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RESEARCH AND EXPERIMENTAL RESULTS REGARDING THE BURNING OF LEAN MIXTURES WITH ADDED ON BOARD PRODUCED HYDROGEN

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***Abstract*—**The paper presents the results of an experimental research in which a comparison is made between the combustion phases of a gasoline engine operating with a lean mixture and addition of hydrogen and a standard operating one.

***Keywords*—** cyclic dispersion, fuel consumption, hydrogen addition, on board production of hydrogen

I. INTRODUCTION

THE burning of lean mixtures in SI (spark ignition) engines is one of the most promising methods to improve the fuel economy and to reduce polluting emissions in the exhaust gases [1]-[6].

Nowadays there are over 200 versions, but still does not give the expected results in terms of reducing fuel consumption, being developed with the aim of reducing polluting emissions of exhaust gases.

The need to save oil fuels and to reduce polluting emissions led to the S.I. engines with mixed fuel, especially gasoline-hydrogen mixture.

The presence of hydrogen in the combustion chamber allows the use of lean mixtures, a linked kinetics of reactions, and all this leads to a reduction to a minimum of the duration of the combustion, an improvement of the dynamic and the economic indicators, because it reduces the propagation of the flame by increasing the speed of propagation of the fire front and by reducing the depth of the combustion zone (or it's thickness) of the turbulent flame, with consequences relating to the larger heat cutting in rapid combustion phase as well as in reducing the moderate burning phase [7].

The device for the on board production of hydrogen was installed on a series car. Starting from theoretical and experimental results obtained by different researchers, the purpose was to study the possibility of improving performance and reducing the amount of exhaust gases through the burning of lean mixtures with hydrogen addition on a series engine.

II. EXPERIMENTAL RESULTS

It had been studied the engine characteristic comparatively in standard condition and with lean mixtures and hydrogen addition under different operating conditions of the engine, respectively. The envisaged research program has been carried out by recording the following parameters:

1) parameters concerning the specific real quantities of the engine;

2) parameters concerning the specific indicated quantities of the engine;

3) coefficient concerning the parameters of the combustion process;

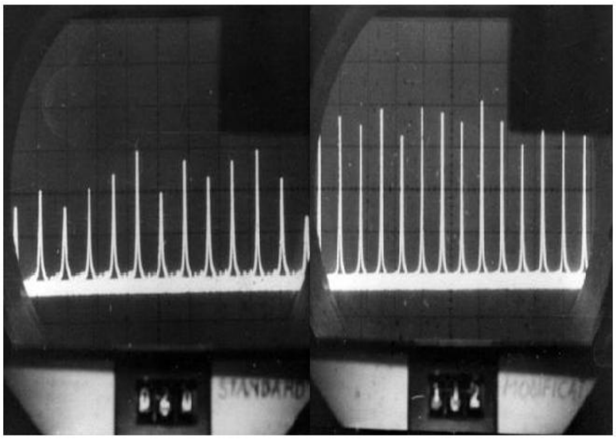
4) quantities concerning the level of the emissions in the exhaust gases.

The correlation of the measured pressure depending on the rotation angle of the crankshaft as well as the opening of the inductive transducers firmly mounted on the cylinder block is made by those transducers which are taking over the impulses from two coaxial disks mounted on the crankshaft.

The denticulate disk, obtained through milling-work, marking the angle has been made with a 20° crank angle division, and near the TDC (top dead centre) has been made symmetrically every 5° in order to accurately pursuit the detachment points.

The influence of hydrogen addition in lean mixtures on the combustion phases and on process parameters was determined by investigating the variation of the combustion delay τ_d (ms), the duration of the fast combustion phase τ_f (ms), the maximum pressure p_{max}

(MPa), the pressure rise speed $(\Delta p/\Delta \alpha)_{med}$ (MPa/CSR), and the cyclic dispersion δ (Fig. 1), depending on the rotational speed, at partial loads: K = 40%, K = 80% and for full load K = 100%.



