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Research Article

MODELING OF THERMAL CYCLE CI ENGINE WITH MULTI-STAGE FUEL INJECTION

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ABSTRACT
This work presents a complete thermal cycle modeling of a four-stroke diesel engine with a three-dimensional simulation program CFD - AVL Fire. The object of the simulation was the S320 Andoria engine. The purpose of the study was to determine the effect of fuel dose distribution on selected parameters of the combustion process. As a result of the modeling, time spatial pressure distributions, rate of pressure increase, heat release rate and NO and soot emission were obtained for 3 injection strategies: no division, one pilot dose and one main dose and two pilot doses and one main dose. It has been found that the use of pilot doses on the one hand reduces engine hardness and lowers NO emissions and on the other hand, increases soot emissions.

Keywords: thermal cycle, diesel engine, multi-stage fuel injection, CFD.

INTRODUCTION

Half of contemporary manufactured cars are powered by compression-ignition engines, called diesel engines. Their advantages include: greater reliability, fuel economy, greater power range, longer service life, faster response to power demand, increased torque, better power characteristics and lower diesel prices [1]. Fuel combustion in a diesel engine is accompanied by emissions of harmful exhaust gases such as nitrogen oxides (NO_x), hydrocarbons (THC), carbon monoxide and carbon dioxide (CO, CO_2), soot and particulate matter (PM). The biggest problem is NO_x emission reduction [2]. High internal cylinder temperatures and excess oxygen are favourable conditions for the formation of these compounds [3].

The challenge for newly engineered engines is the compromise between efficiency and performance as well as the emission of harmful compounds. In diesel engines, the challenge is to develop technology and optimize fuel injection [5, 6]. The impact of different injection strategies on combustion is extremely important. Fuel injection modifications can be implemented in a variety of ways, mainly by varying the injection time and fuel pressure delivered to the engine. The increased fuel pressure allows for better spraying in the combustion chamber, while the injection timing, in strictly defined phases, allows the reduction of harmful emissions [4].

Injection systems for diesel engines have developed in recent years. The most modern of these are Common Rail systems that maintain high injection pressure and precise fuel dose distribution and controlled by the electronic control unit of the ECU. Although high-pressure injection systems have been used for several years now, work is still underway on how to distribute fuel to the engine cycle, developing the optimum injection strategy.

In systems using multi-stage injection, individual doses of fuel fulfil strictly defined functions. Pilot dose reduces noise due to reduced combustion pressure increase. Pre-injection reduces the amount of fuel delivered during the autoignition delay, resulting in an additional reduction in NO_x emissions. The injection intensifies the PM particle combustion in the filters, while the droplet enhances the conversion in the catalytic reactor.

Developing modern engines with high efficiency and meeting increasingly stringent exhaust emission standards is becoming increasingly difficult, laborious and costly. More recent and better research methods are required. More and more frequently, mathematical modeling is used to analyze the processes taking place in the engine working cycle.

The purpose of this paper is to model the compression ignition (CI) engine cycle and to analyze the numerical process of the multi-stage diesel injection in S320 Andoria engine. The study concerns the complete thermal cycle of four-stroke CI engine. Modeling was done using the AVL Fire program designed to simulate combustion engine operation. As a result, information on the effect of the fuel dosage configuration on the parameters of the combustion process, such as ignition delay, combustion duration, indicated mean effective pressure and emission of toxic exhaust gas components of the test engine was obtained.

RESEARCH OBJECT

The modeling facility was a four-stroke, stationary S320 Andoria diesel engine. This engine has a horizontal cylinder system with two valves and a cooling system by evaporating the water jacket. The engine is a stationary unit designed to operate at constant maximum load and at constant rotational speed - 1500 rpm. The engine power supply uses a high-pressure single-sided injection pump. Figure 1 shows the view and cross-section of the test engine. Table 2 shows the basic parameters of the engine.

ENGINE MODEL

Mathematical modeling of the working cycle of a combustion engine begins with establishing a physical model of phenomena occurring during its duty cycle, then requires a mathematical description of the predetermined physical model, acceptance of appropriate initial and boundary conditions, and numerical solution of sets of equations of the accepted mathematical description [5, 6].

