EFFECTS OF USING A DUAL-INJECTOR FUEL SYSTEM ON A PROCESS OF COMBUSTION IN A SPARK-IGNITION ENGINE

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Abstract

This paper analyzes a process of combustion in a spark-ignition engine. The aim of the analysis was to determine the differences in the combustion process between the engine with a classic multipoint injection system and a system which injects fuel directly to the cylinders as well. To aid in the analysis the measurements of the high variable pressure in the engine's cylinder against the crankshaft angle were taken. This allowed us to obtain indicated diagrams for both types of engines. To eliminate distortions, both functions were approximated using combined functions. The other basic parameters were also measured: torque, RPM and instantaneous fuel consumption. The analysis of the indicated diagrams yielded an indicated mean pressure and thermal efficiency for both fuel systems. Then, a comprehensive analysis of pressures in cylinders was performed in order to determine the way the flame spreads in the cylinder for the two types of engines. The results obtained from this analysis show that the speed of combustion is greater for the charge formed by the dual-injector fuel system. The increased speed of the combustion, especially when it reaches a 50% fraction of the exhaust gas in the cylinder, is what accounts for the increase in the indicated mean pressure and the increase in the thermal efficiency of the engine with a dual-injector fuel system. The time of the spreading of the flame, as well as the fast burn period were reduced. The increased efficiency of the combustion process in the cylinder means that the heat losses through the cylinder sleeve are greatly reduced. The results of the research in this paper confirm the purposefulness of using a dual-injector fuel system in a spark-ignition engine.

Keywords: *dual-injector fuel system, gasoline, direct injection, port injection, combustion, spark-ignition engine*

1. Introduction

To be able to perform the analysis of any differences between the combustion processes of the multipoint injection and the dual injector fuel system, the indicating of the working space of one of the tested cylinders was made: the measurement of the high variable pressure were plotted against the position of the crankshaft. To do this, an optical-electronic pressure sensor Optrand C82255-SP (mounted on a specially modified spark plug) and an incremental encoder of the angle position Omron E6B-CWZ3E, were used. Data from both sensors was collected using a PC computer equipped with a National Instruments DAQCard-6062E measuring card and an application created using the LabView environment. The data stream was saved in a format that allowed for it to be imported into an Excel spreadsheet.

Other parameters were measured also. They were: RPM , torque M_o and instantaneous fuel consumption G_e as well as the temperature and outside pressure.

2. Effects of the injector system on an indicating diagram of an engine and on its working parameters

The research was conducted on a test stand in the labs of the Chair of Internal Combustion Engines. The test stand is shown in Fig. 1. The central part of the rig is a four cylinder spark-ignition engine from Toyota Yaris with a 1.3 dm^3 engine displacement.

The engine, besides the standard multipoint injection was also fitted with a direct injection. The aim of using such dual-injector fuel system is to obtain better engine performance and lower fuel consumption with lower emissions of toxic components of the fumes [2, 6]. A more detailed description of the test rig can be found in another paper [7].

