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OPTIMIZATION OF THE IRREVERSIBLE DIESEL CYCLE USING FINITE SPEED THERMODINAMICS AND THE DIRECT METHOD

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Abstract: *The paper presents a Direct Method Approach in estimating the performances of the irreversible Diesel cycle with finite speed. The irreversibilities generated by the finite speed of the processes, throttling and friction were taken into account. The efficiency and the power output of the Diesel cycle were compared for several working gases, while studying the influence of the piston speed and volume increase ratio at combustion. Optimum values, as well as limit values of these variables resulted.*

Key words: *Diesel Cycle Performances, Finite Speed Thermodynamics, Direct Method, Irreversibility Causes.*

1. Introduction

Equilibrium processes can be studied using the Classical Reversible Thermodynamics. However, the equilibrium conditions restrain the study to the reversible approach, which means to processes that occur either in infinite time, or with the speed tending to zero.

In reality, thermal processes are developing in finite time [1-3] or very fast, their speed (w) determining the gap between reversible and real irreversible processes [4-16].

The objective of this paper is to study the performances (Efficiency and Power) of the Irreversible Diesel Cycle, in the framework of the Irreversible Thermodynamics with Finite Speed. The ireversibility factors taken into account were:

- The Finite Speed of the Piston (FSIT) during compression and expansion strokes;
- The Mechanical Friction between the piston and the cylinder, generating *piston friction pressure losses* (PFPL) all around the cycle;
- The Fluid Friction due to Throttling of the Gases during intake and exhaust strokes, generating *throttling pressure losses* (THPL).

2. Irreversible Processes Approach by the Direct Method in Thermodynamics with Finite Speed

Several models for determining the mathematical expression of the *Irreversible Mechanical Work* in Finite Speed Processes were developed, starting with the kinetic-molecular explanation model introduced by W. Macke [17] and A. Sommerfeld [18], and also using the

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Onsager-Prigogine Linear Irreversible Phenomenological Thermodynamics (LIPT) [19,20].

a) S. Petrescu and L. Stoicescu [9,11] have got the first expression of the *pressure on a moving piston (p_p) with finite speed (w):*

$$p_p = p_{m,i}\left(1 \pm \frac{a\,w}{c} + \frac{b\,w^2}{c^2} \pm ...\right) \qquad (1)$$

where:

$p_{m,i}$ - *the instantaneous mean pressure in the gas*;

a and b - coefficients, $a = 2$, $b = 5$;

$c = \sqrt{3RT_{m,i}}$ m/s] - the average speed of the molecules computed at *instantaneous mean temperature, $T_{m,i}$.*

The sign (+) is for compression and the sign (-) is for expansion processes. R is the constant of the gas.

By substituting equation (1) in the expression of the elementary work ($p_p \cdot dV$) the following relationship for *irreversible finite speed work* has been obtained:

$$\delta W_{irr} = p_{m,i}\left(1 \pm \frac{a\,w}{c} + \frac{b\,w^2}{c^2} \pm ...\right) \cdot dV = p_p\, dV \qquad (2)$$

Equation (2) was the first mathematical expression obtained for *Irreversible Work in Processes with Finite Speed*, for simple closed systems (piston machines - like internal combustion engines and piston compressors).

b) The Advanced Kinetic-Molecular Model [9,11,21] based on the Maxwell-Boltzmann distribution of the molecules speed and the *finite relaxation time* in the system, led to the expression:

$$\delta W_{irr} = p_{m,i}\left(1 \pm 2.764\frac{w}{c} + 3\frac{w^2}{c^2} \pm 1.283\frac{w^3}{c^3} + ...\right) \cdot dV \qquad (3)$$

This was the second mathematical expression for the *Irreversible Work in Processes with Finite Speed*

c) The Phenomenological Model of the interaction between piston and gas [9, 19, 20, 22] based on the hypothesis of pressure wave propagation and their relaxation generated by the Finite Speed of the piston and the speed of the sound in the gas [12,23], led to:

$$\delta W_{irr} = p_{m,i}\left(1 \pm \frac{a\,w}{c}\right) \cdot dV \qquad (4)$$

where: $a = \sqrt{3\gamma}$ and γ is the specific heat ratio.

