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APLICATION OF THE DIRECT METHOD TO OTTO AND DIESEL IRREVERSIBLE CYCLES

Traian FLOREA[1]
Stoian PETRESCU[2]
Corneliu MOROIANU[3]
Monica COSTEA[4]
Ligia-AdrianaSPORIS[5]
M. FEIDT[6]
[1]Dep. IMAN, Prof.eng. PhD Naval Academy "Mircea cel Batran", Constanta, 1 Fulgerului Street
[2]Politehnica University Bucharest, Prof.eng. PhD
[3]Dep. IMAN, Associate Prof., PhD Naval Academy "Mircea cel Batran", Constanta, 1 Fulgerului Street
[4]Politehnica University Bucharest, Prof.eng. PhD
[5]Dep. IMAN, Lecturer, PhD Naval Academy "Mircea cel Batran", Constanta, 1 Fulgerului Street
[6]Laboratoire d'Énergétique et de Mécanique Théorique et Appliquée, Vandœuvre, France

Abstract: *The Direct Method from Thermodynamics with Finite Speed (TFS) studies the irreversibility's (internal and external) produced during operation of real thermal machines, through progressive analysis and* ***direct*** *integration of the First Law of Thermodynamics, combined with the Second Law of Thermodynamics for processes with Finite Speed, for each process of the cycle. Thus are obtained analytical expressions for the Efficiency* ***(****for Power cycles****),*** *respectively COP (for Refrigeration Machines and Heat Pumps) and Power (produced, respectively consumed) function of the speed of the processes and of the functional and geometrical parameters of the machine. This paper presents the Main Moments in the Development of Thermodynamics with Finite Speed and Direct Method "invented" in its framework. Recent Progresses in Application of the Direct Method to Otto and Diesel cycles are presented.*

The influence of several *irreversibility factors* on the Direct Cycle *Efficiency* and *Power Output*, was studied in the framework of the ***Irreversible Thermodynamics with Finite Speed***, using the ***Direct Method***:

- The *Finite Speed (w) of the Piston* (FSIT) during compression and expansion [1 - 10].
- The Mechanical Friction between the piston and the cylinder, generating *piston friction pressure losses* (PFPL) all around the cycle [141], [142].
- The Fluid Friction due to Throttling of the Gases during intake and exhaust, generating *throttling pressure losses* (THPL) [141], [142].

One of the studies was conducted onto an irreversible Otto cycle [87]. In fig.1, one can notice that the reversible approach leads to a linear increase of the power output with the piston speed, while in the irreversible approach the power curve is a parabola, its maximum corresponding to the optimum speed. Taking into account step by step, different irreversibility's, the maximum power output is decreasing, as well as the corresponding optimum speed (from 145 kW at about 150 m/s, to about 20 kW at 20 m/s, fig.1).

A similar study conducted on the adiabatic Diesel cycle with constant pressure combustion [88] showed (fig.2)in this case also, that while for the reversible adiabatic cycle the power curve is linear, for the irreversible cycle it gets parabolic in shape.

Another important observed issue was that the difference between the reversible and the irreversible performances (efficiency and power) significantly increases with the increase of the piston speed.

Similar results were obtained in the case of Semi – Diesel cycle (constant volume heating first, then constant pressure), fig.3, taking into account the same initial pressure, the same compression ratio and same combustion heat for both reversible and irreversible cycles[90,91].

