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NEW METHODOLOGY FOR PERFORMANCE COMPUTATION OF SOLAR STIRLING ENGINES DEVELOPED IN THE FRAMEWORK OF THERMODYNAMICS WITH FINITE SPEED AND THE DIRECT METHOD

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A new methodology for computing performance of solar Stirling engines based on achievements of Irreversible Thermodynamics with Finite Speed and the Direct Method is presented. It allows a detailed analysis of thermal and mechanical irreversibilities of Stirling engine cycle. The new concept of isometric irreversible process occurring in the regenerator of Stirling Machines is introduced. Throttling and friction pressure losses are modelled. The computation methodology was validated based on comparison with experimental data (power and efficiency) for 10 Stirling Engines. Prediction errors of 1-3 % allow to consider it extremely accurate, thus very useful in the design and optimization of new Stirling Machines.

Keywords: Stirling Engine, Isometric Irreversible Process, Pressure losses, Friction and throttling, Irreversible Thermodynamics with Finite Speed.

1. Introduction

The previous research on modeling real operation and performance estimation of Stirling engines was focused mainly on the study of cycle's losses due to irreversibilities. Within the framework of Thermodynamics with Finite Speed, a methodology for the computation and optimization of Stirling engines was developed and validated [1-9], whose results were compared with experimental data of 12 operational engines [10-12]. This methodology, based on the expression of the First Law of Thermodynamics for Processes with Finite Speed, takes into account the internal and external irreversibilities of the Stirling engine cycle. In the mathematical formulation of the First Law for complex

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systems (the case of a Stirling engine, due to the presence of the regenerator) within Thermodynamics with Finite Speed appear three causes of internal irreversibility: (1) the finite speed of the piston, during inside machine developed processes - the term *aw/c*, (2) mechanical and gas dynamic friction - term $\Delta P_f / P_{m,i}$, and (3) throttling - term $\Delta P_{thr} / P_{m,i}$. All these losses were included in the equation of the machine efficiency. Then, it was further used to express the machine's power together with the heat received by the gas in the cycle [13-15]. The analytical quantification of the pressure losses as function on the speed of the process was essential to validate the computation methodology of the performance indicators (efficiency and power) using the Direct Method on 12 of the most advanced Stirling engines (and 16 operating modes) in [16], and to subsequently validate the computation methodology on the Sunmachine Stirling engine (Germany) [17, 18] and on the WisperGen Stirling engine (New Zealand) [1, 19].

The computation methodology takes into account the imperfect regeneration of heat (in the regenerator), and the limited heat transfer from the hot source related to the machine's capacity of taking over this heat. Each of these losses led to the inclusion in the computation methodology of an adjustment coefficient, which served for validation. It is remarkable that only two adjustment coefficients were necessary to validate the use of the Direct Method for Stirling engines.

In this paper, the modeling of the Stirling engine operation was extended by taking into account losses on the two isometric transformations of the cycle and highlighting the effect of each type of loss on the produced work. This new computation methodology for the performance indicators of a Stirling engine, was developed within the framework of Irreversible Thermodynamics with Finite Speed by using the Direct Method [15, 20].

By incorporating the effects of irreversibilities during gas flow through the regenerator (heating and cooling processes) as novelty elements, into the study of the irreversible cycle, led to the completion of the new computational methodology that takes into account isometric processes based on the Direct Method and Thermodynamics with Finite Speed.

This new methodology for computation and optimization of Stirling engines was applied for validation as a first step to two Stirling engines of the best performance, and then for 8 types of Stirling engines - classical and solar alike - currently in operation around the world (USA, Germany, Spain etc.).

2. First Law of Thermodynamics for processes with Finite Speed in Complex Systems

Stirling engines have special features because they include two isometric processes, namely processes 2-3 and 4-1 (Fig. 1). The difficulty of understanding the Stirling cycles lies in the inability of the classic *PV* diagram to explain the way in which the isometric processes are carried out. This led to the necessity of conceiving new diagrams, namely *PV / Px* and *PV / Px / Tx*, which are concretized in the works [1, 3]. On their basis, it was possible to explain, illustrate and highlight irreversibilities on finite speed isometric process, including the case of Stirling engines.