Equation (4) can also be written as a function of the *speed of sound*, $c_S = \sqrt{\gamma RT_{m,i}}$, instead of the *average speed of the molecules, c*, leading to:

$$\delta W_{irr} = p_{m,i}\left[1 \pm \gamma\left(\frac{w}{c_S}\right)\right] \cdot dV \qquad (5)$$

d) The Onsager-Prigogine [19,20] Linear Irreversible Phenomenological Thermodynamics (LIPT) model [15], applied to finite speed interactions between the gas and piston, led to the expression [9,10]:

$$\delta W_{irr} = p_{m,i}\left(1 \pm K_1 \cdot w\right) \cdot dV \qquad (6)$$

Here K_1 is a constant (called *the Phenomenological Coefficient*) depending on the properties of the gas and *the instantaneous mean temperature of the gas, $T_{m,i}$.* Comparing equation (6) with equation (2), K_1 can be expresses as a function of R and $T_{m,i}$:

$$K_l = \frac{2}{\sqrt{3RT_{m,i}}} \quad \text{[s/m]} \qquad (7)$$

By consequence, when the expression of the irreversible mechanical work eq. (2) is replaced in the First Law of Thermodynamics ($dU = \delta Q - \delta W_{irr}$), the obtained mathematical expression becomes *a combination of the First and Second Laws of Thermodynamics* for Finite Speed Processes in closed systems [9-12]:

$$dU = \delta Q - p_{m,i}\left(1 \pm \frac{a\,w}{c} + \frac{b\,w^2}{c^2} \pm \ldots\right) \cdot dV \quad (8)$$

In this expression (8) there are different signs for compression (+) and expansion (-) as a consequence of the *primarily cause of irreversibility* in the processes with finite speed.

e) The other two causes of irrevesibilities (friction and throttling), as manifestations of the Second law of Thermodynamics in real irreversible processes, specific to piston machines, were similarly expressed in a *Generalized Equation for Irreversible Work* [12,13]:

$$\delta W_{irr} = p_{m,i}\left(1 \pm \frac{a\,w}{c} + b\frac{\Delta p_{th}}{2p} \pm \frac{\Delta p_f}{p}\right) \cdot dV \quad (9)$$

with Δp_{th} - *the pressure drop due to gas throttling* at intake and exhaust

Δp_f - the effect of *mechanical friction* between piston and cylinder walls.

With equation (9) introduced in the First Law we get a *General Equation for Irreversible Processes in closed systems generated by Finite Speed, Friction and Throttling:*

$$dU = \delta Q - p_{m,i}\left(1 \pm \frac{a\,w}{c} + b\frac{\Delta p_{th}}{2p} \pm f\frac{\Delta p_f}{p}\right) \cdot dV \quad (10)$$

where f is the fraction of heat generated by friction that remains in the working gas.

The last mathematical expression combines the First and Second Laws of Thermodynamics. It contains the main 3 causes of internal irreversibility (Finite Speed, Friction and Throttling) for any piston-cylinder real machine [12,13]. So, we call equation (10) ***The Fundamental Equation of Thermodynamics with Finite Speed.***

3. Irreversible Processes Approach by Finite Speed Thermodynamics

To study the finite speed process, both kinds of pressure are necessary (instead of one for the classical reversible process).