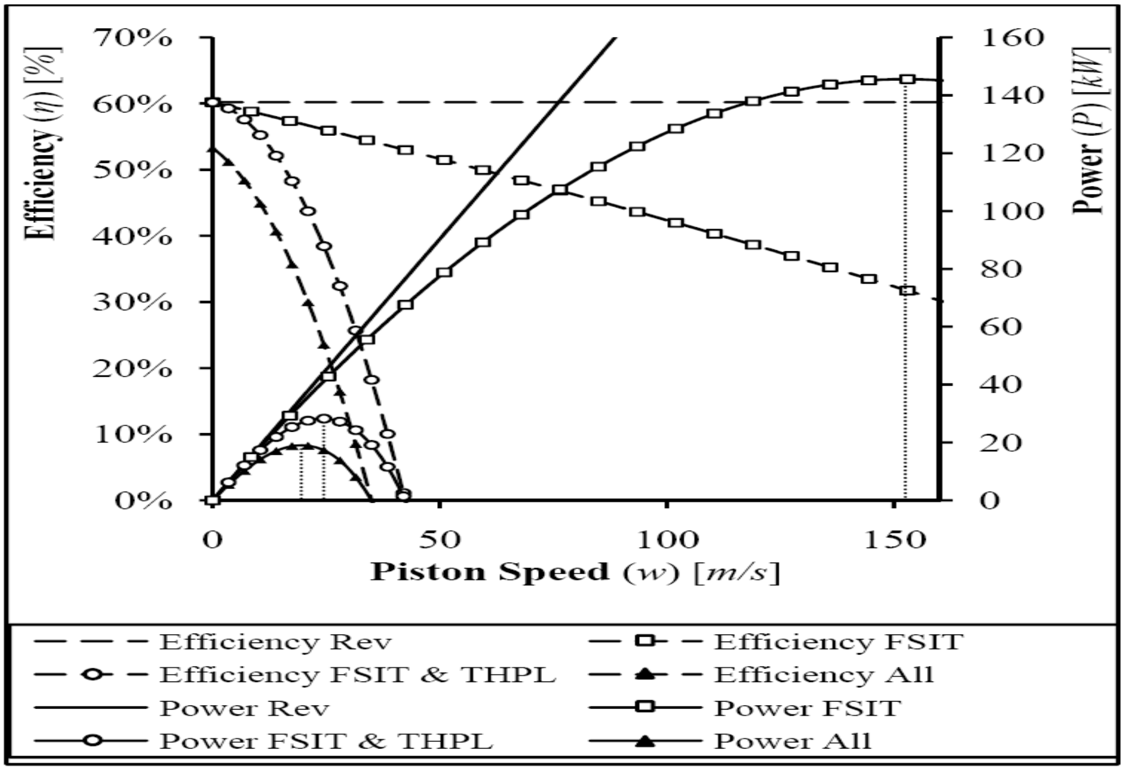

Fig.1. Otto Cycle Efficiency and Power versus Piston Speed [87].

Considering the *throttling* and *friction* irreversibility's, the Semi-Diesel cycle efficiency with finite speed, is given by[90]

