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

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Abstract. Stirling engines are known as high efficient systems that, in theory, can reach the Carnot efficiency. One of the most important sources of its inefficiency is the imperfect heat transfer. In this way, heat exchangers are key components in the design of Stirling engines. This technology has been introduced into the building sector in order to supply the real needs in terms of thermal and electrical energy. Manufacturers have taken a variety of approaches to increase the appeal of Stirling engines: improving efficiency; increasing the maintenance interval periods; extending overall product lifetime; decreasing gas emissions; decreasing cost through less expensive materials; and focusing on inexpensive and renewable fuels. There is an urgent need to use more attractive renewable energy sources (e.g. solar energy). 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. One of the most important thermal components of Stirling engines is the regenerator. The efficacy of the regeneration process has a great effect in the engine performance. The Stirling engine performance depends on the geometrical and operational characteristics of its thermal components. In the present work, a mathematical model, representative of an alpha Stirling engine is presented to study the influence of the heat exchanger's geometrical parameters in the engine performance. 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, heat transfer limitations, as well as pumping losses throughout the system components. A software-code was developed in the MatLab® environment to simulate the Stirling thermodynamic cycle. The study of the geometrical parameters of each thermal component allowed a new combination of geometrical characteristics, which results in the increase of 13.4% in the Stirling engine thermal efficiency for a reasonable power production output.

1 INTRODUCTION

Decentralized energy conversion systems have been recognized as an effective method to improve the efficiency of energy conversion, reducing gas emissions and the impact on climate change [9,10,13]. They are typically designed to provide electricity in a range of 1-10kWe and able to cover similar heat loads, which appears to be a good opportunity to meet the energy needs for the residential sector. The Stirling engine has interesting characteristics for micro-CHP (Combined Heat and Power) applications. Although this technology is not yet fully developed, and thus in very limited use, it shows a great potential due to its intrinsic high efficiency, low emissions and noise/vibration levels and good performance at partial load. Since the presentation of the patent by Robert Stirling in 1816, Stirling engines have been developed for different purposes and applications (such as submarine or aerospace applications). This versatility is due to the fact that the heat source in Stirling engines is external and thus can accept a wide variety of sources, including fossil fuels, biomass, solar, geothermal and nuclear energy. Another good point is that the combustion process can occur in steady state and, therefore, is easier to control. Stirling engines are very flexible and its most outstanding feature is related to their capacity to work at low temperatures [1].

In the design of Stirling engines, two aspects are currently considered: the maximum electrical efficiency and the maximum power production, and these parameters are well studied in the literature. Wu et al. [12] analysed the optimal performance of a Stirling engine. In their Study, the influence of heat transfer and regeneration time on the Stirling engine cycle performance was discussed. Puech and Tishkova [7] performed a thermodynamic analysis of a Stirling engine conducting an investigation about the influence of regenerator dead volume variations. The results showed that the dead volume amplifies the imperfect regeneration effect. Boucher et al. [2] related a theoretical study of the dynamic behaviour of a dual free-piston Stirling engine coupled with an asynchronous linear alternator. The objective was the evaluation of the thermo-mechanical conditions for a stable operation of the engine. Formosa and Despesse [6] developed an analytical thermodynamic model to study a free-piston Stirling engine architecture. The model integrated the analysis of the regenerator efficiency and conduction losses, the pressure drops and the heat exchangers effectiveness. The model was validated using the whole range of the experimental data available from the General Motor GPU-3 Stirling engine prototype. The influence of the technological and operating parameters on Stirling engine performance was investigated. The results from the simplified model and the data from the experiment showed a reasonable correlation. Rogdakis et al. [8] studied a Solo Stirling Engine V161 cogeneration module via a thermodynamic analysis. Calculations were conducted using different operational conditions concerning the heat load of the engine and the produced electrical power. Zarinchang and Yarmahmoudi [14,15] performed a very interesting study in order to optimize the thermal components in a 20 kWe Stirling engine. The main objective of their study was re-designing the heat exchangers by using two programs, the STRENG and the OPTIMUM. Thus, the authors presented an evaluation to the geometrical parameters effect in the Von-Mises stress, engine efficiency and power output.