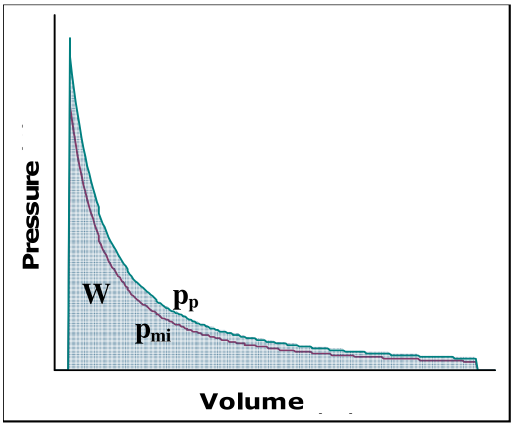

Fig. 1. *Irreversible Work at Compression*

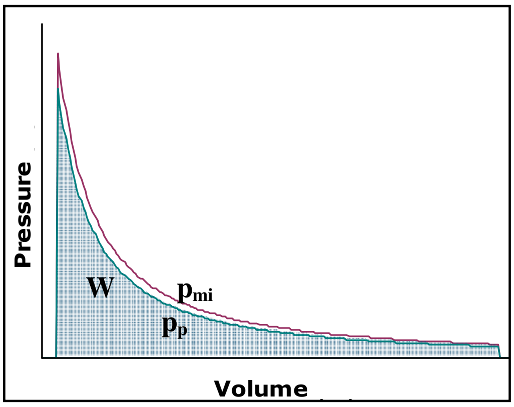

Fig. 2. *Irreversible Work at Expansion*

Their evolution during compression and expansion are described in *p-V* coordinates by the curves [19,20] shown in Figure 1 and 2.

During compression, the pressure on the piston will be higher than the pressure at any other point in the system (reaching its minimum on the cylinder head). During expansion, the opposite happens, namely the pressure on the piston will be less than in any other point of the system (reaching its maximum on the cylinder head).

In the irreversible thermodynamics, the process work is equal with the area under the pressure on the piston (p_p) curve, in *p-V* diagram.

4. Adiabatic Processes with Finite Speed

The equations of the *irreversible adiabatic* ($\delta Q=0$) *finite speed compression (+) and expansion (-) processes* were obtained [5,9,14-16] by analytical integration of equation (8):

$$T_1\left(1\pm\frac{a\,w}{c_1}\right)^2\cdot V_1^{\gamma-1}=T_2\left(1\pm\frac{a\,w}{c_2}\right)^2\cdot V_2^{\gamma-1} \quad (11)$$

$$p_1\left(1\pm\frac{a\,w}{c_1}\right)^2\cdot V_1^{\gamma}=p_2\left(1\pm\frac{a\,w}{c_2}\right)^2\cdot V_2^{\gamma} \quad (12)$$

$$\frac{T_1\left(1\pm\frac{a\,w}{c_1}\right)^{\frac{2}{\gamma}}}{p_1^{\frac{\gamma-1}{\gamma}}}=\frac{T_2\left(1\pm\frac{a\,w}{c_2}\right)^{\frac{2}{\gamma}}}{p_2^{\frac{\gamma-1}{\gamma}}} \quad (13)$$

With these equations the Direct Method is used in order to obtain the performances (Efficiency and Power) of Diesel irreversible cycle with Finite Speed.

5. Work Loss due to FSIT

The work loss due to FSIT is given by the difference between the work for the reversible cycle ($W_{cycle\ r}$) and that for the irreversible one ($W_{cycle\ ,ir}$):

$$W_{cycle,\ loss,\ FSIT}=W_{cycle,r}-W_{cycl,irr} \quad [\text{J}] \quad (14)$$

6. Work Loss due to the Throttling

Throttling occurs when a fluid passes through small orifices like valves. The corresponding loss of pressure during intake and exhaust reduces the actual cycle work. The total throttling pressure loss per cycle is estimated from [13]:

$$\Delta p_{THPL,cycle}=0.45\cdot\left(\frac{w}{100\cdot St}\right)^2 \quad [\text{bar}] \quad (15)$$

where St is the piston stroke.

