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TITLE:

INVESTIGATION OF THE EFFECTS OF HYDROGEN ADDITION ON PERFORMANCE AND EXHAUST EMISSIONS OF DIESEL ENGINE

Topic:

- ○ FUTURE AUTOMOTIVE TECHNOLOGY
- ○ INTELLIGENT TRANSPORTATION SYSTEMS
- ○ USER FRIENDLY AUTOMOBILE
- ○ ADVANCED PRODUCTION AND LOGISTICS
- ● VEHICLES & THE ENVIRONMENT

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National Society: ● YES ○ NO

Name of the National Society: Society of Automotive Engineers in Bulgaria

Abstract:

Experiments were carried out to evaluate the influence of the addition of hydrogen-oxygen mixture (obtained from electrochemically decomposed water) to the inlet air of a single cylinder direct injection diesel engine.
Addition of hydrogen to the intake or delivery into the cylinder of diesel engine can improve combustion process due to superior combustion characteristics of hydrogen in comparison to conventional diesel fuels.
Presented paper describes the dynamometer test results of a study where a small amount of hydrogen-oxygen mixture, produced by hydrogen-oxygen generator is added to the intake of a diesel engine.

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INVESTIGATION OF THE EFFECTS OF HYDROGEN ADDITION ON PERFORMANCE AND EXHAUST EMISSIONS OF DIESEL ENGINE

INTRODUCTION

When a small amount oh hydrogen is added to the intake air, the combustion process of the internal combustion engines could be considerably enhanced. It was found out that the addition of hydrogen has a good influence on the combustion (2, 4). The combustion improvement is due to superior flammable characteristics and higher flame propagation velocity of hydrogen in comparison with the conventional hydrocarbon fuels (4).
The aim of the presented investigation is to evaluate the influence of hydrogen addition on performance and exhaust emissions by the means of indicated, brake, and exhaust engine performance parameters.

DESCRIPTION OF THE TEST INSTALATION

The test engine is a single cylinder direct injection compression ignition engine with *98 mm* cylinder bore and *130 mm* piston stroke.

The engine is loaded with *DC* dynamometer *MEZVETIN MS 2218–4*.
The fuel consumption is measured by mass method.
The air consumption is measured by laminar flowmeter *CUSSONS M79RH*.
The smoke is measured by *HARTRIDGE MK3* smokemeter.
The *NO* emissions are measured by gas analyzer *RADAS 1*.
The indicated pressure data are collected by piezoelectric pressure transducer *"KISTLER 6509"* and crank shaft position encoder *"HEIDENHAIN ROD 428D.163"* that gives the crank angle position and the basic *TDC* impulse.
The engine speed is measured by frequencymeter *FM1100*.

TEST METHODS

During the test procedure the engine run on constant load conditions and the crankshaft speed was changed from *1300* to *1800 rpm*. Two constant load operating characteristics were taken down as follows:

1.Constant load operating characteristic with conventional diesel fuel;
2.Constant load operating characteristic with hydrogen-oxygen addition to the intake air. The hydrogen-oxygen mixture was obtained from hydrogen-oxygen generator.

The hydrogen flowrate was maintained at *160 l/h* (hydrogen-oxygen flowrate is *240 l/h* respectively).
The injection timing during the test was maintained at *18* degrees before *TDC*.

INDICATED PRESSURE DATA PROCESSING METHODS

Following the upper described test procedure, the indicated pressure data were collected for every point of the constant load operating characteristic. The values that were necessary for the indicated pressure curve $p=p(\varphi)$ were collected from data acquisition port and were saved as a text file in the *PC* hard disk. The indicated pressure rate $dp/d\varphi$ was computed trough numerical differentiation of $p=p(\varphi)$ data.