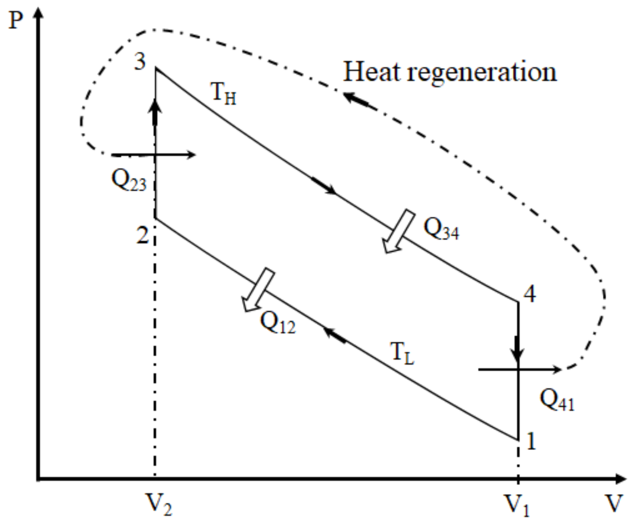

Fig. 1. The Stirling engine cycle in the PV diagram

By cumulating the effects of various internal irreversibility causes for a compression or an expansion process with finite speed, among which it is mentioned: (1) the finite speed of the piston, (2) the throttling of the gas, (3) the gas flow with friction through the heat exchangers of the machine, and (4) the friction between piston and cylinder, the general equation of the First Law of Thermodynamics for Processes with Finite Speed for closed complex systems was obtained by S. Petrescu, G. Stanescu, R. Iordache, A. Dobrovicescu [22-25]:

$$dU = \delta Q - P_{m,i}\left(1 \pm \frac{aw}{c} \pm \frac{b\Delta P_{thr}}{2P_{m,i}} \pm \frac{f\,\Delta P_f}{P_{m,i}}\right)dV\,, \quad (1)$$

where:
w – piston speed [m/s];
a – coefficient, depending on the gas nature, $a = \sqrt{3\gamma}$;
b – coefficient related to the gas throttling, 0 < b < 2;
c – average molecular speed, $c = \sqrt{3RT}$ [m/s];

γ- ratio of specific heats at constant pressure and volume, $\gamma = {c_p}/{c_v}$;

ΔP_{thr} - pressure drop caused by throttling [Pa];

f – fraction of the heat generated by friction between the machine moving parts ($0 < f < 1$) that remains in the system;

$P_{m,i}$ - instantaneous mean pressure of the gas [Pa];

ΔP_f - pressure drop due to friction between the moving parts of the machine [Pa].

The equation for the irreversible mechanical work of the First Law of Thermodynamics for Processes with Finite Speed is thus obtained:

$$\delta W_{irrev} = P_{m,i}\left(1 \pm \frac{aw}{c} \pm \frac{b\Delta P_{thr}}{2P_{m,i}} \pm \frac{\Delta P_f}{P_{m,i}}\right)dV = P_p dV \ . \tag{2}$$

where P_p – pressure on the piston [Pa].

In both equations, the + and – signs inside the parenthesis correspond to compression and expansion process respectively.

2.1. Evaluation of the losses due to the finite piston speed, throttling and friction

The pressure losses due to the finite speed of the piston will be computed for compression and expansion processes in the following way:

$$\Delta P_w = \frac{a \cdot w}{c} \ . \tag{3}$$

These losses are caused by the pressure non-uniformity in the cylinder due to the movement of the piston with finite speed.