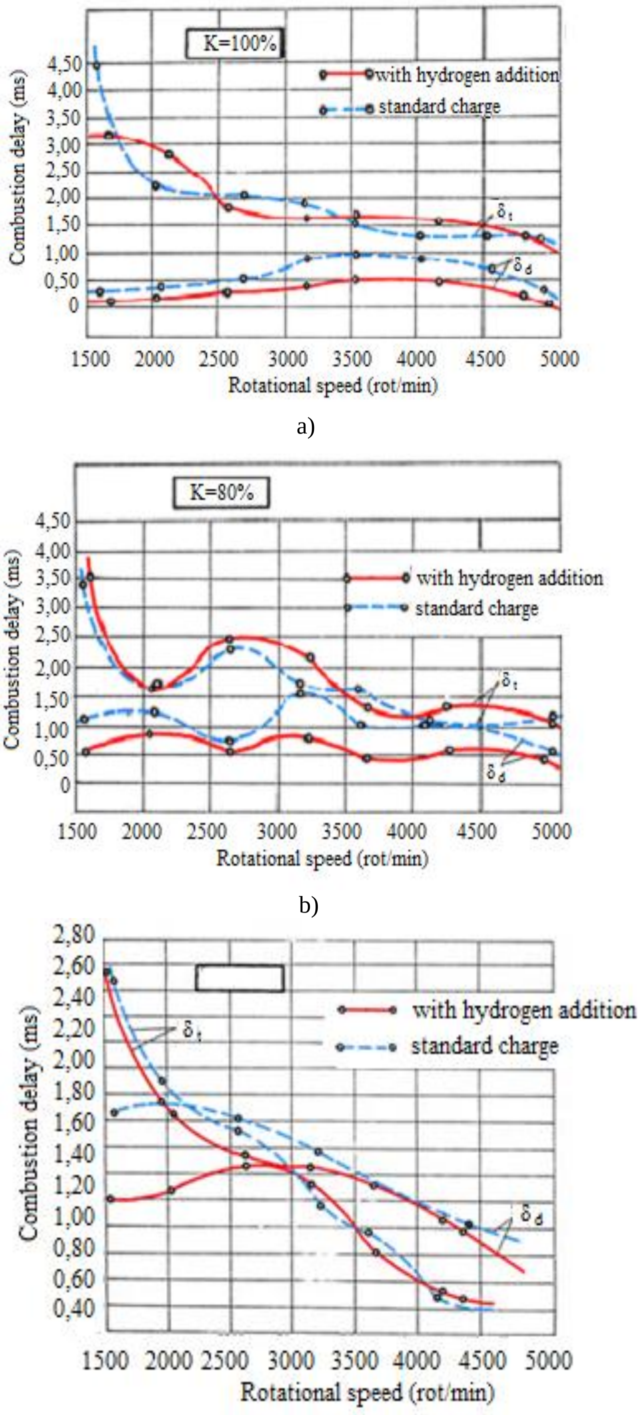
a) b)
Fig. 1. The cyclic dispersion: a) for series engine: b) gasoline-hydrogen engine

The variation of the combustion delay and the fast combustion duration, both expressed in milliseconds, depending on the rotational speed n (rot/min), is shown in Fig. 2.

These diagrams show that the presence of a small quantity of hydrogen in a lean mixture determines a chain-reaction kinetic that reduces the delay to the fast combustion phase with 25 to 50%. So, the combustion rate is influenced only by the speed of the laminar flame propagation, which depends on the mixture's composition, on the air-fuel ratio, on the activation energy and on the turbulence intensity. The favorable change in the mixture's composition and in the activation energy determines the decrease of the fast burning phase.

In the second phase, during the fast combustion, it has been found that the speed of the flame propagation is determined only by the turbulence intensity, because the combustion zone is equal to, or even exceeds, the macro vortices. This is demonstrated by the fact that the plotted diagrams for both the standard and the hydrogen addition engines are very close, even intersecting for some rotational speeds. So, the influence of the physical and chemical properties on the speed of the flame propagation is insignificant.

In conclusion, the duration of the flame propagation decreases for some partial loads and rotational speeds because the speed of the ignition front propagation increases and the combustion zone decreases. This causes a decrease of the steady combustion phase, more heat being released in the fast combustion phase. At partial loads (K = 40% and K = 80%) the combustion duration can be decreased if another spark timing law is chosen, but the standard engine controls were preserved in order to determine only the influence of hydrogen presence into the mixture.



c)
Fig.2. The combustion delay for: a) K=100%; b) K=80%; c) K=40%

For a constant rotational speed the combustion delay increases as the load decreases. As it is shown in Fig. 3, for three rotational speed values the combustion delay is always longer for the standard charged engine at all partial loads.

In this situation is even more evident that the active centers have a favorable influence on the combustion delay at about the same pressure and temperature conditions in the moment of the electric spark triggering.

