# THERMODYNAMICS WITH FINITE SPEED (TFS)
# II. Validation of the direct method for Stirling engine cycles with finite speed

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**Abstract:** This paper has two parts. In the first part we present: "The Development of the Scheme of Computation for Stirling Cycle with Finite Speed Performances (Efficiency and Power)". Because this Computation Scheme of the Performances (Efficiency and Power) is the most important Achievement of the Direct Method utilization, from Thermodynamics with Finite Speed, we describe here the "process of its discovery and invention" from the epistemological point of view, in order to show how important is and was (for me, SP) the connection between teaching and research in generating and developing new ideas. In the Second part of this Chapter we developed this Scheme, showing also its Validation on 12 of the most performing Stirling Engines in the world (USA, Japan, Germany, Sweden), and 16 Operating Regimes.
**Keywords:** validation of direct method for Stirling engines, internal losses in stirling machines, thermodynamics with finite speed

## 1. THE DEVELOPMENT OF THE "SCHEME OF COMPUTATION FOR FINITE SPEED STIRLING CYCLE PERFORMANCES"

The Development of ***The Scheme of Performances Computation and Optimization of Stirling Engines*** (using **TFS**) started in 1991, when Stoian Petrescu published at Helsinki University of Technology, Finland, the book ***Lectures on New Sources of Energy*** [19]. This moment became possible just because of a very long preparation (30 years, starting in 1961) based on many papers and books [1-18] where we actually **build the Fundamentals of "*Thermodynamics with Finite Speed"*** (TFS). I (SP) have to admit retrospectively, that **without the chance to teach a course for PhD Students in Finland, probably this would never happen.**

In October 1991, I (SP) was invited by Prof. Markku Lampinnen (from Helsinki University of Technology), to teach a course for his PhD Students, regarding **"New Sources of Energy".** In his invitation letter he asked me to cover in my lectures the following 4 Chapters:

1 – **Engineering Irreversible Thermodynamics** (where I have developed more on ***Thermodynamics with Finite Speed***);

2 – **Stirling Machines** (Engines, Refrigerators, Heat Pumps);

3 – **Utilization of Solar Energy**;

4 – **High Performances Batteries and Fuel Cells**.

After I have described (in a *letter of intention*) the content of these 4 chapters, Prof. Markku Lampinnen answered that it is OK, but he asked me to prepare in addition to "theoretical aspects" also some "applications" of ***Thermodynamics with Finite Speed***, in order to give some Home-Works for his graduate students (in order to became familiar with some *Computations of Performances and Optimization of Stirling Machines*: Engines, Refrigeration and Heat Pumps). I followed his advice and that was the *beginning* of the ***Development of the Computation Scheme of Stirling Machines Performances.***

The result of preparation of these ***Lectures*** (working 16 hours/day, 3 month in the summer of 1991, in Provita) was 320 pages of notes. When I have arrived in Helsinki (on 1-st October 1991) and he (M.L) saw these notes, he said to me: ***"Let's publish this notes as a textbook for my graduate students here at Helsinki University of Technology".*** The next day each graduate student had in his hands this book [19], and after each lecture they had to prepare for the next day a Home-Work, based on "those Applications" which Prof. M. Lampinnen asked for.

***The Computation Scheme of Stirling Machines Performances*** developed with this occasion conducted us (with graduate students: Roxana Iordache [49], George Stanescu [49, 50, 51], Monica Costea [58, 59, 110], George Popescu [112-120], Traian Florea [60] and Prof. Charles Harman [55, 61, 62, 63, 64,]) eventually to its ***Validation*** and the ***Direct Method*** for 12 Stirling Engines and 16 regimes of functioning [70] in 2002, after 10 years of research and teaching. In the 3 Schemes of Computation for the 3 types of Stirling Machines (Engines, Refrigerators and Heat Pumps) S. Petrescu, did use the following ***main new ideas*** [19]:

- To express all *Performances (Efficiency* and *Power*) of any Stirling Machine as function of the ***Carnot Cycle Efficiency***, multiplied with the ***Second Law Efficiency*** (introduced by Adrian Bejan [95]);
- To **express analytically** the ***Carnot Cycle Efficiency*** as **function of *temperature differences*** (which are the *causes of external irrevesibilities*) at heat sources: $\Delta T_H$ and $\Delta T_L$;
- To **express** these ***temperature differences*** as **function of the *Speed of the Piston w*** (and other *properties of the thermal agent*, and *geometrical dimensions*);
- To **express** the ***Second Law Efficiency*** as **a product of two terms** containing respectively:

  1° - the ***pressure losses in the Regenerator, generated by the Throttling Process,*** $\Delta P_{thr}$, and

  2° - the ***pressure losses generated by the Friction*** between mechanical parts of the Stirling Machine, $\Delta P_f$;
- To **express eventually** these two ***pressure losses*** **as function of the *Speed of the Piston w*** and other ***geometrical characteristics*** (*stroke, diameters, compression ratio, and number of screens in Regenerator*).

This "goal" was achieved later in USA (after 1992) at Duke University in 1992-1993, with the help of Prof. Adrian Bejan and Prof. Charles Harman and at Bucknell University (1993-1998 and 2001-2006) with the help of Prof. Peter Stryker, and my graduate students from there.

The book published in Finland [19] played an essential role in "choosing me out of 200 candidates", mainly because of the "proof" of my capability to teach these courses in an *interesting and attractive mode for students*, which at that moment "*did hate Thermodynamics",* as Prof. James Zaiser explained me later.

Prof. James Zaiser (the Head of Mechanical Engineering Department) and Prof. Peter Striker which visited me at all my lectures, encouraged me and gave me some ideas of improving the course.

Prof. Peter Striker (who was teaching an *Internal Combustion Engines* course) made me aware about how, in the *classical book* of Haywood [94] (on Internal Combustion Engines) the *losses due to friction and throttling* in the valves are taken into account, based on *empirical formulas* as function of the *Number of Rotations per Minute* $N_r$ [rot/min].

With Prof. James Zaiser and Prof. Valeria Petrescu we have published several improved versions of the book from Finland [23-25, 27, 28] which have been developed later in USA, in a two Volume Textbook entitled: **Advanced Energy Conversion** [31, 33, 34, 36, 39, 40, 41, 42] [1].

This book [19] has been developed at Bucknell University where I (SP) was teaching for 10 years a course of "**Advanced Energy Conversion**", to Master students, more and more Problems ("*Applications*" as Prof. Markku Lampinnen called them) for Home-Works and Exams being included.

An essential improvement for the Scheme of Stirling Engine Performances (*Efficiency* and *Power*) Computation was the transformation of these formulas in expressions of the *pressure losses* as function of the *Speed of the Piston w*. Later, we have used these formulas very successfully, first in Monica Costea PhD Thesis (1997) [58], then in Traian Florea PhD Thesis (1999) [60], in Camelia Petre PhD Thesis (2007) [76], in Cuong M. Dang Master Thesis (2002) [69], in Barry Cullen PhD Thesis (Dublin - Ireland 2010) [83] and also in very many other papers (see References).