For the throttling pressure losses, ΔP_{thr}, in the regenerator, a formula was found in the literature [5-8] expressing it as function on the piston speed, w, and the *number of screens, N_s*:

$$\Delta P_{thr} = \frac{15}{\gamma} \cdot \frac{\rho_{gR} \cdot w_g^2}{2} \cdot N_s \ , \tag{4}$$

where: γ - adiabatic exponent; ρ_{gR} – density of the gas in the regenerator; w_g – gas speed in regenerator.

For each part of the cycle it was considered that throttling has a 20% contribution divided between the compression (1-2) and expansion (3-4) processes and on the isometric heating and cooling processes, throttling will have a contribution of 80% divided between them.

This formula is based on a figure from Organ's book [11] where a lot of empirical measurements of ΔP_{thr} have been synthesized as dependent on the Reynolds number (*Re*) of the gas passing through the regenerator screens [1-3, 13, 21].

Another important step was the transformation of the Heywood's empirical formulas [12] which may be written as function of the piston speed, *w*, and stroke, *z*, as follows [5-12]:

$$\Delta P_f = 0.97 + 0.45 \cdot \left(\frac{3w}{100 \cdot z} \right). \tag{5}$$

The above formula's (based on experimental measurements on Internal Combustion Engines - ICE) use for Stirling Engines is justified by the tendency to design and build (technologically) the Stirling Engines in a similar manner as ICE (with similar parts and similar technologies).

In order to compare and highlight the effect of each pressure losses caused by the irreversibilities it was introduced non-dimensional coefficients for each type of losses on the cycle processes. For example, the coefficients of loss due to the finite speed of the piston at compression and expansion are calculated as follows:

$$B_{cpr,w} = \frac{\Delta P_{cpr,w}}{P_{med12}} \;,\; B_{exp,w} = \frac{\Delta P_{exp,w}}{P_{med34}} \,. \tag{6}$$

The loss coefficients for each irreversibility in the isometric cooling process are:

$$B_{C,f} = \frac{\Delta P_{C,f}}{P_{C,41}} \;,\; B_{C,thr} = \frac{\Delta P_{C,thr}}{P_{C,41}} \;,\; B_{C,w} = \frac{a \cdot w}{c'} \,. \tag{7}$$

Also, the loss coefficients due to throttling, friction and the finite speed of the piston will be calculated:

$$B_{H,thr} = \frac{\Delta P_{H,thr}}{P_{H,23}} \;,\; B_{H,f} = \frac{\Delta P_{H,f}}{P_{H,23}} \;,\; B_{H,w} = \frac{a \cdot w}{c''} \,. \tag{8}$$

The medium molecular speed in the isometric cooling (*c'*) and heating (*c"*) process is determined as:

$$c' = \sqrt{3 \cdot R \cdot T_{41}} \;,\; c'' = \sqrt{3 \cdot R \cdot T_{23}} \,. \tag{9}$$

The two expression in eq. (9) will be equal since the medium temperatures in the isometric cooling and heating processes of the cycle are the same.

Also, the pressure drops due to friction in the cooling and heating isometric processes are equal:

$$\Delta P_{H,f} = \Delta P_{C,f} \,. \tag{10}$$

2.2. Evaluation of the total mechanical work, power and efficiency with the losses due to the finite piston speed, throttling and friction

Firstly, the reversible mechanical work is calculated in the isothermal processes 1-2 and 3-4:

$$W_{12,rev} = P_1 \cdot V_1 \cdot \ln\left(\frac{V_2}{V_1}\right),\ W_{34,rev} = P_3 \cdot V_3 \cdot \ln\left(\frac{V_4}{V_3}\right), \tag{11}$$

The total non-dimensional coefficients for the isothermal processes 1-2 / 3-4, will be computed, with each type of loss taken into account successively:

$$\begin{aligned} B_{cpr,total,w,thr,f} &= 1 + B_{cpr,w} + B_{cpr,thr} + B_{cpr,f} \\ B_{exp,total,w,thr,f} &= 1 - B_{exp,w} - B_{exp,thr} - B_{exp,f} \end{aligned} \tag{12}$$