The irreversible work loss due to throttling is determined multiplying equation (15) by the displacement volume (V_S):

$$W_{THPL,\ cycle}=\Delta p_{THPL,cycle}\cdot V_S \quad [\text{J}] \quad (16)$$

7. Work Loss due to the Friction

The friction generated heat during the relative motion of the piston inside the cylinder is another important source of irreversibility. It increases gas temperature and pressure, as compared to the reversible process. The total friction pressure loss per cycle is estimated from [23]:

$$\Delta p_{PFPL,\ cycle}=(0.97+0.045\cdot w) \quad [\text{bar}] \quad (17)$$

which multiplied by the displacement volume (V_S), gives the irreversible friction work loss:

$$W_{PFPL,\ cycle}=\Delta p_{PFPL,cycle}\cdot V_S \quad [\text{J}] \quad (18)$$

8. Cycle Efficiency

The Diesel cycle efficiencies could be determined taking into account, step by step, different kinds of irreversibilities. The following relations have been used:

- Reversible Cycle Efficiency (no irreversibility)

$$\eta_r = \frac{W_{cycle,\, r}}{Q_{in}} \quad (19)$$

- FSIT Irreversible Cycle Efficiency

$$\eta_{irr,\, FSIT} = \frac{W_{cycle,\, irr}}{Q_{in}} \quad (20)$$

- FSIT & THPL Irreversible Cycle Efficiency

$$\eta_{irr,\, FSIT\&THPL} = \frac{W_{cycle,\, irr} - W_{cycle,\, loss,\, THPL}}{Q_{in}} \quad (21)$$

- Actual Irreversible Cycle (FSIT & THPL & PFPL) Efficiency:

$$\eta_{irr,\, FSIT} = \frac{W_{cycle,\, irr,\, act}}{Q_{in}} \quad (22)$$

where the Irreversible Cycle Effective Work is:

$$W_{cycle,\, irr,\, act} = W_{cycle,\, r} - W_{cycle,\, loss,\, FSIT} - W_{cycle,\, loss,\, THPL} - W_{cycle,\, loss,\, PFPL} \quad [J] \quad (23)$$

9. Results and Conclusions

In this work the engine is considered as adiabatically insulated, in accordance with the conditions offered by the ceramic engine, a high efficiency energetic solution under intensive research at present. Using the equations for irreversibile processes previously deduced within the framework of the Finite Speed Thermodynamics, the processes composing the Diesel cycle are covered successively, determining the work (W) and the heat exchange (Q) for each one, finally getting to the performances: the Efficiency (η) and the produced Power (P).

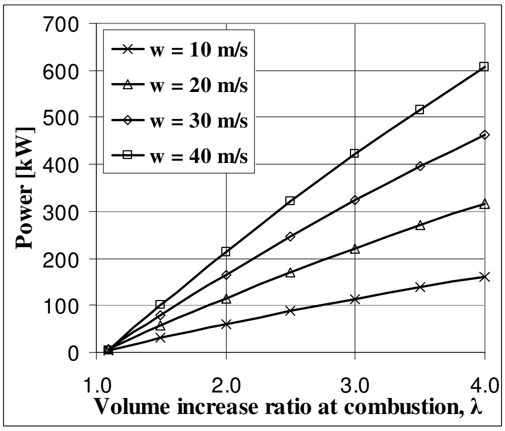

Fig. 3. *Power versus volume increase ratio at combustion, for air, at different mean piston speeds*

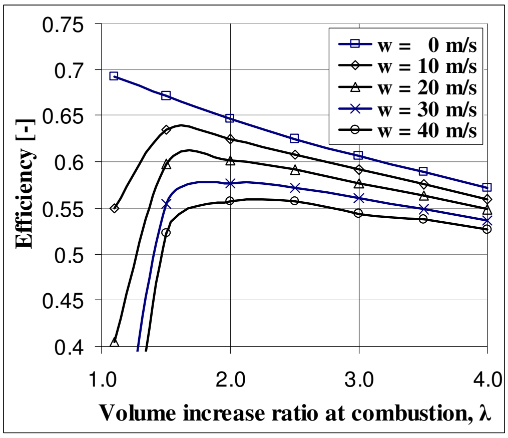

Fig.4. *Efficiency versus volume increase ratio at combustion, at different mean piston speeds (air).*

Figures 3 and 4 are showing the influence of the volume increase ratio at combustion (λ) on the power output and, respectively, on the efficiency of the irreversible Diesel cycle, taking into account the work loss due to the finite speed of the piston alone.