At one of ASME (American Society of Mechanical Engineering) Congress [Atlanta-1992] where Prof. Adrian Bejan invited me to participate, the famous Prof. Berchowitz presented a very interesting paper. He was already famous for his Stirling Scheme of Computation.

I was asking him: *"how does he take into account the losses in Stirling Engine, due to friction and throttling process"*. He answered very simply: *"... as usual in Fluid Mechanics staff"*.

Our approach was different in comparison with Berchowitz's approach (and others - for example Finkelstein's approach) and this was one of the reason of success (in the **final Validation** [60, 70]) of our ***Scheme of Computation for Stirling Engines Performances***. It consists in expressing the *pressure losses* determined by *throttling* and *friction* in the Stirling engine in a "new way", namely **expressing them as function of the *Speed of the Piston w.***

- For the ***throttling pressure losses*** $\Delta P_{thr}$ in the Regenerator we did find a formula as function of the ***Speed of the Piston,*** *w*, and ***Number of the Screens,*** $N_s$, based on a figure from Organ's

[1] I was actually hired as Visiting Professor at Bucknell University to teach two courses: "***Thermodynamics"***, and "***Advanced Energy Conversion".***

famous book [92] where he synthesized a lot of empirical measurements of $\Delta P_{thr}$ as dependent on the Reynolds Number (Re) of the gas passing through the Regenerator screens. Our formula which "describes the best the empirical results" [58 - 61, 64, 70-86] is:

$$\Delta P_{throt} = (15/k)\cdot(\rho_{gR}\,.\,w_{gR}^{\,2}/2)\cdot Ns \qquad [\text{bar}] \qquad (1)$$

where: $k$ is the *adiabatic exponent*; $\rho_{gR}$ – *density of the gas* in Regenerator; $w_{gR}$ – *speed of the gas* in Regenerator; $Ns$ – number of screens in the Regenerator.

A second important moment was the transformation of the Haywood's [94] empirical formulas for ***friction pressure losses*** $\Delta P_f$ , expressed as a function of the *Number of Rotation per Minute Nr*:

$$\Delta P_f = 0.97 + 0.15\cdot(Nr/1000)$$

into a function of the ***Speed of the Piston*** $w$ and the *Stroke S* [58-61, 64,70, 76-86]:

$$\Delta P_f = 0.94 + 0.45\cdot w/(100\cdot S) \quad [\text{bar}] \qquad (2)$$

where: $S$ is the *Stoke* of the piston. Using this formula based on experimental measurements on Internal Combustion Engines (ICE) for Stirling Engines is justified on the tendency to design and build (technological) the Stirling Engines in a similar manner as Internal Combustion Engines (with *similar parts* and *similar technologies*). The best example regarding this tendency is the Stirling Engine designed, built and tested by Ford Company. This Design was based on the Swedish Stirling Engine [93] Square 4-45, which did have 3 shafts, and 4 cylinders, with *double acting pistons.* Using only one crankshaft instead of 3 shafts, based on the V type Internal Combustion Engines (typical for Ford Design and Technology) made it possible to decrease not only the cost of the Stirling Engine, but also the *friction losses*, at the level of the *friction losses* in Internal Combustion Engines. Based on these considerations we have developed and used expression (2) which actually was demonstrated empirical for ICE, also for Stirling Engines with the highest performances (the 12 Stirling Engines for which our Scheme of Computation was Validated). This engine has the highest thermal ***Efficiency*** of 41 %.

Because in ***Thermodynamics with Finite Speed (TFS)***, we have expressed **all losses as function of the *Speed of the Piston w*,** a sort of **Unification of the losses treatment, just function of the most important parameter (the *Speed of the Piston w*) was the key of the success in Optimizing this Speed for *Maximum Efficiency* or for *Maximum Power* of the Stirling Machines (and later for any Irreversible Cycle, as for example: Carnot** [29, 33], **Otto** [46], **Diesel** [47]**, Otto-Stirling** [80-83], **Brayton cycle** [106-109]). The very interesting and challenging interaction with my students from Bucknell University (Lewisburg, PA, USA) "forced me (SP) to invent" new tools for better understanding and more precisely computation for *Performances* of Stirling Machines. One of the most important "tool" was the invention of two PV/Px diagrams [29, 33-44, 60, 62, 63, 64, 66, 70, 71], **a completely new diagram for more intuitive description of the Stirling Cycles** (for Engines, Refrigerators and Heat Pumps). Cooperation with Prof. Adrian Bejan and Prof. Charles Harman (The Director for Graduate Studies from Duke University-NC, USA) in this period was essential. A. Bejan gave me 4 papers of Prof. J.A. Organ [92] regarding the Regenerator in Stiling Machines and one of his famous book [92]. Based on these S. Petrescu and M. Costea [58, 59], were able **to find a formula for estimation the *throttling losses* in Regenerator, based on experimental data**. This formula was essential for the **"initial Validation"** [58, 59, 27-44] and later for **more precisely the "Final Validation"** [70, 60, 29]. Prof. C. Harman (from Duke University, NC, USA) "puts in my hands" the NASA - Code for Free Piston Stirling Engine, in which 9 *adjusting parameters* are used for validation only for one Free Piston Stirling Engine (which was initially designed "to go" on Mars Planet). Many papers [59-86, 110] regarding the Computation of Performances of Stirling Engines and 3 PhD Thesis: G. Stanescu (1992) [50], M. Costea (1997) [58], and T. Florea (1999) [60] followed these researches. Four of these papers [59-86]: **Petrescu, S., Costea, M., C. Harman, T. Florea,** (2000-2002) [62-64, 70], the **Monica Costea** Thesis [58] and the **Book based on it** [110], the **Traian Florea's** PhD Thesis (1999) [60] and the **Book based on it** (2000) [29] were "*the culmination*" of all of previously researches, and for this reason these are the most important for the ***Development of Thermodynamics with Finite Speed and the Direct Method***. Here [70, 60][2], in the period 1999-2002, the V**alidation for 12 Stirling Engines and 16 Regimes of Operation have been obtained.**

This almost "unexpected success" from the point of view of ***Validation of the Direct Method***

[2] PETRESCU, S., COSTEA, M., HARMAN, C., FLOREA, T. ***Application of the Direct Method to Irreversible Stirling Cycles with Finite Speed,*** International **Journal of Energy Research,** USA, Vol. 26, pp.589-609, 2002 [70].