Table 1. The main parameters of the engine

Parameter	Value
Type	four-stroke
Number of cylinders	1
Cylinder arrangement	horizontal
Rotational speed	1500 rpm
Compression ratio	17
Bore x stroke	120 x160 mm
Squish	2.5 mm
Injection angle	15 deg before TDC
Intake valve opening (IVO)	23 deg before TDC
Intake valve closure (IVC)	40 deg after BDC
Exhaust valve opening (EVO)	46 deg before BDC
Exhaust valve closure (EVC)	17 deg after TDC

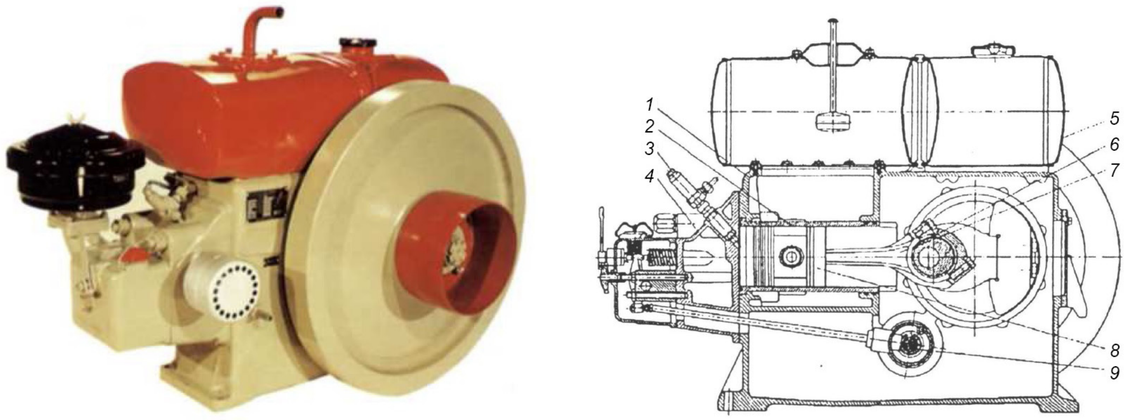

Fig. 1. The view of test engine S320 Andoria and the cross section of the engine, 1 - engine body, 2 - cylinder liner, 3 - injector, 4 - head, 5 - fuel tank, 6 - connecting rod, 7 - crankshaft, 8 - piston, 9 - camshaft

For piston engines, modeling is a highly complex process, because it includes the thermodynamic and gasodynamic phenomena within the variable volume cylinder, including combustion chemistry and charge exchange processes. The complexity of the combustion process itself in a reciprocating engine, inter alia due to its non-stationary nature, requires modeling of the introduction of simplifying assumptions. Mathematical modeling of the working cycle of a piston engine encounters three essential constraints: insufficient description of the actual physical system being analyzed, imperfection of numerical methods used to solve the described equation systems, computer capabilities related to their capacities and speed of operation. Research based on numerical simulations using advanced mathematical models has recently been intensively developed. The development of numerical modeling is possible by increasing the computational power of computers, which allows to model not only the flow processes, but also the combustion process in 3D [7÷9]. One of the more advanced numerical models used for modeling the combustion process in piston engines is AVL Fire [10÷12].

The Fire program provides the ability to model heat-flow processes in the intake and exhaust manifolds and in the combustion chamber of the piston engine. This program allows you to calculate the transport, mixing, ignition and turbulent combustion phenomena in the internal combustion piston engine. Modeling can be made of both homogeneous mixtures prepared in a combustion chamber and heterogeneous (non-homogeneous) mixtures formed by fuel injection into a chamber, and calculations may also involve a spark-ignition engine or a diesel engine [13÷15]. In AVL Fire chemical kinetics is described by combustion models that take into account oxidation processes at high temperatures. Several models are available for autoignition of hydrocarbon fuels, and also for modeling knock in piston engine compartments. The program allows the creation of a three-dimensional grid of computational space for the engine, including inlet and outlet channels with respect to the camshaft phases and cargo exchange processes.

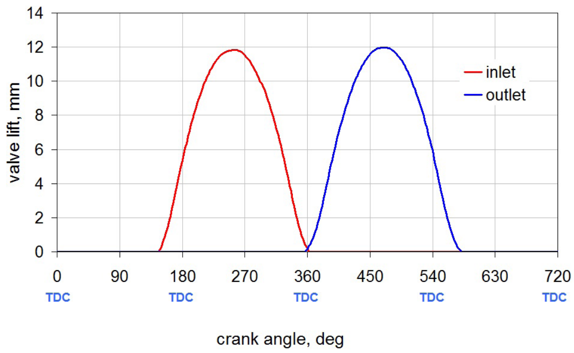

Fig. 2. Timing of the test engine

CFD modeling starts with the creation of a three-dimensional grid. In the case of complete cycle simulation of the engine, account should be taken of the phase of the load exchange with the timing mechanism and the geometry and movement of the valves. Fig. 2 shows the actual valve heights of the test engine used for timing of phase simulation.