Therefore, the Stirling engine performance depends on geometrical and operational characteristics of the engine and its thermal components. The present study presents a sensitivity analysis to the geometrical parameters of the heat exchangers (i.e. heater, regenerator and cooler). The main objective is to analyze the influence of geometrical parameters such as the internal diameter, heat exchanger length, number of tubes for the cooler and heater; the porosity and the diameter of the regenerator matrix wire, in the performance and efficiency of the Stirling engine.

2 NUMERICAL MODEL & INPUT PARAMETERS

This study is focused on the optimization of the thermal components (i.e. the heat exchangers) of a Stirling engine, able to produce 1-5 kW of electricity. For this study, an alpha configuration was chosen to perform the numerical simulations. 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 the compression and expansion spaces by alternate crossing of, a low temperature heat exchanger (cooler), a regenerator and a high temperature heat exchanger (heater).

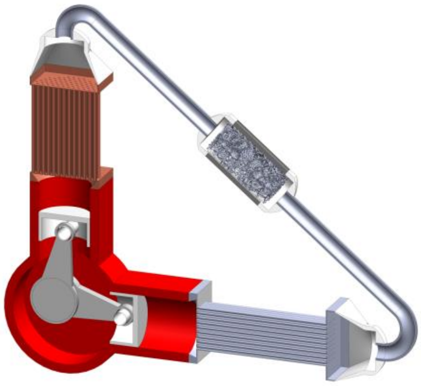

Figure 1: Representation of an alpha Stirling configuration.

Heat is transferred from the external heat source to the working gas in the heater during the expansion process, cyclically stored and recovered in the regenerator during alternate isochoric evolutions, and rejected by the working gas in the cooler during the compression stage. Thus, the working gas temperature varies in a smaller than the hot and cold sink temperatures while flowing between them. The hot and cold sink temperatures correspond to the values of 725 K and 353 K, respectively. The value for the hot temperature was assumed considering that the energy source is concentred solar radiation. The mathematical model in the present works integrates two analyses to reach the numerical solution of the thermodynamic cycle. Firstly, the developed code, called SEA (Stirling Engine Analysis), invokes an Ideal Adiabatic simulation and, sequentially, a Non-ideal simulation where the net enthalpy reduction in heat transfer due to the imperfect regeneration and the pressure losses are evaluated and accounted in the Stirling engine performance. The non-ideal effects of the regeneration are mainly due to the convective thermal resistance between the gas and the regenerator surface, whereas, the pumping losses refer to fluid friction associated with the flow through the heat exchangers, resulting in a pressure drop and power output reduction. The full mathematical description of the numerical model is presented elsewhere [3,4].

Several input parameters are required for the analysis: mean operating pressure, cylinder swept volumes, clearance volumes, hot and cold temperatures. The operating input parameters are presented in Table1.

Table 1: Operating parameter values for the numerical simulation

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

The numerical simulations were carried out considering helium as the working fluid. This gas has been chosen due to its thermal properties: helium has a higher thermal diffusivity and lower dynamic viscosity than air. Previous studies demonstrated that engine specific power is roughly proportional to the engine speed and mean pressure [3,5]. So, the mean operation pressure was fixed at 30 bar and the rotational speed was considered to be 3000 rpm.

Concerning the heat exchangers configuration, the heater and cooler are considered as a bank of parallel thin tubes. This choice results from the need for a large contact area together with the capacity for a large mass flow of working gas with low pumping losses. The regenerator is a special heat exchanger used in Stirling engines to improve its efficiency. Heat is transferred from the regenerator to the working fluid and it is therefore pre-heated when the working fluid moves from the compression to the expansion space. In the present work, a tubular regenerator with a wire mesh matrix was considered. The geometrical input parameters to the base-case configuration are presented at Table 2.