Based on the reversible mechanical work in isothermal processes 1-2 / 3-4 and the total loss coefficients on compression and expansion, we get the mechanical work in the isothermal process taking into account the losses caused by the finite speed of the piston, throttling and friction:

$$\begin{aligned} W_{12,irev,w,thr,f} &= W_{12,rev} \cdot B_{cpr,total,w,thr,f} \\ W_{34,irev,w,thr,f} &= W_{34,rev} \cdot B_{exp,total,w,thr,f} \end{aligned}, \tag{13}$$

From equation (13) the mechanical work is calculated on the isothermal processes:

$$W_{irev,w,thr,f} = W_{34,irev,w,thr,f} + W_{12,irev,w,thr,f}\,. \tag{14}$$

The mechanical work in the isometric cooling and heating processes, considering the losses caused by the finite speed of the piston, throttling and friction is:

$$\begin{aligned} W_{C,41,w,thr,f} &= P_{C,41} \cdot V_1 \cdot \left(B_{C,w} + B_{C,thr} + B_{C,f}\right) \\ W_{H,23,w,thr,f} &= P_{H,23} \cdot V_2 \cdot \left(B_{H,w} + B_{H,thr} + B_{H,f}\right) \end{aligned}, \tag{15}$$

Based on the expressions (14) and (15), the mechanical work on the cycle is calculated with the losses caused by the finite speed of the piston, throttling and friction taken into account:

$$W_{cycle,w,thr,f} = W_{irev,w,thr,f} - W_{C,41,w,thr,f} - W_{H,23,w,thr,f}\,. \tag{16}$$

Furthermore, the total power of the cycle is determined as:

$$Power_{cycle,w,thr,f} = W_{cycle,w,thr,f} \cdot \frac{w}{2 \cdot z} \cdot i\,. \tag{17}$$

where i – number of cylinder.

The Efficiency of the actual Stirling Machines is always less than that of the ideal Stirling cycle operating between the same temperature limits. This is mainly due to heat transfer losses, X being the term that includes all the losses due to incomplete heat transfer in the Regenerator. This coefficient, X, depends on many variables and parameters. Among these are piston speed w, cylinder dimensions (stroke z and diameter D_C), Regenerator dimensions (diameter D_R and length L) properties of the solid screens (d, b_{Rs}, N), gas properties (ρ_g, c_P, γ, R) and the range of operating conditions ($\varepsilon = V_1/V_2$, $\tau = T_{H,g}/T_{L,g}$, T_m, P_m). A relationship for X that includes the effect of these parameters has been evaluated using First

Law and Heat Transfer principles. This analysis of the losses generated differential equations that were then integrated. The integration of the equations was based on either a lumped system analysis, which gives relatively unfavorable results, X_1, or on a linear distribution of the temperature in the regenerator matrix and gas, which gives relatively favorable results, X_2. The resulting expressions for X_1 and X_2 are [5-9]:

$$X_1 = \frac{1+2M+e^{-B}}{2(1+M)}, \; X_2 = \frac{M+e^{-B}}{1+M}, \tag{18}$$

where: $$M = \frac{m_g c_{v,g}}{m_R c_R}, \; B = (1+M)\frac{h \cdot A_R}{m_g \cdot c_{v,g}} \cdot \frac{z}{w}, \tag{19}$$

and:

$$h = \frac{0.395 \cdot \left(4 \cdot P_m / R \cdot T_L\right) \cdot w_g^{0.424} \cdot c_{P,T_m} \cdot \mu_{T_m}^{0.576}}{(1+\tau) \cdot \left[1 - \pi/4\left[\left(b_{Rs}/d\right)+1\right]\right] \cdot D_R^{0.576} \cdot Pr^{2/3}}, \tag{20}$$

with:

m_g – mass of the gas passing through the Regenerator [kg];
m_R – mass of the screens of the regenerator [kg];
A_R – heat transfer surface area of the wires in the Regenerator [m^2];
D_R – diameter of the Regenerator if circular, or Hydraulic Equivalent Diameter for other forms of the Regenerator section [m];
v_{Tm} – viscosity of the working gas at average temperature T_m [m^2/s];
$c_{p,Tm}$ – specific heat at constant pressure at average temperature T_m [J/kg·K];
$c_{v,g}$ – specific heat of the gas at constant volume at T_m [J/kg·K];
c_R – specific heat of the Regenerator material (copper) [J/kg·K];
Pr – Prandtl Number at average temperature T_m [-];
P_m – average pressure of the gas in engine [Pa];
T_L – temperature of the gas at the sink [K];
w – speed of the piston [m/s];
w_g – speed of the gas in Regenerator [m/s];
$\tau = T_{H,g}/T_{L,g}$, ratio of the gas temperatures [-];
R – gas constant [J/mol·K];
b_{Rs} – the distance between the wires in the Regenerator screens [m];
d – diameter of the wires of the Regenerator screens; [m];
h – convective heat transfer coefficient in a porous medium. [W/m^2·K];

The effect on X_1 and X_2 of the operation variables such as the piston speed w was determined. The computed values of X_1 and X_2 were found to accurately predict the values of X determined from experimental data available in the literature using the following equation [3, 11, 21]:

$$X = y \cdot X_1 + (1 - y) \cdot X_2 , \tag{21}$$

where y is the adjusting parameter.

The new computation methodology was used to find the influence of irreversibilities on efficiency and power, by studying 10 Stirling engines listed in Table 1.

For the adjustable parameter y, the optimum value in the above range can be found according to the method used for evaluating the pressure losses: pessimistic or optimistic. This parameter is found in the equation for X (which includes all the losses due to incomplete heat transfer in the Regenerator), which is directly proportional to it. Therefore, it is important to account for many variables and parameters specific to each engine when setting y value.

The results obtained through this new computation methodology for the 10 Stirling engines are comparable or very close to the experimental efficiency and power for each engine. For most engines studied the efficiency and power graphs are almost identical to the experimental values.

The efficiency on the total cycle with all irreversibility taken into consideration, with a coefficient of regeneration taken into account:

$$\eta_{cycle,w,thr,f} = \frac{W_{cycle,w,thr,f}}{z_q \cdot Q_{34,rev} + X \cdot Q_{41}} . \tag{22}$$

It was computed and determined that y may take values in the range 0.25 – 0.8, based on the number of cylinders that each Stirling Engine has, while z_q may take only two values: 0.55 or 0.8. [15, 20].

3. Validation of the performance of the Stirling engines according to the new computation methodology

In this paper it is shown how this new computation methodology was applied to 10 Stirling engines, for which it was computed their power and efficiency. The engines are among the most used worldwide, solar or classical, which means the input heat can come either from solar concentrators, or from other heat sources. The following engines where studied: 4-95 MKII, 4-275, STM4-120, V-160, GPU-3, MP-1002, NS-03M, NS-03T, NS-30S, NS-30A, for which the experimental data are given in Table 1 together with analytically computed data. For the first 3 engines it was used z_q = 0.55. The obtained values (as well as the graphs) indicate that the computed efficiency is very close to the experimental one. For the last 7 engines we used z_q = 0.8. Again, the computed efficiency was very close (almost identical) to the experimental efficiency, as in [25]. Similar results were obtained for the power: the values obtained with the new methodology were very close to the experimentally obtained ones. These results lead to the conclusion that the new computation methodology is valid for the Stirling engines it was tested on and it may be used successfully for other Stirling engines, too.