While the power grows with the volume increase ratio (λ), the influence of the mean piston speed is opposite (see Figure 3).

The efficiency (η) - volume increase ratio (λ) curve (Figure 4) has a different shape: for the studied piston speeds > 0, it shows a maximum point, with a sudden decrease towards lower values of λ.

Another important observation is that there is a certain value of λ under which the produced work cannot cover the irreversibility losses ($\lambda_{min} \approx 1.3$ for air at w = 30 m/s, for instance).

Taking into account step by step, different kinds of irreversibility (Figure 5), the maximum of the efficiency migrates towards higher values of λ, as well as the "extinction" point (λ_{min}).

In the irreversible approach, the power-piston speed curve (Figure 6) is a parabola, too. By consequence, the same power can be reached at 2 different piston speeds (with lower efficiency at the higher piston speed).

Taking into account progressively different kinds of irreversibility (Figure 6), the maximum of the power curve, is decreasing, as well as the value of the corresponding optimum speed.

This diagram (Figure 6) reveals also that there is a higher limit corresponding to the maximum affordable speed over which the engine couldn't cover anymore the work loss due to irreversibility.

A similar influence of the piston speed on the power and the efficiency of the Diesel engine have been obtained for He, CO_2 and H_2 (see Figures 7 and 8).

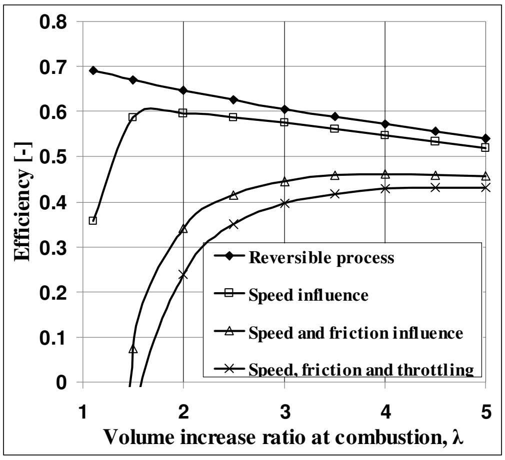

Fig. 5. *Efficiency versus volume increase ratio at combustion for air at w* = 20 *m/s*

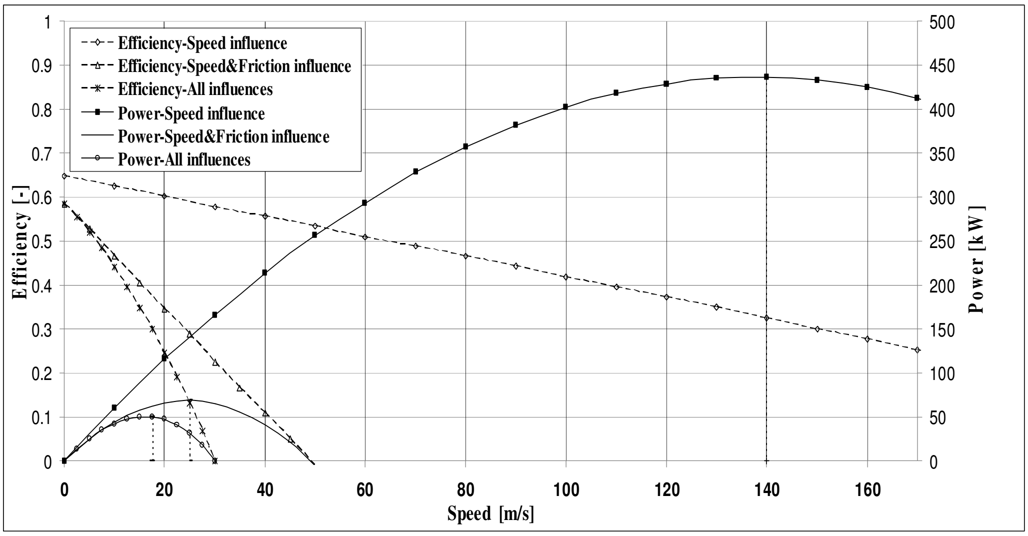

Fig. 6. *Piston speeds' influence on power & efficiency of a Diesel* (*air,* λ = 1.8)

