



HAL
open science

Experimental and theoretical investigation of Stirling engine heater: Parametrical optimization

Ramla Gheith, Houda Hachem, Fethi Aloui, Sassi Ben Nasrallah

► **To cite this version:**

Ramla Gheith, Houda Hachem, Fethi Aloui, Sassi Ben Nasrallah. Experimental and theoretical investigation of Stirling engine heater: Parametrical optimization. *Energy Conversion and Management*, 2015, 105, pp.285-293. 10.1016/j.enconman.2015.07.063 . hal-03448409

HAL Id: hal-03448409

<https://uphf.hal.science/hal-03448409v1>

Submitted on 16 Jul 2024

HAL is a multi-disciplinary open access archive for the deposit and dissemination of scientific research documents, whether they are published or not. The documents may come from teaching and research institutions in France or abroad, or from public or private research centers.

L'archive ouverte pluridisciplinaire **HAL**, est destinée au dépôt et à la diffusion de documents scientifiques de niveau recherche, publiés ou non, émanant des établissements d'enseignement et de recherche français ou étrangers, des laboratoires publics ou privés.

Experimental and theoretical investigation of Stirling engine heater: Parametrical optimization

R. Gheith ^{a,*}, H. Hachem ^a, F. Aloui ^b, S. Ben Nasrallah ^a

^a Université de Monastir, École Nationale d'Ingénieurs de Monastir, Laboratoire LESTE, Avenue Ibn El Jazzar 5019, Monastir, Tunisia

^b Université Lille Nord de France, UVHC, Département de Mécanique, LAMIH CNRS UMR 8201, Campus Mont Houy, 59313 Valenciennes Cedex 9, France

The aim of this work is to optimize γ Stirling engine performances with a special care given to the heater. This latter consists of 20 tubes in order to increase the exchange area between the working gas and the hot source. Different parameters were chosen to evaluate numerically and experimentally the heater. The selected four independent parameters are: heating temperature (300–500 °C), initial filling pressure (3–8 bar), cooling water flow rate (0.2–3 l/min) and frequency (2–7 Hz). The amount of energy exchanged in the heater is significantly influenced by the frequency and heating temperature but it is slightly enhanced with the increase in the cooling water flow rate. The thermal and the exergy efficiencies of the heater are very sensible to the temperature and pressure variations.

1. Introduction

The worldwide economy and heavy industries are based on fossil fuels specially coal, petroleum and natural gas. These energies are depleted and more over their costs are very influenced by the world political situation. In addition they release gaseous or liquid pollutants during operation. For example the coal plants provide 40% of our electricity, but it presents important losses from 6.4% to 64%, with an average of 31.25% at 815 °C [1]. Some installation techniques can be used to optimize the combustion of the coal. The thermal losses can be valorized by the use of cogeneration or combined heat and power (CHP) systems [2,3]. In this context the Stirling engines are presented as possible solution to convert thermal losses either into electricity or to produce hot water for domestic or industrial installations.

The Stirling engines technology are suitable for micro-cogeneration. These installations are promising for industrial or residential applications thanks to their high efficiency, their simplicity to connect to existing installation, and their low emissions with respect to alternative technologies [2]. A cogeneration unit that is principally composed of a Stirling engine, an alternator and a heat exchanger used to produce electricity can save 30% of fuel and moreover, if the unit is adapted to work at higher pressures and temperatures, power over 1.5 kW can be obtained in the alternator [2].

Cacabelos et al. [4] proposed a dynamic model of a commercial micro-combined heat and power unit and analyze its dynamic behavior when the engine is running at different mass flow inputs. The simulation predicts the most important output from the cogeneration unit during the starting, steady-state and stopping periods. The transit behavior of the Stirling engine cogeneration unit was studied by [3,5] they discretized the performance time into different running modes. They proposed thermodynamic models predicting both produced electricity and power. De Paepe et al. [6] studied 3 types of micro-CHP systems for residential uses. Two types of houses (detached and terraced) are compared with a two storey apartment. The different technologies are technically mature, installation costs should at least be reduced by 50% before CHP systems become interesting for residential use. Condensing gas boilers, now very popular in new homes, prove to be economically more interesting and also have a modest effect on primary energy consumption.

Heat exchangers are essential elements of cogeneration or CHP units. The phenomena occurring in Stirling heat exchangers are quite complex, due to the combination of a transient nature flow together with heat transfer and pressure drop. As the experimental studies are limited and expensive, in terms of facilities and manpower, and it is proportional to the number of data and the number of configurations tested, numerical analysis is an adequate alternative for the analysis of Stirling heat exchangers.

A Stirling engine includes three heat exchangers. The cooler improves the compression space heat transfer.

* Corresponding author.

E-mail address: ramla2gheith@yahoo.fr (R. Gheith).

Nomenclature

Variables

A	area (m ²)
T	temperature (°C)
V	volume (m ³)
L	length (m)
P	pressure (bar)
N	engine rotation speed (rpm)
S	stroke (m)
Q	energy (J)
W	work (J)
d	diameter (m)
\dot{m}	flow rate (kg s ⁻¹)
NTU	number of heat transfer units
f	friction factor
h	convective heat transfer coefficient (W m ⁻² K ⁻¹)
r	perfect gas constant (287.058 J kg ⁻¹ K ⁻¹)
Eff	net efficiency of the engine (%)
ρ	density (kg m ⁻³)

λ	thermal conductivity (W m ⁻¹ K ⁻¹)
μ	dynamic viscosity (kg m ⁻¹ s ⁻¹)

Indices

c	compression space
e	expansion space
k	cooler
h	heater
r	regenerator
$r1$	first section of regenerator
$r2$	second section of regenerator
v	viscous
$c-k$	interface between compression space and cooler
$k-r1$	interface between cooler and regenerator
$r1-r2$	interface between first and second section of regenerator
$r2-h$	interface between regenerator and heater
$h-e$	interface between heater and expansion space

The heater increases the exchange area between the working fluid and the heat source (solar, geothermal, burner, coal plants losses, etc.).

And the regenerator presents a thermal barrier between the hot source and the cold source and assigns a thermal sponge that absorbs thermal calorie or restores it to working fluid according to the flow direction [7]. The regenerator losses are proportionate to difference of the temperature between sources of heat [8].

Hachem et al. [9] showed that the maximum losses are recorded in regenerator. It is the set of 44% of viscous loss, 33% of internal conduction loss and 22% of imperfection loss respectively from the total losses inside it.