Table 2: Heat exchangers base-case input geometric parameter

Stirling Component	**Heater**	**Regenerator**	**Cooler**
External Diameter, mm	n.a.	56	5.0
Internal diameter, mm	3.0	46	3.0
Number of Tubes	80	n.a.	150
Length, mm	150	60	100
Matrix Porosity	n.a.	0.7	n.a.
Matrix Wire Diameter, mm	n.a.	0.3	n.a.

n.a. – non applicable

To investigate the influence of these parameters on the engine performance, they were studied individually and, for each simulation, the other parameters were kept unchanged.

3 RESULTS AND DISCUSSION

The goal to be pursued in this sensitivity analysis is an optimization of engine thermal efficiency against efficiency power output. The modifications to be considered in this paper will be restricted to the heat exchangers. These components can be re-designed to improve the engine performance. The method to be employed involves simulating different input geometrical parameters and assesses the engine efficiency, power output and the respective effectiveness of each heat exchanger.

3.1 Heater

The physical model only considers the convective heat transfer from the internal wall of the heater tubes to the working gas and assumes a constant temperature at the outer surface of the heater. Thus, the heat transfer will be significantly affected by the heater inner diameter, the heater length and the number of tubes, which are used to define the contact heat transfer area. Figure 2a) shows the engine efficiency and the heater effectiveness variation as a function of the number of the heater tubes. Results show that the engine efficiency slightly increases with the number of the heater tubes, varying between the values 23.6% and 28.1%. The heater effectiveness also increases with the number of tubes due to the increase of the heat transfer area with the increment of the heater tubes number. Nevertheless, it was also verified that the convective heat transfer coefficient decreases due to lower flow velocities.

a) **b)**

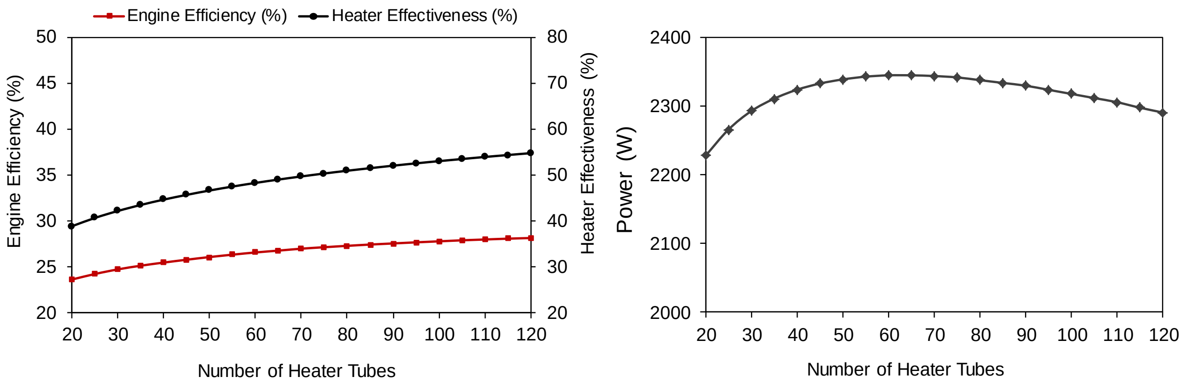

Figure 2: **a)** Engine efficiency, heater effectiveness and **b)** Power production as a function of the number of heater tubes.

The power production was also calculated, as shown in Figure 2b). Results show that the power production increases with the number of the heater tubes until a certain value. This values corresponds to a power production of 2345 W obtained for a number of heater tubes between 60 and 65. This means that, for a heater configuration with a set of smooth tubes, the optimum value of tubes corresponds to 60-65 tubes to achieve the maximum power production and an engine efficiency of 26.6 - 26.8%, respectively. Increasing the number of the tubes also reduces the fluid frictional work in the heater. Nevertheless, at some point, the increase in the void volume at the heat exchanger overlaps the benefits obtained from a lower friction and better heat transfer. From the analysis, it was found out that the heat transfer, and hence the overall performance, is improved only very slowly by increasing the number of tubes.