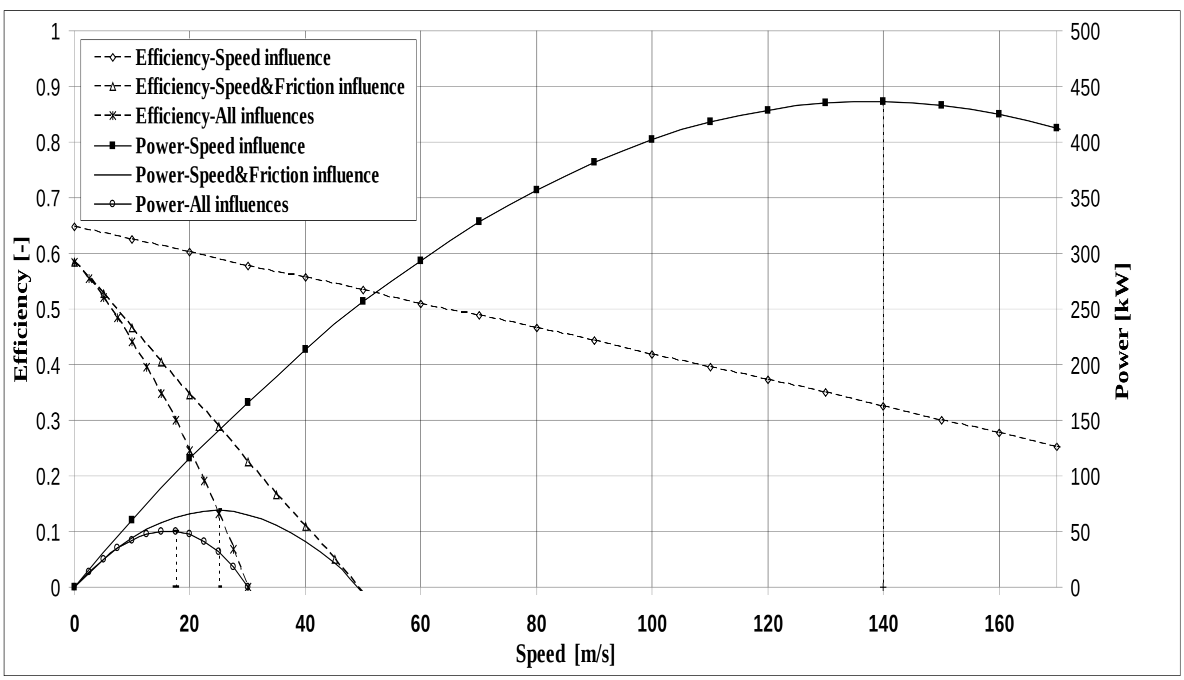

Fig.2. Diesel Cycle Efficiency and Power versus Piston Speed [88]

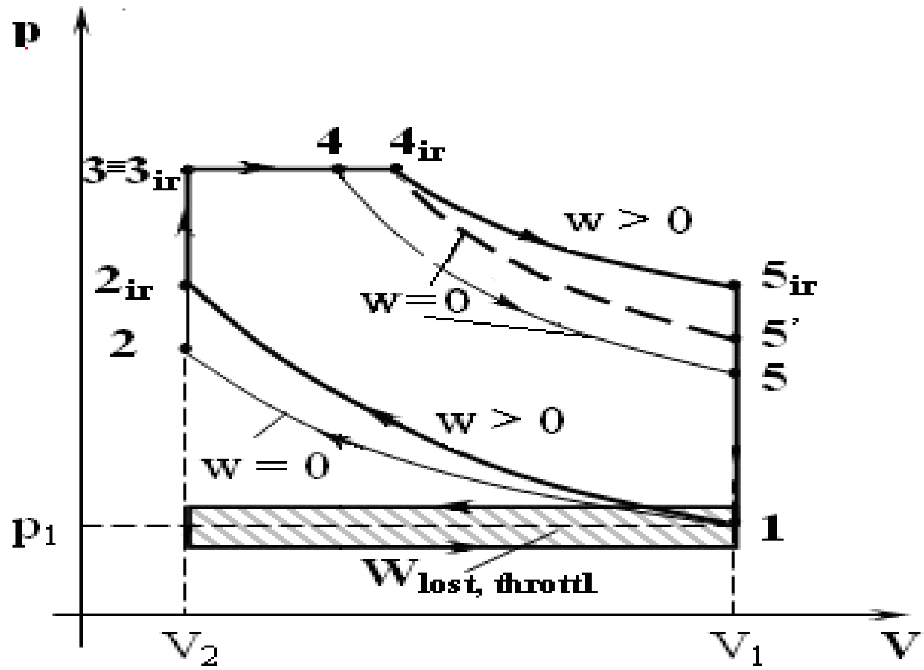

Fig.3. Comparison between Semi – Diesel reversible (w = 0) and irreversible (w > 0) cycle [90, 91]

$$\eta_{ir,\,f,\,thr} = 1 - \frac{1}{\varepsilon^{k-1}} \frac{\dfrac{{\alpha_{ir}}^{k} \cdot \lambda \cdot T_1}{T_1}\left[1 - \dfrac{a \cdot w}{\sqrt{3R \cdot \alpha_{ir} \cdot \lambda \cdot \varepsilon^{k-1} \cdot T_1}}\left(1 - \left(\dfrac{\varepsilon}{\alpha_{ir}}\right)^{\frac{k-1}{2}}\right)\right]^2 - 1}{k \cdot \lambda(\alpha - 1) + \lambda - 1} -$$

$$- \frac{\left(\Delta P_f + \Delta P_{thr}\right)\Delta V}{mc_v\left[\left(T_3 - T_2\right) + k\left(T_4 - T_3\right)\right]} \quad (4)$$

while the Power output in all reversible and irreversible cases, (for a 4 cylinders - 4 strokes Diesel engine) has been determined using relation:

$$\left.\begin{aligned} P_{ir} &= 4W_{cycle} \cdot \frac{n_r}{2 \cdot 60} \\ W_{cycle} &= \eta \cdot Q_{in} \\ n_r &= \frac{30 \cdot w}{z} \end{aligned}\right\} \Rightarrow P_{ir} = 4\eta \cdot Q_{in} \cdot \frac{30 \cdot w}{120 \cdot z} \Rightarrow \quad P_{ir} = \eta \cdot Q_{in} \cdot \frac{w}{z} \quad (5)$$