Hosseinzade and Sayyaadi [10] proposed a novel thermal model which considered effect of finite speed of piston, pressure throttling in heat exchangers and regenerator and piston's mechanical friction to determine the pressure drops in regenerator and heat exchangers and to propose a new correlation to estimate the regenerator's effectiveness based on the transfer unit (NTU) correction.

The installation of the heater, the regenerator, and the cooler is necessary in Stirling engine in order to maintain the temperature difference necessary to the operation of this kind of engine. These three components and their internal dead volumes are actually inevitable, and in this regard, a study of the optimal value of the total dead volume ratio becomes an important issue [11].

Some machines are equipped by tubular coolers and heaters (or inclined coils) for increasing the exchange area with the working fluid. El-Ehwany et al. [12] have proposed a new cooler and heater design; their designs are formed by a large number of tubes in parallel without deviation or bend. This new design can increase the power of their machine by 13%.

Scollo et al. [13] studied the possibility of a Stirling engine heater improvement. After construction and testing they remark that there was a deficit of heat exchange area in the outside of expansion exchanger. As solution, they propose to increase the exchange area.

As described by Cheng and Yang [14], the temperature ratio is a function of thermal characteristics of the heater and the cooler, effectiveness of the regenerator, mass of the working gas, and engine speed. Formosa et al. [15] estimated the Reynolds numbers fluid flow in the Stirling engine heat exchangers. They concluded that the variations of the Reynolds numbers for the heater and

the cooler are significant and the associated heat exchange coefficients are likely to be modified accordingly. Consequently, for a given external temperature conditions, the internal temperatures of the engine will change and so will do the operating variables.

The Stirling engine power output increases and the efficiency reaches a maximum which is about 40% when the working fluid mass is equal to 0.8 g [16]. When the mass increases, the efficiency reduction is due to the increase of the pressure loss and to the limitation of the heat exchange capacity in the regenerator and the heater [17]. Gang et al. [18] investigated the heat transfer characteristics of oscillating flows of a 36-tubes Stirling engine heater, whose operating conditions were close to the flow regions of real engines, and derived an empirical oscillating heat transfer correlation to predict oscillating flow heat transfer performance for promoting practical design of tubular heaters.

In this paper, the role of the heater will be underlined by proposing the influence of its parameters of the Stirling engine performances. Initially the quasi-steady model including the Thermodynamic phenomenon and Thermal mechanical losses will be presented. The final model will be applied to a Gamma type Stirling engine. A global characterization of the heater will be elaborated. Several operation parameters are selected (heating temperature, initial filling pressure, cooling water flow rate and frequency) and varied. The theoretical and the experimental influence of every parameter on the heater exchanged energy and efficiencies are presented and discussed.

2. The experimental facility

2.1. The Stirling engine set-up

The experimental set-up showed in Fig. 1 is composed of a Stirling engine linked to an external compressor giving the initial filling pressure and acquisition system recorded all data (temperature, pressure, flow rate, frequency, etc.).

The engine use compressed air as working fluid with a maximum charge pressure of 10 bar. The maximal speed frequency is about 10 Hz and the maximal output brake power about 500 W. In addition to the expansion and the compression spaces, the engine includes three heat exchangers: the regenerator, the cooler and the heater. The regenerator is a porous medium (i.e. made of

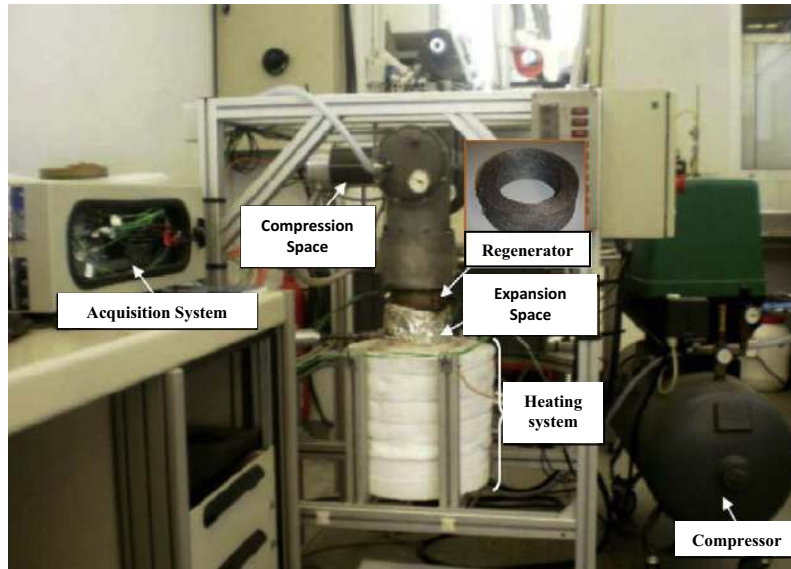


Fig. 1. The experimental set-up.

Stainless with 85% porosity). The cooler acts as a cold source formed by an open water circuit and the heater consists of 20 tubes in order to increase the exchange surface between the working gas and the hot source (electrical resistance). Pressure transducers were installed in the compression space and the area after the expansion space. A wide range of frequency responses was chosen to ensure good instantaneous measurements of the pressures inside the working spaces.

2.2. The heater

The heater (Fig. 2) is formed of 20 curved pipes of a 1 cm of internal diameter; each pipe has a length of 0.49 m. They allow increasing the exchange area between the working fluid inside the engine and the heat source. Three thermocouples are placed respectively inside the heater, outside the heat and in heating system. The heat source of the Stirling engine is formed by an electrical resistance surrounding the heater tubes. The system is insulated to minimize thermal losses (Fig. 3). The electrical resistance can be replaced by any source of heat (Fig. 4) (biomass or bio-carburant burner, thermal losses from coal plant, etc.).

2.3. Acquisition system

The thermocouples have $\pm 1.1\%$ of incertitude for the measured temperatures. The pressure transducer has 0.15% of incertitude



Fig. 3. Stirling engine heating system.

for measured pressure and the force transducer has a 0.25% N.C. (nominal charge) of incertitude. All sensors and thermocouples are recorded to fast acquisitions system (ADwin Pro-II). The ADwin Pro-II is an external processing system with modular expansion options. Our system includes eight modules. It has an

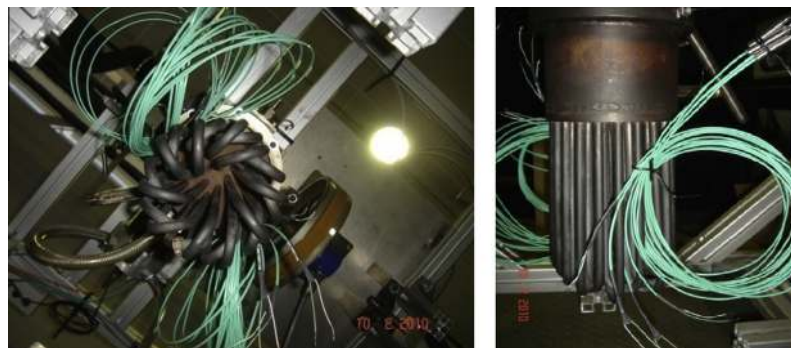


Fig. 2. The Gamma Stirling engine Heater.

