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The injector location impact on the fuel combustion process in a direct gasoline injection system

The article contains an analysis of the fuel dose combustion phenomena and exhaust emissions in a direct injection system of an SI engine for variable injector location in the combustion chamber. The research performed is a continuation of the research presented in the article CE-2018-104. The tests were performed using the AVL Fire 2017 simulation environment. 27 injector placement combinations in three planes were analyzed: axial distance from the cylinder axis, injector depth relative to the head and angular position relative to the cylinder axis. An optimal solution was chosen, taking into account the significance of individual indicators. It was shown that the greatest impact in terms of the most advantageous combustion process indicators is the injector setting depth in the combustion chamber cavity, while the distance from the cylinder axis is of secondary importance. The smallest changes in the combustion and emission factors values are seen with the change of the injector placement angle (in the value range used in this study).

Key words: *gasoline direct injection, fuel combustion, exhaust emission, simulation software*

1. Introduction

Combustion process tests carried out on real combustion engines require high financial and material investment relative to the same type of research conducted using the CFD (computational fluid dynamics) technique. The initial stage of combustion system design relies on data from simulation projects [1, 13, 16, 22]. Despite this fact, the final verification of such results are tests performed on real test objects and prototypes [2, 7, 15, 20].

Numerous research works conducted by scientists around the world confirm the usefulness of research on the hydrocarbon fuel mixtures combustion process. Huang et al. [9] found that direct injection of a dose of ethanol into a homogeneous gasoline-air mixture has a positive effect on the combustion knock prevention, but it also has a negative impact on the exhaust emission values. The compromise can, however, be achieved by a more careful choice of the ethanol injection angle. The earlier it is, the smaller is the negative impact on exhaust emissions, but then the ethanol cooling properties are not fully exploited to prevent knock.

Research on fuel mixing is also one of the subjects in the development of CI engines. Lee et al. [14] have demonstrated the high potential of using additional gasoline injection for the SI engine at low loads to significantly reduce nitrogen oxides emissions.

The results presented below are a continuation of research on the fuel injection process shaping by modifying the injector placement in the combustion chamber. The results of these tests can be found in [18]. These previous tests allowed to determine the most advantageous position of the injector in the aspect of injection and fuel atomization. The same variants of injector placement settings were used in this article to analyze the combustion process and exhaust emissions.

The presented considerations are aimed overall at analyzing a dual-fuel system in which both injectors are placed in the combustion chamber. Similar studies of the dual-fuel system in the PFI-DI configuration have already been carried out earlier [4, 5, 8, 16, 19]. These studies, however, do not encompass the same methods and measurement parameters as intended for this article, and subsequent tests are aimed at determining the fuel mixing indicators in the combustion chamber immediately before ignition.

2. Research aim and goals

The proposed tests constitute the injection and combustion process study stage for a direct injection system of liquid hydrocarbon fuels [17]. This stage focuses on determining the engine processes indicators using one direct gasoline injection injector while changing its location in the engine combustion chamber.

The goal of the research is to determine the optimal spatial position of the injector relative to the spark plug and the angular position of its axis relative to the cylinder axis. The optimal location will be defined as such a position, at which the sum of thermodynamic combustion and emission indicators will be the highest, while remaining within the set limits and considering the weight of each indicator.

3. Research methodology

3.1. Combustion chamber geometry

The combustion process tests were done using the AVL Fire 2017.1 simulation software. The shape of the combustion chamber was modeled (Fig. 1a) and imported into the simulation software (Fig. 1b). The displaceable mesh with a square side of 1 mm was automatically condensed in the vicinity of the spark plug to a value of 0.1 mm. The engine specifications are shown in Table 1.

a) b)

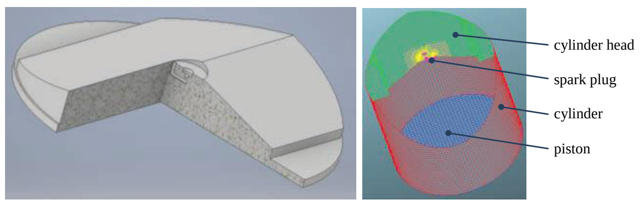

Fig. 1. The combustion chamber including the spark plug: a) 3D drawing, b) the mesh in AVL Fire program

Each injector position is described by a code describing the position change with respect to the y-axis, with respect to the z-axis and a change of angle with respect to the axis of the cylinder:

$$y(i)z(j)alpha(k) \tag{1}$$

where: i = 7 mm, 8 mm and 9 mm, j = 9 mm, 10 mm and 11 mm, and k = 15, 17.5 and 20 deg.

Table 1. Modeled engine technical data

Parameter	Unit	Value
Type	–	Piston engine, 4-stroke, spark ignition
Cylinder number	–	1
Displacement	cm^3	385
Compression ratio	–	10.2
Bore	mm	83
Stroke	mm	71.2
SOI	deg	670
Injection duration	ms	0.6
Injected fuel dose	mg	13,1
λ-value	–	1
Ignition CA	deg	690

Test conditions include 27 injector placement configurations. These configurations are shown in Fig. 2, and the matrix of test settings in Fig. 3.