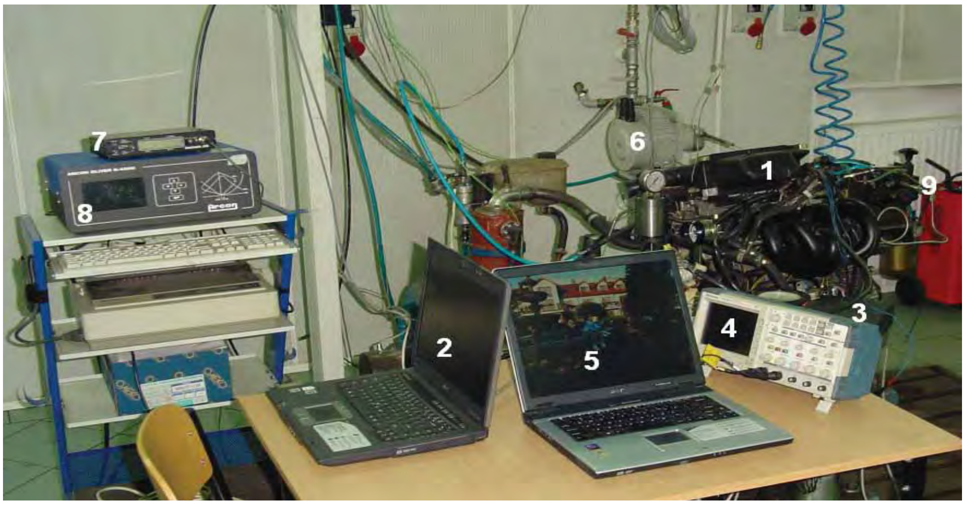

Fig. 1. The test stand 1 - Engine, 2 - PC computer to control the injector system , 3 - Controller of the injector-ignition system, 4 - Digital oscilloscope, 5 - PC computer used to take the measurements of the pressure inside the cylinder, 6 - Throttle valve controller, 7 - Fuel flow meter for the direct injection system 8 - Exhaust gas analyzer Arcon Oliver K-4500, 9 - Fuel pump of the direct injection system, with the fuel pressure controller

The results which were used to analyze the combustion process were taken at 2000 RPM and with 20% of throttle opening and for stoichiometric charge. The advanced angle of the spark ignition was 14° of crankshaft angle (CA). The intended absolute pressure for the inlet pipe was 0.079 MPa. The direct fuel injection pressure was set at 8 MPa, and its advanced angle was 281° of CA before the top dead centre (TDC). The fraction of the dose of the direct injection x_{DI} when combined injection was used was 0.62. It was for this value that the minimal unit fuel consumption was used at a given point of the engine's work.

The open indicated diagrams obtained for the engine with the multipoint injection and for the dual-injector are shown in Fig. 2.

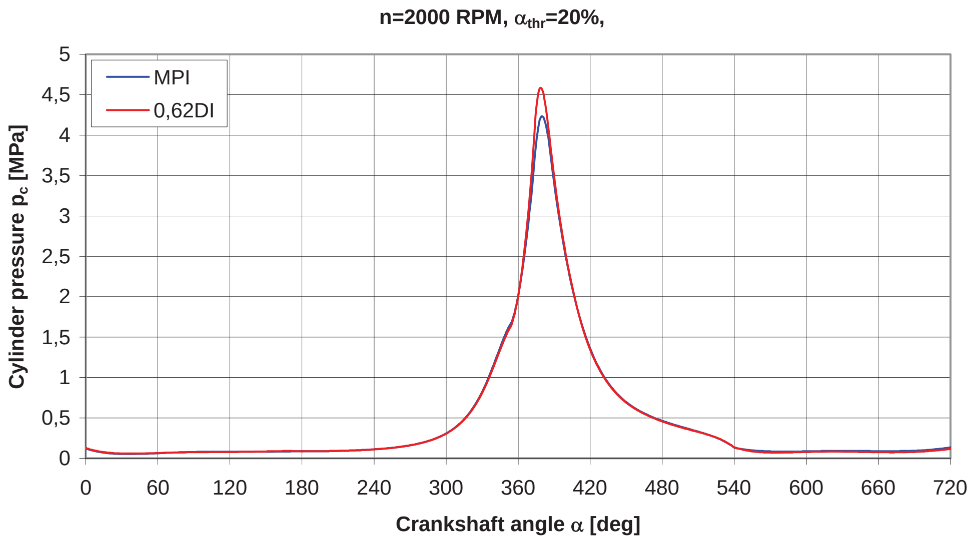

Fig. 2. Open indicated diagrams for the multipoint injection and for the combined injection with 0.62 fraction of the direct injection, 2000 RPM, 20% of throttle opening, stoichiometric charge

Based on information from Fig. 2, the closed indicated diagrams were obtained by calculating the instantaneous volume of the cylinder V_c as a function of the crankshaft angle α by using formula (1):

$$V_c = V_{ks} + \frac{\pi \cdot d_c^2}{4 \cdot 1000} \cdot \left[r \cdot (l - \cos\alpha) + l - \sqrt{l^2 - r^2 \sin^2\alpha} \right], \tag{1}$$

where:

V_c - volume of the cylinder, cm^3,
V_{ks} - combustion chamber volume, cm^3,
d_c - cylinder diameter, mm,
r - crank radius, mm,
l - crankshaft length between the axes of the cylinders, mm,
α - crankshaft angle, °,
1000 - divisor used to obtain the results in the chosen units of measure.

