AN IMPACT OF USING A DUAL-INJECTION SYSTEM ON A COMBUSTION ENGINE'S WORKING PARAMETERS

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Abstract

The paper analyses the impact of using a dual- injection system on the eng ine's working parameters. The analysis is based on the results of the test stand research on a four-stroke four-cylinder, spark-ignition engine. Dual-injection system combines the multipoint injection with additional injection directly into the combustion chamber. It is used to increase the performance of an engine without affecting the fuel consumption or the low toxicity of exhaust gas. For the purposes of this research an en gine from Toyota Yaris wa s fitted with th e direct injection by insta lling high pressure injectors on th e cylinder head. This engine is factory-equipped with MPI. Based on the tests performed four controlling characteristics of the torque and brake specific fuel consumption were pre pared. The con trolling value was the proportion of the fuel injected into the inlet channel to the fuel injected directly into the combustion chamber. The same rela tionships were also described in relation to temperature of the exhaust gas and the fraction in volume in the exhaust gas of ingredients such as carbon monoxide, carbon d ioxide, unburned hydrocarbons and nitric oxid e. Points of measurement were decided upon based on the existing literature on the subject. The analysis of the test results shows a possibility of a 2-4% decrease in the brake specific fuel consump tion for the dual- injection engine in the low and middle RPM range and for the partial loads. The dual-injection engine also shows a decrease in the nitric oxide's fraction in the exhaust gas and a lower temperature of the exhaust gas. There was a small increase in the carbon monoxide and hydrocarbons.

Keywords: *dual-injector fuel system, MPI, direct injection, combustion, spark-ignition engine, experimental investigations*

1. Introduction

The aim of this research was to prepare controlling characteristics for chosen points of the engine's work where the variable was the fraction of the doses of fuel injected directly into the combustion chamber and by MPI system. The tests were performed on a test rig in the Chair of Internal Combustion Engines of the Cracow University of Technology. To perform these tests, the engine from Toyota Yaris with capacity 1.3 dm^3 was fitted with a direct injection system. More information about this system can be found in [2-5]. Fig. 1 shows how the direct injectors were mounted on the cylinder head of this engine.

Fig. 1. The cylinder head set-up: 1 - Common rail, 2 - High pressure injector, 3 - Cylinder head, 4 - Spacer plate of the inlet channel, 5 - Inlet channel

2. Test stand trials

The choice of the research points of measurement was decided upon based on the existing literature on the subject [2, 4, 5]. These papers point to partial loads at low and average RPMs, as the areas where the potential for using a dual-injection system should be the greatest.

Table 1 contains the points of areas of the engine's work used to prepare the controlling characteristics of the fraction of the direct injection x_{DI} in the total fuel mass per one work cycle of the engine. This fraction x_{DI} is defined by (1).

$$x_{DI} = \frac{G_{eDI}}{G_e}, \tag{1}$$

where:
x_{DI} - fraction of the direct injection x_{DI} in the total fuel mass injected, -,
G_{eDI} - fuel consumption of the fuel injected directly, kg/h,
G_e - total fuel consumption, kg/h.

Tab. 1. Points of the engine's work used to prepare the controlling characteristics of the fraction of the direct injection xDI

No.	Throttle opening position, α_{thr},%	Rotational Speed n, RPM	Brake mean effective pressure for the MPI, p_e, MPa
1.	13	2000	0.51
2.	20	1800	0.79
3.	20	2000	0.75
4.	25	2500	0.71

The minimum load, representing the brake mean effective pressure p_e, which is possible to obtain at the current configuration of the test rig was 0.51 MPa for dual-injection. Below this value there were issues with instability of the engine's work. This was caused by the unevenness of the doses of fuel delivered by each direct injector for short opening intervals. Shortening the time of injection below 0.3 ms is necessary to obtain lower x_{DI} fractions at partial loads cause the above phenomenon to occur.

The main values measured during the research were: crankshaft rotational speed, torque, fuel consumption, volume concentrations of the ingredients of the exhaust gas and their temperatures. The outside pressure p_{ot} and outside temperature T_{ot} was also measured so that the results could be recalculated for standard conditions.

At each trial constant RPMs were maintained as well as the stoichiometric mixture composition (with accuracy $\Delta\lambda = \pm 0.007$) and constant throttle opening. The determination of the mixture composition after the change in dose of the fuel injection was done automatically through an engine controller working in a closed loop with a wide-spectrum lambda probe. Based on the signal from the lambda probe the engine's control mechanism, depending on the need, would shorten or lengthen the time the MPI injectors were open, to make sure that the mixture was stoichiometric.