The first law of thermodynamics for closed system was applied to compute the differential characteristic of net heat release rate *(Fig. 3)* according to Heywood – pp. 388 (3):

$$\frac{dQ_a}{d\varphi} = 100000.\frac{k}{k-1}.p\frac{dV}{d\varphi} + \frac{1}{k-1}.V\frac{dp}{d\varphi}, J/\deg, \qquad (1)$$

where Q_a is net heat (the part of heat obtained as a result of fuel combustion which was spent on the internal energy increase and on mechanical work), *J*;
φ–crank angle position, *deg*;
κ–averaged specific heat ratio, $\kappa=1,3$ (3);
p–current value of cylinder pressure for crank angle position φ, kgf/cm^2;
V–current value of cylinder volume for crank angle position φ, m^3;

$$V = \frac{\pi.D^2}{4}\left[\frac{S}{\varepsilon - 1} + r\left[(1-\cos(\varphi)) + \frac{\lambda}{4}(1-\cos(2\varphi))\right]\right], m^3, \qquad (2)$$

where S is piston's stroke, *m*;
D–cylinder bore, *m*;
ε–compression ratio;
r–crank radius, *m*;
λ–ratio of crank radius and connecting rod length;

The values of Q_a were computed for *120* degrees of crank angle rotation (duration of combustion), starting from the beginning of injection.
The net heat Q_a *(Fig. 4)* was computed by numerical integration of equation *(1).*

The values of cylinder temperature were computed using ideal gas equation of state:

$$T = \frac{p.V}{m_c R}, K \qquad (3)$$

where m_c is cycle amount of air, *kg/cycle, R* – gas constant, *J/kg.K*

RESULTS AND DISCUSSIONS

Figures *1, 2, 3, 4, 5,* and *6* show respectively indicated pressure data, indicated pressure rate, net heat release rate, net heat release, cylinder temperature, *P-V* diagram, and cylinder temperature at engine speed *1500 rpm.*