Table 1

Results for the Solar Stirling engines using the new method for calculation: 4-95 MKII, 4-275, STM4-120, V-160, GPU-3, MP-1002, NS-03M, NS-03T, NS-30S, NS-30A

Engines	Working gas	Power (kW)		Efficiency (%)		X	z_q	y	i
		Experim data	Calc. data	Experim. data	Calc. data				
4-95 MKII	H	25	25.016	41	35.2	0.191	0.55	0.25	4
4-275	H	50	50.089	42	43.5	0.208	0.55	0.25	4
STM4-120	He	25	25.030	40-45	39.4	0.167	0.55	0.30	4
V-160	He	9	9.037	30	33.7	0.310	0.80	0.60	2
GPU-3	He	3.96	4.016	12.7	14.6	0.430	0.80	0.80	1
MP-1002	Air	0.20	0.204	15.6	15.8	0.509	0.80	0.80	1
NS-03M	He	2.03	2.200	35.9	34.8	0.410	0.80	0.80	1
NS-03T	He	3.08	3.490	32.6	34.7	0.414	0.80	0.80	1
NS-30S	He	30.90	31.800	37.2	37.4	0.280	0.80	0.55	4
NS-30A	He	23.20	23.470	37.5	36.4	0.305	0.80	0.60	4

The graphic material presented below shows the validation results closely.

In Fig. 2, 3 and 4 were plotted curves of power and efficiency for 3 Stirling engines: 4-95 MKII, 4-275, STM4-120. At their plotting were used different values of the coefficient z_q: 0.55; 0.7; 0.8, consequently resulting different values of the Stirling engine efficiency.