Creation of a computational grid begins with plotting, based on the actual engine dimensions, the geometry of a 3D object in a CAD program.

In order to make the calculation results independent of the density of the computational mesh, initial simulations were made for several different grid resolutions. The impact of grid density on the pressure and temperature of the combus-

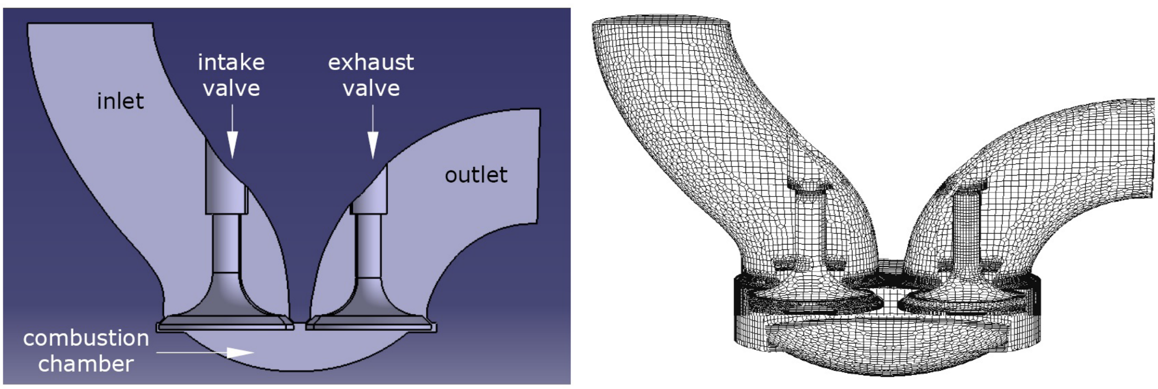

Fig. 3. CAD geometry and computational mesh of the test engine

Table 2. Modeling parameters

Parameter	Value
Factory angle fuel injection	15 deg before TDC
Fuel	Diesel
Dose per cycle	68 mg
Initial pressure	0.095 MPa
Initial temperature	365 K
Air density	1.19 kg/m^3

Table 3. Used AVL Fire submodels

Model	Name
Combustion model	Coherent Flame Model ECFM
Turbulence model	k-zeta-f
NO formation model	Extended Zeldovich Model
Soot formation model	Lund Flamelet Model
Evaporation model	Dukowicz
Breakup model	Wave

tion process was analyzed. The mesh density was increased sequentially until it had no significant effect on the pressure and temperature. As a result of these calculations, a grid of about 160k was chosen. Cells dimension was from 0.25 to 2 mm.

Depending on the researched phenomenon, AVL Fire provides the user with a variety of models that are appropriate for the modeled and analyzed process. The selected modeling parameters indicating the assumed initial conditions and models used in the calculation are shown in Table 2 and 3.

RESULTS OF ANALYSIS

Literature analysis [4,16,17] on multistage fuel injection research shows that the most commonly used injection strategy is a main injection split, where most of the fuel is delivered to the cylinder, about 60-90% and one or two pilot doses - each containing about 7-15% of the fuel delivered during one cycle.

In the present study, it was assumed that the weight of the total fuel injected into the cylinder would be 68 mg. The total injection time measured by the crankshaft rotation angle was 18 deg in each case. The pilot dose was set at 8% of the total mass of the fuel injected and was delivered at 3 deg sufficiently in advance. The time interval between individual doses was 8 deg. It was assumed that the end of the injection was constant while the start of the injection was changed. For one dose of fuel, the injection started 15 deg before the TDC and ended at 3 deg after the TDC, according to the factory settings of the engine. For one pilot dose and main