The influence of the *piston speed w*, *compression ratio, ε,* the *pressure increaseratio,λ* (at constant volume) and the *cut-off ratio,α*, has been studied, *too.*

The diagram form fig. 4 shows the influence of the finite speed (between 0 and 50 m/s) on the Efficiency of the Diesel engine. While for the reversible cycle the efficiency remains constant (at about 65 %), the irreversible cycle efficiency is decreasing with the piston speed obviously and it may even vanish, when the maximum affordable mean piston speed is reached (around 42 m/s, with all irreversibility's considered).

The power output diagram from figure 5 shows a linear increase with the mean piston speed in the case of the reversible cycle, while in the irreversible approach, the Power-Piston speed curve is a parabola. By consequence, the same power can be reached at two different piston speeds (with lower efficiency at the higher piston speed).

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

The same diagram (fig. 5) also reveals the existence of the maximum affordable speed where the Power output drops to zero, in good correspondence with that observed in the Efficiency diagram (42 m/s with all irreversibility's considered, see fig. 4).

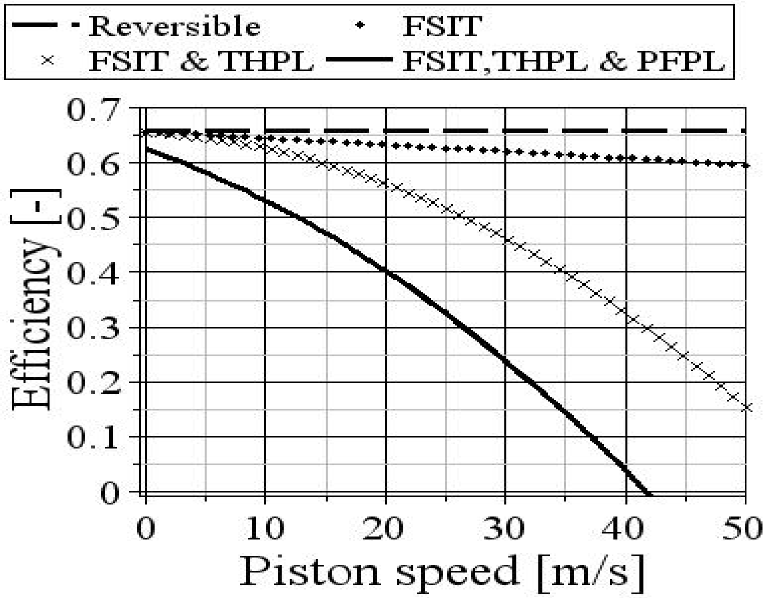

Fig.4. Efficiency of reversible and irreversible Diesel cycles at ε = 20, λ = 1.5 and α = 2,[91]

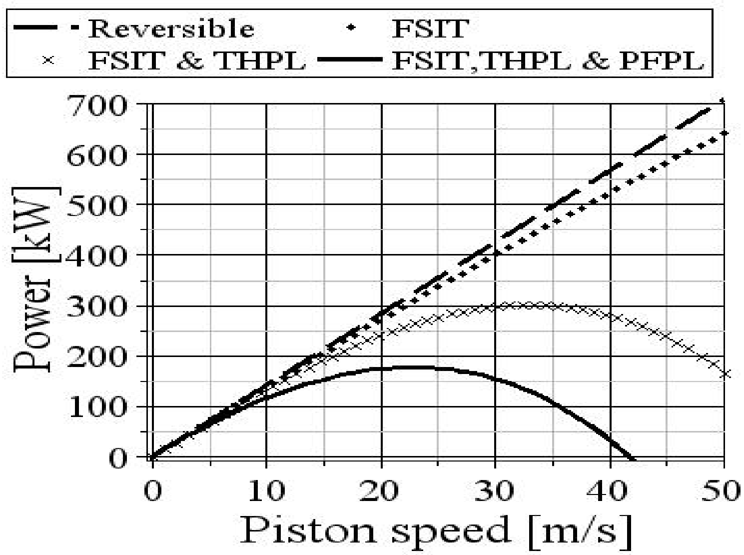

Fig.5. Power of reversible and irreversible Diesel cycles at ε = 20, λ = 1.5 and α = 2 [91]

Figure 6 shows that the Efficiency and the Power output grow with the *compression ratio* ε, as well as the maximum affordable speed, the maximum power and its corresponding optimum mean piston speed.