The changes in pressure as a result of the changing the crankshaft angle α against the instantaneous volume of the cylinder V_c are shown in Fig. 3.

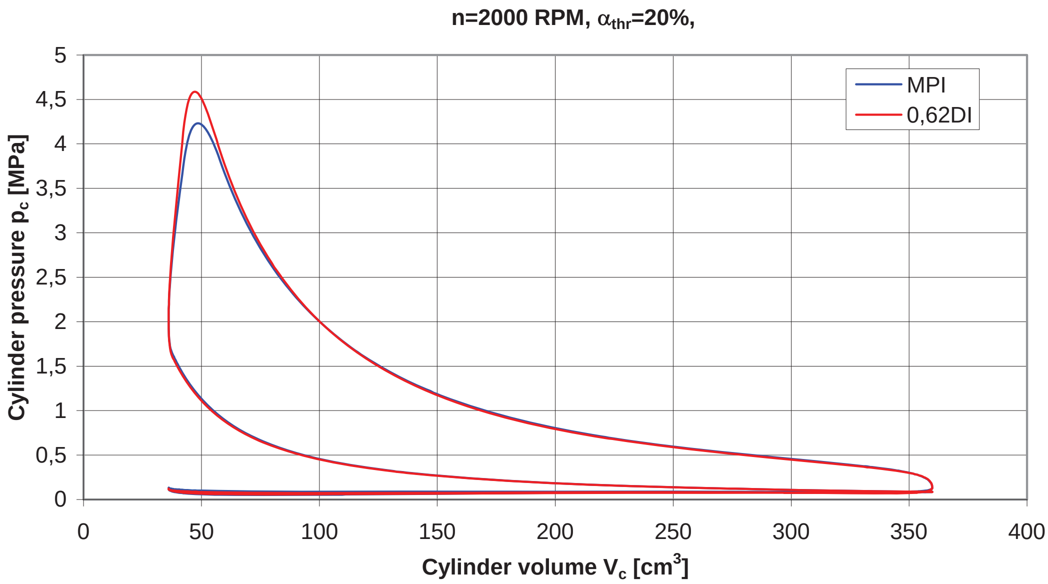

Fig. 3. A comparison of the indicated diagram for the multipoint injection and combined injection with 0.62 fraction of the direct injection; 2000 RPM , angle position of the throttle 20%

The diagrams presented above were approximated with combined functions from 10 averaged working cycles obtained during the test. The method used to calculate the speed with which the load is burned requires the indicated pressures to be free of the noise. The approximation of the real indicated diagrams was done in an Excel spreadsheet.

To achieve this, the polynomials of various degrees were used. For parts of the indicated diagram the compression and decompression stages were assumed to occur in a polytrope way with $p_c(V_c)^k$ = const. The value of the polytrope index was determined from the closed indicated diagrams described with logarithmic co-ordinates. The average deviation of the diagram derived from the approximated combined functions from the results registered by the sensor was 8.89×10^{-4} MPa for multipoint injection and 9.73×10^{-4} MPa for the dual injector system. In the range most important for analyzing the combustion process that is from 352 ° to 416 ° CA the average deviation is even smaller: 8.58×10^{-4} MPa for the multipoint injection and 9.45×10^{-4} MPa for the combined injection.

In both of the above diagrams, and especially in Fig. 3, one can see an enlargement of the area of the function representing the positive work of the engine's working cycle. The maximum pressure during the combustion process was 4.23 MPa at 21° CA after TDC for the multipoint injection and 4.60 MPa at 19.5º CA after TDC for the combined injection with 0.62 fraction of the direct injection. The maximum pressure is even higher for the dual injector system (by 0.37 MPa) when compared to the result for the just the multipoint injection system.

Figure 4 shows the increase of the cylinder pressure δp_c for the dual-injector system as a function of the crankshaft angle.