The engine's variable valve timing system was deactivated for these trials. It was done to make the measured values independent of the influence of the position of the camshaft. The advance ignition angle, for the direct injection, to the TDC (top dead centre) was also unchanged. Its value was determined during the pre-trials at 281° CA before TDC, which means that the direct injection occurs during the intake stroke. During those trials the pressure of the direct injection was set at 8 MPa.

3. The effect of using a dual-injection system on the engine parameters and fuel consumption

Based on the results of the research described in the introduction, the characteristics of torque M_o, fuel consumption g_e as a function x_{DI} (the fraction of the fuel mass injected directly into the combustion chamber in the total fuel mass per one working cycle of the engine) were taken.

The figures below show graphical representations of four controlling characteristics where the variable was the fraction of direct injection. The measurement results in all cases were approximated using second degree polynomials.

Figure 2 shows the torque and brake specific fuel consumption for the throttle opening of 13% and 2000 RPM.

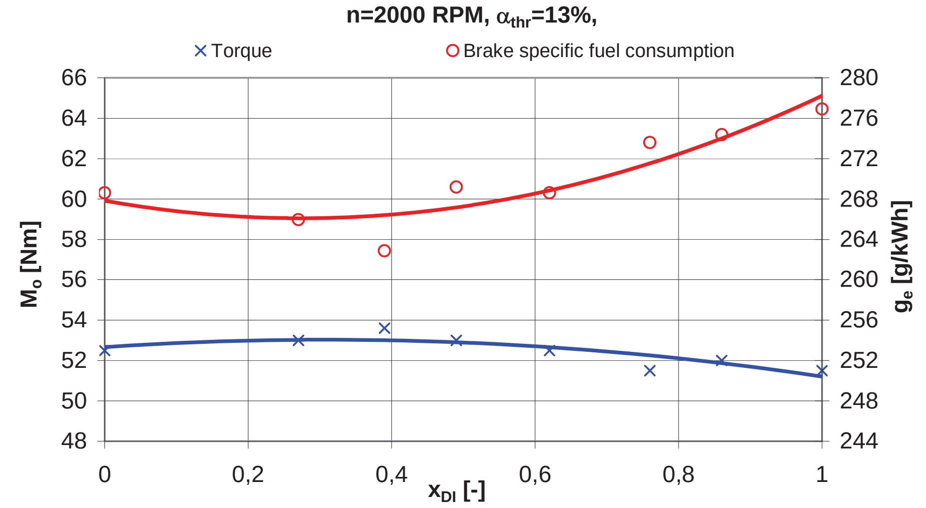

Fig. 2. Torque Mo and brake specific fuel consumption ge as a function of xDI (fraction of the fuel injected into the combustion chamber in the total fuel mass) for throttle opening at 13% and 2000 RPM

The above figure illustrates that the maximum torque and minimum brake specific fuel consumption was achieved for the fraction of the direct injection x_{DI} close to 0.4. The research results of this fraction of direct injection are significantly different from other results, especially those when the whole dose of fuel is directly injected into the combustion chamber.

Figure 3 illustrates results of test done at 1800 RPM with throttle opening at 20%.

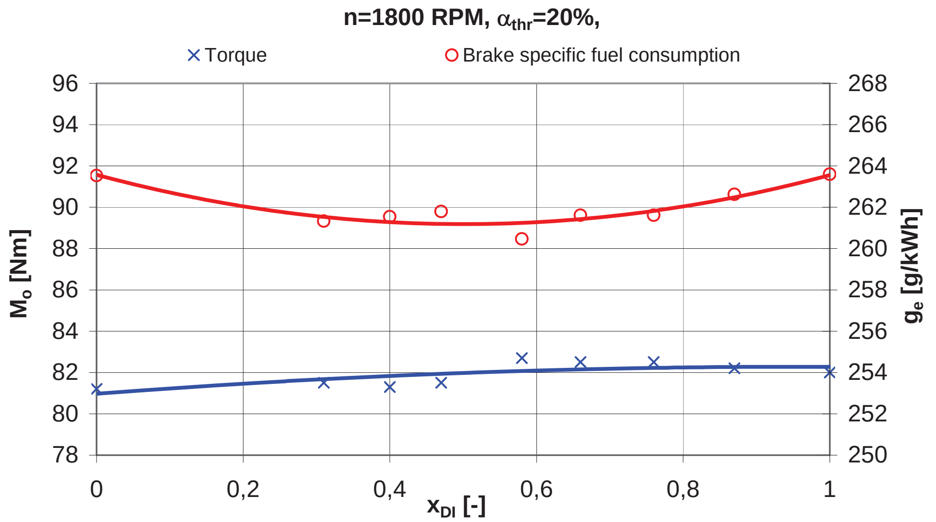

Fig. 3. Torque Mo and brake specific fuel consumption ge as a function of the direct-injection's fraction xDI at 1800 RPM and throttle opening at 20%

For this lower RPM speed and a larger throttle opening, the biggest torque and the smallest brake specific fuel consumption occur when the fraction of direct injection x_{DI} is 0.58. It should be pointed out that the differences in results for the different proportions of injection through the two injection systems are somewhat smaller in this case than was observed on Fig. 1.