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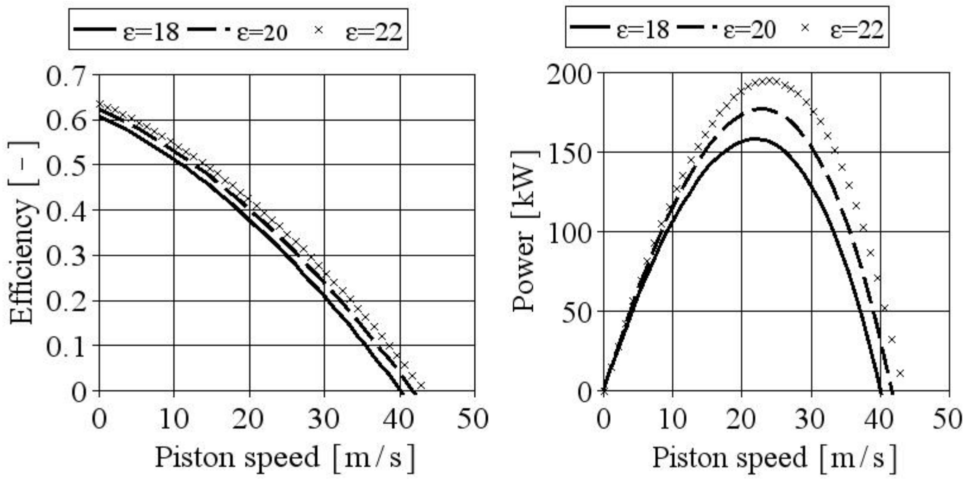

Fig.6. Efficiency and Power output of reversible and irreversible Semi-Diesel cycles at α = 2, λ = 1.5 and ε = 18 - 20 [91]

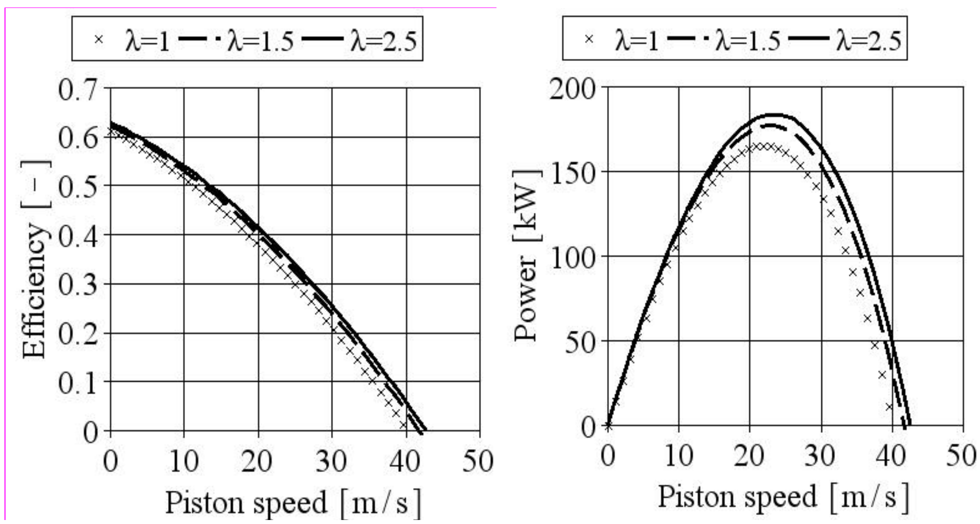

Fig.7. Efficiency and Power output of reversible and irreversible Semi-Diesel cycles at α = 2, ε = 20 and λ = 1 - 2.5 [91]

Figure 7 shows that the growth of the *pressure increaseratio, λ* from 1 to 1.5 has a significant contribution to the growth of the engine Efficiency and the Power output, too.
The influence of the *cut-off ratio,α* on the performances of the Diesel engine is presented in the diagrams from figure 8, where their growth is obvious. It is to be noticed, however, that α and λ are in competition, as one's growth implies the diminution of the other (to keep constant the total heat input).

