TWO STEPS FOR AN ENVIRONMENTAL FRIENDLY PROPULSION ENGINE

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ABSTRACT

A logical way to preserve the natural resources and lowering the pollution level is the improving of the thermal engine efficiency in order to reduce the fuel consumption. Considering the weight factor of the spark ignition engine in automotive market, the authors propose a combined solution to improve the performances of this thermal engine.

First, in order to improving the combustion efficiency, we are trying to obtain a closer approach to the ideal constant volume combustion cycle, specific to the spark ignition engine, by developing a variable sequential ratio engine.

Secondarily, by modifying the architecture of the compression ring, the theoretical model developed allows the determination of a new transversal profile of the compression ring in order to obtain better lubrication conditions, to lower friction ware and increase the mechanical efficiency.

This theoretical and experimental study regards the both modification of these parameters in various situations, aiming at the optimization of this environmental friendly propulsion engine.

KEYWORDS: combustion and mechanical efficiencies, environmental friendly propulsion engine

1. Introduction

The spark ignition engines for automotive propulsion still represent about 65%-70% of the world market, and about 85%-90% of the American market. On the other side, although the alternative (also called hybrid) propulsion solutions are becoming more and more popular, they are not widely spread. The American market, one of the largest fuel consumers in the world, is dominated by medium-large sized spark ignition engines with high fuel consumption. These types of engine have been improved step by step.

The fueling systems have been optimized; improved solutions for gases distribution have been used, the compression ratio has been modified, etc. This is an old area of interest and specialists proposed several solutions. Currently, the most popular solution [1], introduced in small production numbers, is the one developed by SAAB.

On the other hand, in the purpose of improving the spark ignition engines efficiency, the actual researches taken into account the amelioration of constructive and technological solutions.

2. Proposed Engine Solutions

2.1. Improving the operating cycle basis efficiency

The study developed by the authors has the goal of improving the efficiency of the spark ignition engine, in the first step, by improving the basis efficiency of its operating cycle. Moreover, one of the ways consists in obtaining a higher relative efficiency, i.e. a closer match to the ideal thermodynamic cycle (constant volume combustion). This goal can be achieved by assuring an almost constant volume during the combustion process.

An engine functioning in this manner will lead to a higher compression ratio close to the TDC, during the burning process (when the lowest heat quantity is developed). The compression ratio varies only in certain moments of the cycle, remaining unchanged for the rest of the cycle.

The solution proposed by the authors achieves the variation of the compression ratio by moving a small piston in the opposite direction to the main piston, during a 90 degree angular interval after the TDC, thus achieving an almost constant volume for the main part of the burning process. In the Fig. 1 this solution are shown for a position near the TDC.

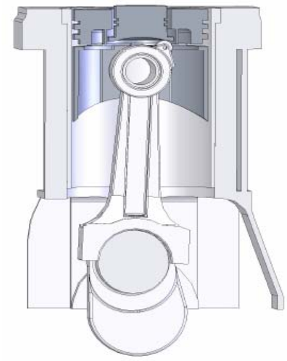

Fig.1. *Position near the TDC.*

2.2. Increase of the mechanical efficiency

The second step of the researches consist in the increase of the mechanical efficiency by improving the lubrication condition for the compression ring; this is the consequences of increased values for the minimum film thickness and reduced friction forces inside the cylinder liner-piston ring couple.

The modified compression rings developed by the authors are based upon the rectangular ring (ISO 6621-1); the rings are modified as their peripheral surface has a bi-conical shape.

3. Analysis and Modeling

Aiming to obtain results that would allow a comparison between real conditions and the proposed solution (in terms of differences between efficiencies), we developed a detailed analysis of this engine solution. For improving the basis efficiency of operating cycle, two separate cases were taken into account: the first one presumes a constant bore of the small piston, while displacing it between 1 and 10 mm. The second case preserves the stroke of the same small piston, while its bore varies between 44 and 97 mm. An analysis of the obtained efficiencies led us to the optimization of the mechanical parameters.