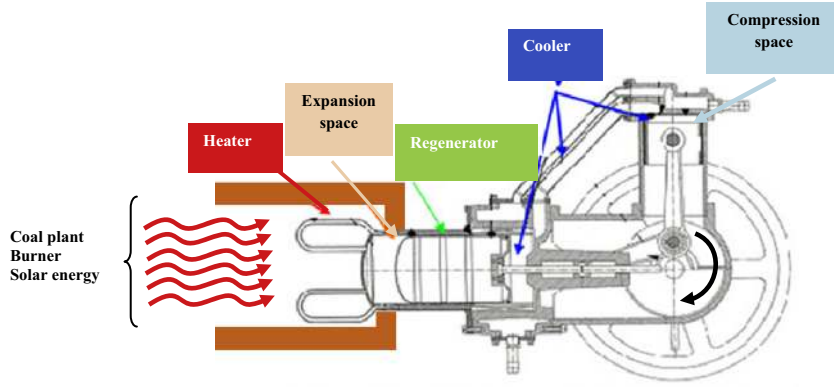


Fig. 4. Proposal of coupling the Stirling engine with a micro-cogeneration system.

internal memory of 768 KB and has an external trigger input that enables the processor module to recognize an external signal. A crank angle transducer is located in the end of the Stirling engine shaft. On every crankshaft degree, this crank angle transducer sends a transistor-transistor logic external signal to the processor module, which allows the acquisition of all signals by all used modules.

3. Theoretical evaluation of the heater by quasi-steady model

The quasi-steady model proposed by Berchowitz et al. [19] was adopted. The basic assumptions consider the working gas as an ideal gas with constant properties. The most important heat losses governing the Stirling engine, described below, are added to the classic quasi-steady model. The following diagram present losses recorded in each Stirling engine compartment (Fig. 5).

3.1. Losses due to pistons movement

3.1.1. Shuttle loss $\delta\dot{Q}_{st}$

This loss is due to the shuttle movement of the piston between the top dead center and the bottom dead center of the cylinder. The latter has a temperature of the expansion space higher than the temperature of the compression space. Therefore, the piston material is heated in the hot side and dissipates the heat in the cold side. In the case of a Gamma Stirling engine, both workspaces are separated in independent cylinders. So this loss must be calculated for each piston using Longsworth and Zimmermann formula [20]:

$$\delta\dot{Q}_{stc} = \frac{\pi S_c^2 \lambda_g d_c}{8x_c L_{pc}} (T_{c-k} - T_c) \quad (1)$$

$$\delta\dot{Q}_{ste} = \frac{\pi S_e^2 \lambda_g d_e}{8x_e L_{pe}} (T_e - T_{h-e}) \quad (2)$$

where T_{c-k} and T_{h-e} are respectively the temperature of the interface between compression cylinder and cooler, the expansion cylinder and heater, S (m) is the stroke of displacer or power piston, d (m)

is the internal diameter of the cylinder, L_p (m) is the length of displacer or power piston, x (m) is the annular gap between the piston and the cylinder and λ_g ($W m^{-1} K^{-1}$) is the gas thermal conductivity.

3.2. Heat exchangers losses

In order to know net power, we need to calculate losses in each heat exchanger (heater, cooler and regenerator).

3.2.1. Internal conduction loss $\delta\dot{Q}_{cd}$

An important gradient of temperature exist between Stirling engine interfaces leading to an internal conduction loss through regenerator, heater and cooler. This loss is calculated within the two sections of the heat exchangers as follows [20]:

$$\delta\dot{Q}_{cdh} = \frac{\lambda_m A_h (T_{h-e} - T_{r2-h})}{L_h} \quad (3)$$

$$\delta\dot{Q}_{cdr2} = \frac{\lambda_m A_{r2} (T_{r2-h} - T_{r1-r2})}{L_{r2}} \quad (4)$$

$$\delta\dot{Q}_{cdr1} = \frac{\lambda_m A_{r1} (T_{r1-r2} - T_{k-r1})}{L_{r1}} \quad (5)$$

$$\delta\dot{Q}_{cdk} = \frac{\lambda_m A_k (T_{k-r1} - T_{c-k})}{L_k} \quad (6)$$

where λ_m is the conductivity of the material through which heat conduction occurs ($W m^{-1} K^{-1}$), T_{k-r1} , T_{r1-r2} , T_{r2-h} , T_{h-e} and T_{c-k} are respectively interface temperature cooler-regenerator, regenerator-regenerator, regenerator-heater, heater-expansion space, compression space-cooler (K).

The regenerator thermal conductivity depends on the porosity ϕ , on the thermal conductivity of the working fluid λ_g and of the material λ_m . It can be calculated by the Gotting formula [21]:

$$\lambda_{r1,r2} = \lambda_g \frac{\lambda_g \phi + \lambda_m (2 - \phi)}{\lambda_g (2 - \phi) + \lambda_m \phi} \quad (7)$$

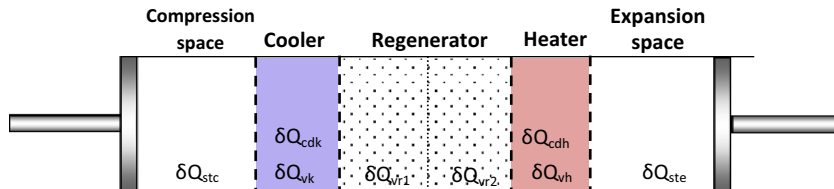


Fig. 5. Location of the various energy losses in a Stirling engine.

3.2.2. Loss due to imperfect heat exchanger $\varepsilon_{h,k,r}$

Heater, cooler and regenerator like any heat exchangers are not perfect. The average air temperatures T_h , T_k , T_r respectively inside heater, cooler and regenerator are lower than their wall temperatures T_{wh} , T_{wk} , T_{wr} . The efficiency of the hot end heat exchanger ε_k and the cold end heat exchanger ε_c are determined from the following equation [22]:

$$\varepsilon_{k,h} = 1 - \exp^{-NTU_{k,h}} = 1 - \exp^{-\frac{hA_{wg}}{C_p \dot{m}}} \quad (8)$$

It depends on the overall heat transfer coefficient h , on the wetted area of the heat exchanger surface A_{wg} .