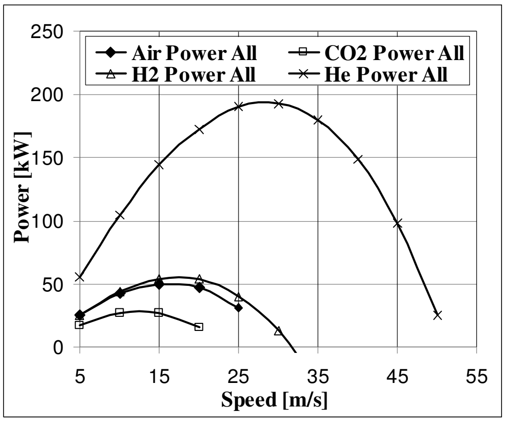

Fig. 7. *Power output versus mean piston speed for different working gases (all three irreversibility causes)*

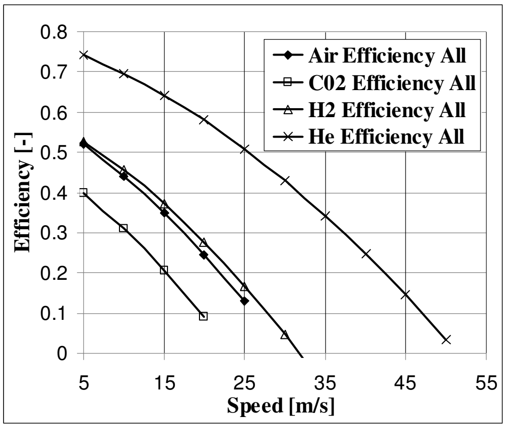

Fig. 8. *Efficiency versus mean piston speed for different working gases (all three irreversibility causes)*

The same behavior has been observed at different values of $\lambda = 1.1 \div 4$.

For the same λ, each working gas has its own optimum speed, corresponding to the maximum of the power output (Figure 7), as well as a maximum (unsurpassable) working speed.

This kind of analysis gives to the designer the means to improve the engine performances by insisting on the reduction of its most significant work losses. It opens, also, the way towards sensitivity studies revealing the influence of different parameters (temperatures, speed, compression ratio, dimensions, etc.) on the engine performances and giving the opportunity for optimizing the working cycle with respect to these parameters [1-16]. It has also the "power" to compute the *Internal Source of Entropy* and correlate it with the *Performance Parameters of the Cycle.*

References

1. Petrescu, S., Harman, C., Costea, M., Feidt, M.: *Thermodynamics with Finite Speed versus Thermodynamics in Finite Time in the Optimization of Carnot Cycle*. In: Proceedings of the 6-th ASME-JSME Thermal Eng. Joint Conference, Hawaii, 2003.
2. Andersen, B.: *Finite-Time Thermo-dynamics, Physics Laboratory II*, University of Copenhagen, 1983.
3. Feidt, M.: *Thermodynamique et optimisation énergétique*. In : *Technique et Documentation Lavoisier*, Paris, 1996.
4. Petrescu, S., Costea, M., Harman, C., Florea, T.: *Application of the Direct Method to Irreversible Stirling Cycles with Finite Speed.* In: *International Journal of Energy Research*, **26** (2002), p. 589.
5. Petrescu, S., Harman, C., Costea, M., Florea, T., Petre, C.: *Advanced Energy Conversion*, Bucknell University, Lewisburg, 2006.
6. Petrescu, S., Stănescu, G., Costea, M.: *A Direct Method for the Optimization of Irreversible Cycles using a New Expression for the First Law of Thermodynamics for Processes with Finite Speed.* In: *Proc. of the 1st Int. Conf. on Energy* ITEC'93, Marrakech, 1993, p. 650.
7. Petrescu, S., Feidt, M., Harman, C., Costea, M.: *Optimization of the Irreversible Carnot Cycle Engine for*