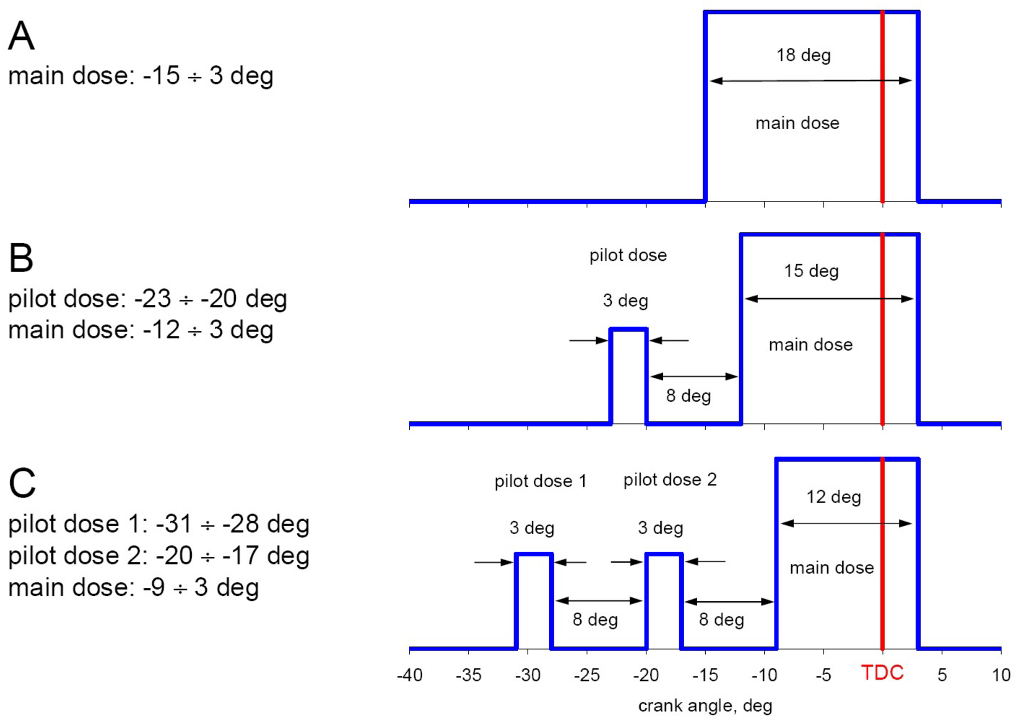

Fig. 4. Diagrams of the fuel injection strategy analyzed in the research

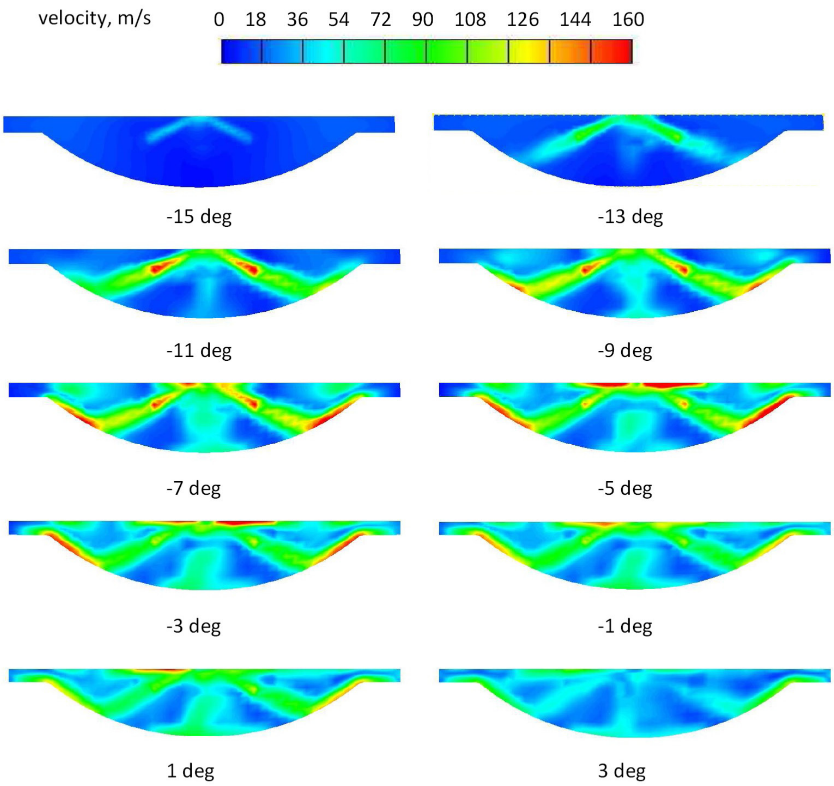

Fig. 5. Changes in fuel injection velocity for the dose without division

dose, injection started 23 deg before TDC and finished 3 deg after TDC. For two pilot doses and main dose, injection started 31 deg before TDC and finished 3 deg after TDC. Fig. 4 shows the patterns of analyzed injection strategies.