The regenerator role is to store energy from the hot gas and restores to him during his passage in the opposite direction toward the cooler. The amount of heat stored or restored by the regenerator matrix depends on the characteristics of the material used and it is never equal. To estimate this loss, we have to calculate its efficiency ε_r as follow:

$$\varepsilon_r = \frac{NTU_r}{1 + NTU_r} \quad (9)$$

3.2.3. Friction loss in heat exchangers $\delta \dot{Q}_v$

The fluid friction associated with the flow through the heat exchangers will in fact result in a pressure drop Δp_{vr} , Δp_{vh} , Δp_{vk} across the three heat exchangers which greatly reduces efficiency and power output of the engine [23]. To evaluate pressure drop through heat exchangers, we use this correlation:

$$(\Delta p_v)_{r,h,k} = -\frac{2f\mu_g u_g V}{Ad_{hyd}^2} \quad (10)$$

where f is the viscous friction coefficient, μ is the dynamic viscosity of the working fluid ($\text{kg m}^{-1} \text{s}^{-1}$), u is the axial velocity (m s^{-1}), V is the volume of the heat exchanger (m^3), A is the fluid passage section through heat exchanger (m^2), d_{hyd} is the hydraulic diameter (m).

We quantified then energy loss by viscous friction respectively in heater, cooler and regenerator as follows:

$$(\delta \dot{Q}_v)_{h,k,r} = -\frac{\dot{m}_{h,k,r}(\Delta P_v)_{h,k,r}}{\rho_m} \quad (11)$$

The obtained model with all considered losses was applied to the gamma Stirling engine presented in Section 2.

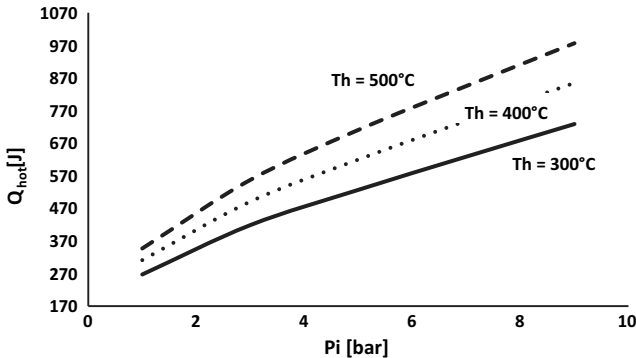


Fig. 6. Heater energy for one Stirling cycle vs. heating temperature and initial filling pressure.

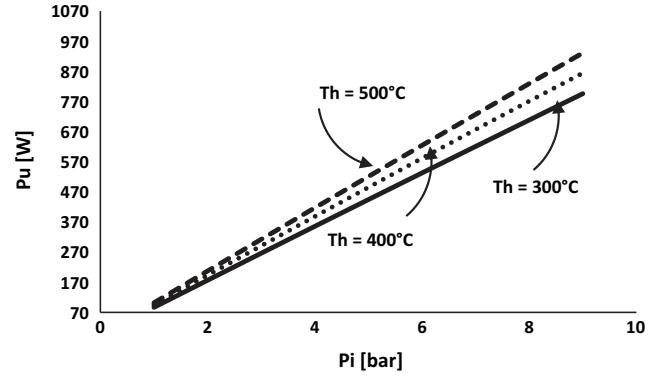


Fig. 7. Output brake power for one Stirling cycle vs. heating temperature and initial filling pressure.

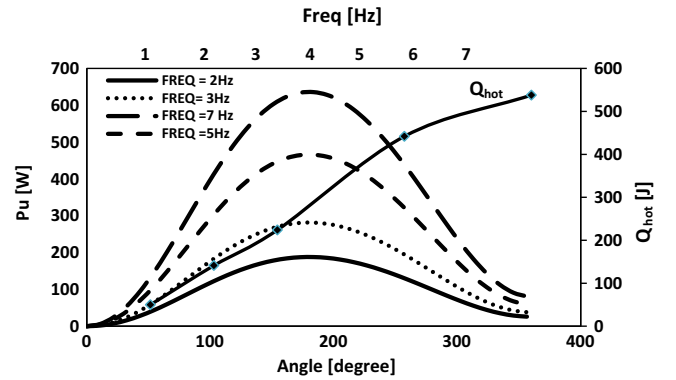


Fig. 8. Output brake power and heater energy for one Stirling cycle vs. engine frequency.

4. Parametric studies

4.1. Theoretical results

Figs. 6 and 7 present respectively the effect of heating temperature and initial filling pressure on the heater exchanged energy and engine output brake power. The augmentation of the hot end temperature brings additional thermal energy to the system so it increases the energy exchanged in the heater with the working fluid. The mass of working fluid involved in compression, expansion and heat exchange processes increases with the initial filling pressure. It is beneficial to increase these parameters to enhance the parameters of a Stirling engine. The engine frequency increases the amount of heat exchanged in the heater and the output brake power too (Fig. 8). For higher frequency the heater energy seems to become constant, the exchange process is so fast; the working fluid does not have enough time to absorb all the heat delivered by the hot end source.

The thermal efficiency of the heater represents the ability to reject the heat to the working gas when it passes through the heater. High heater efficiency is difficult to achieve experimentally. Efficiency near 70% is commonly obtained for Stirling engine heater. This is significantly lower than the efficiency of a heat exchanger used in boilers for example. The increase of the heater efficiency has a relevant influence on the amount of heat exchanged between the hot end source and the working fluid. The engine output brake power and the heater exchanged energy are presented in Figs. 9 and 10 functions of the heater efficiency and the engine frequency. An increase from 0.2 to 0.8 of the heater

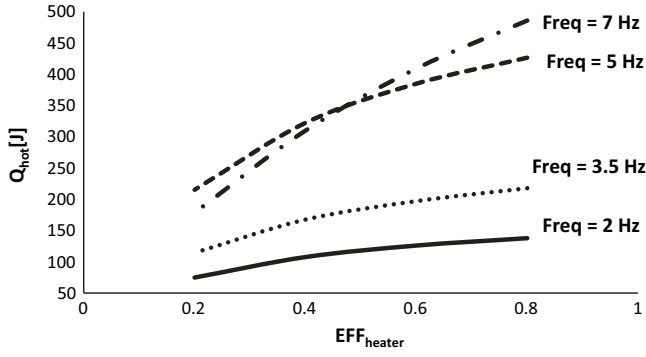


Fig. 9. Output brake power and heater energy for one Stirling cycle vs. heater efficiency.

efficiency increase the amount of heat absorbed by the working fluid in the heater by 2 times and 3 times for respectively an operation frequency of 2 Hz and 3.5 Hz. The output brake power rises from 100 W to a mean of 340 W for respectively 2 Hz and 7 Hz. The heater efficiency variation from 0.2 to 0.8 does not have a detectable influence on the engine brake power for frequency lower than 7%. For a frequency of 7 Hz, an increase of 8% of the brake power is observed for an increase of heater efficiency from 0.2 to 0.8.