The torque and specific fuel consumption as a function of the fraction of fuel injected directly into the combustion chamber at 2000 RPM and 20% throttle opening are presented in Fig. 4.

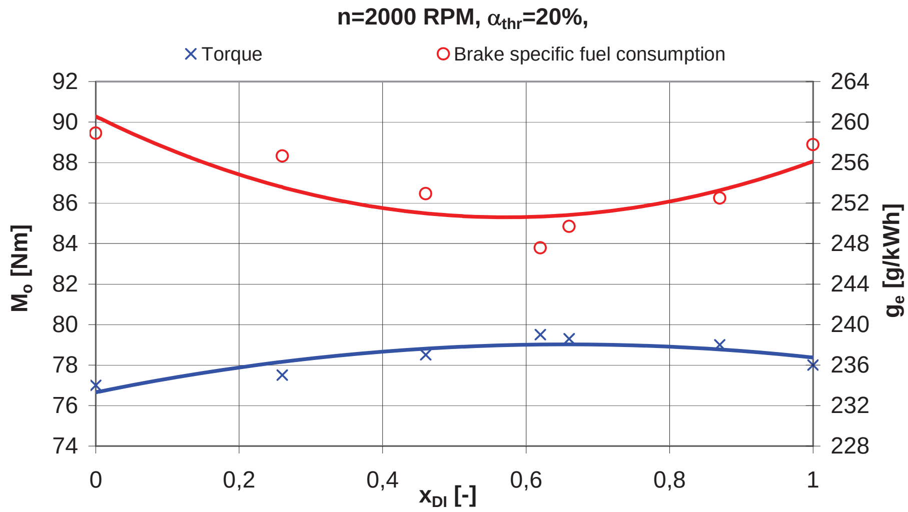

Fig. 4. Torque Mo and brake specific fuel consumption ge as a function of the fraction of fuel injected directly into the combustion chamber at 2000 RPM and 20% throttle opening

For the throttle opening α_{thr} of 20% and 2000 RPM the best brake specific fuel consumption and torque results were observed for the fraction of direct injection of 0.62. In this situation the working parameters of the engine show a marked improvement when compared to the situation where the whole dose of fuel is delivered through the MPI system.

Figure 5 shows the controlling characteristics for tests done at 25% of the throttle opening and 2500 RPM.

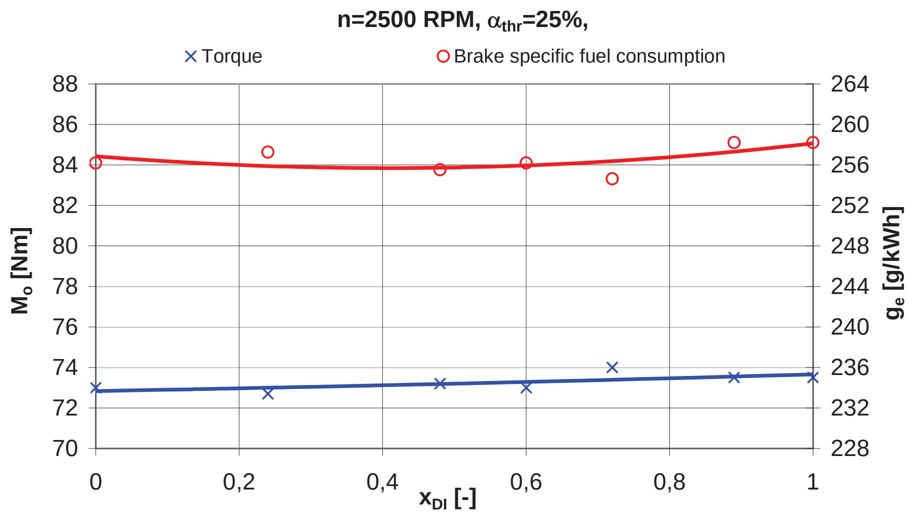

Fig. 5. Torque Mo and brake specific fuel consumption ge as a function of direct injection's fraction xDI at 2500 RPM and throttle opening of 25%

In these conditions the best torque and brake specific fuel consumption results were obtained for the fraction of direct injection of 0.72. The differences in results for other proportions of fuel mass are not significantly marked. Torque and brake specific fuel consumption are characterized by relatively flat changes, especially when compared with the previous results.