***and the Scheme of Computation and Optimization of Stirling Machines is exceptional important,*** because in comparison with **NASA - Code** ***we have used only 3 and finally only 2 adjusting parameters, (much less than 9 adjusting parameters used in NASA- Code for only one Free Piston Stirling Engine).***

In addition to that, our Scheme of Computation from this paper [70] ***deserves a special attention for all interested in understanding The Direct Method from TFS***, because it is valid for 12 engines (in comparison with NASA-Code valid only for one engine) and it is useful also for Stirling Refrigeration and Heat Pumps Machines [65, 67, 68]. Using this Method, at least two new PhD Thesis have been developed: **Camelia Petre** [76] - where the **Scheme of Computation was applied and extended for Solar Stirling**, and **Barry Cullen** [83] - where **The Scheme of Computation was used in order to Optimize an assemble of Otto and Stirling Engine**, and many papers have been published [80, 81, 82, 103-109]. More and more researchers recognized the **importance of the Direct Method from TFS and the utility of it in the Treatment, Optimization and Design of Irreversible Cycles with Finite Speed** [98-102].

## 2. VALIDATION OF THE PERFORMANCES COMPUTATIONAL SCHEME FOR 12 STIRLING ENGINES AND 16 REGIMES OF OPERATION.

In this paragraph, a technique for calculating the ***Efficiency*** and ***Power*** of Stirling Engines is presented (based on the papers [19, 23, 27, 29, 31, 33, 34, 59, 60, 62, 63, 70, 71]. We called it **"The Computation Scheme of Stirling Machines Performances".** Despite of the fact that initially, it was validated only for Stirling Engines, it can be used also for Refrigeration and Heat Pump Stirling Machines [86].

This technique is based on the **First Law of Thermodynamics for processes with Finite Speed for Complex Systems** [19, 49, 55] and the Direct Method for closed systems. In order to apply the Direct Method to Stirling Cycles, a new and novel PV / Px diagram was invented [62, 63, 64]. The merit of this new diagram is its capability to show the effects of *pressure losses* due to Friction, Finite Speed and Throttling Processes in the Regenerator of the Stirling Engine.

The method used for the analysis of this irreversible cycle with Finite Speed involves the *direct integration* of equations based on the First Law for processes with Finite Speed to obtain the cycle ***Efficiency*** and ***Power*** directly.

This technique is termed the **Direct Method**. The results predicted by this analysis are in very good agreement with the actual engine performance data of twelve different Stirling Engines, over a range of output from economy to maximum power. This provides a solid verification that this analysis, based on the Direct Method, can accurately predict actual ***Stirling Engine Performance***, particularly with regard to ***Efficiency*** and output ***Power.*** In addition to the powerful predictive capabilities of the Direct Method, the new PV / Px diagram for the Stirling cycle is both an effective and an intuitive tool for explaining the Operation and Design of Stirling Machines.

In 1992 Organ [92] makes **a remarkable analysis of the different methods for treatment of the details of Stirling Cycle Engines**. He presents the existing methods of analysis and points out the advantages and disadvantages of each. He places each of these methods between two extremes of complexity. The simplest presents the Stirling cycle as only a diagram on PV coordinates consisting of two *isothermal* process lines combined with two *isometric* process lines. This provides a grossly insufficient description of even the ideal cycle and it is even more inadequate as a basis for understanding real machines. However, this is the typical introductory textbook representation.

At the other extreme, Organ (1992) [92] considers the most complex methods to be those based on the *numerical integration of differential equations* written to account for as many of the actual variables in Stirling cycle machines as possible. These complex analyses are capable of a high degree of mathematical precision, but they have a severe disadvantage in that they tend to completely obscure the physical processes involved. They are not effective in aiding the understanding of the basic cycle nor are they effective aids to designers in their efforts at optimization of actual Stirling cycle machines.

The 4 objectives of this paragraph are:

(1) **- to present two new and novel *PV / Px* diagram for the Stirling Cycle;**

(2) **- to use this diagram to provide a basis for a clear understanding the processes of the actual Stirling Engine** and

(3) **- to provide an analysis that includes calculations of the *pressure and heat transfer losses* in the Stirling cycle.**

Once these *pressure losses* have been determined and the way the machine actually functions is clear, **an analysis can be made that makes possible the**

**prediction** of the ***Power*** and ***Efficiency*** of the machines that, in terms of complexity, **lies somewhere between the extremes cited above. This analysis conducted us [70] to the Computation Scheme of Stirling Machines Performance (Efficiency and Power).**

(4) - **to validate this Computation Scheme of Stirling Machines Performance for as many as possible Stirling Engines**.

### 2.1. Two new and novel PV / Px Diagram for Stirling Cycle Processes Description

The objective of this section is to provide an analytic model that closely simulates the operation of actual Stirling engines and that does so without losing insight to the mechanisms that generate the irreversibilities in the actual engine. In order to accomplish this, a new *PV / Px* diagram for the Stirling cycle is presented. [43, 56, 60, 62, 64, 66, 70].

The two basic configurations that are characterristic of most actual Stirling engines (Walker, 1983) [83] are considered. These are first the *split engine with displacer and piston* which is referred to as ***Type 1*** and which is shown in **Figure 1** and second the *two piston engine* which is referred to ***Type 2*** and which is shown in **Figure 2.** The ***PV / Px* diagram** for engine ***Type 2*** is presented in [62] and that for ***Type 1*** is presented, in greater detail in [64].