To maximize the heat exchanger efficiency, a larger transfer area and thus larger volume are primordial. The transfer area will determine the amount of heat energy to reject or to supply to or from the engine. The influence of the heater volume on the engine output brake power and on the heater exchanged heat are presented in Fig. 11. The increase of the heater volumes has contradictory effects on the engine performances. It increases the area of exchange between the working fluid and the hot end source but it increases the total dead volumes of the Stirling engine and the friction losses. As demonstrated by Tlili [22] the increase in the dead volume produces an exponential drop in the net power, which in turn decreases the maximum pressure. The dead volume increases deteriorate the engine performances (Fig. 13).

4.2. Experimental evaluation of the heater

4.2.1. Thermal and exergy efficiencies of a Stirling engine heater

The thermal efficiency can be calculated as the output work devised by the amount of heat received by the working gas. Therefore, it is given as:

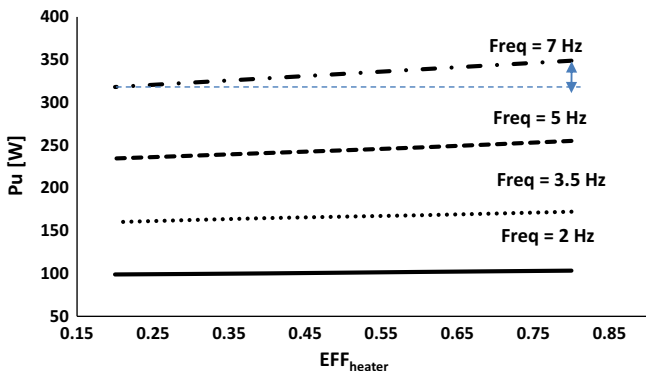


Fig. 10. Output brake power for one Stirling cycle vs. heater efficiency and engine frequency.

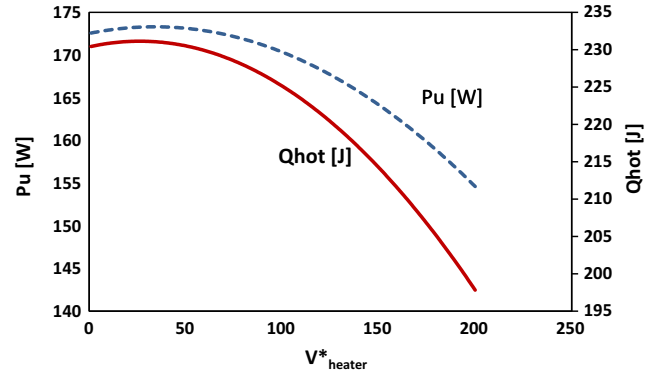


Fig. 11. Output brake power and heater energy for one Stirling cycle vs. heater volume.

$$\eta_{th} = \frac{W_{out}}{Q_H} \quad (12)$$

The Carnot efficiency is the maximum possible (reversible) thermal efficiency of the engine. It is given by:

$$\eta_c = 1 - \frac{T_C}{T_H} \quad (13)$$

The exergy efficiency, called the second law efficiency, is defined as the ratio of the actual thermal efficiency to the maximum possible (reversible) thermal efficiency under the same conditions:

$$\eta_{Ex} = \frac{\eta_{th}}{\eta_c} \quad (14)$$

It can also be expressed as the ratio of the exergy recovered (as useful input) and the exergy supplied (as total input):

$$\eta_{Ex} = \frac{\text{Energy recovered}}{\text{Energy supplied}} = 1 - \frac{\text{Energy destroyed}}{\text{Energy supplied}} \quad (15)$$

For a heat engine, the exergy supplied is the decrease in the exergy of the heat transferred to the engine, which is the difference between the exergy of the heat supplied and the exergy of the heat rejected. The net work output is the recovered exergy. Finally, the exergy efficiency is calculated as follows:

$$\eta_{Ex} = \frac{W_{out}}{Ex_{Q_H}} \quad (16)$$

The exergy efficiency η_{Ex} frequently gives a finer understanding of performance than the energy efficiency η_{th} . In evaluating efficiency, the same weight is assigned to energy whether it is shaft work or a stream of low-temperature fluid. The parameter η_{Ex} points out that there is a need to deal with all external and internal irreversibilities to improve the machine performance [24].

Finally, the overall efficiency of the Stirling heat engine is calculated as follows:

$$\begin{aligned} \eta_g &= \frac{\text{Output Work}}{\text{Electric Energy supplied}} = \frac{-W}{Q_{in}} = -\frac{W}{Q_H} \cdot \frac{Q_H}{Q_{in}} \\ &= \eta_{th} \cdot \eta_{cal} \end{aligned} \quad (17)$$

where η_{cal} is the calorific efficiencies of the hot heat exchanger.

4.2.2. Global study of the heater

The heating temperature is maintained at 500 °C for an initial filling pressure of 5 bar. The temperature in the heater follows a periodic evolution (Fig. 12). During the first phase, the working fluid preheated by the regenerator pass through the heater. This mass of working fluid is mixed to the mass of the working fluid

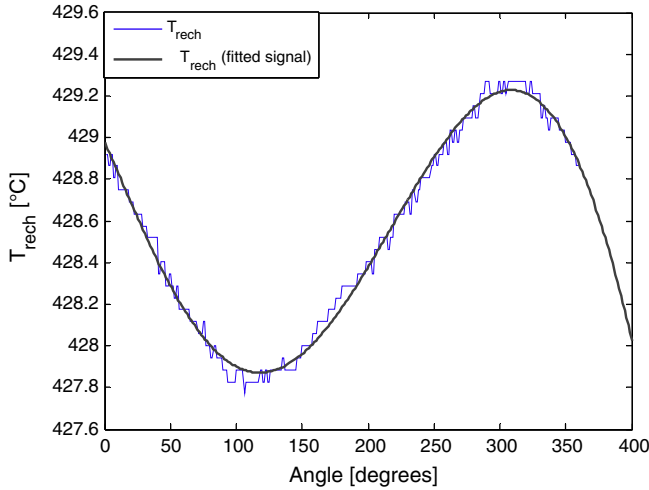


Fig. 12. Temperature evolution of working fluid inside the heater ($T_{\text{heat}} 500\text{ }^{\circ}\text{C}$).

initially presented in the heater. The mixture temperature decreases initially then increases by the constant flow of heat given by the hot source.