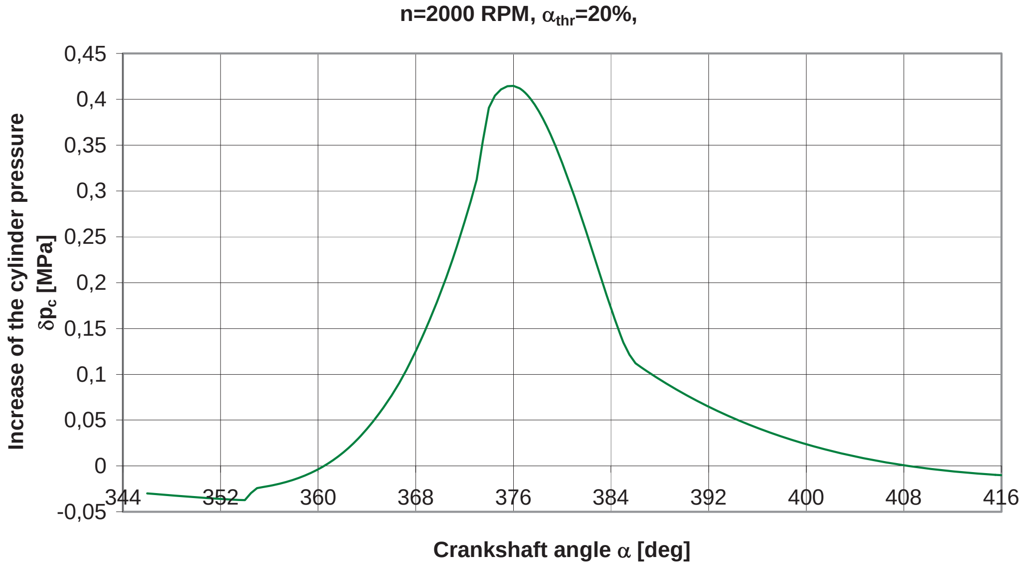

Fig. 4. The increase of the cylinder pressure δpc for the dual-injector system as a function of the crankshaft angle at (0.62DI) compared to the pressure in the cylinder of the multipoint injection (MPI).

To be able to further determine the differences in the course of the obtained indicated diagrams, the indicated mean pressure p_i was calculated based on the obtained results for both types of fuel injection. The numeric integration method was used on the appropriate areas of Fig. 3 (using a spreadsheet). To increase the accuracy of the calculations the trapezium method was used (with height of dV_c and the width of the base equal to the function values of $p_c = p_c(V_c)$. The results of this calculation are shown in Tab. 1.

The brake mean effective pressure p_e was calculated for both types of injection using the following formula (2):

$$p_e = \frac{\pi \cdot \tau \cdot M_o}{500 V_{ss}}, \tag{2}$$

where:

p_e - brake mean effective pressure, MPa,
τ - coefficient, for a four stroke engine equals 2, -,
M_o - torque, Nm,
V_{ss} - engine displacement, dm^3,
500 - divisor used to obtain the results in chosen units of measure.

Using equation (3) it was possible to calculate the thermal efficiency of the engine in both cases. The calorific value of fuel was 44000 kJ/kg.

$$\eta_c = \frac{N_i}{N_c} = \frac{30 \cdot p_i \cdot V_{ss} \cdot n}{G_e \cdot W_d}, \tag{3}$$

where:

η_c - engine's thermal efficiency, -,
N_i - indicated power, kW,
N_c - power delivered to the engine by fuel, kW,
p_i - indicated mean pressure, MPa,

V_{ss} - displacement, dm^3,
n - crankshaft rotational speed, RPM,
G_e - total fuel consumption per one hour, kg/h,
W_d - calorific value of fuel, kJ/kg,
30 - constant, as a result of chosen units of measure.

The results of the brake mean effective pressure and the thermal efficiency obtained using formulas (2) and (3) and of the indicated mean pressure is displayed in Tab. 1.