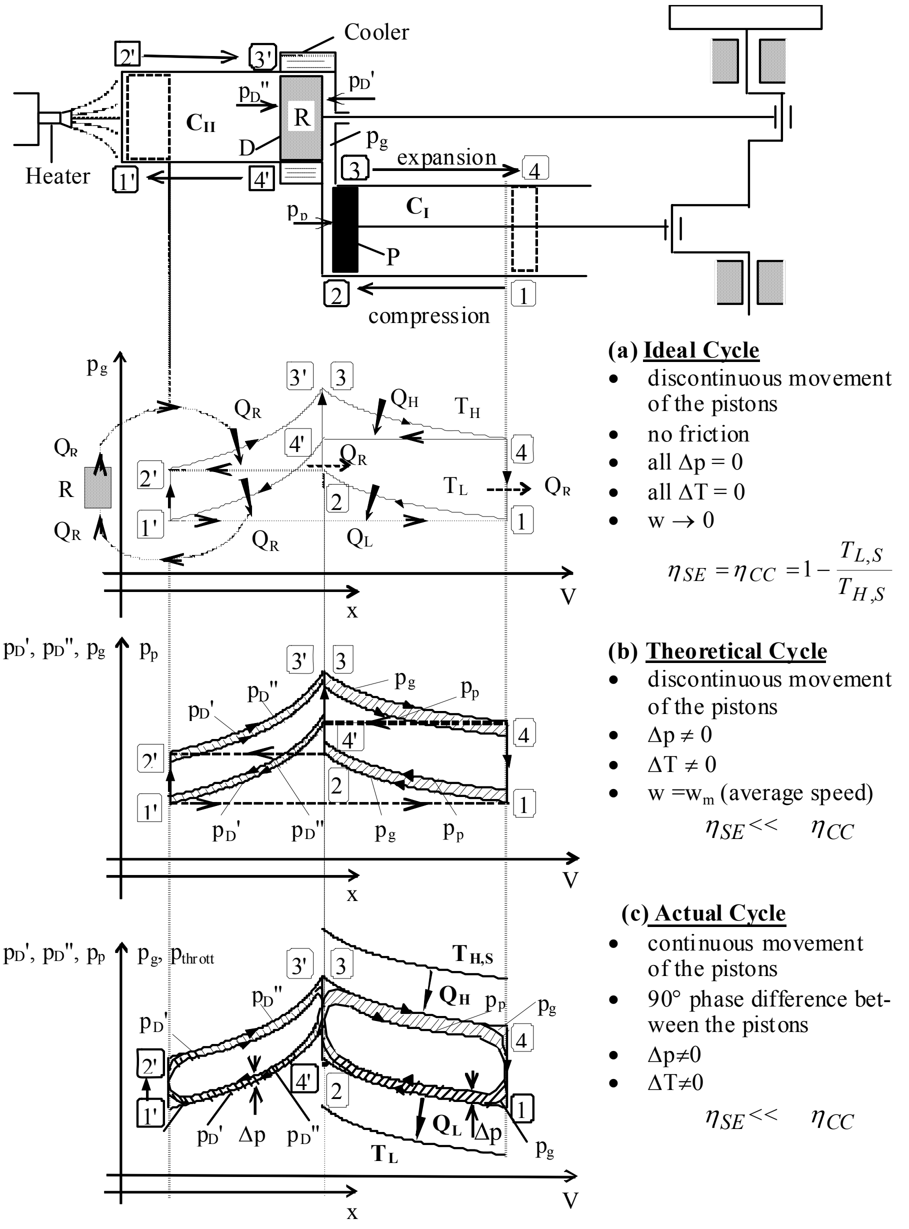



**Fig. 1.** The *PV / Px* diagram for Stirling engine ***Type 1*** [27, 28, 42, 43, 60, 64, 70].

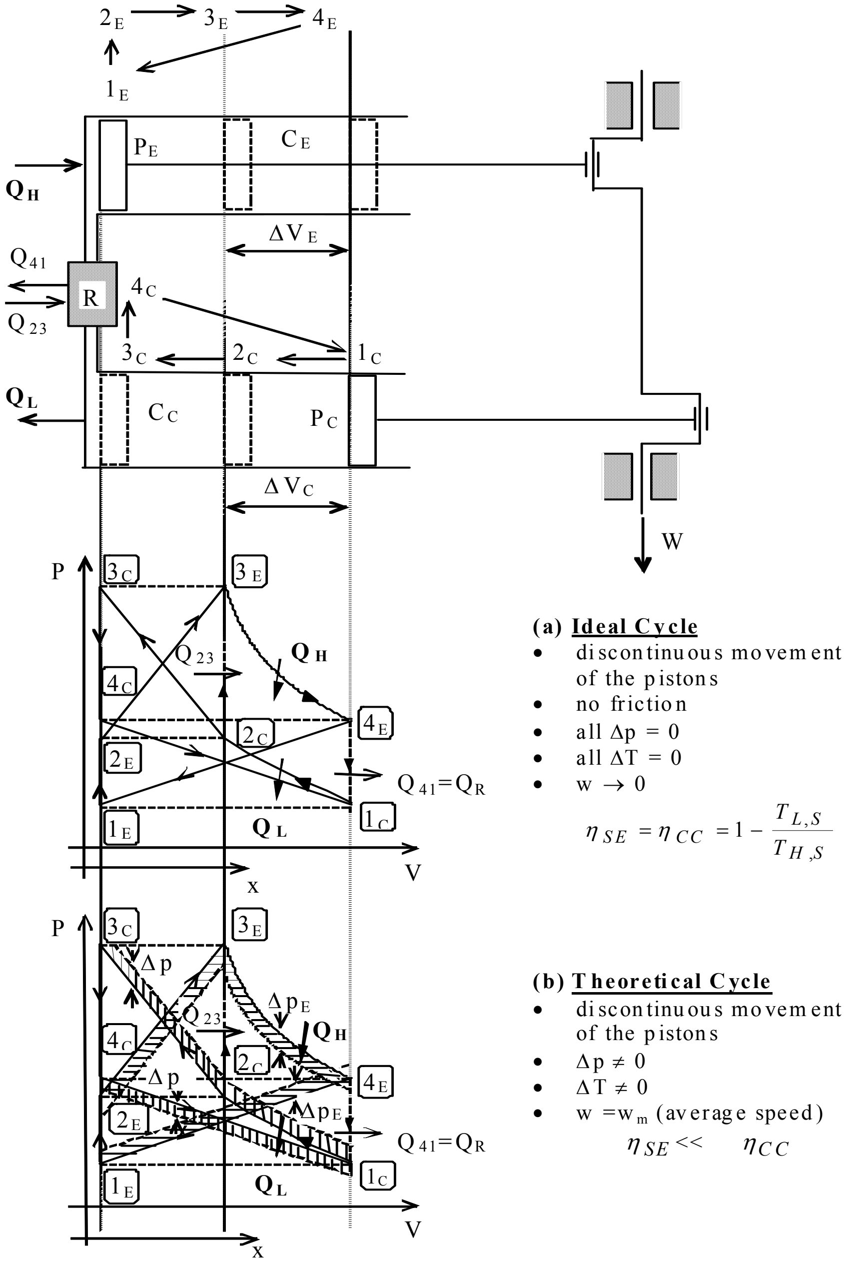



**Fig. 2.** The *PV/ Px* diagram for Stirling engine ***Type 2*** [27, 28, 42, 43, 60, 62, 64, 70].

## 2.2. Calculation of *Pressure Losses* and *Second Law Efficiency*, using the Direct Method from TFS

***Pressure losses*** for the processes shown on the *PV/Px* diagrams are calculated using the ***First Law of Thermodynamics for Processes with Finite Speed*** (which actually is combined with the second part of the Second Law) **for processes with *Finite Speed, Throttling and Friction*** [19, 49, 55]:

$$\mathrm{d}U = \delta Q - P_{m,i}\left(1 \pm \frac{a\,w}{c} + \frac{b\,\Delta P_{thr}}{2\,P_{m,i}} \pm \frac{f\,\Delta P_f}{P_{m,i}}\right)\mathrm{d}V \quad (3)$$

Then, the ***irreversible work for processes with finite speed*** in closed systems is:

$$\delta W_{irr} = P_{m,i}\left(1 \pm \frac{a\,w}{c} + \frac{b\,\Delta P_{thr}}{2\,P_{m,i}} \pm \frac{\Delta P_f}{P_{m,i}}\right)\mathrm{d}V \quad (4)$$

when applied to processes with finite speed for the Stirling engine as shown in the ***PV / Px* diagrams**.

In these equations, the (+) sign corresponds to compression and the (–) sign to expansion.