Figure 6 presents the graphs of the engine's total efficiency η_o and the plots of the relative increases of total efficiency $\Delta\eta_{DI+MPI}$ for the dual-injection system when compared to the MPI system, based on Fig. 2 and Fig. 4. The curves in Fig. 6 are a result of square polynomial approximations.

The engine's total efficiency was calculated from equation (2). A calorific value of petrol was 44000 kJ/kg [1].

$$\eta_o = \frac{3{,}6 \cdot 10^6}{g_e \cdot W_d}, \tag{2}$$

where:

η_o - engine's total efficiency, -,

g_e - brake specific fuel consumption, g/kWh,

W_d - petrol's calorific value, kJ/kg,

$3.6 \cdot 10^6$ - constant, resulting from the chosen units of measure.

The increase in the total efficiency $\Delta\eta_{DI+MPI}$ for the dual-injection system when compared to the MPI system was calculated using equation (3):

$$\Delta\eta_{DI+MPI} = \frac{\eta_o - \eta_{MPI}}{\eta_{MPI}} \cdot 100, \tag{3}$$

where:

$\Delta\eta_{DI+MPI}$ - increase in the total efficiency in relation to the efficiency when MPI system is used,%,

η_o - total engine efficiency for dual-injection system,%,

η_{MPI} - total engine efficiency with MPI system in given conditions (α_{thr}, n),%,

100 - coefficient used to get the result as a percentage,%.

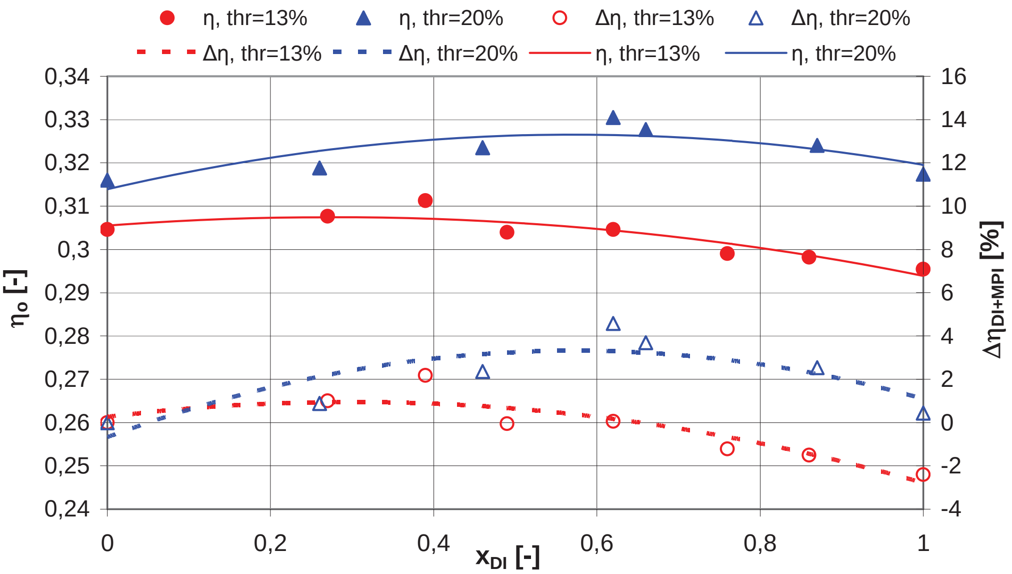

Fig. 6. Engine's total efficiency ηo as a function of the direct injection' s fraction xDI and t he relative increase in the total efficiency ΔηDI+MPI for the dual injection system when compared to the MPI

The biggest increase in the total efficiency $\Delta\eta_{DI+MPI}$ was 4.58% for the first and 2.18% for the second area of choice. In the first case the biggest improvement in the engine's economy was for the direct injection fraction of 0.62. In the second case analyzed in Fig. 6, the biggest improvement in the total efficiency when compared to the MPI was when the fraction of the fuel injected directly to the combustion chamber was 0.39.