Tab. 1. A comparison of the engine's working parameters for the multipoint injection and the dual-injector system

	Indirect injection $x_{DI} = 0$	Dual-injector system $x_{DI} = 0.62$	Percentage increase to x_{DI}=0,%
Brake mean effective pressure p_e, MPa	0.745	0.769	3.22
Indicated mean pressure p_i, MPa	0.931	0.955	2.585
Thermal efficiency η_c, -	0.395	0.410	3.797

The final engine parameter in this part of the analysis of the indicated diagrams is the rate of the pressure rise $dp_c/d\alpha$. The changes of this parameter as a function of the crankshaft angle (for the most relevant to the analysis of the combustion process part of the indicated diagram) are shown in Fig. 5. The rate of pressure rise during the combustion process was assumed to be the main indicator of the possibility of an occurrence of an adverse phenomenon of knocking.

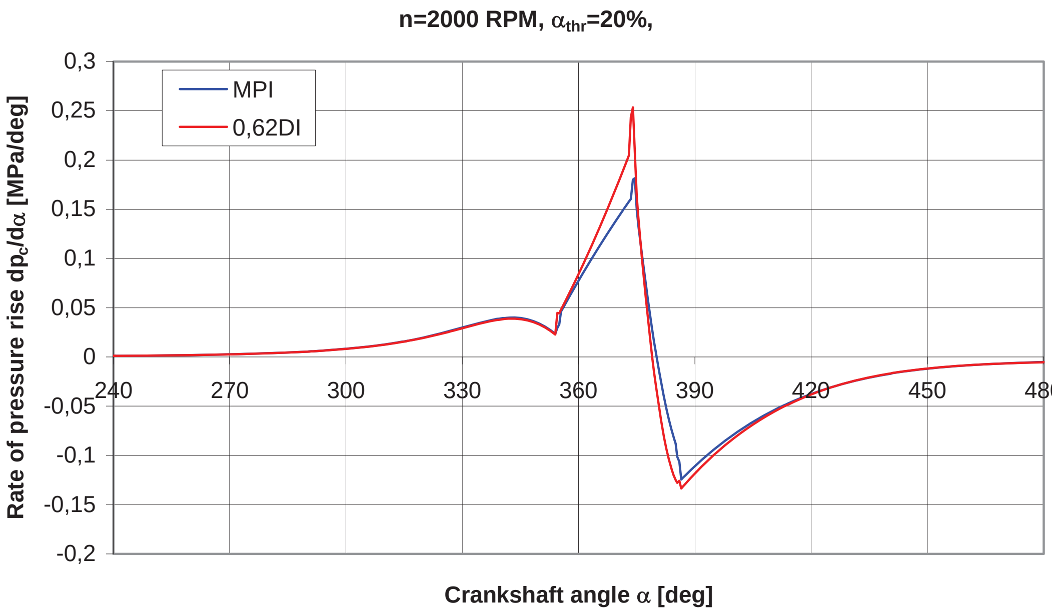

Fig. 5. The rates of pressure rise as a function of the crankshaft angle for the two fuel injection systems

The analysis of the results indicates a rise in the rate of pressure rise in the case of the engine with combined injection. The maximum value of the positive rate of change was 0.181 MPa/CA for multipoint injection and 0.253 MPa/°CA for the dual-injector system. It is worth mentioning here that knocking combustion usually occurs when maximum rates of pressure rise are larger than 0.5 MPa/°CA [4].

3. Effects of the fuel injection system used on the rate of combustion of the charge

The second stage of the analysis of the diagrams of the chamber's working volume in the cylinders of the two systems is to describe the process of combustion of the charge. The method employed allows determining the fraction of exhaust fumes in the cylinder as a function of the crankshaft angle; it is described in more detail in [5].

After a spark goes through the electrodes of the spark plug the rate of increase in pressure Δp_c in the cylinder's working volume when the crankshaft angle changes by $\Delta\alpha$ is made up of two components (4):

$$\Delta p_c = \Delta p_{sp} + \Delta p_V , \tag{4}$$

where:
Δp_{sp} - increase in pressure as a result of the combustion,
Δp_V - change in pressure resulting from the change in cylinder's working volume.

Fig. 6 shows the change in pressure in the cylinder p_{sp} as a result of combustion as a function of the crankshaft angle for the MPI injection and the dual-injector system.