For the *finite speed isothermal irreversible compression process* 1-2, we can integrate directly by applying the **Direct Method** [50 - 52, 55] to Eq. 4:

$$W_{12,irr} = \int_1^2 P_{m,cpr,i}\, \mathrm{d}V + \int_1^2 \left( \frac{a\,w}{c} + \frac{b\,\Delta P_{thr}}{2\,P_{m,cpr,i}} + \frac{\Delta P_f}{P_{m,cpr,i}} \right) P_{m,cpr,i}\, \mathrm{d}V \quad (5)$$

$$W_{12,irr} = W_{12,rev} + \left( \frac{a\,w}{c_{cpr}} P_{m,cpr} + \frac{b\,\Delta P_{thr}}{2} + \Delta P_f \right) (V_2 - V_1) \quad (6)$$

The *work losses* are calculated for the compression process 1-2 in **Figure 2** by using Eq. (6).

These include losses due to the ***pressure drops caused by finite piston speed, throttling of the gas through the regenerator displacement piston and mechanical friction***.

$$W_{losses,cpr} = \left| W_{12,irr} - W_{12,rev} \right| = \sum \Delta P_{cpr} (V_2 - V_1) = \sum \Delta P_{cpr}\, \Delta V_c \quad (7)^3$$

Similarly, the *work loss* during the *isothermal expansion process with finite speed* 3-4 is calculated using:

$$W_{losses,\exp} = W_{34,rev} - W_{34,irr} = \sum \Delta P_{\exp}\, \Delta V_e \quad (8)$$

with $$\Delta V = V_4 - V_3 = V_1 - V_2 = \Delta V_c = \Delta V_e \quad (9)$$

The ***work loss*** is computed similarly, for the *isometric regeneration process* 2'-3' that corresponds to the movement of the *displacer D* containing regenerator R, as shown in cycle (b) of **Figure 1** through the integration of Eq. (4).

$$W_{losses,R,\,2'3'} = \sum \Delta P_R^{'}\, \Delta V_R = \left( \frac{a\,w}{c_{T_m}} P_{m,2'3'} + \frac{b\,\Delta P_{thr}}{2} + \Delta P_f \right) \cdot \Delta V_R^{'} \quad (10)$$

The loss in the ***isometric regenerative cooling process*** 4'-1' that occurs with the movement of the *displacer / regenerator* in cycle (b) in **Fig. 1,** is calculated by integrating Eq. (4):

$$W_{losses,R,\,4'1''} = \sum \Delta P_R^{''}\, \Delta V_R = \left( \frac{a\,w}{c_{T_m}} P_{m,4'1} + \frac{b\,\Delta P_{thr}}{2} + \Delta P_f \right) \cdot \Delta V_R^{''} \quad (11)$$

For the Stirling Engine of ***Type 1*** in **Figure 1** the *volume change during regeneration* is:

$$\Delta V_R^{'} = \Delta V_R^{''} \cong V_2 \quad (12)$$

The loss due to the *finite speed* of the pistons, *throttling processes* and *mechanical friction* for the ***Type 1*** Stirling Engine is obtained by summing the above four losses:

$$W_{losses,\sum \Delta P_i} = \Delta V \left( \sum \Delta P_{cpr} + \sum \Delta P_{\exp} \right) + V_2 \left( \sum \Delta P_R^{'} + \sum \Delta P_R^{''} \right) \quad (13)$$

In the following, **Two Methods for evaluating the pressure losses are presented**: one gives a ***pessimistic prediction*** and the other an ***optimistic prediction*** of the engine *Work* and *Efficiency.*

In the ***Pessimistic Method we make the hypothesis***, that the ***pressure losses*** could be estimated as **an average which is taken on the 4 *pressure losses*:**

$$\sum \Delta P_{cpr} \cong \sum \Delta P_{\exp} \cong \sum \Delta P_R^{'} \cong \sum \Delta P_R^{''} = \sum \Delta P_m \quad (14)$$

and developing with have got:

$$\sum \Delta P_m = \Delta P_{thr} + \Delta P_f + \underbrace{0.5 \frac{aw}{c_{cpr}} P_{m,cpr}}_{\Delta P_{cpr}^{w}} + \underbrace{0.5 \frac{aw}{c_{\exp}} P_{m,\exp}}_{\Delta P_{\exp}^{w}} \quad (15)$$

where: $\Delta P_{thr}$ and $\Delta P_f$ are expressed as a function of the *average speed of the gas* through the Regenerator $w_R$ and respectively the *average speed of the pistons w*.

The ***Optimistic method***, in comparison, takes into account the pressure losses due to throttling in the Regenerator only during two processes 2'-3' and 4'-1'. This occurs when the *displacer / regenerator* is moving and the working piston P is stationary. Also, the losses generated by the *finite speed* of the power piston *w* are taken into account only twice and not four times, as in the pessimistic method. That means, that these *pressure losses* appear only during the compression process with *finite speed* 1-2 and expansion process with finite speed 3-4, when the displacer/regenerator is stationary. As a consequence, the expression for the *work losses* generated by the *finite speed w* of the pistons using the ***optimistic hypothesis*** is:

$$W_{losses,\Sigma \Delta p_i} = (V_1 - V_2) \cdot \Delta P_{cpr}^{w} + (V_1 - V_2) \cdot \Delta P_{\exp}^{w} + \Delta P_{R,23} \cdot V_2 + \Delta P_{R,41} \cdot V_1 + \Delta P_f \cdot V_S \quad (16)$$

The ***efficiency*** of Stirling Engines, upon substitution of the results of pressure loss calculations using the *pessimistic hypothesis*, for ***Type 1*** and ***Type 2***, respectively is calculated by Eq. (17), (18) and (19).

The ***efficiency*** using the ***optimistic hypothesis***, for ***Type 1*** and ***Type 2*** Stirling Engines, respectively [19, 29, 60, 70] is calculated by Eq. (20) and (21).

[3] Remark: ΔV's in Eqs. (7) – (12) assume that the *dead volume* is negligible.

$$\eta_{II,irrev\Sigma\Delta P_i}=1-\frac{\frac{w}{w_{SL}}\gamma\frac{\varepsilon}{\varepsilon-1}\left(1+\sqrt{\tau}\right)\ln\varepsilon+\frac{15}{2}\cdot(\varepsilon+1)N\left(\frac{w_R}{w_{SL}}\right)^2+\frac{0.94+0.045w}{P_1}10^5}{\tau\cdot\eta'\cdot\ln\varepsilon} \tag{17}$$

$$\eta_{II,irrev\Sigma\Delta P_i}=1-\left(3-\frac{1}{\varepsilon}\right)\frac{\frac{w}{w_{SL}}\frac{\gamma}{2}\frac{\varepsilon}{\varepsilon-1}\left(1+\sqrt{\tau}\right)\ln\varepsilon+\frac{15}{4}\cdot(\varepsilon+1)N\left(\frac{w_R}{w_{SL}}\right)^2+\frac{0.94+0.045w}{2P_1}10^5}{\tau\cdot\eta'\cdot\ln\varepsilon} \tag{18}$$

$$\eta'=\left(1-\sqrt{\frac{T_L}{T_{H,S}}}\right)\cdot\left[1+\frac{X\left(1-\sqrt{T_L/T_{H,S}}\right)}{(\gamma-1)\ln\varepsilon}\right]^{-1}. \tag{19}$$