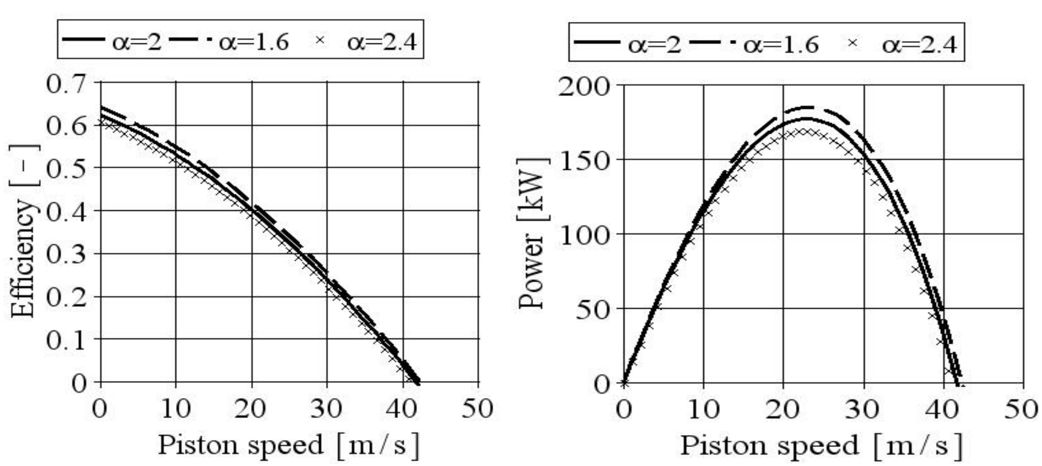

Fig.8. Efficiency and Power output of reversible and irreversible Diesel cycles at ε = 20, λ = 1.5 and α = 1.8 - 2.2 [91]

The last part of the study focused on the Power and Efficiency of the Diesel cycle for various work fluids as air, H_2, He and CO_2.
On the graphs from figure 9 it is possible to remark similar performances for Hydrogen and Air. Helium yields the maximum efficiency, but the material requirements do not suggest it as a solution yet. However, based on this study, Helium seems to be a very good working fluid for an external combustion cycle.
An interesting conclusion is also that the Hydrogen's irreversible properties are closer to the reversible properties than those of the other three work fluids.

By emphasizing the most significant work losses, this kind of analysis gives to the designer the tools to efficiently improve the engine performances.
It opens the way towards sensitivity studies revealing the influence of different parameters (temperatures, speed, compression ratio, dimensions etc.) on the engine performances and the optimization of the engine cycle with respect to these parameters.

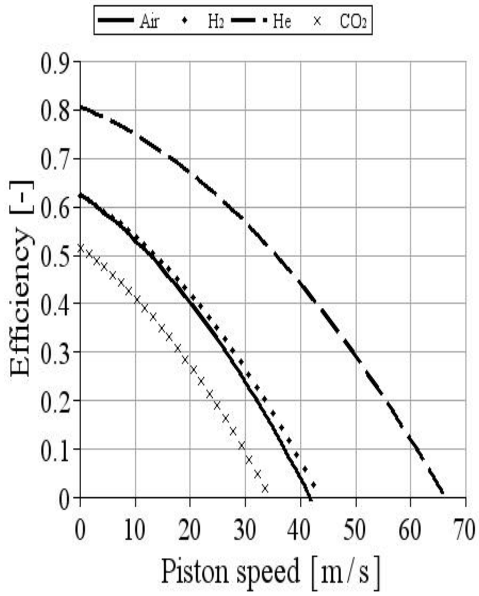

Fig.9. Efficiency Power output of Diesel cycle as function of piston speed for different work fluids [91]