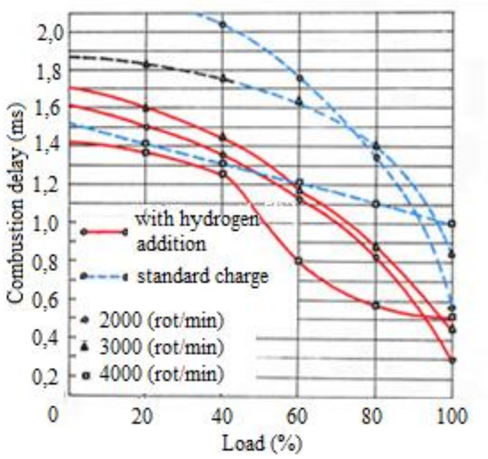
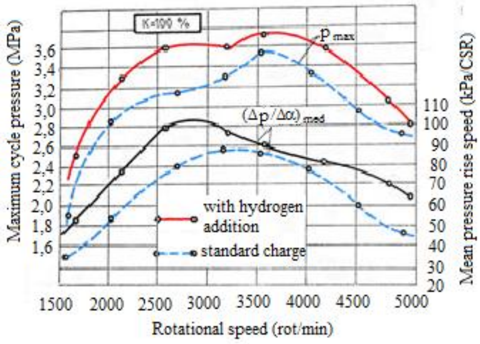
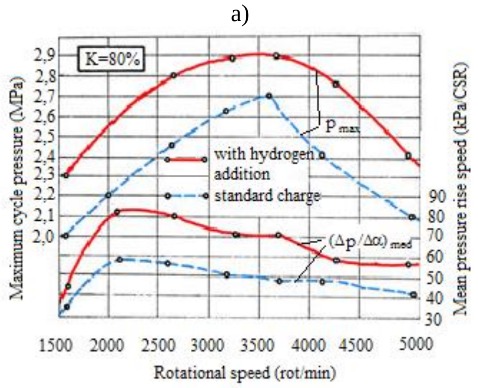


Fig. 3. Combustion delay at three different rotational speeds

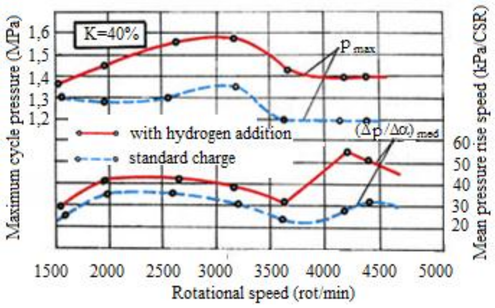
This determines the increasing of the maximum pressure of the cycle and also the increasing of the mean pressure rise speed for all partial loads considered, as it is shown in the diagrams presented in Fig. 4.



a)



b)



c)

Fig. 4 The maximum engine cycle pressure for: a) K=100%; b) K=80%; c) K=40%

The increasing of the pressure rise mean speed is about 11,6%, without any effects on the engine's smooth running.

At low rotational speed values (under 2.500 rpm) the maximum pressure decrease is caused mainly by the considerable increase of the fast combustion duration, the point in which the maximum pressure is registered is moving towards the expansion stroke (Fig. 5).

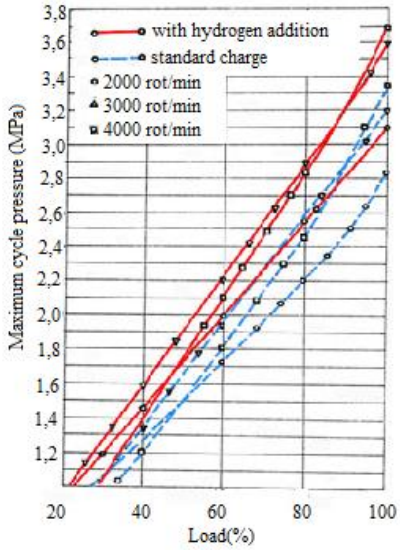


Fig. 5 The maximum engine cycle pressure for different rotational speeds

As the rotational speed increases, the initial phase $\Delta\alpha_d$ (crankshaft angle) will be longer because the combustion delay does not vary with the rotational speed and $\Delta\alpha_d=6\cdot n\cdot\tau_d$. So the maximum pressure peak will move towards the expansion stroke. For small loads (K = 40%) this fact is not so evident, because as the rotational speed increases the fast combustion duration decreases (see Fig. 2.c).

In lean mixture with hydrogen addition the induction-time is shorter, because more active centers exist and so the number of the initiation reaction increases. This is very well shown by the cyclic dispersion δ.

As shown in fig. 6, the pressure rise speed is greater in the modified engine.