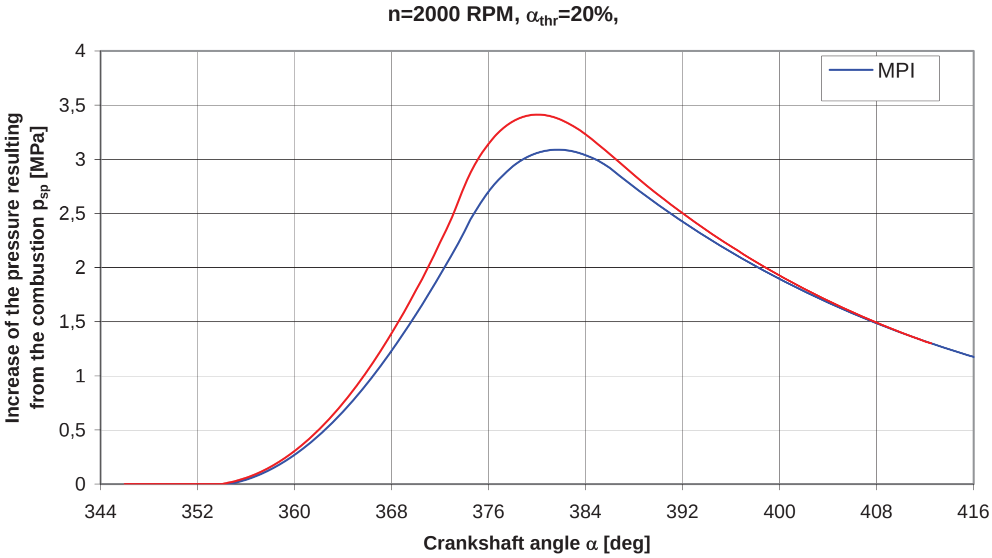

Fig. 6. The rate of increase in pressure in the cylinder's working volume resulting from the combustion process as a function of the crankshaft angle

The change of the angle of the crankshaft from α_i do α_{i+1} is associated with a change in the working volume of the cylinder from V_i ti V_{i+1} and in pressure from p_{ci} to p_{ci+1}. It was assumed that the change in pressure as a result of the change in the cylinder's volume is of a polytrope kind, with a previously calculated index of k. These assumptions mean that it is possible to simplify formula (4) to formula (5):

$$p_{ci+1} - p_{ci} = \Delta p_{sp} + p_{ci} \cdot \left[\left(\frac{V_i}{V_{i+1}} \right)^k - 1 \right]. \tag{5}$$

To calculate Δp_{sp} , the pressure change resulting from the combustion process, it can be changed to the following form (6):

$$\Delta p_{sp} = p_{ci+1} - p_{ci} \cdot \left(\frac{V_i}{V_{i+1}} \right)^k . \tag{6}$$

It should be remembered that the increase in pressure as a result of combustion cannot be directly proportional to the mass of the burned fuel, because the combustion process in an engine does not occur at constant volume. This is why it is necessary to relate the increased pressure to a given volume, which may be the volume of the combustion chamber V_{ks}. This is represented in formula (7):

$$\Delta p_{sp}' = \Delta p_{sp} \cdot \frac{V_i}{V_{ks}}. \tag{7}$$

After N changes of the crankshaft angle the rise in pressure as a result of the combustion process approaches zero. That means the end of the process of combustion. Putting forth the proviso that the adjusted change in pressure as a result of combustion Δp_{sp}' is proportional to the mass of the burned charge x_b we obtain (8):

$$x_b = \frac{\sum_0^i \Delta p_{sp}'}{\sum_0^N \Delta p_{sp}'}. \tag{8}$$

Fig 7. shows the changes in the mass fraction burned x_b as a function of the crankshaft angle α for both injection systems. The characteristic for the combustion process values of the mass fraction burned in the cylinder of 0.1 and 0.9 were marked. The value of the angle of the flame propagation $\Delta\alpha_r$ is determined by the moment in which the fraction of the mass of the burned charge is 0.1, according to the following formula (9):

$$\Delta\alpha_r = \alpha_{10\%} - \alpha_{ign}, \tag{9}$$

where:
$\Delta\alpha_r$ - flame propagation angle, °CA,
$\alpha_{10\%}$ - angle at which 10% of the charge is burned, ºCA,
α_{ign} - angle of ignition, ºCA.

