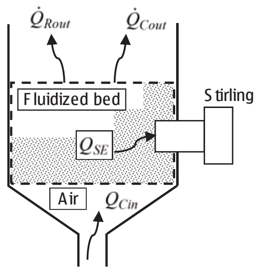

Fig. 1. Schematic of the FBC control volume with main thermal contributions to the energy balance.

In previous applications, Stirling engines have been placed in direct contact of the hot flues gases, thus exploiting the gas-solid heat transfer mechanism [5]. A large surface area is required for the heat exchanger to ensure the transfer of the amount of heat needed for a given mechanical power. In fact, with the efficiency measured in practical devices, the mechanical power obtained is usually lower than 1/5 of the thermal power absorbed. The required surface area can be achieved only with relatively long and closely disposed tube banks, that inevitably introduce negative effects for the performance of the SE itself, both because of the large dead space volume and of the high friction losses during the passage of the working fluid in the tubes.

Ongoing investigations study the global efficiency of the CHP system [6] by looking at the efficiency of the combustion process, both experimentally [7] and theoretically [8]. In this paper, we explore the possible means to achieve higher SE efficiency by taking advantage of the higher heat exchange coefficients. The assessment is performed adopting a dynamic model of the SE, introduced earlier [9] and adopted to analyse the performance of a micro-CHP system [6] coupled with a model for the FBC to attempt the optimal design of the heater when immersed in the FBC.

2. Numerical Model

Model for the FBC. The development starts from the definition of the energy balance in the control volume, hereinafter CV, identified by the surface that surrounds the volume of the fluidized bed, that interacts with the external environment in different ways, as shown schematically in Figure 1. Empirical models are adopted to evaluate the coefficients of all terms contributing to the energy balance. The source term $\dot{Q}_b$ in the FBC block accounts for the heat released by the combustion of the biomass, assumed complete. Details about the numerical model of the FBC can be found in [1].

Model for the SE. Following Bandurich, Chen and Normani [10,11], the engine is modelled as partitioned into five compartments (Figure 2), i.e. the compression space V_c, the expansion space V_e, the heater volume V_h, the cooler volume V_k, and the regenerator volume V_r. Cooler, heater and regenerator volumes sum up for the dead volume V_d. The mathematical model consists of mass and energy balances, making up a set of differential-algebraic equations (DAEs), under the following assumptions: the system is closed (no fluid leakage); the working fluid is ideal; effects of inertia of the working fluid are negligible; the dead space is isothermal; pressure in the dead space is uniform; compression and expansion spaces are adiabatic; the volumes of the compression and expansion spaces vary in time with sinusoidal law; wall temperatures of heater and cooler are uniform and constant, and equal to the respective gas temperatures; 80% of total pressure drop occurs inside the regenerator [11]. The mechanical power output is overestimated, since the model of the SE does not take into account mechanical friction [11].

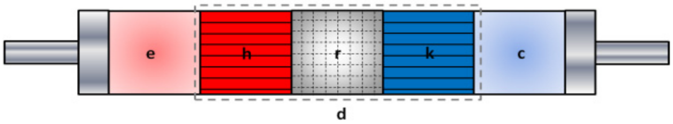

Fig. 2. Control volumes of the Stirling engine model..

Coupling of the FBC and SE models. The coupled model consists of a steady-state lumped CSTR that describes the FBC, and the Bandurich-Normani (BN) model that describes the SE. Coupling is achieved by equating the heat flux pairs:

- the heat flux $\dot{Q}_{SE} = U_b A_{ext,h}(T_{bed} - T_h)$ from the FBC to the SE, and $\dot{Q}_h$, the heat flux absorbed by the SE as it shows in the BN model [10];
- the heat flux $\dot{Q}_c = U_c A_{ext,c}(T_k - T_{w,0})$ from the cooler of the SE to the cooling water stream, and $\dot{Q}_k$, the heat flux output as it shows in the BN model [10].

Therefore, the coupling conditions of the model can be written as:

$$\dot{Q}_{SE} = \dot{Q}_h = U_b A_{ext,h}(T_{bed} - T_h) \quad ; \quad \dot{Q}_c = \dot{Q}_k = U_c A_{ext,c}(T_k - T_{w,0}) \tag{1}$$

where U_b is the heat transfer coefficient between the FB and the SE heater and U_c is the heat transfer coefficient between the Stirling engine and the water stream. These coupling conditions are justified since, at steady state, the thermal power absorbed and released by the engine are respectively equal to that transferred from the bed to the heat exchanger of the heater and that transferred from the heat exchanger of the cooler to the water stream, respectively. The problem consists of finding the temperatures of the heater, T_h, and cooler, T_k, which satisfy Equations (1), by means of an iterative method. The heat dissipated by the SE is partially recovered for production of Domestic Hot Water, $\dot{Q}_{DHW,SE} \equiv \dot{Q}_c$. The mechanical efficiency $\eta_{cog,mech}$, and part of the global cogeneration efficiency, $\eta_{cog,SE}$, are given by:

$$\eta_{cog,mech} = \frac{P_{SE}}{\dot{m}_{fuel} LHV_{fuel}} \quad , \quad \eta_{cog,SE} = \frac{P_{SE} + \dot{Q}_{DHW,SE}}{\dot{m}_{fuel} LHV_{fuel}} \tag{2}$$