3.1 The basis efficiency of operating cycle analysis

First case modeling

The momentary stroke of the main piston is:

$$s(\alpha)=r\cdot\left[1-\cos(\alpha)+\frac{1}{\lambda}\left[1-\left(1-\lambda^2\sin(\alpha)^2\right)^{\frac{1}{2}}\right]\right] \quad (1)$$

and the momentary volume generated by the main piston is:

$$Vp(\alpha)=\frac{Vs}{S}\cdot s(\alpha) \quad (2)$$

The momentary volume of the cylinder is:

$$Vp(\alpha)=Vc+Vp(\alpha) \quad (3)$$

The momentary volume occupied by the small piston is:

$$Vm(\alpha m,Sm)=\frac{Vsm(Sm)}{Sm}\cdot s(\alpha m,Sm) \quad (4)$$

The momentary volume of the cylinder, as affected by the movement of the small piston during the angular interval of its stroke, becomes:

$$V2(\alpha m,Sm)=V(\alpha m)-Vm(\alpha m,Sm) \quad (5)$$

and is shown in Fig. 2, in comparison with the one of standard (unmodified) engine. Further on this engine solution will be named shortly **VSCR** (Variable **S**equential **C**ompression **R**ation).

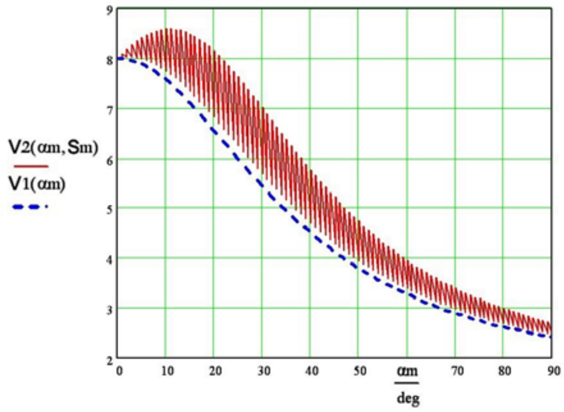

Fig. 2. *Variation of the cylinder volume as affected by the movement of the small piston, for different strokes.*

The momentary compression ration, for the standard engine is given by the expression:

$$\varepsilon1(\alpha m)=1+\frac{Vs}{V(\alpha m)} \quad (6)$$

while, for the **VSCR** engine, we get:

$$\varepsilon2(\alpha m,Sm)=1+\frac{Vs}{V2(\alpha m,Sm)} \quad (7)$$

Overlapping the two compression ratios, for the small piston working domain, results in the curves shown in Fig. 3.

Considering an adiabatic coefficient k=1.3 the efficiencies for the two studied cases become:

$$\eta tv1(\alpha m)=\left(1-\frac{1}{\varepsilon1(\alpha m)^{k-1}}\right)\cdot 100 \quad (8)$$

For the working range of the small piston we get the thermal efficiency gain of the **VSCR** solution as:

$$A\eta tv(\alpha m,Sm)=\left|\eta tv2(\alpha m,Sm)-\eta tv1(\alpha m,Sm)\right|\% \quad (10)$$

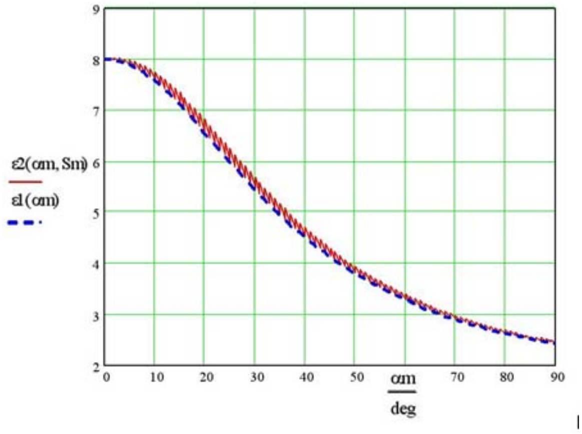

Fig. 3. *Overlapped variation of the compression ratios.*

The efficiency gain variation range for different strokes or diameter of the small piston is shown in Fig. 4 and Fig. 5.