The internal diameter of the heater tubes is also an important parameter that affects the pressure drop in heater. A variation between 1.0 mm and 8.0 mm was studied for the internal diameter of the heater tubes. Figure 3a) presents the engine efficiency and the heater effectiveness variation as a function of the internal diameter variation. Results show that the increase of the internal diameter leads to a strong decrease in the heater effectiveness (80.0% to 29.0%). This is probably due to the reduction of the convective heat transfer coefficient. Regarding the engine efficiency, the maximum value is achieved for an internal diameter of 4.0 mm. Nevertheless, the power production (Figure 3b)) reaches the highest value for lower diameters (i.e. maximum power of 2560 W for an internal diameter of 1.5 mm).

Increasing the internal diameter also reduces the friction between the gas and the heater walls, which results in lower pumping losses. This outcome is in agreement with the analysis of Zarinchang and Yarmahmoudi [14] who claim that an increase in tube diameter produces large reductions in pressure drop.

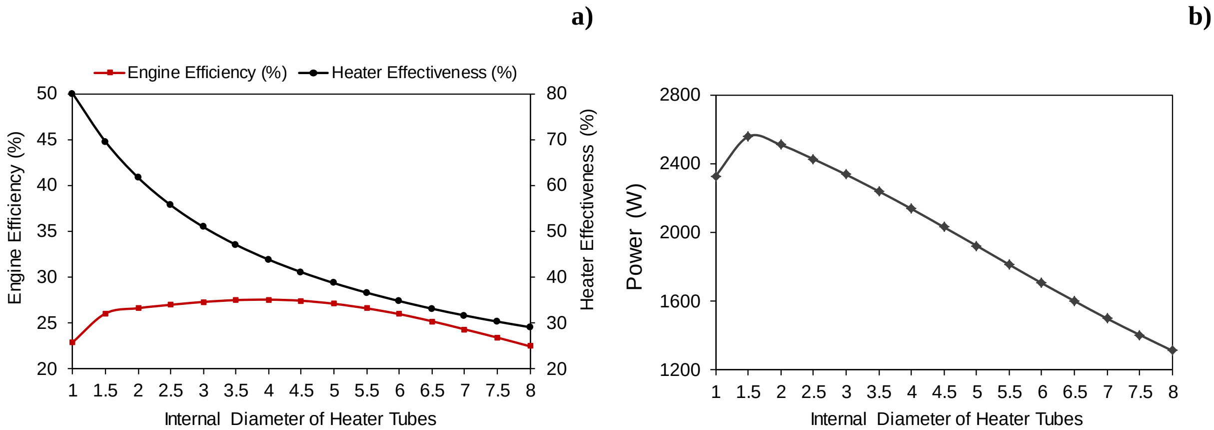

Figure 3: **a)** Engine efficiency, heater effectiveness and **b)** Power production as a function of the internal diameter of the heater tubes.

The length of the heat exchanger is also an important parameter to investigate. The simulations have been performed considering a variation in the range of 20 mm to 200 mm in length. Figure 4a) presents the results for the engine efficiency and heater effectiveness considering different values for the heater length.

Results show that the heater effectiveness rises almost linearly with the increase of the heater length. The increase of the heater length leads to an increase of the heat transfer area, which explains the increase in the heater effectiveness. The engine efficiency also increases, but its value almost stabilizes for heaters with 100 mm of length. According to the data from the Figure 3b), it is verified that the power increases continuously with the heater length and this can be explained by the implicit increase in the heat effectiveness versus smaller increases in void volume and pumping losses.