Maximum Efficiency and Maximum Power through Use of Finite Speed Thermodynamic Analysis, In: Proceedings of the ECOS'2002 Conference, Berlin, 2002, Vol. II, p. 1361.

8. Stoicescu, L., Petrescu, S.: *Thermo-dynamic Cycles with Finite Speed*, In: Polytechnic Institute of Bucharest Bulletin **XXVII** (1965) No. 2, p. 82.
9. Petrescu, S.: *Contributions to the study of interactions and processes of non-equilibrium thermal machines*. In: Ph.D. Thesis, Polytechnic Institute of Bucharest, 1969.
10. Petrescu, S.: *An Expression for Work in Processes with Finite Speed Based on Linear Irreversible Thermo-dynamics*, In: *Studii şi Cercetări de Energetică şi Electrotehnică*, **19** (1969) No. 2, Romanian Academy, p. 249.
11. Stoicescu, L., Petrescu, S.: *The First Law of Thermodynamics for Processes with Finite Speed, in Closed.* In: Polytechnic Institute of Bucharest Bulletin, **XXVI** (1964), No. 5, p. 87.
12. Petrescu, S., Harman, C.: *The Connection between the First and Second Law of Thermodynamics for Processes with Finite Speed. A Direct Method for Approaching and Optimization of Irreversible Processes*, In: *Journal of the Heat Transfer of Society of Japan*, **33-128** (1994), p. 60.
13. Petrescu, S., Stănescu, G., Iordache, R., Dobrovicescu, A.: *The First Law of Thermodynamics for Closed Systems, Considering the Irreversibilities Generated by the Friction Piston-Cylinder, the Throttling of the Working Medium and Finite Speed of the Mechanical Interaction.* In: ECOS'92 Conference, Zaragoza, Spain, 1992, p. 33.
14. Stoicescu, L., Petrescu, S.: *Experimental verification of the new expression of the First Law of Thermodynamics for processes with finite speed.* In: Polytechnic Institute of Bucharest Bulletin, **XXVII** (1965) No. 2, p. 97.
15. Stoicescu, L., Petrescu, S.: *Thermodynamic Processes with Constant Finite Speed.* In: Polytechnic Institute of Bucharest Bulletin, **XXVI** (1964) No. 6, p. 79.
16. Stoicescu, L., Petrescu, S.: *Thermo-dynamic Processes with Variable Finite Speed.* In: Polytechnic Institute of Bucharest Bulletin, **XXVII** (1965) No. 1, p. 65.
17. Macke, W.: *Thermodynamik und Statistik*. Leipzig, 1963.
18. Sommerfeld, A., *Thermodynamics and Statistical Mechanics*. Vol. 5, New York. Academic Press, 1967.
19. Onsager, L.: *Phys. Rev.* **37** (1931), p. 405; **38** (1938), p. 2265.
20. Prigogine, I.: *Introduction to thermodynamics of irreversible processes*. Charles Thomas, Publisher, Springfield, IL, 1955.
21. Petrescu, S.: *Kinetical considerations regarding the pressure on a piston with finite speed.* In: *Studii şi Cercetări de Energetică şi Electrotehnică* **21** (1971) No. 1, Romanian Academy, p. 93.
22. Carnot, S.: *Reflections sur la puissance motrice du feu et sur les machines propres à développer cette puissance*. Paris. 1824.
23. Heywood, J.B.: *Internal Combustion Engine Fundamentals*. New York. McGraw-Hill Book Company, 1988.