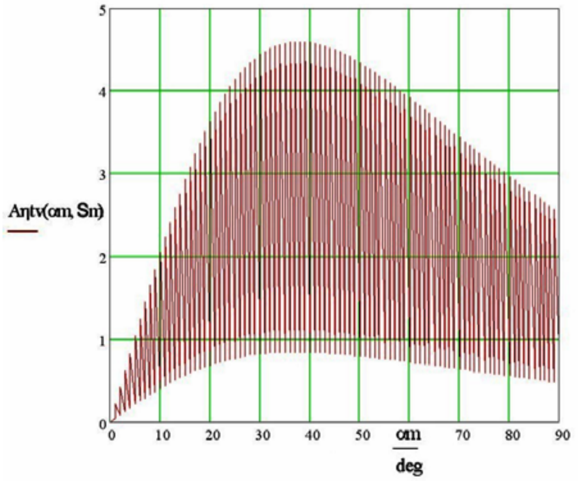

Fig. 4. *Efficiency gain variation for different strokes of the small piston.*

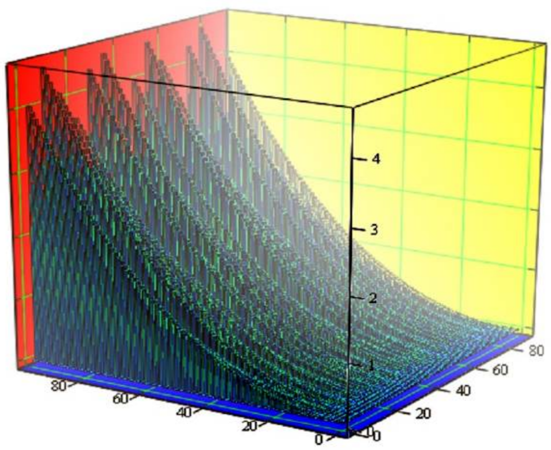

Fig. 5. *Efficiency gain by modifying the diameter of the small piston.*

Second case modeling

In this case the highest value for the small piston stroke (10 mm) is used, which leads to the most significant efficiency increase, while the diameter of the small piston is modified, starting with the lowest value, D_m=44 mm, and ending when the diameter of main piston is reached (D_m=D=97 mm). The previous formulae are modified accordingly, only the intermediate and final results being presented.

For the both cases, stresses and strains of the main piston of the **VSCR** engine were evaluated, using the **F**inite **E**lement **M**ethod (**FEM**); some results are shown in Fig. 6.

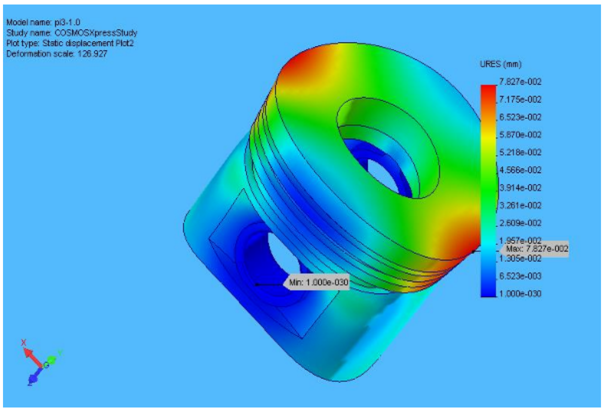

Fig. 6. *Strain for the main piston of the* ***VSCR*** *engine obtained using* ***FEM.***

3.2. Compression rings profile optimization

When the peripheral surface of the piston ring is correctly aligned with the cylinder liner surface, the piston – piston rings-cylinder assembly acts as a labyrinth, insuring an efficient seals [2].

Starting from the p_0 pressure inside the combustion chamber, the pressure decreases to p_{s1} behind the first compression ring, to p_1 after the first ring, to p_{s2} and p_2 behind and after the second ring etc.

Due to the high value of the chamfer angles h_1, only the surface with the length $h_r = h - 2h_1$ [m] is considered as the hydrodynamic active surface of the piston ring-cylinder liner couple.

Thus, for the downward piston stroke, the length of the wedge shaped interstice, marked $h_{efr(d)}$, is:

$$h_{efr(d)} = (1 - X)(h - 2h_1) \text{ [m]} \qquad (11)$$

and accordingly for the upward stroke we have hefs(u)

$$h_{efr(u)} = X(h - 2h_1) \text{ [m]} \qquad (12)$$

The specific relations for the hydrodynamic lubrication regime, correlated with the combustion chamber pressure, are used in order to establish the conditions for the oil intake and exhaust inside the piston ring-cylinder liner couple, thus leading to the best values for the angles of the conical surfaces that insure a preponderant hydrodynamic lubrication regime.

The oil flow through the cylinder liner-piston ring is calculated using the flow equation for the hydrodynamic reciprocating couples [3].