3. Results

The range investigated for the parameter values is reported in Table 2. Fuel consumption bounds are dictated by the design target power of the system. In the computations, the engine rotation speed is 900 rpm, and the heat transfer coefficient between the bed and the tubes of the heater is 200 W/m^2 K.

Table 2. Lower and upper bounds for the decision (design and operation) variables.

Range of investigated parameters
2 kg/h $\leq \dot{m}_{fuel} \leq$ 16 kg/h
4 $\leq N_h \leq$ 40
5 mm $\leq D_h \leq$ 8 mm
2 cm $\leq L_h \leq$ 40 cm

Geometrical constraints are suggested by the overall obstruction of the heat exchanger with respect to the size of the bed, and by the need to not reduce too much the cross section of the tubes of the heat exchanger to avoid excessive friction losses. These constraints enter in the model in two ways. Firstly, they are used directly to determine the surface area of the heater exposed to the bed. Secondly, these parameters enter the correlations adopted to evaluate the pressure drop in the heater. A decrease of the pumping losses will result in a significant increase of the SE efficiency. The FBC parameter values here adopted are the same as in [1], while the parameter values of the Stirling engine are the same as in [9].

Table 3. Value of optimal parameters and corresponding model variables assumed as starting points for the analysis.

$\dot{m}_{fuel}$	rpm	D_h	L_h	N_h	V_h	$A_{ext,h}$	$\dot{Q}_{SE}$	T_h	T_{bed}	η_{SE}	$\eta_{cog,m}$	P
kg/h		**mm**	**cm**	**-**	**cm^3**	**cm^2**	**kW**	**K**	**K**	**-**	**-**	**kW**
19.74	**899.8**	**5.59**	**8.031**	**23**	**45.36**	**324.51**	**10.96**	**1030**	**1089**	**0.25**	**0.03**	**2.76**
9.84	**899.9**	**6.10**	**8.032**	**21**	**49.30**	**323.27**	**10.88**	**869**	**933**	**0.22**	**0.05**	**2.43**
4.00	**899.0**	**5.77**	**8.035**	**34**	**71.34**	**494.87**	**10.88**	**698**	**738**	**0.16**	**0.09**	**1.79**

In this work, the objective is to elucidate the effect of the geometry of the heater. To this aim, solutions are computed starting from Pareto optimal points previously evaluated in a multiobjective optimization process [9], reported in Table 3. Given the fact that optimal values in Ref. 9 hit the lower bounds for tube length, such bounds were here relaxed to verify whether shorter tube lengths could lead to ever better solutions.

Since CHP systems are designed to operate close to maximum performance, the influence of the heater design parameters on the efficiency of the SE at maximum fuel mass flow rate was firstly studied. Subsequently, the influence of the fuel mass flow rate for various design parameters of the heater was also investigated. Figure 3 reports the influence of heater tube lengths on the SE efficiency for various diameters and numbers of tubes at a fuel load rate of 16 kg/h. In the investigated parameter ranges, a strong drop of the efficiency of the SE occurs for tube lengths less than 4-5 cm. This is due to the decrease of the exchange surface area, that corresponds to insufficient heat absorbed by the heater. The efficiency of the SE shows a slowly decreasing trend for large tube lengths. For long heater tubes, the increase of both the friction losses and the dead volume become dominant on the increase of the absorbed thermal power allowed by the increase of the exchange surface area, thus reducing the efficiency of the engine.

Hence, the optimal length of the heater tubes to maximize the efficiency is found to range between 6 and 8 cm, for any combination of the other heater design parameter investigated. For some configurations, the optimal tube length is shorter than the lower bound length employed in Ref. [9]. It can be also observed that, by increasing the tubes diameter, the efficiency of the SE becomes less sensitive to the tube length. Particularly, for $D_h = 8$ mm and $N_h=10$, the efficiency of the SE shows a trend approximately constant versus tube lengths between 6-12 cm. These results suggest the use of tube diameters of about 8 mm to achieve a flexible design with respect to the choice of the number and length of the heater tubes. Indeed, too small diameters (e.g. D_h = 2 mm) restrict the range of feasible tube lengths. Therefore, the diameter of D_h = 8 mm is selected as optimal also in sense of design flexibility for the other heater parameters.