4. Exhaust gas analysis for the dual-injection system

The analysis of the exhaust gas was done without the catalytic converter to be able to ascertain the concentrations of the gases leaving the cylinders. An efficient catalytic converter should reduce the concentration of harmful exhaust gas ingredients by about 98%, which makes the measurement of toxic ingredients after they have passes through the catalytic converter not as reliable as those done without it.

The temperature of exhaust gas in the exhaust channel was also taken. This data was used to construct the graphs on the following pages. They contain the concentrations of carbon monoxide, carbon dioxide CO_2, nitric oxide NO, unburned hydrocarbons HC and the exhaust temperature t_{exh}. The overall fraction of the hydrocarbons HC in the exhaust gas was calculated in hexane C_6H_{14} which is the substance commonly used for this purpose.

Figure 7. shows the volumetric concentrations of carbon monoxide CO, carbon dioxide CO_2 as a function of the fraction of direct injection, at 2000 RPM and throttle opening of 13%.

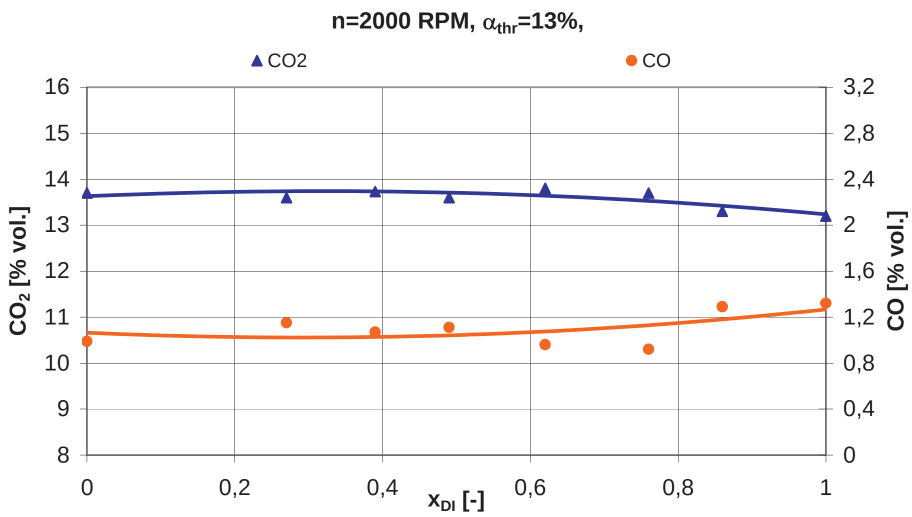

Fig. 7. Volumetric concentrations of carbon monoxide CO, car bon dioxide CO2 as a function of the fraction of direct injection, at 2000 RPM and throttle opening 13%

Graphs of exhaust gas temperature changes t_{exh}, nitric oxide NO and hydrocarbon HC concentrations in the exhaust gas, for the same as above conditions are illustrated graphically in Fig. 8.

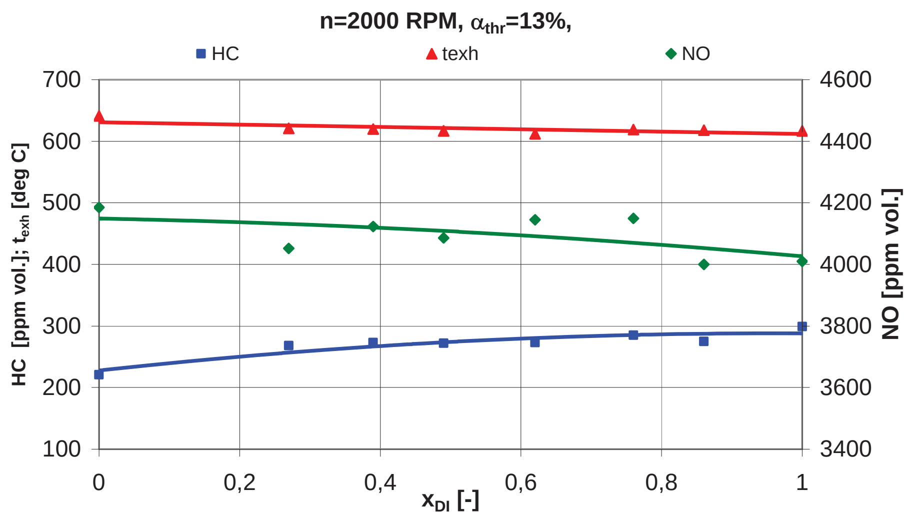

Fig. 8. Exhaust gas t emperature and the concentr ations of hydrocarbons and nitric oxide in the exhaust gas a s a function of the direct injection's fraction at 2000 RPM, throttle opening at 13%

A close analysis of Fig. 7 and 8 reveals that as the fraction of the direct injection increases, so do, to a certain extent, the concentrations of carbon monoxide and hydrocarbons in the exhaust gas, while the concentrations of nitric oxide and carbon dioxide decrease. There is also a slight decrease in the temperature of the exhaust gas leaving the cylinders. The difference in concentration of nitric oxide for MPI and dual injection is not large and is about 170 ppm. The fraction of hydrocarbons for the direct injection increases more significantly in this comparison, but it is still not very large at 290 ppm.