The equations are:
- for the downward stroke:

$$Q_{L(d)} = \pi \cdot D \cdot \left[\frac{h_1 h_2}{h_1 + h_2} v_p + \frac{1}{6} \frac{(h_1 h_2)^2 (p_1 - p_0)}{\eta (h_2^2 - h_1^2)} \right] [\mathrm{m^3/s}] \quad (13)$$

- for the upward stroke:

$$Q_{L(u)} = \pi \cdot D \cdot \left[\frac{h_1 h_2}{h_1 + h_2} v_p + \frac{1}{6} \frac{(h_1 h_2)^2 (p_0 - p_1)}{\eta (h_2^2 - h_1^2)} \right] [\mathrm{m^3/s}] \quad (14)$$

In order to evaluate the overall oil flow during one engine cycle we define the overall oil circulation Q_{Lt} [m^3/s], calculated by graphical integration of the oil flow according to Fig. 7.

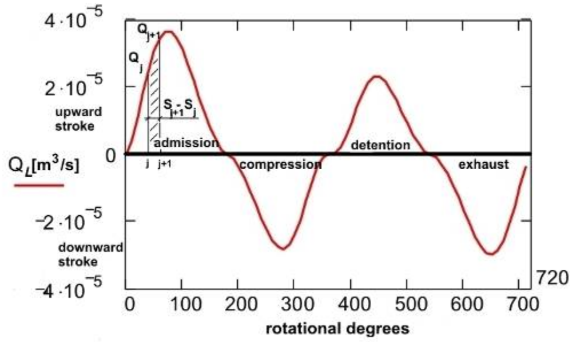

Fig. 7. *The oil flow through the cylinder liner-piston ring and the overall oil circulation QLt calculated by graphical integration.*

The relation for the overall oil circulation is:

$$Q_{Lt} = \sum_{j=0}^{71} \frac{Q_{Li+1} + Q_{Li}}{2S} \left| S_{i+1} - S_i \right| [\mathrm{m^3/s}] \quad (15)$$

A positive value of the overall oil circulation means that the oil flow is directed towards the combustion chamber.

Starting from the shear forces in the reciprocating couple [4], for the hydrodynamic lubrication regime, we evaluate the friction forces inside the cylinder liner-piston ring couple using the equations:
- for the downward stroke:

$$F_{f(d)} = \frac{2 \cdot \pi \cdot D \cdot \eta \cdot v_p}{k_d} \cdot \left[3 \cdot \frac{h_2 - h_1}{h_1 + h_2} - 2 \cdot \ln \frac{h_2}{h_1} \right] + \ldots$$

$$\ldots + \pi \cdot D \cdot k_d \cdot \frac{h_1 \cdot h_2}{h_1 + h_2} \cdot (p_1 - p_0) [\mathrm{N}] \quad (16)$$

- for the upward stroke:

$$F_{f(u)} = \frac{2 \cdot \pi \cdot D \cdot \eta \cdot v_p}{k_u} \cdot \left[3 \cdot \frac{h_2 - h_1}{h_1 + h_2} - 2 \cdot \ln \frac{h_2}{h_1} \right] + \ldots$$

$$\ldots + \pi \cdot D \cdot k_u \cdot \frac{h_1 \cdot h_2}{h_1 + h_2} \cdot (p_0 - p_1) [\mathrm{N}] \quad (17)$$

Defining the piston ring mechanical work of the friction forces L_{fr}[J] we may evaluate the mechanical losses, using a graphical integration method for the variation shown in Fig. 8.

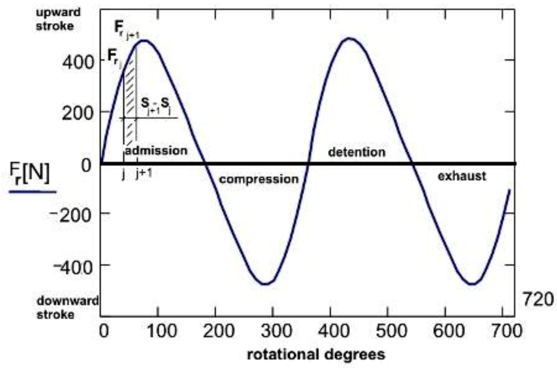

Fig. 8. *The friction forces inside the cylinder liner-piston ring couple and piston ring mechanical work of the friction forces L_{fr}[J] calculated by graphical integration.*

Using the general expression of the mechanical work, we get the relation:

$$L_{fr} = \sum_{j=0}^{71} \frac{\left| F_{ri+1} + F_{ri} \right|}{2} \left| S_{i+1} - S_i \right| \; [\mathrm{J}]. \quad (18)$$

A high value of this mechanical work means high friction forces inside the couple and diminishes the engine's mechanical efficiency.