To improve the engine performance, a certain combination of input geometrical parameters is required. From the results, probably the best combination of parameters relies on heater tubes with a diameter of 4.0 mm, with a length above 100 mm. In terms of number of tubes, if the maximization of the power production is one of the required improvements, the number of tubes should not be more than 80 (considering the commitment between the engine efficiency increase and the power production reduction).

a) **b)**

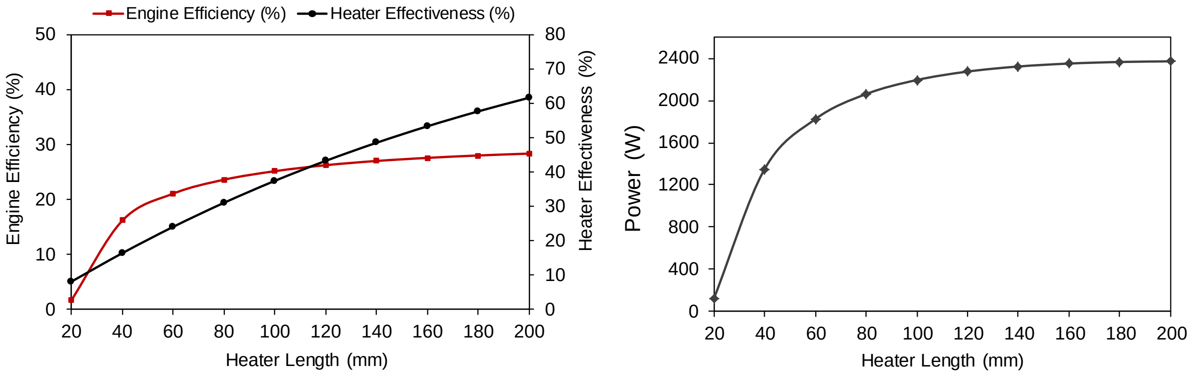

Figure 4: **a)** Engine efficiency, heater effectiveness and **b)** Power production as a function of the heater length.

3.2 Regenerator

The efficacy of the regeneration process is very important in the engine performance because along the thermodynamic cycle, heat transfer losses in the fluid pre-heating and pre-cooling leads to increases in the hot and cold energies and thus to an important decay in engine efficiency. The maximum efficiency of this process would only be achieved if the heat transfer coefficient or the area of heat transfer is infinite. However, the working gas does not have a null heat transfer capacity neither the regenerator matrix has an infinite heat capacity. To improve the regenerator heat transfer, it is of utmost importance to establish a commitment between the heat transfer and the fluid friction. In that way, the regenerator geometry and the variations in matrix wire diameter and porosity are studied in order to understand their influence in engine and regenerator performance. Figure 5a) presents the results of the engine efficiency and regenerator effectiveness for different regenerator lengths.

a) **b)**

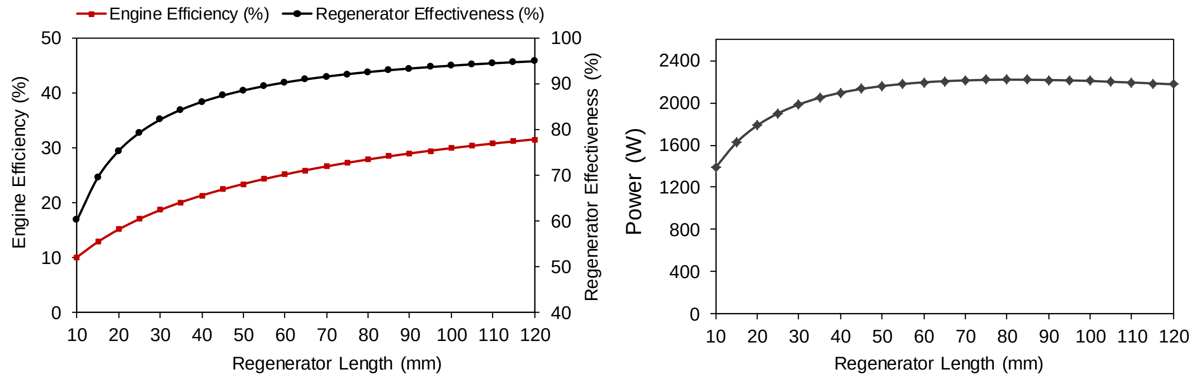

Figure 5: **a)** Engine efficiency, regenerator effectiveness and **b)** Power production as a function of the regenerator length.