Figure 9 presents the concentrations of carbon monoxide and carbon dioxide occurring at throttle opening 20% and 1800 RPM.

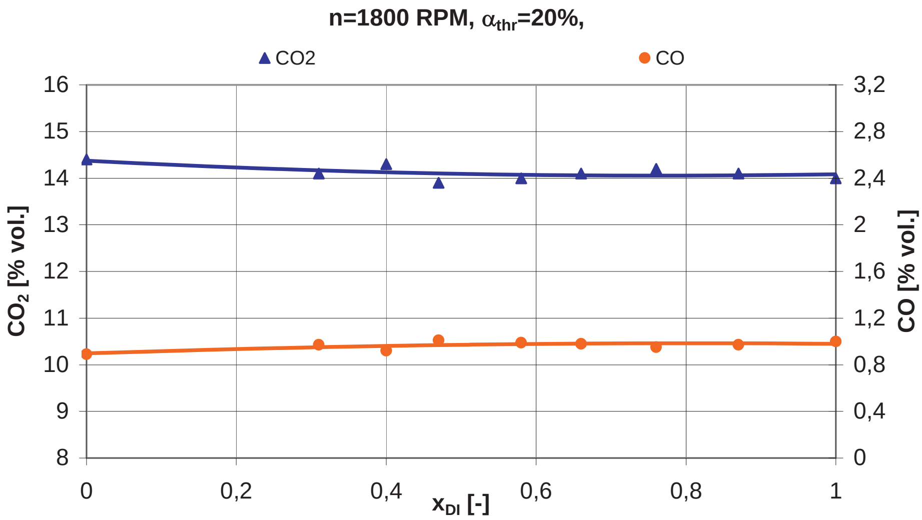

Fig. 9. Concentrations of carbon monoxide and carbon dioxide occurring at throttle opening 20% and 1800 RPM as a function of direct injection's fraction

Concentrations of nitric oxide and hydrocarbons in the exhaust gas and the exhaust gas temperature at 1800 RPM and throttle opening of 20% are presented in Fig. 10.

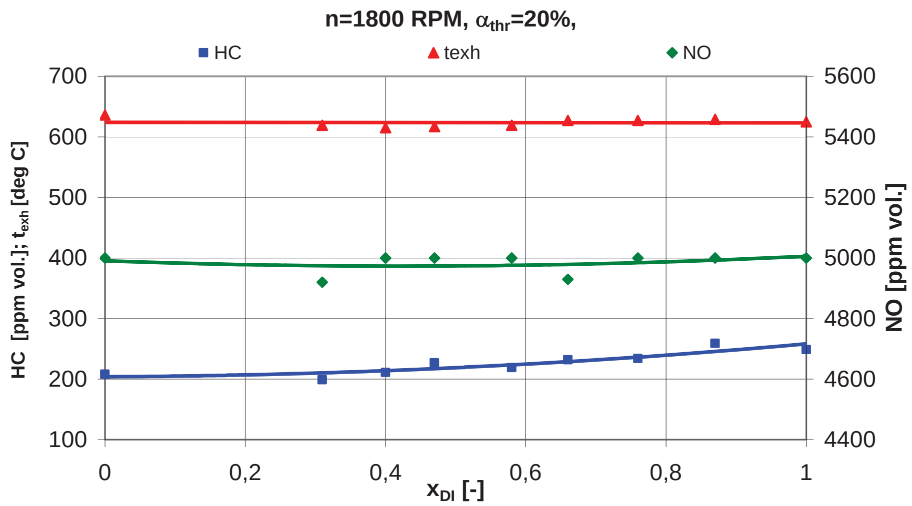

Fig. 10. Concentrations of nitric oxi de and hy drocarbons in the exhaust gas and the exhaust gas te mperature at 1800 RPM and throttle opening of 20%

The changes of the parameters of the exhaust gas at 1800 RPM and throttle opening of 20% are similar to the ones described above.

Figure 11 shows the results for 2000 RPM and throttle opening of 20%: concentrations of carbon monoxide, carbon dioxide as a function of direct injection's fraction x_{DI}.