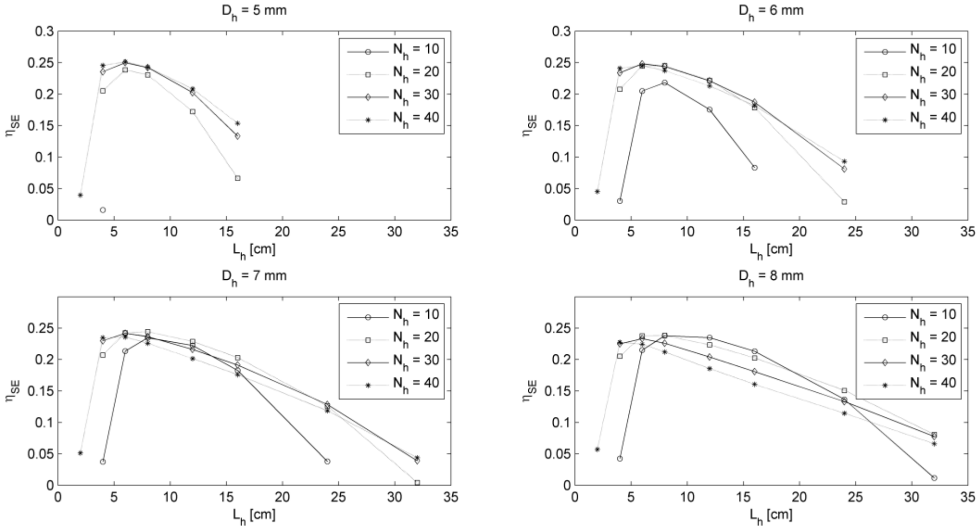

Fig. 3. SE efficiency with respect to the geometrical parameters of the heater at fuel feeding rate of 16 kg/h.

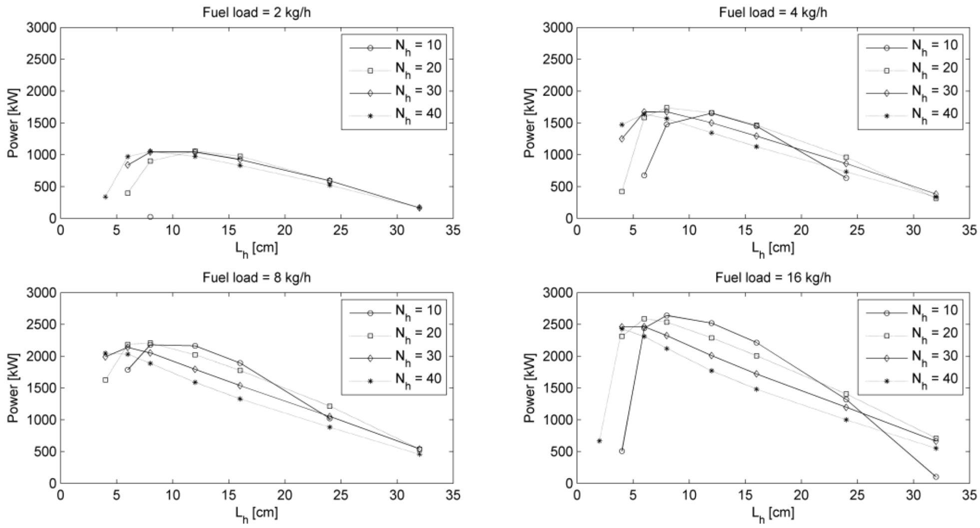

Fig. 5 (4). Mechanical power output from the SE with respect to L_h and N_h with D_h = 8 mm, at different fuel feeding rate.

Figure 4 reports the influence of L_h, for different N_h and fuel mass flow rates, fixed D_h at 8 mm, on the mechanical SE power, P_{SE}. Comparison of the parameter values of maximum efficiency with those of maximum power indicates that with the chosen diameter it is possible to select lengths L_h and number N_h of tubes that permit to match the maximum of efficiency with the maximum of the mechanical power at least at full load operating conditions. Actually, the maximum of the mechanical power at low load (2 kg/s) occurs approximately around the optimal set of parameters obtained at full load (16 kg/s).

4. Conclusions

This work focused on the effect of the geometric parameters of the SE heater on the performance (power and efficiency) of a microCHP system, starting from the optimal values obtained in a previous investigation [9]. Particularly, the results have shown that the diameter of the heater tubes is an important design parameter in view of the performance. The analysis shows that an appropriate choice of the length of the heater tubes allows to obtain a higher flexibility on the other design parameters, which can be sized to obey design constraints, given by the specific application, without high losses of efficiency and power.

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Biography

Simone Lombardi is a PhD student in Mechanical Engineering at University of Naples Federico II, under the supervision of prof. Gaetano Continillo. He was graduated with laude in Energy Engineering at University of Sannio. Since September 2013 to May 2014, he was visiting student at University of Cambridge.