Results for the engine efficiency and the regenerator effectiveness show a similar trend. According to the results, both rise with the increase of regenerator length. Regarding the power production, it is observed that there is no significant variation from 70 mm of length awards. The additional dead volume and the increase of the pressure losses lead to a maximum power for a length of 80 mm. However, for small regenerator lengths, i.e., below 30 mm, the power, as well as, efficiency quickly decreases, as presented in Figure 5b).

The porosity of the regenerator is an important parameter for engine performance. It affects the hydraulic diameter, dead volume, velocity of the gas, regenerator heat transfer surface and regenerator effectiveness; and thus, affects the losses [10,11]. Figure 6 presents the performance results for several combinations of matrix porosities and matrix wire diameters.

a) **b)**

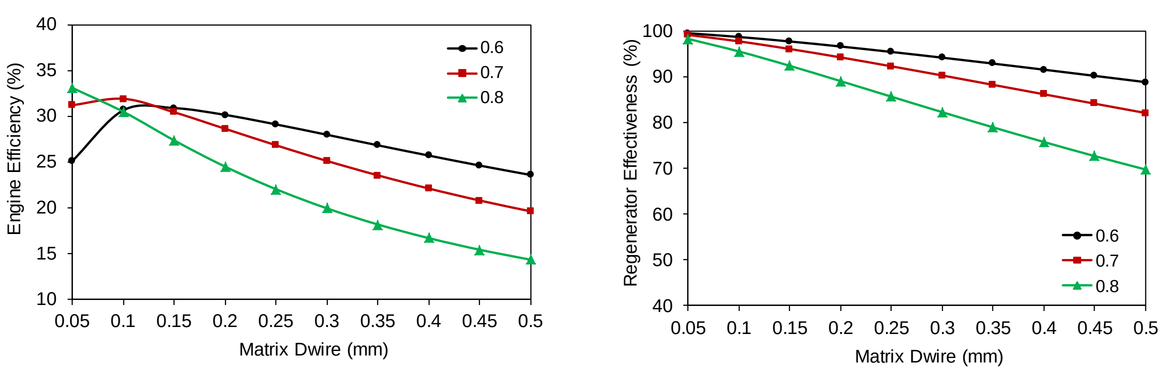

Figure 6: **a)** Engine efficiency and **b)** Regenerator effectiveness variation for different combinations of matrix wire diameter and mesh porosity.

The best matrix should compromise between high effectiveness and low-pressure drop in order to obtain minimal losses in the regenerator. Results shows that the best engine performance is obtained for lower wire mesh diameters (0.1 mm) combined with a mesh porosity of 0.7 for higher wire diameters a lower porosity (0.6) gives better results.

3.2 Cooler

The cooler tubes internal flow conditions are quite similar to the heater but at lower temperatures. To reduce the temperature of the working fluid an outside flow of water is used as a cold sink. Thus, heat transfer phenomenon includes the convective heat transfer from working gas inner to the inner wall of the cooler tubes to the, conductive heat transfer through the inner to outer tube wall surface and outside convective heat transfer to the coolant, the water. Figure 7a) shows the engine efficiency and the cooler effectiveness variation as a function of the number of the cooler tubes. Results show that the engine efficiency increases with the number of the cooler tubes, showing a variation of 16.8% in the tested range (40 to 160 tubes). Likewise, the results from the cooler effectiveness show that the increase of the number of cooler tubes does not affect it significantly. There is a variation of only to 8.0% in the cooler effectiveness. Power production presents a great increase with the rise of the tubes. This results may be explained by the counterbalance between the gains in heat transfer (i.e. increase of the heat transfer area) and the reduced pumping losses.

a) b)