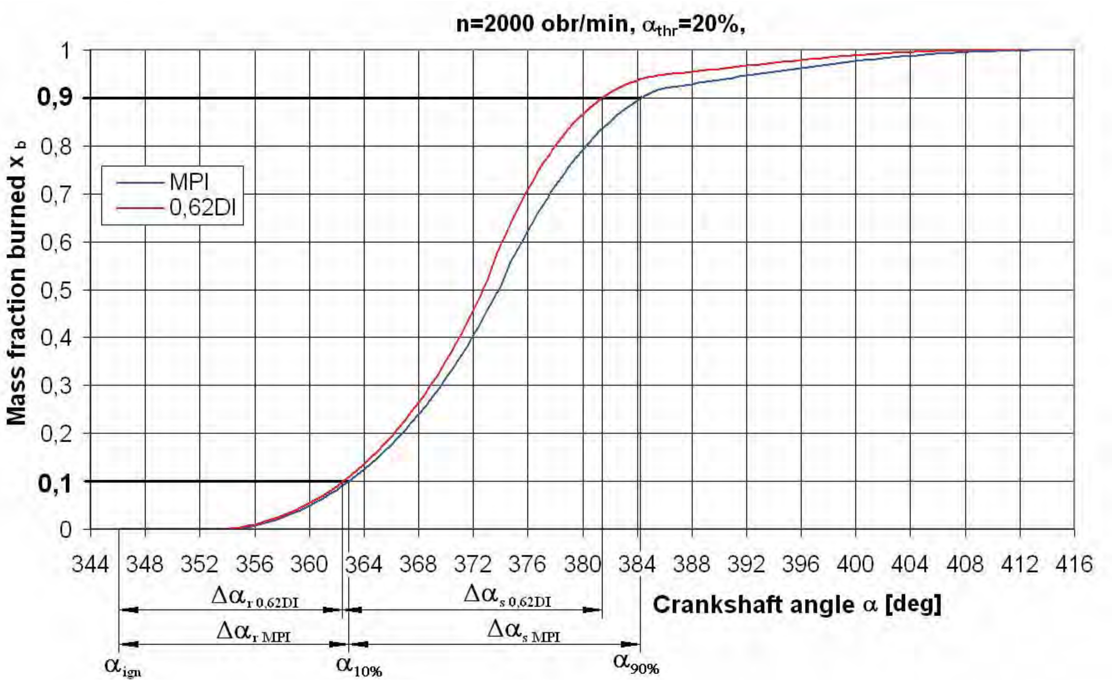

Fig. 7. Mass fraction burned xb as a function of the crankshaft angle α for both injection systems

Fast burn angle $\Delta\alpha_s$ - [3] – is defined by formula (10) as a difference between the value of the CA at which the fraction of the mass of the combusted charge is 0.9, and the angle at which the mass of the combusted charge is 10% of the cylinder's total charge.

$$\Delta\alpha_s = \alpha_{90\%} - \alpha_{10\%}, \tag{10}$$

where:
$\Delta\alpha_s$ - fast burn angle, °CA,
$\alpha_{10\%}$ - angle at which 10% of the charge is burned, °CA,
$\alpha_{90\%}$ - angle at which 90% of the charge is burned, °CA.

Total angle $\Delta\alpha_o$ of combustion is described as the sum of the flame propagation angle and the fast burn angle – formula (11):

$$\Delta\alpha_o = \Delta\alpha_r + \Delta\alpha_s \,. \tag{11}$$

The values of the angles that characterize the combustion process, which have been graphically represented in Fig. 7 are displayed in Tab. 2 for both the multipoint injection and the combined injection with 62% fraction of doses mass injected directly.