$$\eta_{II,irrev\Sigma\Delta P_i}=1-\frac{\frac{w}{w_{SL}}\gamma\left(1+\sqrt{\tau}\right)\ln\varepsilon+\frac{15}{2}\cdot\left(1+\frac{1}{\varepsilon}\right)N\left(\frac{w_R}{w_{SL}}\right)^2+\frac{0.94+0.045w}{P_1}10^5\left(1-\frac{1}{\varepsilon}\right)}{\tau\cdot\eta'\cdot\ln\varepsilon} \tag{20}$$

$$\eta_{II,irrev\Sigma\Delta P_i}=1-\frac{\frac{w}{w_{SL}}\gamma\left(1+\sqrt{\tau}\right)\ln\varepsilon+15\cdot N\left(\frac{w_R}{w_{SL}}\right)^2+\frac{0.94+0.045w}{P_1}10^5\left(1-\frac{1}{\varepsilon}\right)}{\tau\cdot\eta'\cdot\ln\varepsilon} \tag{21}$$

The *performance trends* predicted by these equations have been compared to *actual performance data* obtained from a variety of the most performing operating Stirling engines [88, 89, 90, 91, 93] in the World (USA, Japan, Sweden). The actual Stirling Engine data was combined with the above analysis and **a single equation** for the ***Second Law Efficiency*** that takes into account the ***pressure losses*** was discovered that formulated as follows, is "good enough", and we do not need two equations:

$$\eta_{II,irrev\Sigma\Delta P_i}=$$
$$=1-\frac{\frac{w}{w_{SL}}\gamma\left(1+\sqrt{\tau}\right)\ln\varepsilon+5\cdot N\left(\frac{w_R}{w_{SL}}\right)^2+\frac{3(0.94+0.045w)}{4P_1}10^5}{\tau\cdot\eta'\cdot\ln\varepsilon} \tag{22}$$

### 2.3. Calculation of *Efficiency and Power of Stirling Engines*, using the Direct Method from TFS

The ***Stirling Engine Efficiency*** will then be given by [19, 59, 60-64]:

$$\eta_{SE}=\eta_{CC}\cdot\eta_{II,irrev}=\underbrace{\left(1-\frac{T_L}{T_{H,S}}\right)}_{\eta_{CC}}\cdot$$
$$\cdot\underbrace{\left[1+\sqrt{\frac{T_L}{T_{H,S}}}\right]^{-1}}_{\eta_{II,irrev,\Delta T_{opt,power}}}\underbrace{\left[1+\frac{X\left(1-\sqrt{T_L/T_{H,S}}\right)}{(\gamma-1)\ln\varepsilon}\right]^{-1}}_{\eta_{II,irrev,X}}\eta_{II,irrev,\Sigma\Delta P_i} \tag{23}$$

where: $\Delta T_{opt,power}$ is the *temperature difference* associated to the ***Maximum Power output*** of the Stirling Engine.

The ***Efficiency*** of actual Stirling Machines is always less than that of the idealized Stirling cycle operating between the same temperature limits. This is due principally to heat transfer losses that occur in the regenerative processes. The coefficient of regenerative losses, *X*, in Eq. (23) is the term that includes all of the losses due to *incomplete heat transfer* in the Regenerator. An analysis for determining these losses has been made by Petrescu et al. [19, 60, 62-64,70]. This coefficient, X, clearly depends on a large number of parameters and variables. Among these are *piston speed w, cylinder dimensions* (diameter $D_C$ and stroke *S*), *Regenerator dimensions* (*diameter* $D_R$ and *length L*) *properties of the solid porous material* in the Regenerator (*density* $\rho_R$, *specific heat* $c_R$), *dimensions of the screens* (*d, b, N*), *gas properties* ($\rho_g$, $c_P$, $\gamma$, *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 these losses produced differential equations that were then integrated. The results are shown in **Figure 2**.

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 $X$ are:

$$X_1 = \frac{1 + 2M + e^{-B}}{2(1 + M)}, \; X_2 = \frac{M + e^{-B}}{1 + M} \tag{24}$$

where: $M = \frac{m_g c_{v,g}}{m_R c_R}$, $B = (1 + M)\frac{h A_R}{m_g c_{v,g}} \cdot \frac{S}{w}$ (25)

$$h = \frac{0.395\left(4P_m / RT_L\right)w_g^{0.424} \cdot c_P\left(T_m\right) \cdot \nu\left(T_m\right)^{0.576}}{(1+\tau)\left[1 - \frac{\pi}{4\left[(b/d)+1\right]}\right]D_R^{0.576} \cdot \Pr^{2/3}} \tag{26}$$

*with: $m_g$ – mass of the gas* passing through the Regenerator; *$m_R$ – mass of the screens* of the Regenerator; *$A_R$ – heat transfer surface area* of the wires in the Regenerator; *$D_R$ – Diameter of the Regenerator if is circular,* or *Hydraulic Equivalent Diameter* for *other forms of the Regenerator section; $\nu$ ($T_m$) – viscosity of the working gas at average Temperature $T_m$; $c_p$ ($T_m$) – specific heat at constant pressure at average Temperature $T_m$; $c_{v,g}$ – specific heat of the gas at constant volume at $T_m$; $c_R$ – specific heat of the* Regenerator *material* (copper, stainless steel); *Pr – Prandtl Number at average temperature $T_m$ ;$p_m$ – average pressure of the gas in the engine; $T_L$ – temperature of the gas at the low heat source;S – Stroke of the Piston; w – speed of the piston; $w_g$ – speed of the gas in Regenerator; $\tau = T_{H,g} / T_{L,g}$, ratio of the gas temperature; R – gas constant;b – distance between wires in the Regenerator screens;d – diameter of the wires of the Regenerator screens; h – convective heat transfer coefficient* in a porous medium [96].

The effect on $X_1$ and $X_2$ of the operating variables such as *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 [88, 89, 90, 91, 93] using the following equation:

$$X = yX_1 + (1 - y)X_2 \tag{27}$$

where the *adjusting parameter y* is equal to 0.72.

The final loss to be considered in the analysis is the *loss due to incomplete regeneration* as indicated in Eq. (23). The ***Second Law Efficiency*** due to irreversibilities from *incomplete regeneration* is from Eq. (23) with Eq. (27) and taking into account *variable specific heats with temperature*):

$$\eta_{II,irrev,X} = \left[1 + \frac{\left[X_1 \cdot y + X_2 \cdot (1-y)\right]\left(1 - \sqrt{T_L / T_{H,S}}\right)}{\frac{R}{c_v(T)}\ln\varepsilon}\right]^{-1} \tag{28}$$

In the papers [70, 60, 62 - 64] **we have studied:**

– The variation of the ***coefficient of regenerative losses*** with the piston speed is represented for several values of the analysis parameters (d-*wire diameter*, S-*stroke*, *porosity*);

– The variation of ***regenerative losses coefficient***, *X*, versus the *piston speed w* for several values of the *average pressure* of the working gas;

– The ***convective heat transfer coefficient*** *h* dependence on *piston speed w* for several values of the *average pressure* of the working gas.