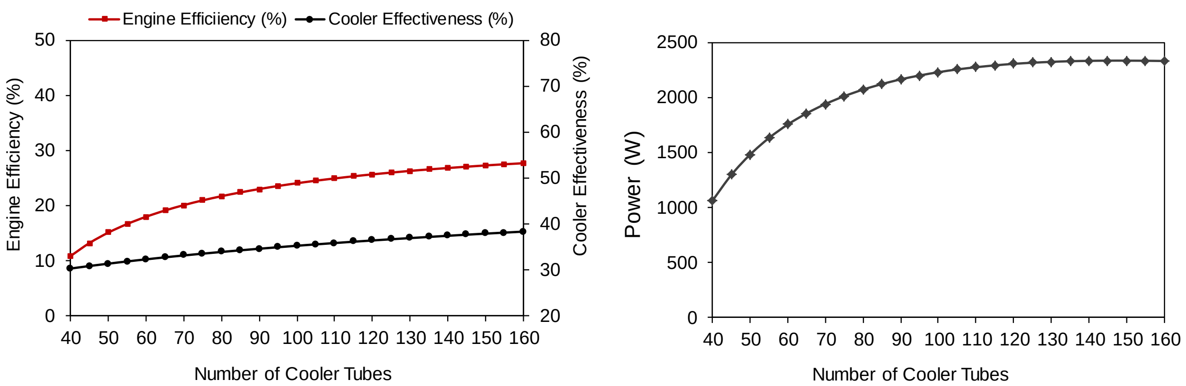

Figure 7: **a)** Engine efficiency, cooler effectiveness and **b)** Power production as a function of the number of cooler tubes.

Figure 8a) presents the engine efficiency and the cooler effectiveness variation for different values of the internal diameter of cooler tubes. The cooler effectiveness, similarly to the heater results, decreases with the rise in internal diameter value. Also, the engine efficiency reaches its maximum value for an internal diameter of 4.0 mm. The simulations demonstrated that for the lowest values of tube internal diameter, the pumping losses diminish until a certain minimum; after this value, the fluid friction effect outweighs the gains from the increase of heat transfer. It is also noted that increasing the internal diameter produces a reduction in power production (see Figure 8b)).

a) b)

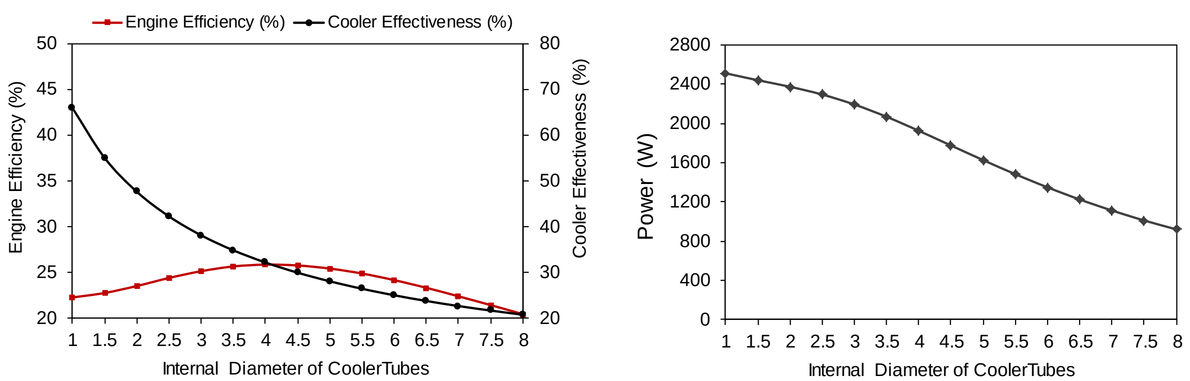

Figure 8: **a)** Engine efficiency, cooler effectiveness and **b)** Power production as a function of the internal diameter of the cooler tubes.

Figure 9a) presents the engine efficiency and the cooler effectiveness variation for different values of the cooler length. A range between 40 mm and 240 mm was considered to study the length influence in the Stirling efficiency. It is observed that the engine efficiency and the cooler effectiveness increases with cooler length, probably due to the gains in heat transfer with the increase of the contact area. Nevertheless, from 120 mm length, the produced power production starts to decrease. The maximum power is achieved at 120 mm of length, which corresponds to the value of 2244 W (see Figure 9b).

