Over compression influence to the performances of the spark ignition engines

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Abstract. This paper presents the theoretical and experimental results of some procedures used in improving the performances of the automobile spark ignition engines. The study uses direct injection and high over-compression applied to a standard engine. To this purpose, the paper contains both the constructive solutions and the results obtained from the test bed concerning the engine power indices, fuel consumption and exhaust emissions.

1. Introduction
This study aims at improving the engine performances by implementing the direct injection of gasoline. Moreover, in addition to the main direct and well-known benefits of the direct injection supply of gasoline, we tried to turn into good account the absence of the fresh load pre-heating in the admission manifold, on the one hand, as well as the additional cooling of both the load and the combustion chamber walls, due to the fuel vaporization upon its injection in the engine cylinder, on the other hand. They lead to an extended delay of self-ignition and, finally, to a decrease in the detonation tendency. Hence the possibility to increase the compression ratio \( \varepsilon \). By \( \varepsilon \) rate rising, the relative quantity of residual burnt gas decreases, which, on the one hand, improves filling and, on the other hand, increases the engine energetic performances. From this point of view, the high compression ratios represent a certain and efficient solution for the reduction of the specific consumption, \( c_{em, min} \). However, we aimed to use fully the possibilities that the proposed energetic solution can offer, by adopting a compression ratio \( \varepsilon_{Sc} = 11 \), superior to the ratio at the detonation limit, imposed by the ignition chamber architecture and the octane number. This tendency of high over-compression represents a modern procedure, yet, less used in the engine construction industry.

2. Theoretical basis of the adopted solutions
The method, rather well-known, maintains the octane number, but alters the power and specific consumption with full loads, by decreasing the pre-ignition to the detonation limit, \( \beta_{Id} \), thus reaching the condition: \( \beta_{Id} < \beta_{opt} \), as illustrated in figure 1. Obviously, when reducing the load, the tendency towards detonation decreases and \( \beta_{Id} \) rises. The condition: \( \beta_{Id} = \beta_{opt} \) defines the highest load at which this over-compressed engine develops the maximum actual efficiency, \( \eta_{max} \), without detonating. With lower loads than the detonation limit load, i.e. \( k < k_{Id} \), the direct injection engine will operate with the characteristic of optimum advance, the condition: \( \beta_{opt} < \beta_{Id} \) being satisfied.

This way, an operation with a high compression ratio is achieved, which will lead to the specific consumption decrease. The use of the over-compression procedure for this proposed engine solution is justified, on the one hand, by an insignificant reduction of the rated power – given the reserve...
appeared because of using direct gasoline injection, and, on the other hand, the fact that the car engine operates with a low frequency in full load regime. In this sense, data are available about such engines being installed on auto-vehicles; such data, presented in figure 2, indicate that the frequency of large loads is, in fact, negligible and the regimes used with the highest frequency – about 85%, represented by the hatched area, cover a small part of the engine running domain, i.e. that of partial loads of up to 18%, and the low speeds. Figure 3 illustrates the influence of 25% over-compression level upon the engine. It can be noticed that, within the load range $k < k_{ld}$, an average reduction of the specific consumption by 15% can be obtained in the load range [1].

Figure 1. Advance variation.

Figure 2. Load frequencies.

Figure 3. Over-compression influence.

In order to obtain maximum benefits from maintaining or even increasing the power of the engine supplied by direct gasoline injection, when using high over-compression, we tried to achieve an improved engine cylinder filling. For this purpose, we proceeded to the modification of the intake manifold therefore having rather reduced transversal cross-sections, intended to ensure the transport of mixtures of air and various sizes fuel drops, it also having lengths imposed by a spatial configuration, in accordance with the cylinder position and the criterion of assuring route equality to the served
cylinders. It is a well-known fact that an increased cross-sections of passing obtained by re-

dimensioning of the intake manifolds, leads to a net improvement of the filling coefficient \( \eta_v \).

Based on certain data presented in the specialized literature [2, 3], we aimed obtaining a proper

filling for a rather wide speed regime, by using – as much as possible – a favourable overlapping of

the pressure waves. For this purpose, we correlated the form and size adopted for the manifold, based

on some previous researches and with the recommendation of other authors. We also had in mind to obtain combined changes of the pressure in the cylinders and manifold, capable to lead to favourable effects on filling. These effects will be found in the most important engine regimes, being recorder an increase of maximum torque and maximum power.

With our solution we aim to improve the engine performance in what concerns the environment pollution.

3. Experimental results. Interpretation

Figure 4 presents the engine speed characteristics with various loads.

![Figure 4. Speed characteristics.](image)

The curve of the effective power variation corrected for total load, emphasizes a rated power of 66.24 [kW], developed with a 4000 [rpm] speed. Yet, the minimum effective consumption with total load rises to a 308 [g/kWh] rate, as compared to 280 [g/kWh] rate at standard solution. It is, in fact, a phenomenon typical for engines having a high over-compression degree which, as shown above,
appears with full load and higher partial loads, due to the reduction of the pre-ignition, from optimum rates to rates situated under the knock limit.