The heater efficiency is very high for this kind of heater (tubular) it is higher than 70% [25]. The heater efficiency is not constant during one Stirling cycle (Fig. 13). Its evolutions are similar to the temperature ones. The heater efficiency is linked to the temperature of the working fluid passing through the heater. In flame or hot gas heated Stirling engines, the heat transfer from hot gas to heater of the engine is one of the most important reasons of low efficiency [26]. It is beneficial to improve the efficiency of the heater to enhance the whole engine efficiency.

For a heating temperature of $500\text{ }^{\circ}\text{C}$, an initial filling pressure of 8 bar and a cooling water flow rate of 8 l/min, the global heating energies exchanged in each heat exchanger of the Gamma type Stirling engine are calculated and presented. All recorded energies present a periodic evolution (Fig. 14). The maximal energy is exchanged through the regenerator. This latter is filled with a stainless steel porous material which economizes an important amount of heat. The performances of a Stirling engine are closely linked to its regenerator properties [17]. However, it is the seat of important internal and external loss by conduction and pressure drops [23] (caused by friction of the working fluid with the porous medium). These losses are a function of the geometrical and physical

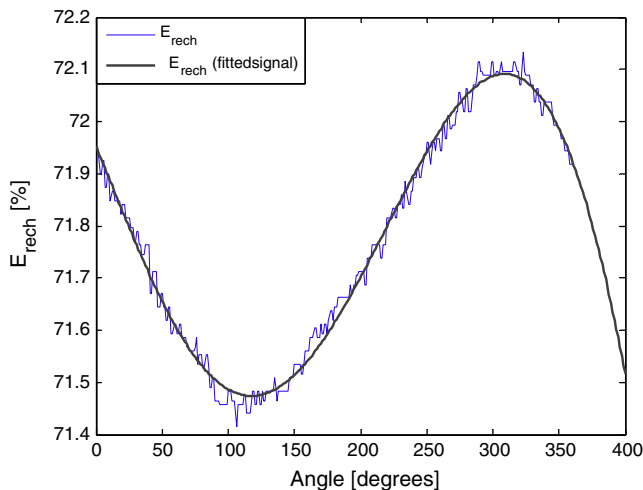


Fig. 13. Heater efficiency evolution for one Stirling cycle.

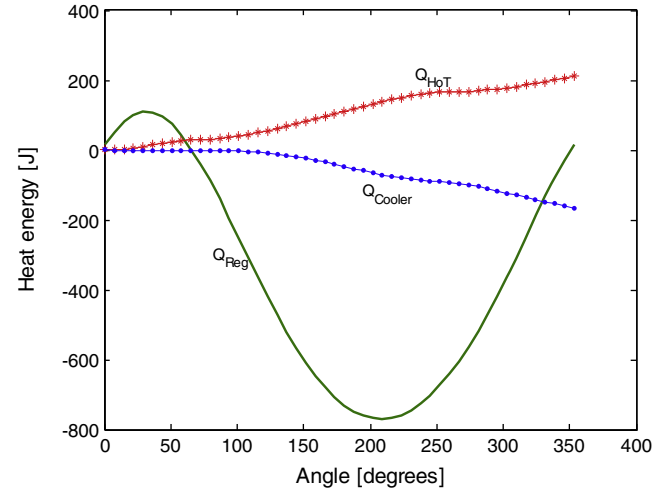


Fig. 14. Evolution of the thermal energy exchanged in all heat exchanger for one Stirling cycle.

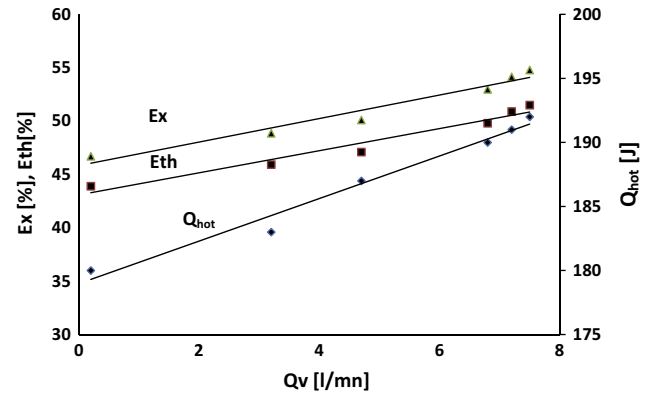


Fig. 15. Evolution of exergy and thermal efficiencies and heater energy vs. cooling water flowrate ($T_h = 400\text{ }^{\circ}\text{C}$, $P_i = 5\text{ bar}$).

properties of the material constituting the regenerator [17]. For an ideal Stirling cycle, the cyclic energy flow through the regenerator sums to zero. It can be seen in Fig. 14 that the heat energy recorded in the regenerator is of the order of four times more than that of the heater and five times more than that of the cooler. The heat transfer capacity of the regenerator is higher respectively four and five times than that of the heater and the cooler. This statement is in accordance with Reader Ref. [20] which stated that the adequate regenerator for a Stirling engine must be able to deal with four to five times the heat load of the heater.

For the same operation conditions theoretical total heat energy exchanged in the Gamma Stirling engine is about 300 J for every Stirling cycle. The experimental value is about 210 for every Stirling cycle. The difference can be explained by the addition of thermal losses which are not considered in the theoretical model. The simplifying assumptions adopted in the model consider conditional temperatures at the engine interfaces and variable mass flow and imply that at any given moment the pressure is constant throughout the engine.

4.2.3. Parametric optimization of the heater

The optimization of the heat exchanged in a Stirling engine heater can be made by proposing new design (number, diameter and average length of the tubes, etc.) or getting optimal functioning parameters (fuel mass flow rate, heating temperature, initial filling

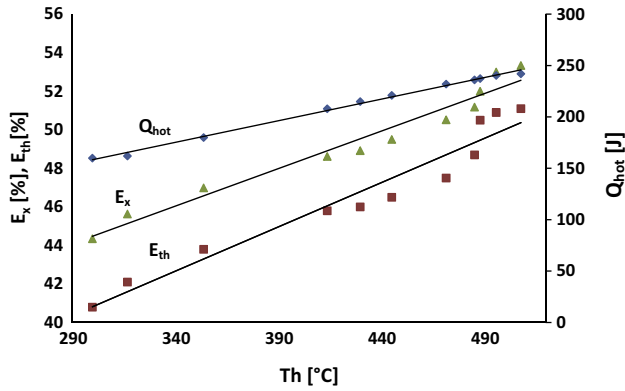


Fig. 16. Evolution of exergy and thermal efficiencies and heater energy vs. heating temperature ($Q_v = 8$ l/min, $P_i = 5$ bar).

temperature, engine frequency, etc.). For both solutions the goal is to maximize the output brake power and the heater efficiency and to minimize the thermal losses.