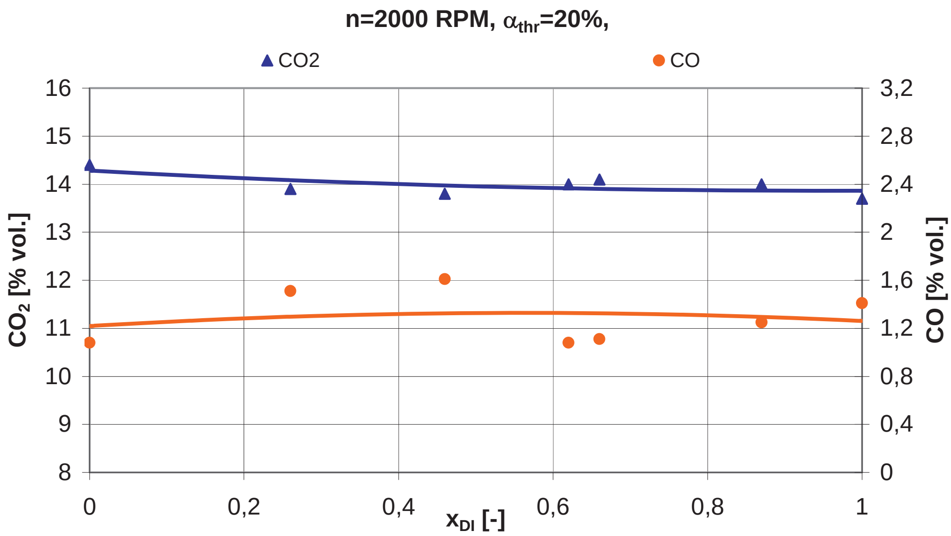

Fig. 11. Concentrations of car bon monoxide, carbon dioxide as a f unction of direct injection' s fraction xDI (2000 RPM and throttle opening of 20%)

Figure 12 shows the concentrations of nitric oxide and hydrocarbons and the exhaust gas temperature as a function of the direct injection's fraction for 2000 RPM and throttle at 20%.

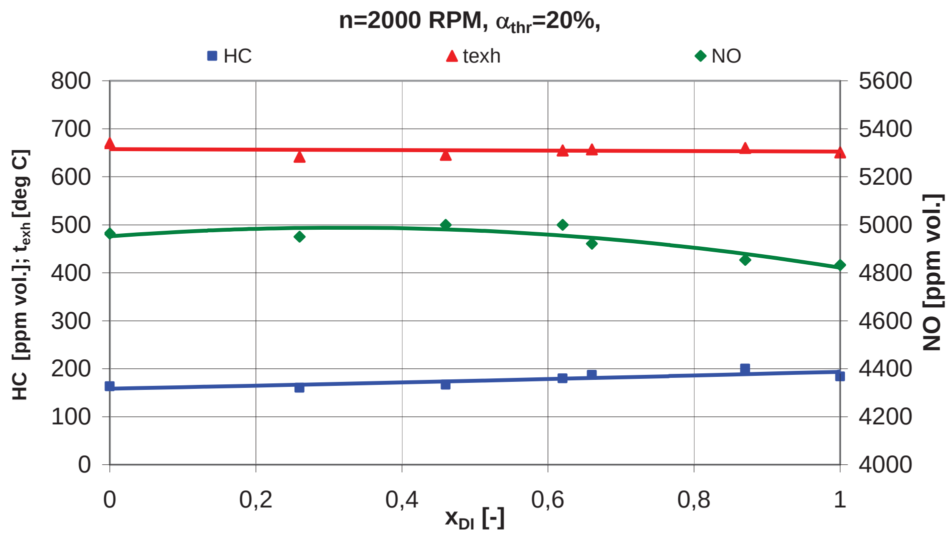

Fig. 12. Concentrations of nitric oxide and hydrocarbons and the exhaust gas temperature as a function of the direct injection's fraction for 2000 RPM and throttle at 20%

The results in Fig. 11 and 12 significantly differ from the results obtained before.

Figure 13 shows the curves of the carbon monoxide and carbon dioxide concentrations at 2500 RPM, throttle opening at 25%.

The changes in exhaust gas temperature, concentrations of nitric oxide and unburned hydrocarbons in the exhaust gas as a function of the direct injection's fraction at 2500 RPM and 25% throttle opening are presented Fig. 14.