Based on such *sensitivity studies* a better Design of Stirling Machines could be achieved, using the *analytical results of the Direct Method* presented here.

The final **analytic expression** for the ***Power*** output is:

$$Power_{SE,irrev} = \eta_{SE} \cdot z \cdot mRT_{H,g} \cdot \frac{w}{2S}\ln\varepsilon \tag{29}$$

This **expression accurately predicts the *Power* output of twelve Stirling engines** [88 - 91, 93], when the ***adjusting parameter z* equals 0.8.**

## 2.4. Validation of Efficiency and Power output for Stirling Engines with Finite Speed

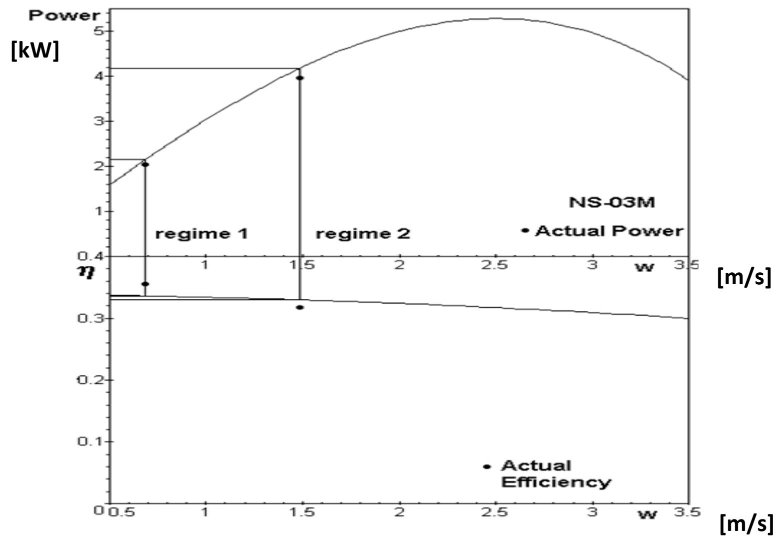
**The calculated and experimental values**, together with the errors of COMPUTATION VALUES IN COMPARISON WITH REAL VALUES are presented in **Tables 1 and 2** [70, 60]. **The calculations for Validation** are based on DATA (regarding the most preferment Stirling Engines from: USA, Germany, Sweden, Japan) and were collected by S. Petrescu and T. Florea [70, 60] and presented in the **Tables 1.A and 1.B**. in **section Annex**.

The **high degree of correlation between the analytic and the operational data** shown in **Figures 3** and **4** [70, 60] and **Table 1** [70, 60] and **Table 2** [70, 60] indicates that the **analysis accurately predicts the performance in terms** of ***Power*** and ***Efficiency*** for Stirling Engines over the range of conditions. **This capability has value in the Design of new Stirling Engines and in predicting the *Power* and *Efficiency* of a particular Stirling Engine.**

In **Figure 3 we can see a very good concordance between computed and measured values of Power output and efficiency for two of those 12 Engines studied in the papers** [70, 60].

In this way the **Scheme of Computation of Stirling Engines Performances is Validated, and at the same time, also, the Direct Method from**

**Thermodynamics with Finite Speed which becomes one of the most powerful "tool" from all the Branches of Irreversible Thermodynamics is Validated.**



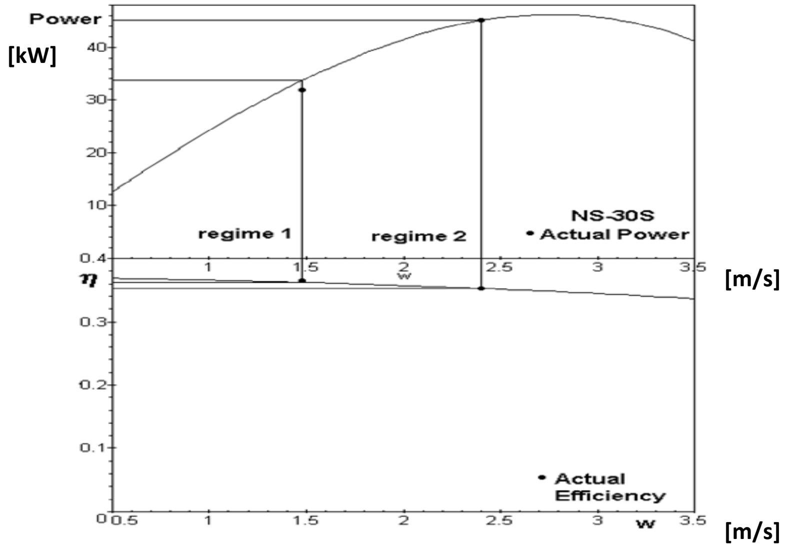




**Fig. 3**. Comparison of the analysis results with actual performance data for the NS-03M and NS-30S Stirling Engines [60, 70].

*Table 1*

***Computed Power*** **of Stirling Engines with formula (29) in comparison with** ***actual Power*** [87-93], **from the results in** [70, 60]

| Stirling Engine | Actual Power [kW] | Calculated Power [kW] | Absol. Error [kW] | Relat. Error [%] |
|---|---|---|---|---|
| NS-03, economy | 2.03 | 2.182 | 0.152 | 7.4 |
| NS-03, max.power | 3.81 | 4.196 | 0.386 | 10.1 |
| NS-03T, economy | 3.08 | 3.14 | 0.06 | 1.94 |
| NS-03T, max. power | 4.14 | 4.45 | 0.31 | 7.48 |
| NS-30A, economy | 23.20 | 29.45 | 6.25 | 26.9 |
| NS-30A, max. power | 30.40 | 33.82 | 3.42 | 11.25 |
| NS-30S, economy | 30.9 | 33,78 | 2.88 | 9.32 |
| NS-30S, max. power | 45.6 | 45.62 | 0.02 | 0.043 |
| STM4-120 | 25.00 | 26.36 | 1.36 | 5.4 |
| V-160 | 9.00 | 8.82 | 0.18 | 2 |
| 4-95 MKII | 25.00 | 28.40 | 3.40 | 13.6 |
| GPU-3 | 3.96 | 4.16 | 0.2 | 5.0 |
| MP1002 CA | 0.200 | 0.194 | 0.006 | 3.0 |
| 4-275 | 50 | 48.61 | 1.39 | 2.7 |
| Free Piston Stirling Eng. | 9 | 9.165 | 0.165 | 1.83 |
| RE-1000 | 0.939 | 1.005 | 0.066 | 7.02 |