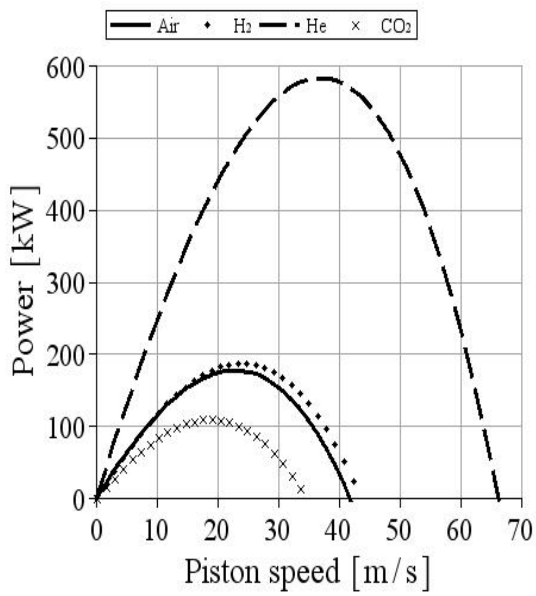

Fig.10**.** Schematic fluid flow through the orifice of the valve

The internal source of entropy can also be determined and correlated with the performance parameters of the cycle.
Based on experimental results, J.B. Heywood presented in [142], an integrated mathematical relationship expressing *the total friction mean effective pressure* loss (*mtf.mep*) as dependent on *the mean piston speed* (w_p) and *the crankshaft rotational speed(N)* has been deduced:

$$mtf.mep\ (kPa) = C_1 + 48 \cdot \left(\frac{N}{100}\right) + 0.4 \cdot w_p^2 \quad (6)$$

Heywood considered the losses due to the mechanical friction as linearly dependent on w_p(or N), while frictional losses in pumping fluids (cylinder gas and cooling air, cooling water and lubricating oil) are taken as proportional to w_p^2 (or N^2).
The *compression-expansion process with finite speed* is expected to add, also, a significant contribution in these *pressuredrops*. FST expresses these supplementary *losses* as proportional to w_p^2, too [100].
For the throttling intake Pressure Losses of the working fluid in the air filter, manifolds and valves, considering a *stationary and isentropic flow* the following relation has been deduced [100]:

$$\Delta P_{ud.i}[kPa] = 0.076 \cdot w_p^2 \quad (7)$$

Comparing with the experimentally obtained relation [100]:

$$\Delta P_f[kPa] = 0.4 \cdot w_p^2 \quad (8)$$

we get:

$$\Delta P_{ud.a}[kPa]/= 0.076/2/0.4 \cdot \Delta P_f[kPa] = 0.095 \cdot \Delta P_f[kPa] \quad (9)$$

For the Exhaust Pressure Losses the following relation was deduced [100]:

$$\Delta P_{ud.e}[kPa] = 0.083 \cdot w_p^2 \quad (10)$$

that is, using (8):

$$\Delta P_{ud.e}[kPa] /= 0.083\ /\ 2\ /\ 0.4 \cdot \Delta P_f[kPa] = 0.1 \cdot \Delta P_f[kPa] \quad (11)$$

The deduced equations showed [100] that the *Pressure Rates* of the *intake process* ($\chi_{u.d.i}=\Delta P_{u.d\,i}/P_{u\,i}$)and *exhaust process* ($\chi_{u.d.e} = \Delta P_{u.d\ e}\ /\ P_{u\ e}$) arefunction of '*geometrical factors*''*fluid characteristic factors*'and of the *squared speed of the piston* (w_p^2).
This work can be seen as a way of finding a correlation between the *empiricparameters* and the *phenomenological assessment* of the *intake and exhaust process*.