Figure 5 shows cylinder sections showing time-space velocity distributions of cylinder load during fuel injection into the combustion chamber. These distributions refer to the first injection strategy - one-stage injection without pilot fuel injection and ranges from -15 to 3 deg.

The modeling resulted in temporal spatial pressure distributions, the rate of pressure increase and the rate of heat release in the test engine cylinder for the three analyzed injection strategies. Figure 6 shows the pressure patterns in the engine cylinder as a function of the angle of rotation of the crankshaft. For the sake of clarity, the results are presented only in the angular range of interest, from -30 to 40 deg. The characteristics of cylinder pressure changes show the impact of the injection strategy used. There is a noticeable difference between conventional injection, no dose sharing, and pilot dose injection. The use of one and two pilot doses causes the initiation of autoignition of fuel and the increase of pressure in the first kinetic phase of the combustion process. In the rest of the process, combustion is similar for all analyzed injection strategies. The maximum pressure in the cylinder reaches a value of about 6.6 MPa.

In the ignition engine, the combustion process is accompanied by a sudden increase in the rate of heat release and a sudden increase in pressure. The pressure rise rate - dp/dφ, has a significant effect on the nature of the engine. The high values of the pressure rise rate indicate the hard work of

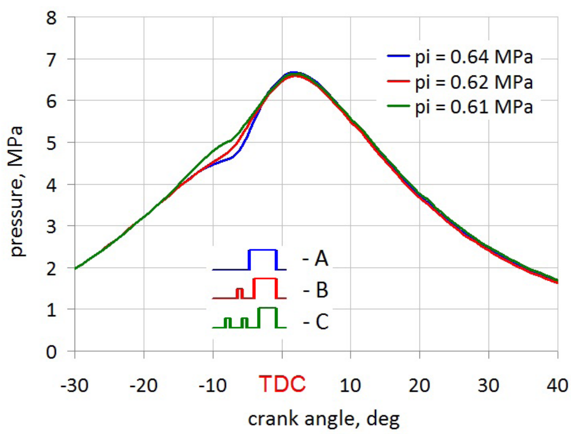

Fig. 6. In-cylinder pressure traces of the engine

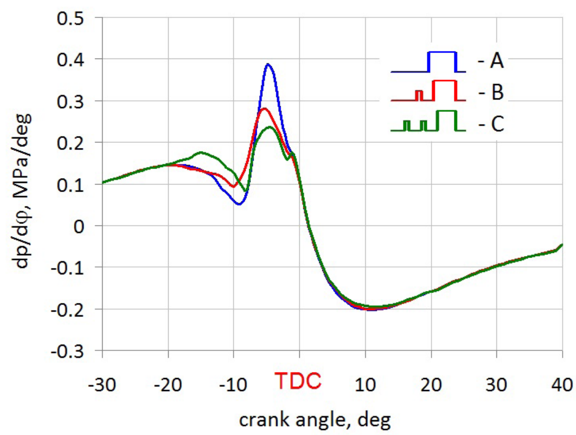

Fig. 7. Pressure rise traces of the engine

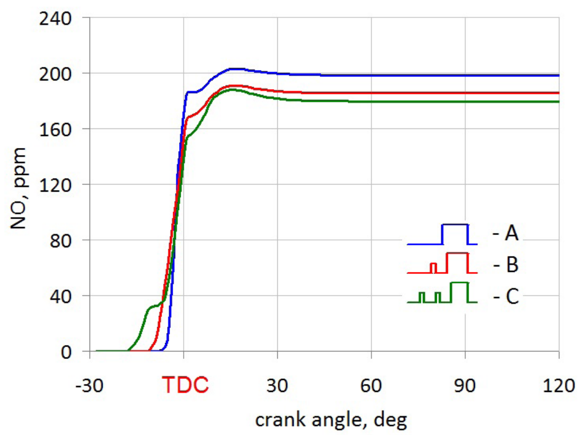

Fig. 9. NO emission

the engine, which results in increased noise and adversely affect the engine design as they cause an increased mechanical load on the crankcase system [18]. Figure 7 depicts changes in pressure rise rates for three analyzed injection strategies. The use of pilot doses positively affected engine performance. For the two pilot doses, the lowest dp/dφ values were obtained.