Fig. 15 shows that the energy exchanged in the heater increase with the cooling water flow rates (evacuate the maximum amount of heat generated in the compression space). Increasing the flow rate of cooling water decreases the temperature of the compression space and consequently the difference of temperature between both working spaces. The flow rate augmentation has a slight influence on the heat transfer recorded in the heater. Only an increase of 6% of Q_{hot} is recorded. Thermodynamic imperfections can be quantified as exergy destructions, which represent losses in energy quality or usefulness [24]. The exergy efficiency is presented to judge the ideality of a system. The heater exergy efficiency increase with the cooling water flow rates without reaching a maximal value (Fig. 15). The heater thermal efficiency presents the same evolution but it is lower than the exergy efficiency; the different is due to anergy (energy destruction) which characterize how internal and external irreversibilities reduce the performance of the Stirling heat engine.

The increase of the hot end temperature will bring additional amount of heat to the engine. In fact, the resistibility of the engine against the external load is enhanced when the heating temperature increases [26]. Consequently the amount of heat giving to the working fluid inside the heater will increase and the heater

efficiency will increase too (Fig. 16). The temperature of the working fluid increase too. So the difference of temperature between both working spaces increases and enhances the performances of the Stirling engine. The increase of the hot end temperature is limited by the heat capacity of the working fluid. An increase of the hot end temperature from 300 °C to 500 °C increases the Q_{hot} respectively from 160 J to 240 J. The thermal efficiency increased rather linearly with the increased temperature of the heater section and reached 48% at the maximum temperature of 500 °C.

For P_i lower than 6 bar (Fig. 17A) the amount of heat absorbed by working fluid enhance the heater efficiencies, increases the operating pressure and decrease irreversibilities inside the regenerator which enhances the Stirling cycle efficiency [27]. For P_i higher than 6 bar, the amount of heat energy added by the hot end temperature is not sufficient to heat the mass of the working fluid participating to the compression and expansion processes. It causes the decrease of the amount of energy exchanged in the engine and decreases the engine performances. The increase of initial filling pressure increases the pressure drops, so it deteriorates the engine performances. At high values of the charge pressure the thermal efficiencies are low [28]. An optimal filling pressure appears as 6, 7 and 7.5 for respectively 300 °C, 400 °C and 500 °C. Fig. 17B. illustrates thermal and exergy efficiencies–filling pressure profiles for different heating temperature. At optimum charge pressures corresponds optimal thermal and exergy efficiencies.

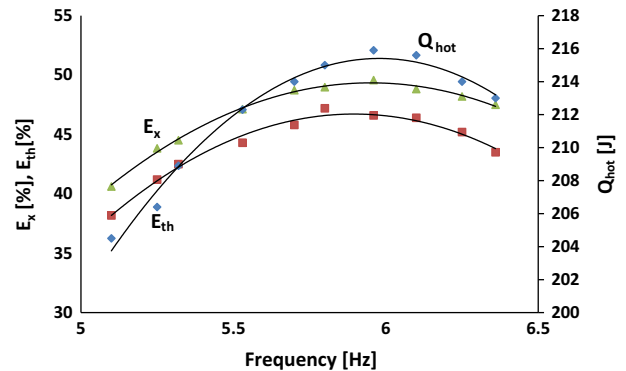


Fig. 18. Evolution of heater energy and exergy and thermal efficiencies vs. engine frequency.

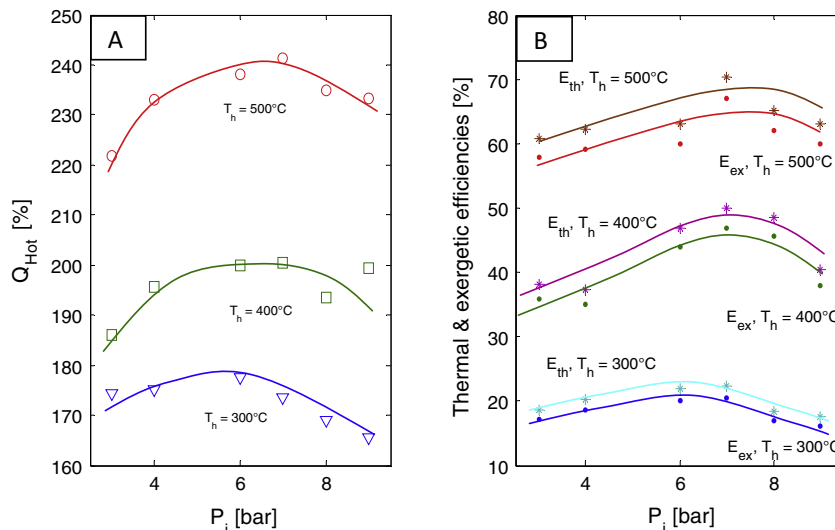


Fig. 17. Heater thermal and exergy efficiencies and energy vs. cooling initial filling pressure ($Q_v = 8$ l/min).

The evolution of heater energy, thermal and exergy efficiencies is devised into two parts (Fig. 18). When frequency is lower than 5.8 Hz, the efficiencies decrease. Lower engine frequency may increase the leakage of the used working fluid (air) and therefore decline the produced brake power [27]. Lower frequency increase the duration heat exchange and therefore to decrease of the irreversibilities caused by the thermal losses. Consequently, the engine brake power increases with the amount of heat exchanged in the heater [27]. Beyond a critical frequency value the exchange process in the engine proceeds very rapidly, the residence time of the working fluid in the heater is cut short. The working fluid does not have time to absorb all the amount of energy delivered by the hot end source. Heat exchanges are deteriorated thereby causing decrease of the heater thermal efficiency.

5. Conclusion

A γ type Stirling engine heater was numerically and experimentally studied. The role of the heater to enhance the Stirling engine performances was highlighted in this article. The classical quasi-study model was ameliorated by adding thermal losses generated in the different compartment of a Stirling engine. The heater exchanged energy increases with the hot end temperature, the initial filling pressure and the heater thermal efficiency without reaching maximal values. It increases too with engine frequency and the heater volumes but until critical values beyond it decreases.