a) **b)**

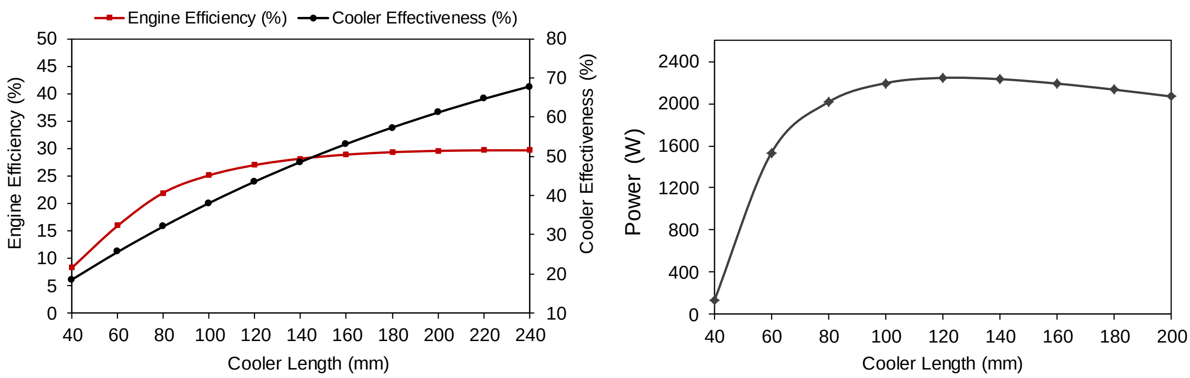

Figure 9: **a)** Engine efficiency, cooler effectiveness and **b)** Power production as a function of the cooler length.

From the results analysis, it can be concluded that the performance of the cooler can be improved increasing the internal diameter to 4.0 mm and raising its length in 20 mm (i.e. a cooler with 120 mm of length). Regarding the number of cooler tubes, it seems that 150 tubes is a reasonable value.

The study of each geometric parameter of the Stirling engine thermal components gives an insight into the relations of those parameters in the engine performance. Therefore, this knowledge can be used to produce significant improvement in the general performance of a particular engine. Regarding this, a comparison of the engine performance results between the base-case previously presented and the optimized configuration resulting from the analysis should be performed, as in Table 3.

Table 3: Engine performance results for the base-case and enhanced configuration

	Base-case Configuration	**Enhanced-case Configuration**
Engine Efficiency, %	17.3	30.7
Power, W	1915	1663
Pumping Losses, W	84	155
Heater Effectiveness, %	34.4	29.4
Regenerator Effectiveness, %	80.4	95.7
Cooler Effectiveness, %	25.2	23.4

Results suggest that the combination of new values for the geometric parameters turns into a higher thermal efficiency of the engine, which corresponds to a relative increase of 56.4% from the base-case. Nevertheless, this large increase in the thermal engine efficiency also translates into a reduction in power output, as well as, an increase in total pumping losses. Also, it should be noted the important increase in the regenerator effectiveness.

4 CONCLUSIONS

A sensitivity analysis to the geometrical parameters of the heat exchangers is presented in this study. The influence of geometrical parameters such as the internal diameter, the heat exchanger length, number of tubes, porosity and the diameter of the regenerator matrix wire in the performance the efficiency of the Stirling engine is then evaluated.

Results for the heater and cooler are quite similar. The engine the heat exchanger effectiveness and the efficiency increase with the number of tubes and with their length. The increase of the internal diameter results into the effectiveness reduction but, for both heat exchangers the maximum engine efficiency is achieved for a diameter of 4.0 mm. Results show that there is a geometrical limit for which the increases in heat transfer is overlapped by the void volume increase and by the increase in pumping losses. Regarding the regenerator, he porosity of the regenerator is an important parameter for engine performance because it affects the hydraulic diameter, dead volume, velocity of the gas, regenerator heat transfer surface and regenerator effectiveness. According to results, the best engine performance is obtained for lower wire diameters (0.1 mm) combined with a mesh porosity of 0.7. The combination of new values for the geometric parameters was then used to produce significant improvement in the general performance. Thus, the outcome resulted in a higher thermal efficiency of the engine, an increase from 17.3% to 30.7%, for a power production of 1663 W.

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