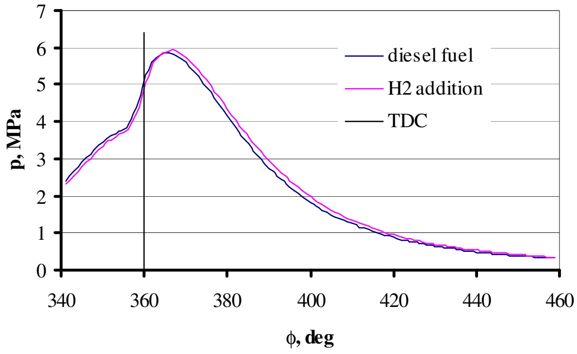

Fig. 1. Indicated pressure data

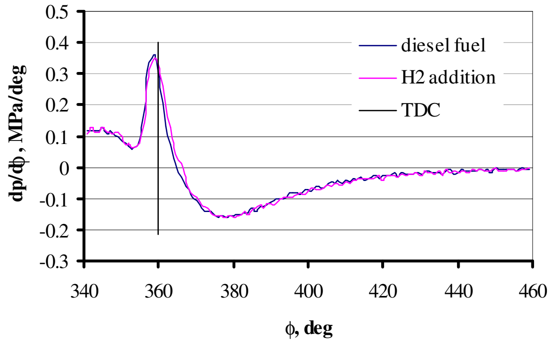

Fig. 2. Indicated pressure rate

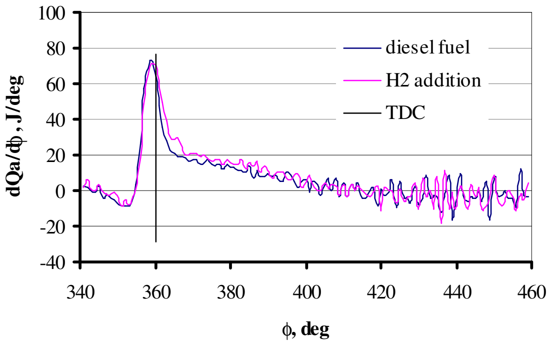

Fig. 3 Net heat release rate

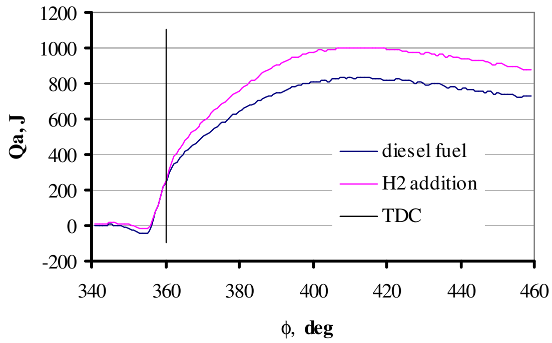

Fig. 4 Net heat release

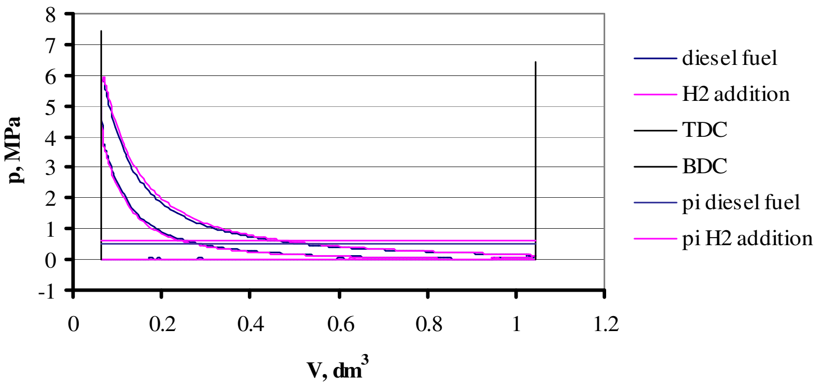

Fig. 5 P-V diagram

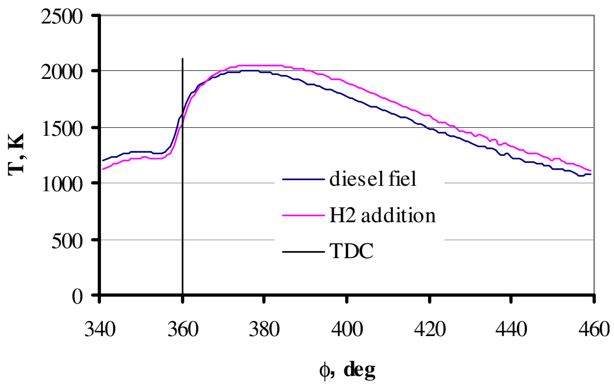

Fig. 6 Cylinder temperature

Figures *7, 8, 9* and *10* show respectively brake power, brake specific heat consumption, *NO* emissions and smoke.

The term brake specific heat consumption is used instead of brake specific fuel consumption because fuels with different lower heating values (diesel fuel and hydrogen) were burned.