The exchanged heat and the efficiency in the heater present periodic evolution. All investigated parameters are significant for the heater performances. The amount of heat absorbed by the working fluid in the heater increases with:

- (i) the heating temperature,
- (ii) the difference of temperature between both working spaces,
- (iii) the cooling water flow rate,
- (iv) the mass for working fluid involved to the process of heat exchange in the heater increases with the initial filling pressure.

A preliminary study must be established to determine the optimal operating values to provide the highest performance of the heater and therefore of the Stirling engine. In our case a frequency of 6.1 Hz, an initial filling pressure of 7 bar, a heating temperature of 500 °C and a maximum cooling water flow rate offer the maximal heater performances.

Acknowledgments

This work was supported by the laboratory TEMPO (University of Valenciennes, France), the laboratory LESTE (ENIM, Monastir, Tunisia) and the European Commission within the International Research Staff Exchange Scheme (IRSES) in the 7th Framework Programme FP7/2014-2017/ under REA grant agreement n°612230. These supports are gratefully acknowledged.

References

[1] Chen J, Liu G, Kang Y, Wu B, Sun R, Zhou C, et al. Atmospheric emissions of F, As, Se, Hg, and Sb from coal-fired power and heat generation in China. *Chemosphere* 2013;90:1925–32.

[2] Valenti G, Silva P, Fergnani N, Di Marcoberardino G, Campanari S, Macchi E. Experimental and numerical study of a micro-cogeneration Stirling engine for residential applications. *Energy Procedia* 2014;45:1235–44.

[3] Thiers S, Aoun B, Peuportier B. Experimental characterization, modeling and simulation of a wood pellet micro-combined heat and power unit used as a heat source for a residential building. *Energy Build* 2010;42(6):896–903.

[4] Cacabelos A, Eguía P, Luís Míguez J, Rey G, Elena Arce M. Development of an improved dynamic model of a Stirling engine and a performance analysis of a cogeneration plant. *Appl Therm Eng* 2014;73:606–19.

[5] Conroy G, Duffy A, Ayompe L. Validated dynamic energy model for a Stirling engine m-CHP unit using field trial data from a domestic dwelling. *Energy Build* 2013;62:18–26.

[6] De Paepe M, D'Herdt P, Mertens D. Micro-CHP systems for residential applications. *Energy Convers Manage* 2006;47:3435–46.

[7] Weiqing X, Jia W, Maolin C, Yan S. Liquid air fueled open–closed cycle Stirling engine. *Energy Convers Manage* 2015;94:210–20.

[8] Ahmadi MH, Ahmadi MA, Bayat R, Ashouri M, Feidt M. Thermo-economic optimization of Stirling heat pump by using non-dominated sorting genetic algorithm. *Energy Convers Manage* 2015;91:315–22.

[9] Hachem H, Gheith R, Aloui F, Ben Nasrallah S. Numerical characterization of a c-Stirling engine considering losses and interaction between functioning parameters. *Energy Convers Manage* 2015;96:532–43.

[10] Hosseinzade H, Sayyaadi H. CAFS: the combined adiabatic–finite speed thermal model for simulation and optimization of Stirling engines. *Energy Convers Manage* 2015;91:32–53.

[11] Cheng C-H, Yang H-S. Optimization of rhombic drive mechanism used in beta-type Stirling engine based on dimensionless analysis. *Energy* 2014;64:970–8.

[12] El-Ehwany A-A, Hennes G-M, Eid E-I, El-Kenany E-A. Development of the performance of an alpha-type heat engine by using elbow-bend transposed-fluids heat exchanger as a heater and a cooler. *Energy Convers Manage* 2011;52:1010–9.

[13] Scollo L-S, Valdez P-E, Santamarina S-R, Chini M-R, Baron J-H. Twin cylinder alpha stirling engine combined model and prototype redesign. *Int J Hydrogen Energy* 2013;38:1988–96.

[14] Cheng C-H, Yu Y-J. Dynamic simulation of a beta-type Stirling engine with cam-drive mechanism via the combination of the thermodynamic and dynamic models. *Renewable Energy* 2011;36:715–25.

[15] Formosa F, Badel A, Lottin J. Equivalent electrical network model approach applied to a double acting low temperature differential Stirling engine. *Energy Convers Manage* 2014;78:753–64.

[16] Tlili I. Thermodynamic study on optimal solar stirling engine cycle taking into account the irreversibilities effects. *Energy Procedia* 2012;14:584–91.

[17] Gheith R, Aloui F, Ben Nasrallah S. Study of the regenerator constituting material influence on a Gamma type Stirling engine. *J Mech Sci Technol* 2012;26(4):1251–5.

[18] Xiao Gang, Chen Conghui, Shi Bingwei, Cen Kefa, Ni Mingjiang. Experimental study on heat transfer of oscillating flow of a tubular Stirling engine heater. *Int J Heat Mass Transf* 2014;71:1–7.

[19] Berchowitz D-M, Urieli I. *Stirling Cycle Engine Analysis*. Bristol: Adam Hilger Ltd; 1984.

[20] Reader G-T, Hooper C. *Stirling engines*. London & New York: E&FN Spon; 1983.

[21] Stouffs P. Dimensionnement Optimal des Volumes de Compression et de Détente des Moteurs Stirling. *Congrès français de thermique SFT* 2000;8:851–6.

[22] Tlili I. Finite time thermodynamic evaluation of endoreversible Stirling heat engine at maximum power conditions. *Renew Sustain Energy Rev* 2012;16:2234–41.

[23] Tlili I, Timoumi Y, Ben Nasrallah S. Analysis and design consideration of mean temperature differential Stirling engine for solar application. *Renewable Energy* 2008;33:1911–21.

[24] Dincer I, Rosen M-A. *Exergy, energy environment and sustainable development*. Elsevier; 2007.

[25] Hazami M, Kerkeni C, Farhat A, Kooli S, Ben Jemaa F, Maalej M. Etude Expérimentale d'un Echangeur Thermique en Polypropylène, Application au Conditionnement des Aquariums. *Rev Energy Ren: J de Therm* 2001:103–8.

[26] Sripakagorn A, Srikam C. Design and performance of a moderate temperature difference Stirling engine. *Renewable Energy* 2011;36:1728–33.

[27] Gheith R, Aloui F, Ben Nasrallah S. Experimental investigations of a gamma Stirling engine. *Int J Energy Res* 2012;36:1175–82.

[28] Solmaz H, Karabulut H. Performance comparison of a novel configuration of beta-type Stirling engines with rhombic drive engine. *Energy Convers Manage* 2014;78:627–33.