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ANALYSIS OF THE GEOMETRICAL PARAMETERS OF THERMAL COMPONENTS IN A STIRLING ENGINE

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Stirling engines are known as high efficient engines that in theory can reach the Carnot efficiency. One of the most important sources of inefficiency is the imperfect heat transfer. In this way, heat exchangers are key components in the design of Stirling engines. These engines have been introduced into the building sector in order to supply the real needs in terms of electrical and thermal energy. Manufacturers have taken a variety of approaches to increase the appeal of Stirling engines: improving efficiency; increasing the maintenance interval; extending overall product lifetime; decreasing gas emissions; decreasing cost through less expensive materials; and focusing on inexpensive and renewable fuels [1]. Also, there is an urgent need to use more attractive renewable energy sources (*e.g.* solar energy) that can be used as energy source. In fact, any hot energy source can be used with a Stirling engine. The solar energy can be focused onto the heater of a Stirling engine, creating a solar-powered prime mover through direct conversion of solar power into mechanical power. In this study, a mathematical model, representative of an alpha Stirling engine is presented. The thermodynamic model of the Stirling cycle is based on the model presented by *Urieli* and *Berchowitz,* considering the non-ideal adiabatic engine conditions by further including non-ideal heat exchangers and heat transfer limitations, as well as the pumping losses throughout the system. A software-code has been developed in the MatLab® environment [2].

The alpha configuration (Figure 1) has two mechanically linked pistons located in separate cylinders that define the compression and expansion spaces. In the mathematical model, the engine is considered as a set of five components connected in series, consisting of: the compression space (*c*), cooler (*k*), regenerator (*r*), heater (*h*) and the expansion space (*e*). Each engine component represents an entity endowed with its respective volume (*V*), temperature (*T*), absolute pressure (*p*) and mass (*m*). The working gas flows between these two spaces by alternate crossing of, a low temperature heat exchanger (cooler), a regenerator and a high temperature heat exchanger (heater). Thus, the engine works between two temperatures T_h and T_c (hot and cold source, respectively). Several input parameters are required for the analysis: mean operating pressure, cylinder swept volumes, clearance volumes, hot and cold temperatures. The heat transfer process takes place in compact heat exchangers for a

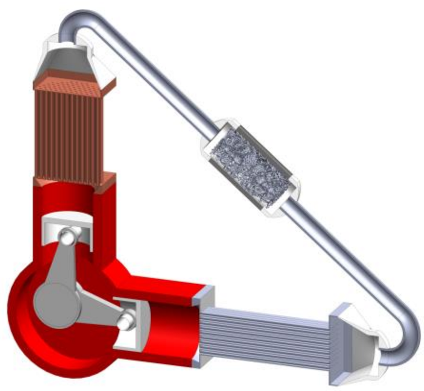

Figure 1. Representation of α-Stirling.

specific power output of the engine. The pumping losses refer to the work required to move the working gas through the heat exchangers and regenerator, thus reducing the net power output of the engine. The non-ideal effects of the regeneration are mainly due to the convective thermal resistance between the gas and the regenerator surface. To calculate the wetted heat transfer area, both heater and cooler are considered as smooth pipes, whereas regenerator was considered as a cylindrical mesh of fine metal wire.
The objective of the present work is to perform a sensitivity analysis to study the influence of geometrical parameters of the heat exchangers in the performance and efficiency of the Stirling engine for micro-cogenerations applications. The operating input parameters are presented in Table1.

Table 1. Values for the operating parameters for the numerical simulation

Parameter	Value	Parameter	Value
Mean pressure, bar	30	Engine swept volume, cm^3	130
Working fluid	Helium	Engine dead volume, cm^3	25
Rotation speed, rpm	3000	Cold sink temperature, K	353
Drive engine configuration	Sinusoidal	Hot sink temperature, K	725

Figure 2 presents the variation of engine efficiency and heater effectiveness as a function of the heater tubes internal diameter. Figure 3 shows the variation of engine efficiency and regenerator effectiveness as a function of regenerator matrix porosity.

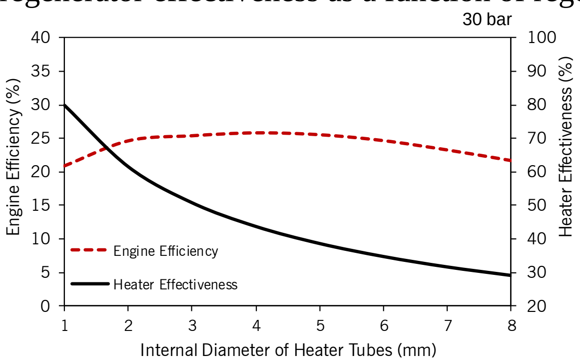

Figure 2. Engine efficiency and heater effectiveness as a function of the internal diameter of the heater tubes.

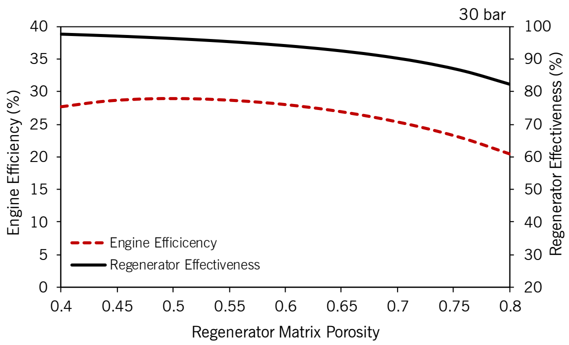

Figure 3. Engine efficiency and regenerator effectiveness as a function of the regenerator matrix porosity.

Figure 2 shows that the engine reaches the maximum efficiency with 4 mm internal diameter heater tubes, corresponding to a heater effectiveness of 43.7%. According to Figure 3, the Stirling engine efficiency is higher when the regenerator matrix porosity is equal to 0.48, which corresponds to a regenerator effectiveness of 96.7%. For the heat-exchangers, especially for the regenerator, a balance is needed between the minimization of flow friction and the maximization of heat transfer.

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