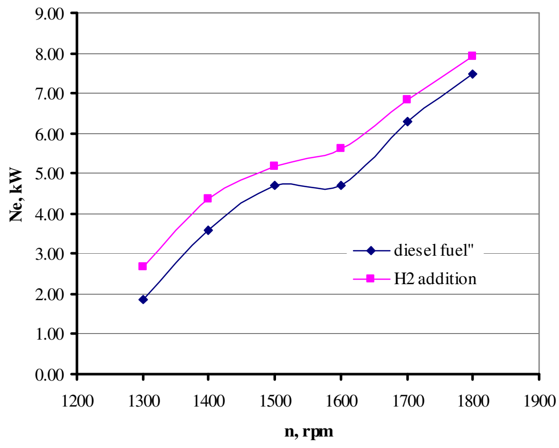

Fig. 7 Brake power

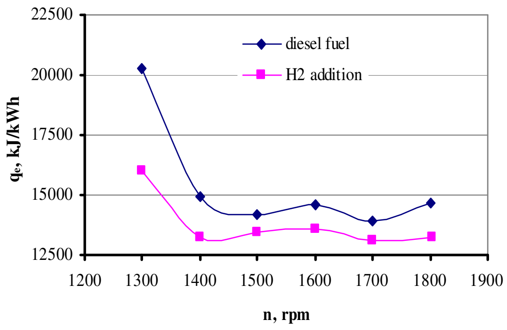

Fig. 8 Brake specific heat consumption

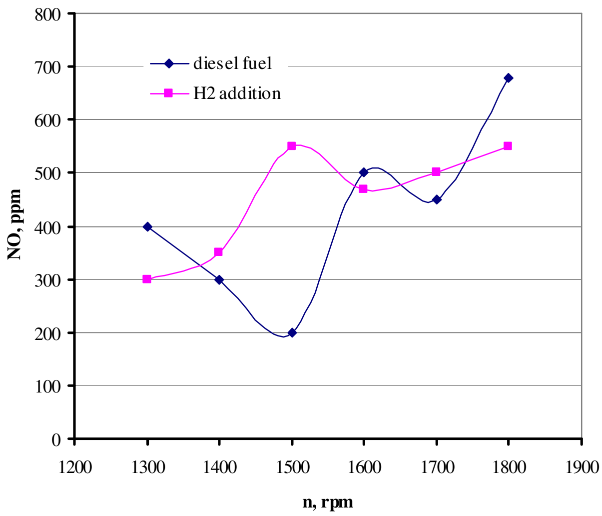

Fig. 9 NO emissions

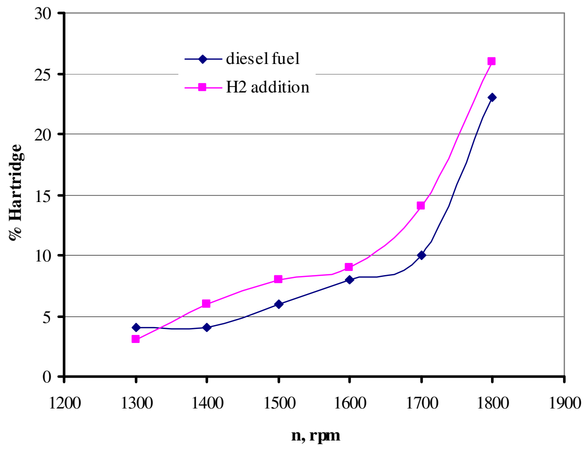

Fig. 10 Smoke of exhaust gases

The experimental results demonstrate that the hydrogen addition in the intake air has an influence on improvement of engine power, and energy consumption *(Fig. 5, 7, 8). NO* emissions are very complicated *(Fig. 9).* Higher smoke of exhaust gases is observed *(Fig. 10).*

The higher engine power with hydrogen addition *(Fig. 5, 7)* is due to the additional heat released from hydrogen combustion and to the reduced combustion duration (1). The combustion of hydrogen addition provides additional heat energy, which is one of the reasons for a higher engine output. The shorter combustion duration causes lower heat transfer rate thought the combustion chamber walls and higher diesel fuel heat utilization. As a result of the calculations that were done, the increase of the whole heat added to the engine cycle due to hydrogen combustion is 2,44% averaged over the entire investigated engine speed region. But the averaged power improvement obtained as a result of the experimental investigation is *15% (Fig. 7).* The middle indicated pressure improvement at *1500 rpm* is *14.8% (Fig. 5).* These facts show that the power improvement comes not only from the energy added as a result of hydrogen addition. The greater part of power increase is due to combustion process improvement by the means of combustion duration reduction because of superior combustion and flame propagation properties of hydrogen. The proofs of this statement are the curves of net heat release and net heat release rate shown respectively on *(Fig. 4)* and *(Fig. 3)*.

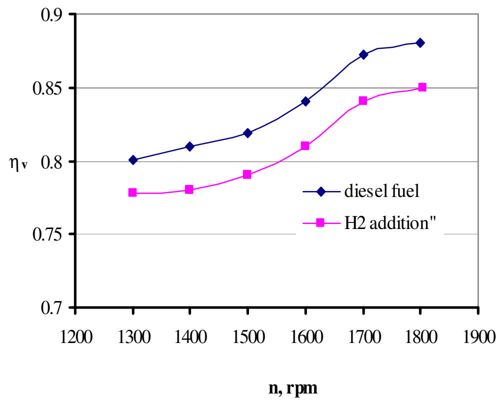

Fig. 11 Volumetric efficiency

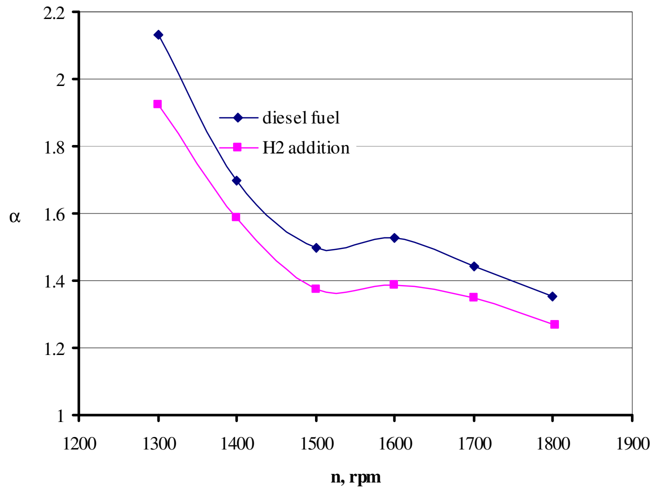

Fig. 12 Air-fuel ratio

The combustion duration reduction is due to reduced mixing-controlled combustion phase invoking higher net heat release rate *(Fig. 3)* and higher net heat release respectively *(Fig. 4)*.