*Table 2*

***Efficiency* of Stirling Engines with formulas (22-28) in comparison with *actual Efficiency*** [87 - 93] **from the results in** [70, 60]

| Stirling Engine | Actual Efficiency [%] | Calculated Efficiency [%] | Absol. Error [%] | Relat. Error [%] |
|---|---|---|---|---|
| NS-03, economy | 35.9 | 33.92 | 1.98 | 5.5 |
| NS-03, max. power | 31.0 | 32.97 | 1.97 | 6.3 |
| NS-03T, economy | 32.6 | 31.9 | 0.7 | 2.1 |
| NS-03T, max. power | 30.3 | 30.9 | 0.6 | 1.9 |
| NS-30A, economy | 37.5 | 35.7 | 0.2 | 0.5 |
| NS-30A, max. power | 33.0 | 33.6 | 0.6 | 1.8 |
| NS-30S, economy | 37.2 | 36.6 | 0.6 | 1.6 |
| NS-30S, max. power | 35.2 | 35.26 | 0.06 | 0.17 |
| STM4-120 | 40.0 | 40.1 | 0.1 | 0.25 |
| V-160 | 30.0 | 30.8 | 0.8 | 2.6 |
| 4-95 MKII | 29.4 | 28.9 | 0.5 | 1.7 |
| GPU-3 | 12.7 | 12.6 | 0.1 | 0.7 |
| MP1002 CA | 15.6 | 15.3 | 0.3 | 1.9 |
| 4-275 | 42.0 | 41.19 | 0.81 | 1.92 |
| Free Piston Stirling Engine | 33.0 | 33.1 | 0.1 | 0.3 |
| RE-1000 | 25.8 | 22.85 | 2.95 | 11.4 |

**Because of that, L. Chen et al. [98 - 102] from China and other researchers like B. Cullen and J. McGovern [80 - 83] from Ireland, started to believe that this Direct Method is one of the most powerful tool in Irreversible Engineering Thermodynamics, and started to use it in order to Optimize Irreversible Cycles.**

**Actually L. Chen et al., in recent papers [98-102] started "a very important process of Unification" between *Thermodynamics with Finite Time* [TFT] and *Thermodynamics with Finite Speed* [TFS], in order to take into account on a *fundamental bases, introduced only in TFS,* the *internal irrevesibilities* (*Finite Sped*, *Friction* and *Throttling*).**

**We do believe that this process of "Unification" and other sort of "Unifications" (between very "different" branches of Science) is going to continue in the near future, in order to obtain an *Irreversible Engineering Thermodynamics more general and capable to optimize the Speed w, Time (duration), Temperature, Geometrical Parameters, Thermal Agent Properties etc., giving the Designers a much better tool for conceiving new and challenging Thermal Machines and also, Electrochemical Devices (See* [111] *Chapter 14) or combinations between different Thermal Machines (like Otto with Stirling* [80-83]*) or like SEHE Systems (See* [84, 111] *Chapter 15).***

*ANNEX*

## DATA FOR VALIDATION OF DIRECT METHOD APPLIED TO STIRLING ENGINES

In **Table 3** and **Table 4** are presented the DATA used for **Validating the Direct Method of TFS for 12 Stirling Engines, at 16 Working Regimes.**

**These DATA** were collected from [89, 90, 91, 93] by S. Petrescu (as Visiting Professor at Bucknell University, Lewisburg, PA, USA in 1993-1998 and 2001-2006) and used in T. Florea PhD Thesis [60], in IJER paper S. Petrescu et all. [70], and in the Book: S. Petrescu, M. Costea et all. [111].

*Table 3*

**Data for Japan Stirling Engines [91] used in T. Florea PhD Thesis [60] and in IJER Paper by S. Petrescu et al. [70]**

| DATA for Validation of TFS and Direct Method | NS-03M [91] | NS-03T [91] | NS-30A [91] | NS-30S [91] |
|---|---|---|---|---|
| No. of cylinders | 1 | 1 | 4 | 4 |
| $\varepsilon=V_2/V_1$ | 1.53 | 1.53 | 1.60 | 1.52 |
| D cylinder [mm] | 80 | 82 | 60 | 68 |
| Stroke [mm] | 32 | 36 | 52.4 | 40 |
| P med [bar] | 62 | 64 | 147 | 155 |
| No. screens in Reg./unit | 700 | 480 | 794 | 631 |

Table 3 (continued)

| DATA for Validation of TFS and Direct Method | NS-03M [91] | NS-03T [91] | NS-30A [91] | NS-30S [91] |
|---|---|---|---|---|
| No. Regenerators | 1 | 1 | 4 | 4 |
| TH [K] | | | 956 | |
| TL [K] | | | 333 | |
| Regenerator type | screens | screens | screens | screens |
| **REGIME 1** | | | | |
| Nr [rot/min] | 650 | 850 | 1000 | 1120 |
| Power, real [kW] | 2.03 | 3.08 | 23.2 | 30.9 |
| Efficiency [%] | 35.8 | 32.6 | 37.5 | 57.2 |
| **REGIME 2** | | | | |
| Nr [rot/min] | 1400 | 1300 | 1500 | 1800 |
| Power, real [kW] | 3.81 | 4.14 | 30.4 | 45.6 |
| Efficiency [%] | 31 | 30.3 | 33 | 35.2 |

*Table 4*

**Data used for Validation of TFS and Direct Method [89, 90, 93]**

| DATA | STM-4-120 [93] | V-160 [93] | 4-95 MKII [93] | 4-275 [93] | FP-9 kW [93] | GPU-3 [90] | MP-1002 CA [90] | FP RE-1000 [89] |
|---|---|---|---|---|---|---|---|---|
| No. cylinders | 4 | 1 | 4 | 4 | 1 | 1 | 1 | 1 |
| $\varepsilon = V_2/V_1$ | 1.53 | 1.53 | 1.4 | 1.68 | 1.56 | 1.21 | 1.53 | 1.45 |
| Diam. cyl. [mm] | 56 | 68 | 55 | 60 | 140 | 69.9 | 55 | 220 |
| Stroke [mm] | 48.5 | 44 | 40 | 40 | 14.1 | 31.2 | 27 | 18 |
| Nr [rot/min] | 1800 | 1500 | | 1500 | 3600 | 2500 | 2220 | 1700 |
| P med [bar] | 120 | 150 | 200 | 150 | 40 | 41.3 | 15 | 40 |
| No. screens in Reg. | 400 | | 200 inox | | | 308 | 200 | |
| TH [K] | 993 | 903 | 993 | 893 | 902 | 977 | 873 | 798 |
| TL [K] | 343 | 333 | 323 | 338 | 333 | 288 | 333 | 313 |
| Regenerator type | Scr. | | Scr. | | | Scr. | Scr. | |
| Cooling | $H_2O$ | $H_2O$ | $H_2O$ | | | | | |
| Power, real [kW] | 25 | 9 | 28.27 | 50 | 9 | 3.96 | 0.20 | 0.94 |
| Power, comp. [kW] [60,70] | 26.1 | 8.93 | 27.1 | 47.4 | 9.00 | 3.58 | 0.199 | 0.93 |
| Effic., real [%] | 40 | 30 | 41 | 42 | 33 | 12.7 | 15.6 | |
| Gas in the Engine | | He | | He | | He | Air | |

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