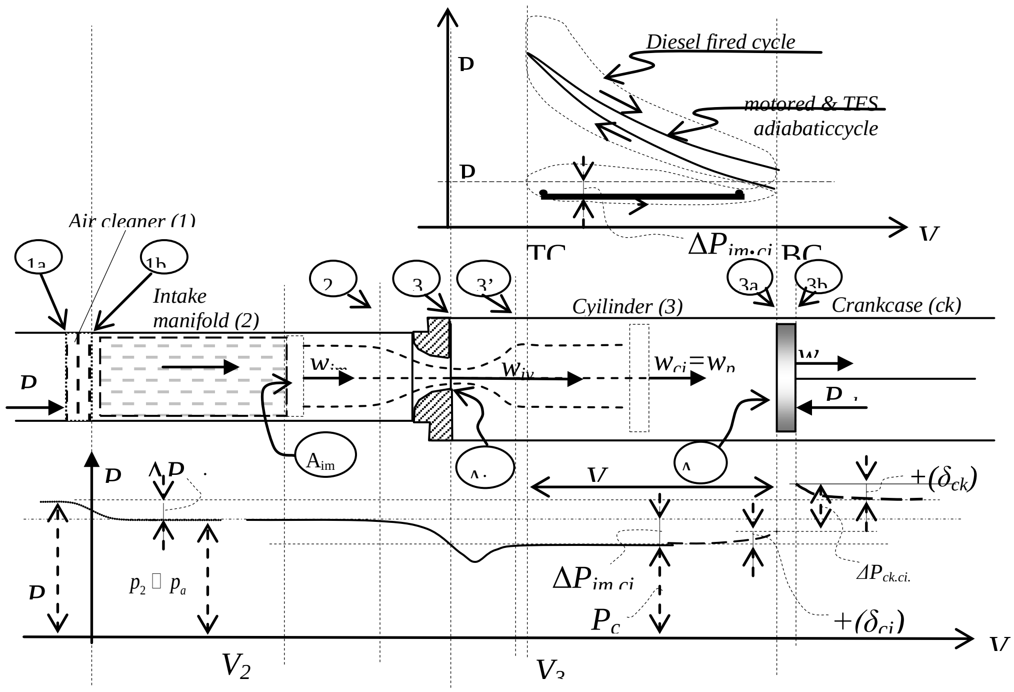

Fig.11. Fluid flow through Diesel intake premises [100]

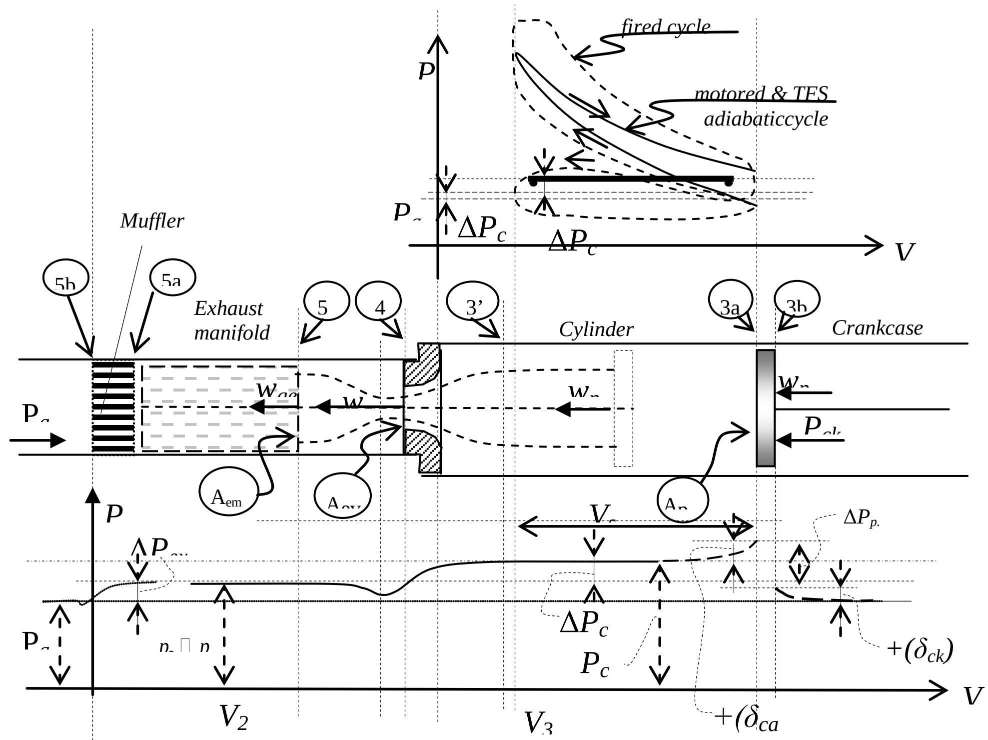

Fig.12. Fluid flow through the Diesel exhausts valve [100]

In these theoretical developments for modelling the *throttling in the valves*, a *stationary flow* was assumed and only the *average* *piston speed* was used. At *high piston speeds,* these assumptions may not be realistic, and the analysis needs to be improved in correlation with new experimental data, in order to emphasize, for instance, *cylinder gas* (*cg)* effects.

At this moment, this type of research is only at the beginning and it should be further developed in order to get a final Validation of all the relations, obtained using the Thermodynamics with Finite Speed and the Direct Method.

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