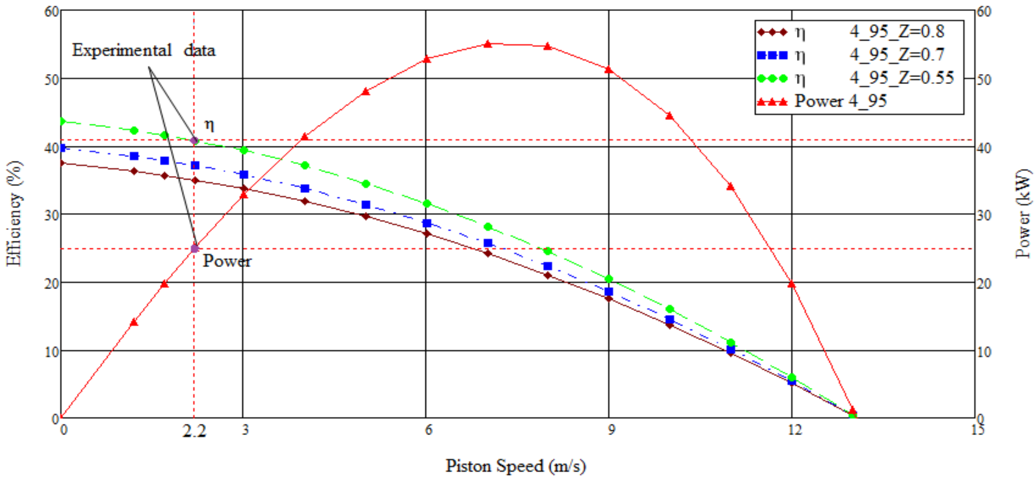

Fig. 2. Stirling engine 4-95MKII, with H_2 as working gas

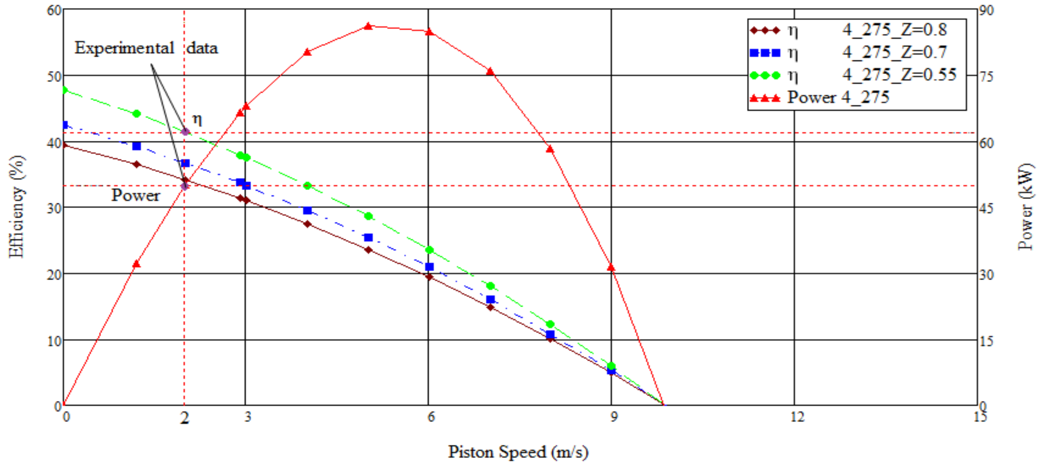

Fig. 3. Stirling engine 4-275 with H_2 as working gas

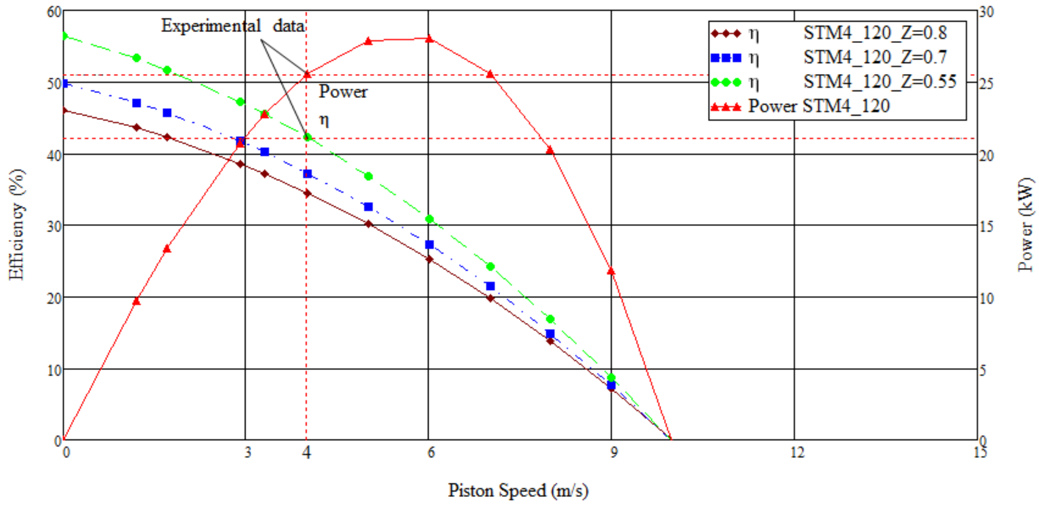

Fig. 4. Stirling engine STM4-120 with He as working gas

For z_q = 0.55, efficiency values resulted for the 3 engines were very close to the experimental data [25].

On the other hand, curves of power and efficiency of other 7 Stirling engines: V-160, GPU-3, MP-1002, NS-03M, NS-03T, NS-30S, NS-30A were plotted for values of the coefficient z_q: 0.6; 0.7; 0.8, consequently resulting different values for the efficiency and power of the Stirling engine. For z_q = 0.8, insignificant differences between the calculated values of the efficiency and power and experimentally obtained data contained in [25] resulted, for the 7 engines.

4. Conclusions

The paper presents a new computational scheme for the performance of Stirling engines. The developed model takes into account the irreversibilities on the isometric processes in the cycle, due to the finite speed of the piston, throttling and friction. These irreversibilities by the gas flowing through the regenerator determine pressure losses in the heating and cooling process, which reduce the engine performance.

The calculation of the real mechanical work revealed that, during isometric processes, the mechanical work is not zero, as considered in reversible ones, but it is negative, thus reducing the power produced by the engine.

Scheme validation, according to the working fluid, and the design characteristics of the engine, has been found possible on groups of 10 Stirling engines used in solar energy applications. The successful validation was obtained by considering only two adjustment coefficients, one taking into account the imperfect regeneration of the heat and the other, the engine capacity to receive only a part of the available heat at the source.

Due to its high precision in predicting real operation Stirling engines performance, the new calculation methodology can be further used for other Stirling engine groups of different powers.

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