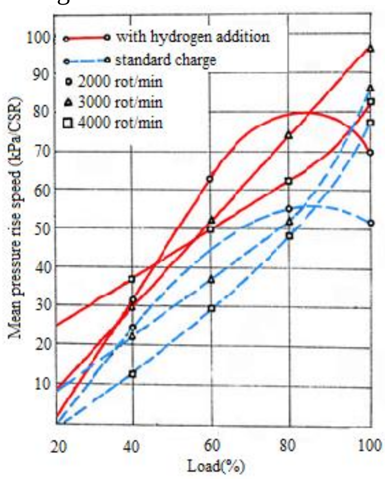


Fig. 6 The variation of the pressure rise speed

The indicated diagrams for the three rated loads were obtained through scale photos taken on the oscilloscope. They served as base for the cyclic fluctuation depending on the rotational speed. The diagrams are shown in Fig. 7.

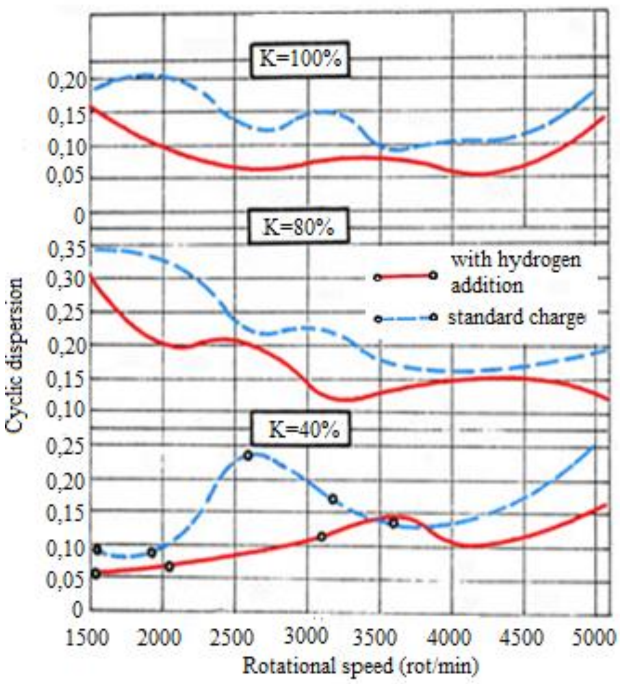


Fig. 7 The variation of the cyclic dispersion

In order to obtain a better precision on data processing, scale photos for 10 cycles on every rated load for every rotational speed value were taken.

It is interesting to observe the diagrams for the rated load K = 80% and for the full load K = 100% between 2.000 and 4.500 rpm. The pressures are in these cases clearly higher for the engine with hydrogen addition, although the mixture is leaner. It is expected that even better results could be obtained if an optimum correlation is acquired between the quantity of added hydrogen, the lean mixture quality and the spark timing. So the cyclic fluctuation could be reasonably diminished for lower partial loads, with direct mechanical, energetically and economical implications.

III. Conclusions

Adding small quantities of hydrogen in SI engines with allows the burning of lean mixtures, improves the performances, and reduces fuel consumption and emissions in exhaust gases, as shown in fig. 8.

The last test was made on the roll stand in accordance with the European norms as well as the polluting emissions standard. As it can be seen in Fig. 8 the test has been performed by comparison with a series car with the same loading at constant speeds.

At speeds of 40-50 km/h allowed by the traffic norms one can see a substantial reduction of the fuel consumption as well as of the CO between 0,8 - 1,2%. As a result of the tests performed there arises the necessity for modifying the advance characteristics.

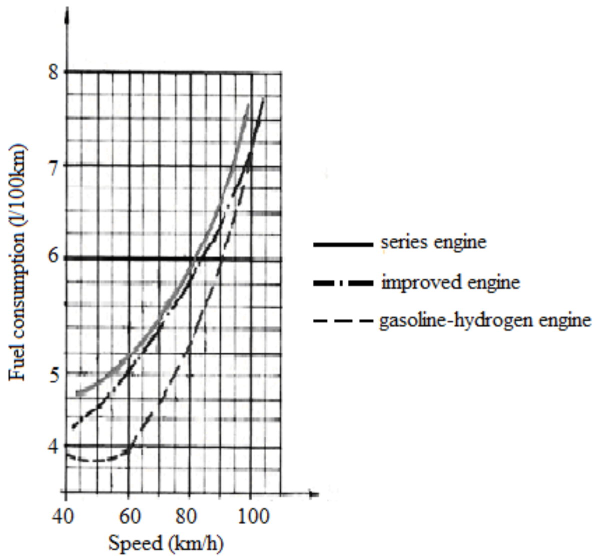


Fig. 8. Fuel consumption for three different engines

Such researches are currently made at the University of Oradea. In this case, a specialized program is used in order to study the influence of some parameters on the combustion process.

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