Further to the analysis of these characteristics, one can notice that the performance difference decreases in the same time with the speed rise, perhaps due to the slight increase of pre-ignition based on the reduction of the detonation tendency. Significant but advantageous differences appear in what concerns the rates of the effective specific consumptions obtained for regimes characterized by partial loads. Thus, a slight reduction already appears at 85%, but the consumption rates corresponding to the 70% load are visibly diminished. The lowest rates appear for this load within the 2500 – 3500 [rpm] range, it being about 267 [g/kWh]. The minimum consumption rate emphasized is 261 [g/kWh] and can be obtained for the same load and a 3000 [rpm] speed. Substantial reductions appear obvious for the 40% and 25% loads.

The idle running characteristic presented in figure 5, indicates consumption rates 11% lower as compared to the standard engine. The burnt gas temperature evolution depending on speed with various engine loads indicates higher rates in low partial loads, with which the pre-ignition rates are optimum, i.e. superior to those corresponding to big loads and mainly to the total engine load. The increase tendency is somewhat slowed down by the mixtures which in this case are even poorer.

The excess air factor, $\lambda$, also illustrated for various engine speeds and loads, are presented in figure 6, by means of engine map. The rates emphasized by these representations are related to the consumption reduction in partial loads, obtained with this engine and presented above. Thus, the $\varepsilon_{SC}$ high rate probably ensures the remake of the burning speed, through pressure and temperature effect upon the reaction speed, which leads to the reduction of the burning area thickness behind the ignition front. The increase of the reaction speed by using this over-compression level allowed, in the same time, for the adjustment of a poorer mixture, in which the fuel burns completely.

These issues, finally, prevent the mixture enriching with partial loads, used right in order to remake the reaction speed, and previously diminished by the high proportion of residual gases in the mixture. Under the circumstances, with a compression ratio $\varepsilon_{SC} = 11$ and the corresponding pre-ignition rates, the engine will run stably, under partial load regimes and excess air factor rates of up to $\lambda \approx 1.22 \div 1.23$. 

Figure 5. Idle running characteristic.
In order to visualize the major phenomena in the engine, here below are presented the diagrams corresponding to the burning process. Considering the limited space of this text, we selected the cyclical modifications of the parameters in the cylinder corresponding to a 25% engine load rate, for various speeds. These modifications are shown in figure 7.

First of all, the analysis of the figures above indicates, the influence of pre-ignition simultaneous with the engine speed rise, in constant loads [4].

Secondly, the curve shape representing the pressure modification in the combustion chamber indicates, initially, a slowed down tendency of pressure rise, yet, rapidly compensated by the reaction speed rise – as shown above – due to the high compression ratio [4].
Although the maximum rates of the pressure in cylinder, for trial speeds, are superior to those recorded for the standard supplied engine, they are, nevertheless, modulated by the air lamination effect due to the small throttle opening within this low load operation. For speed rates of about 2500 – 3000 [rpm], when the engine develops, with this load, its maximum torque, a slight tendency of detonation can be easily emphasized, however, controlled by the limit value of the pre-ignition.

The comparative variations of the carbon content at idle running, for various engine speeds, are shown in figure 8.(a). The obtained average reductions are of 35%. The improvement, obvious with this regime, results from the fact that – for this gasoline injection solution there is no need for a quite reach mixture in order to provide ignition and burning stability. By analogy, the variations corresponding to a 25% load at various speeds are presented in Figure 8.(b), which emphasizes an average reduction of the carbon oxide content in the exhaust gases of 67%.

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The comparative variations of the hydrocarbon content in the exhaust gases, depending on speed – for the same two regimes mentioned above – are presented in figure 9. The results of this study shows, for the gasoline injection engine, hydrocarbon content reductions which, in idle running, average to 23%, while for 25% of the total load, the hydrocarbon content diminishes by average percentage of 42%.

The obtained results can, probably, be explained by the fact that, together with the increase of the excess air up to a $\lambda$ rate $\approx 1.2 \div 1.23$, on the one hand, the hydrocarbon concentration in the limit bed
decreases, and, on the other hand, the oxygen quantity available for the post-burning phenomenon of the exhaustion process rises. From this point of view, a further increase of the excess air ratio rates, i.e. a mixture leaning, would be unfavorable as it might cause a major reduction of the exhaust gas temperature, thus slowing down the post-burning, finally leading to an increase of the exhausted gas hydrocarbon emission content.

4. Conclusions
The energetic solution-making subject of the study ensures the engine a rated power 8.6% higher than the standard version power, the maximum torque becoming 13% higher. The minimum specific consumption obtained is 261 [g/kWh], when, for partial loads of 25% and 40% respectively, the average reduction is of 10.83%; corresponding to the idle running regime, the consumption diminishes by an average percentage of 11%. The results obtained as for consumption are equally related to the possibility to run the engine with poorer mixtures, in certain regimes, such dosages being characterized by rates ≈ 1.2. Besides the advantages resulting from using the direct gasoline injection and over-compression, significant results are obtained in what pollution is concerned.

5. References
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