The most important characteristic characterizing the combustion process in a piston engine is the rate of heat release - dQ/dϕ. This affects, inter alia, the variation in cylinder pressure and thus the engine performance and is determined by the amount of fuel burned for each stage of crankshaft rotation during the combustion period [19]. Figure 8 shows the effect of applied injection strategies on the rate of heat release. There was a clear division of combustion into two stages by delaying combustion and lowering the maximum dQ/dϕ.

The amount of nitrogen oxide formed depends primarily on the amount of free oxygen and the increase in temperature and rate of heat release in the first kinetic stage of the combustion process [20,21]. Figure 9 shows the NO emissions for the three analyzed injection strategies. The use of fuel injection has a positive effect on reducing nitric oxide emissions. The best result was achieved with two pilot doses. The NO concentration for the two pilot doses was lower compared to the 10% unadjusted dose. Reduction of NO concentration in the flue gas is associated with a reduction in the rate of combustion of the charge and heat generation due to pilot doses.

Soot particles in exhaust gas come from incomplete or partial combustion of fuel. Ozone depletion and low temperature of the diffusion stage of the combustion process lead to increased soot emissions. Figure 10 shows the soot particle concentrations in the engine exhaust for the ana-

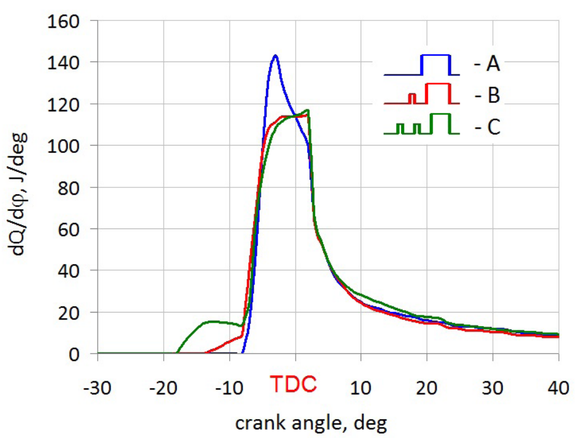

Fig. 8. Heat release rate traces of the engine

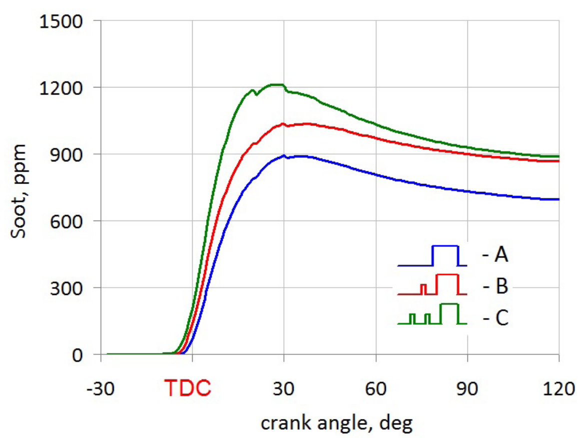

Fig. 10. Soot emission

lyzed fuel injection strategies. The use of pilot doses negatively affects soot emissions, which is higher in both cases, achieving a higher value for two pilot doses. The increase in the concentration of soot in the pilot flue gas compared to the unrestricted dose was 27%.

SUMMARY

In this study was to analyze the effect of fuel dose distribution on selected parameters of the combustion process in CI engine. The analysis was performed for 3 injection strategies: no division, one pilot dose and one main dose and two pilot doses and one main dose.

Based on the made calculations, it can be stated that:

- The use of pilot doses reduces the rate of engine pressure build-up, reducing its hard work and reducing the noisiness,
- Fuel injection dosage has a positive effect on lowering NO,
- Reduction of NO concentration in the exhaust gas is associated with a reduction in the rate of combustion and heat generation,
- The use of pilot doses negatively affects soot emissions, achieving a higher value for two pilot doses.

The program used in this work is a research tool used for modeling the combustion engine work cycle. Applied combustion and emission models allow for spatial analysis of phenomena occurring in the engine combustion chamber.

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