The reduction of mixing-controlled combustion phase is due to the flame propagation of homogeneous hydrogen-air mixture trough the combustion chamber. The flame propagation improves the diffusion process (incurring many molecular collisions) between the hot air and diesel fuel vapors causing it's faster completing. The diffusion process improvement causes higher heat utilization and higher net heat release respectively.

It makes impression that the net heat release curve of hydrogen addition has higher values (immediately after injection) in comparison with that of diesel fuel *(Fig. 4)*. The decrease of net heat release curve is due to heat consumption necessary for diesel fuel vaporization. When hydrogen is added in the intake air ignition of hydrogen occurs immediately after the start of injection. The heat released from hydrogen combustion compensates the heat necessary for diesel fuel vaporization.

The brake specific heat consumption decrease *(Fig. 8)* is due to brake power increase.

The *NO* emissions are compared in *Fig. 9* for the two cases: with and without hydrogen addition. The experimental results that are shown on this figure are very complicated. This fact prevents any final analysis and conclusions about *NO* emissions. When a diesel engine runs with a small amount of hydrogen addition (repeatedly smaller than in the present investigation) the NO_x emissions are very lower in comparison with the case without hydrogen addition (4). The NO_x reduction with hydrogen addition might be attributed to superior combustion characteristics of hydrogen that burns more rapidly and cleanly than hydrocarbon fuels (4), because its amount is smaller and enters combustion reactions at higher velocity, has lower activation energy, and incurs more molecular collisions than heavier hydrocarbon molecules. These characteristics may not only improve combustion process but also enhance transport processes reducing hot spots in combustion chamber that are one of the major contributors to NO_x emissions in *IC* engines.

As shown on *Fig 10*, with hydrogen addition the engine smoke is higher. This behavior could be explained by air-fuel ratio decrease *(Fig. 12).* The lower air-fuel ratio (in the case of hydrogen addition) is due to the volumetric efficiency decrease *(Fig. 11)* due to the higher thermal loading of engine elements. The higher thermal loading is due to heat utilization efficiency improvement *(Fig. 4)*. The engine elements with higher average temperature cause air density decrease. In the present investigation particulate matter *(PM)* emissions were not measured but such a measurement will be done in future investigation. Using the *PM* amount *(g/h)* in the exhaust gases we would be able to evaluate the brake specific *PM* emissions *(g/kWh).* It could be said with confidence that in case of hydrogen addition *PM* emissions *(g/kWh)* would be lower due to brake power improvement.

CONCLUSION

Taking account the experimental results from the brake, emission and indicated diesel engine performances the following conclusions can be drawn:

1. When the engine runs with hydrogen addition heat utilization efficiency improvement was observed. The hydrogen addition influences the power improvement not only quantitatively but **qualitatively** by the means of combustion improvement.

2. *NO* emissions are very complicated and more careful future investigation is needed.

3. When the engine runs with hydrogen addition the smoke increases because of air-fuel ratio decrease. Future investigation is necessary to evaluate the specific particulate matter content *(g/kWh).*

REFERENCES

(1) “**Mihaylov M., K. Barzev**, Influence of hydrogen addition on single cylinder direct injection compression ignition engine parameters”, Proceedings, Volume 39, series 8, pp. 27 – 32, Transport and Mechanics, Rousse 2002.
(2) **Matiewskii D. D**, “Realization and influence of hydrogen addition on diesel engine performance, “Двигателестроение”, pp. 53 – 56, February, 1985.”
(3) **Heywood J**., “Internal Combustion Engine Fundamentals”, McGraw-Hill Book Company, 1988
(4) **Sherestha S. O., Bade A** “Before Treatment Method for Reduction of Emissions in Diesel Engines”, SAE Paper, 2000-0102791, 2000.