The changes in the parameters characterizing the exhaust gas in Fig. 13 and Fig. 14 are not significantly different to the ones observed so far. Concentration of hydrocarbons in the exhaust gas increase by more than few dozens of ppm for direct injection fractions higher than 0.85. Until that point it hovers slightly above 100 ppm. The difference between the maximum and minimum nitric oxide concentration is 110 ppm.

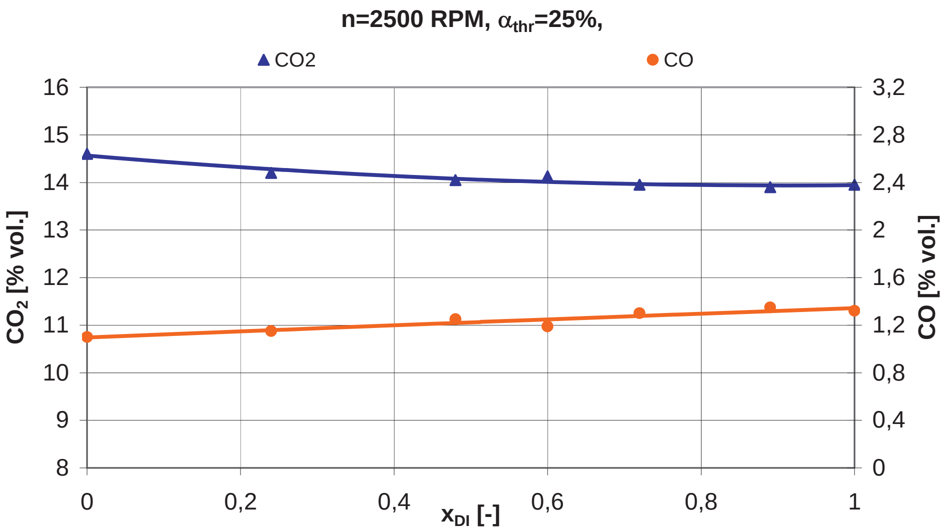

Fig. 13. Carbon monoxide and carbon dioxide concentrations at 2500 RPM, throttle opening at 25%, dual-injection system

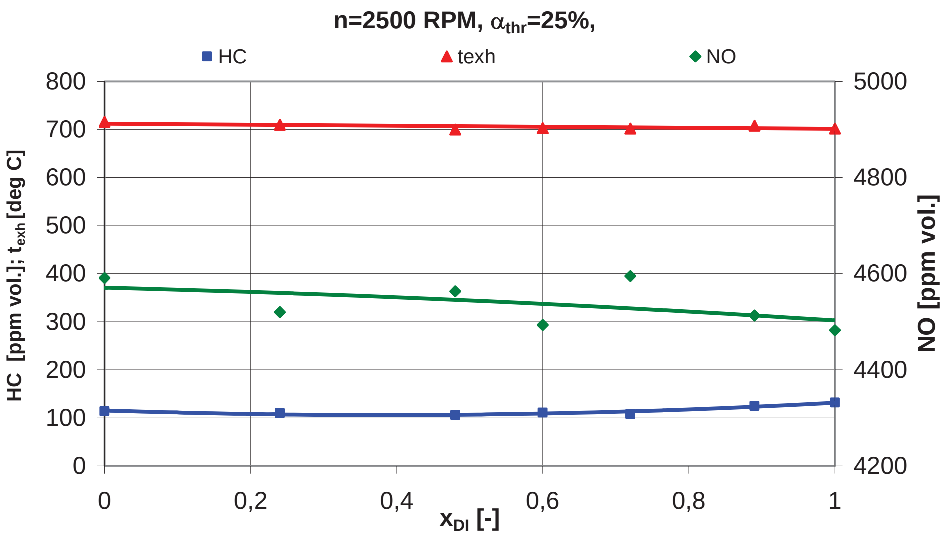

Fig. 14. Changes in exhaust gas temperature, concentrations of nitric oxide and unburned hydrocarbons in the exhaust gas as a function of the direct injection's fraction at 2500 RPM and 25% throttle opening

5. Conclusions

- The introduction of dual-injection system increases the torque of the engine and even more importantly decreases the brake specific fuel consumption. This means an increased total efficiency of an engine.
- For increased RPMs and loads the fraction of the direct injection, which results in the most beneficial increase in torque, increases.
- As the fraction of the direct injection increases there are no significant changes in the exhaust gas composition.
- The exhaust gas temperature decreases as the direct injection's fraction increases when compared to the MPI.
- When the fraction of direct injection is increased there is a small increase in the concentrations of carbon monoxide and hydrocarbons in the exhaust gas. The concentrations of carbon dioxide and nitric oxide decrease on the other hand.

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