Computational procedure

The theoretical model we have developed allows the determination of the transversal profile of the first and the second compression ring in order to obtain a different profile but similar lubrication conditions, to reduce oil consumption, to obtain a lower friction forces and to increase the mechanical efficiency of the piston ring – cylinder line coupling.

So, the slope repartition will be optimized in order to reduce oil flow towards the combustion chamber.

Using the relation (13), (14) and (15) and giving values comprised between 0 and 1 for the slope repartition X, the calculus are made in order to obtain a zero value for the overall oil circulation Q_{Lt}. In that case, the oil flow towards the combustion chamber

and the oil consumption is reduced to the minimum values. The upper and bottom slope angle of the peripheral surface of the rings will be optimized in order to insure a preponderant hydrodynamic lubrication regime and diminish the friction forces and piston rings and cylinder liner wear.
Calculus, using the relation (16), (17) and (18), are made starting at 0 value for the upper and bottom slope angle and is considered ended when we obtain a maximum percentage from entire engine cycle with the hydrodynamic lubrication regime for the piston ring and the minimum value for the piston ring mechanical work of the friction forces L_{fr} [J].

In Fig. 9 will expose the oil flow through the cylinder liner-piston ring and, in Fig. 10, the friction forces inside the cylinder liner-piston ring couple variation for the entire engine cycle for the both compression piston ring.

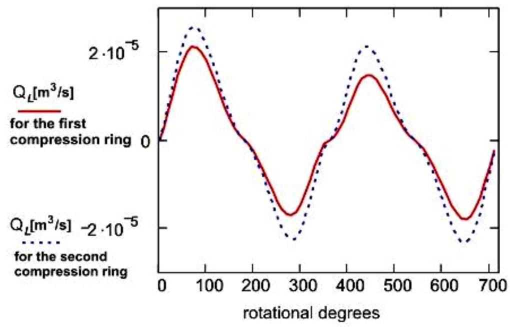

Fig. 9. *The oil flow through the cylinder liner-piston ring for the first and the second modified compression ring of a four-stroke S.I. engine.*

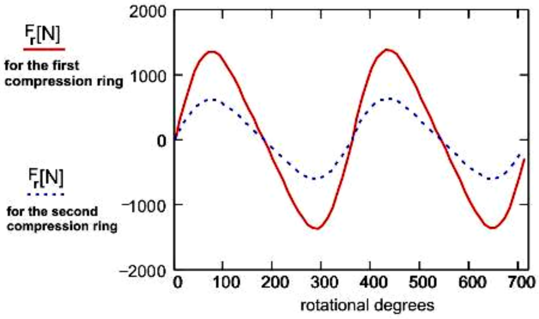

Fig. 10. *The friction forces inside the cylinder liner-piston ring couple for the first and the second modified compression ring of a four-stroke S.I. engine.*

4. Conclusions

- For all the studied cases, the models have revealed a smoother drop in the thermal efficiency of the **VSCR** engine, during the combustion process.
- In the meantime, higher values of the thermal efficiency were recorded during the main phase of the combustion process and towards its end.
- The maximum thermal efficiency increase (up to 4%) was obtained when the small piston's stroke is 10 mm and when its diameter equals the one of the engine's main piston (the "fake piston head" case).
- A study of the efficiency increase shows that, in the first case, its maximum value is obtained at 35 degrees CA after the TDC, while for the second case, the maximum value is attained at 37-38 degrees CA after the TDC, thus showing the advantage of this method in terms of a fuel consumption decrease.
- The lubrication conditions for the second compression ring are better to the first piston ring.
- Several new notions were defined (*overall oil circulation* and *mechanical work of the piston ring friction forces*), in order to improve the profile of the compression rings of an internal combustion engine, to reduce oil consumption, insure a preponderant hydrodynamic lubrication regime and diminish piston rings and cylinder liner wear.
- The tests carried out with modified rings showed a decrease in gas pressure escaped in the crankcase, the situation exposed in Fig. 11, and lower consumption of lubrication oil.
- In conclusion, the modification of those parameters leads to an optimized engine solution for lowering the fuel consumption and the pollution level, an environmental friendly propulsion engine.

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