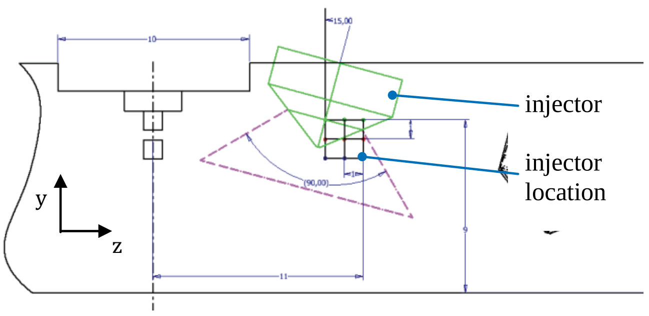

Fig. 2. The combustion process tests configuration considering the change of the linear and angular position of the injector

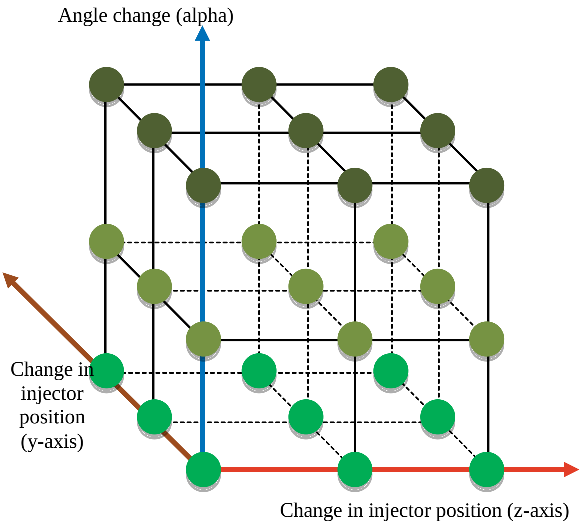

Fig. 3. The combustion process test with injector placement variants matrix

3.2. Initial conditions

The initial conditions of the compression process and the combustion process were adopted in accordance with the values listed in Table 2. Such conditions correspond to a spark-ignition engine with direct petrol injection operating at the speed of 2000 rpm.

Table 2. Initial conditions accepted for simulation calculations

Parameter	Unit	Value
Pressure	bar	0.6
Temperature	K	300
Turbulence kinetic energy (specific)	m^2/s^2	10
Turbulence integral length scale	mm	3
Turbulence dissipation rate	m^2/s^3	1732.05
Tumble rotational speed	rpm	3000
Engine speed	rpm	2000
Crank angle	deg CA	570–800

3.3. Combustion process and exhaust emission modeling

3.3.1. Combustion modeling

The conditions inside the combustion chamber before combustion were determined according to the process described in [18]. The Extended Coherent Flame Model (ECFM) with the spherical shape of the initial flame nucleus was adopted for combustion simulation calculations [3].

Coherent Flame Model (CFM) is built on the basis of a laminar flamelet concept, whose velocity S_l and thickness δ_l are mean values, integrated along the flame front, only dependent on the pressure, the temperature and the fresh gases content.

ECFM is often used to model combustion in spark-ignition engines. Ji et al. [10] compared the combustion process of gasoline and gasoline mixtures with hydrogen modeled using ECFM with a simulated combustion process. The error was deemed to be less than 6%. This model is also used to analyze the combustion process in hydrogen-only engines. Knop et al. [11] used this model to simulate combustion in an engine with indirect and direct hydrogen injection. Colin et al. [6] showed a good correlation between the ECFM model and the combustion process in the first 1.8 GDI engine used in a passenger car.

3.3.2. Modeling the exhaust emission

The use of the Extended Coherent Flame Model results in a two-step fuel combustion reaction according to reactions (2) and (3).

$$C_nH_mO_k + \left(n + \frac{m}{4} - \frac{k}{2}\right) O_2 \rightarrow n\,CO_2 + \frac{m}{2} H_2O \tag{2}$$

$$C_nH_mO_k + \left(\frac{n}{2} - \frac{k}{2}\right) O_2 \rightarrow n\,CO + \frac{m}{2} H_2 \tag{3}$$

In the above formulas n, m and l represent the number of carbon, hydrogen and oxygen atoms of the considered fuel.

The mean laminar fuel consumption rate is the sum of the reaction rates of the above reactions, whereas their respective values are dependent on the local equivalence ratio and the number of carbon and hydrogen atoms.

The Extended Zeldovich Model, which utilizes the reaction formulas (4)–(6), was used to describe the NO formation.

$$N_2 + O \underset{k_{1b}}{\overset{k_{1f}}{\rightleftarrows}} NO + N \tag{4}$$

$$N + O_2 \underset{k_{2b}}{\overset{k_{2f}}{\rightleftarrows}} NO + O \tag{5}$$

$$N + OH \underset{k_{3b}}{\overset{k_{3f}}{\rightleftarrows}} NO + H \tag{6}$$