Tab. 2. Summary of angles characterising the combustion process from Fig. 7

No.	Angle	Symbol	MPI, °CA	0.62DI, °CA	Difference to MPI, °CA
1	of ignition	α_{ign}	346	346	0
2	at which 10% of the charge is burned	$\alpha_{10\%}$	363	362.5	-0.5
3	at which 90% of the charge is burned	$\alpha_{90\%}$	384.3	381.4	-2.9
4	of flame propagation	$\Delta\alpha_r$	17	16.5	-0.5
5	of fast burn	$\Delta\alpha_s$	21.3	18.9	-2.4
6	of complete combustion	$\Delta\alpha_o$	38.3	35.4	-2.9

In the case of the dual-injected system there was a decrease in the angles of flame propagation $\Delta\alpha_r$ from 17 to 16.5° CA and what is more important, fast burn angle $\Delta\alpha_s$ from 21.3 to 18.9° CA. The complete combustion angle which is the sum of those two angles was 38.3° CA for multipoint injection and 35.4° CA for the combined injection system. This results in a decrease of an angle, at which the most important part of the burning process takes place, by 2.9° CA or 7.6%. This is undoubtedly the reason for the increase in indicated mean pressure p_i and the thermal efficiency η_c, which were analyzed earlier.

Fig. 8 shows the curves of the instantaneous speed of combustion $dx_b/d\alpha$ as a function of the crankshaft angle for both injection systems. The speed combustion was derived through differentiation of fraction of charge mass burned x_b shown Fig. 7, against the crankshaft angle α.

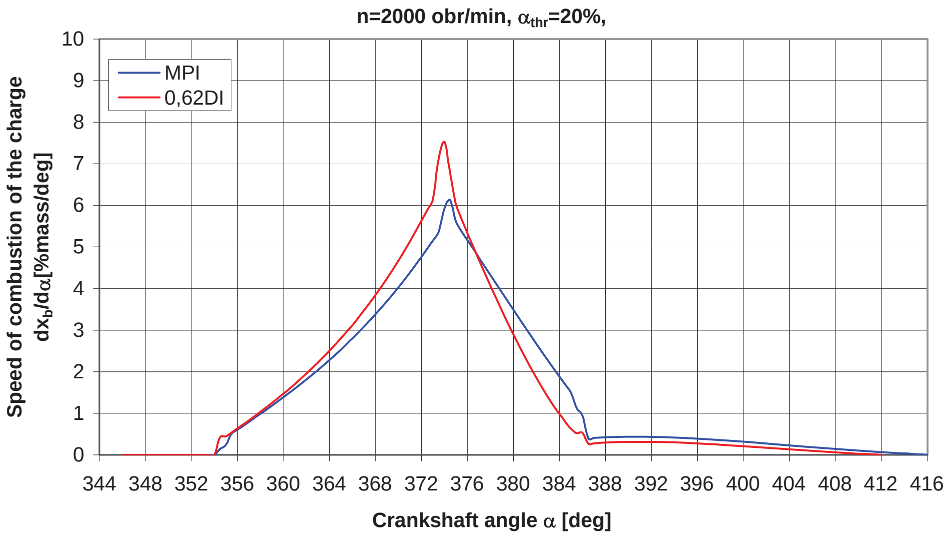

Fig. 8. The speed of combustion of the char ge in the cylinder $dx_b/d\alpha$ as a function of the crankshaft angle for bot h injection systems

Instantaneous speed of charge combustion for the most part of the fast burn period has values greater, on average, by 0.54% of the combusted charge's mass for every 1° turn of the crankshaft in the case of the dual-injector fuel system. The absolute difference in speed of combustion reaches the maximum value of 1.76% of mass/ºCA at 373.5ºCA. In the second part of the fast burning period when the fuel is injected to port injection the process is intensified. The increased speed of charge combustion in the first part of the process (until 50% of the mixture mass is burned) is the biggest influence on increasing the thermal efficiency of the engine η_c [3].

This confirms the positive effects of using dual-injector fuel system on the combustion process in the chosen range of engine's work.

4. Conclusions

The following conclusions can be made based on this research:

- A 2.6% increase of the indicated mean pressure and a 3.8% increase in the thermal efficiency were obtained as a result of using a dual-injector fuel system when compared to multipoint injection.
- These increases indicate an increased efficiency of the combustion process when a dual-injector system is used.
- When a mixture is burned in a shorter time there are smaller heat losses through the cylinder walls, because its surfaces that are in contact with a combusting charge are smaller.
- As a result of using a dual-injector system in a spark-ignition engine, the engine's work parameters such as: indicated mean pressure p_i and thermal efficiency $\eta_{c,}$ are improved. This directly effects engine's total efficiency η_o.

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