Article citation info:
KOWALSKI, J. The influence of the fuel spray nozzle geometry on the exhaust gas composition from the marine 4-stroke diesel engine. *Combustion Engines*. 2018, **172**(1), 59-63. DOI: 10.19206/CE-2018-107

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CE-2018-107

The influence of the fuel spray nozzle geometry on the exhaust gas composition from the marine 4-stroke diesel engine

The paper presents experimental research on a 4-stroke, 3-cylinder, turbocharged AL25/30 Diesel engine. Research consisted in investigating the effect of the geometry of the fuel injectors on the exhaust gas composition from the engine. During measurements, the engine was operated with a regulator characteristic of a load range from 40 kW to 280 kW, made by electric water resistance. The engine was mechanically coupled to the electric power generator. Three observations were made for each engine load, operating with fuel injectors of varying geometry. All considered types of injectors were installed on all engine cylinders. Mentioned injectors differed in the size of the nozzle holes diameters, holes numbers and angles measured between the holes axis. Engine performance data were recorded with a sampling time of 1 s. Cylinder pressure and fuel injection pressure on the front of each injector were collected also. The composition of the exhaust gas was measured using an electrochemical analyzer. According to the results, the change of fuel nozzle geometry results in a change in fuel spraying and evaporation and consequently changes in the course of the combustion process. The effect of this is the change of the composition of the exhaust gas.

Key words: *marine Diesel engine, fuel injector geometry, combustion process, exhaust gas composition, emission*

1. Introduction

Diesel 4-stroke engines are used on ships both for main propulsion and as power generators, including emergency generators. The source of energy in such engines is the combustion process of liquid fossil fuels and the side effect is the emission of toxic compounds including carbon, sulfur and nitrogen oxides into the atmosphere. In the classic construction of marine Diesel engines, the fuel is delivered to each engine cylinder directly using multi-hole fuel injectors. The fuel dosage, its delivery time to the cylinder and the injection timing are regulated by the injector opening pressure and the regulation of the Bosch type fuel pump. In modern Diesel engines constructions, fuel delivery parameters can be controlled by electronic control systems and electromagnetic or piezoelectric valves in a fuel injector (common-rail systems). The main purpose of the fuel dose regulation is the reduction of the fuel consumption and the reduction of emissions of toxic compounds into the atmosphere for the entire engine load range. Fuel injected into the combustion chamber of the engine is broken-up and evaporated and mixed with air at the same time. The rise in temperature and pressure of the fuel and air mixture compressed by the piston causes it to auto-ignite in areas where the composition of the mixture is within its flammability limits. As a result of auto-ignition, flame propagation occurs in the combustion chamber. Note that the described processes occur simultaneously and their course is determined by the shape of the injected fuel spray [1]. The shape of the fuel spray depends on the pressure of the injected fuel, the pressure in the engine cylinder (back pressure), and the position and geometry of the fuel nozzle. Typically, the diameter of fuel injector holes is determined by the fuel pressure in the fuel system. The increase of fuel injection pressure to the cylinder results in an increased intensity of atomization and evaporation, but also an increased fuel dose penetration [2]. Reducing fuel injection holes reduces the initial diameter of fuel droplets, which promotes the intensification of fuel atomization and the evaporation [3], and prevents fuel from burning on cylinder walls. The effect of this phenomenon is shortening the auto-ignition delay [4]. It should be noted, that reducing the diameter of the fuel nozzle holes reduces their cross-sectional area. In the case, in classic construction with the Bosch pump, the reduction of the diameter of the fuel nozzle holes, with unchanged quantitative characteristics of the injection pump causes the increase of the injection pressure. In the case of common-rail systems, the reduction in the diameter of the holes results in the decrease of the fuel flow rate through the nozzle holes with relatively stable fuel rail pressure. We may counteract both phenomena by changing the number of holes in the injector. The increase of the number of holes in the nozzle promotes homogeneous combustion. Excess number of holes in the injector can cause overlap the fuel spray. Based on the study [5], it can be concluded that there is a minimum distance between fuel sprays for which the fuel evaporation is the most intense. The value of the angle of the fuel injection cone is also affected by the fuel combustion process. Too high value of this angle causes the fuel evaporation and combustion near the cylinder head walls and cylinder valves. Decreasing the value of the mentioned angle causes fuel injection towards the piston head. Moreover, according to [6], changing the fuel injection cone angle causes the change of the fuel flow rate from the nozzle.

Mentioned conditions have a significant impact on the composition of the exhaust gas. For example, according to [7] the increase of the fuel injection cone angle results in the increase of the nitric oxides (NO_x) fraction, while according to [8] the increase of the number of holes does not significantly affect the NO_x emission. A thorough analysis of this problem, based on the 3-dimensional CFD model of the combustion process, can be found in [9] and [10].

The presented analysis shows that the geometry of the fuel injector nozzle has a significant effect on the composition of the exhaust gas. Unfortunately, there are very few publications on this topic, based on experimental research on marine engines. Therefore, the aim of the study is to analyze the parameters of the combustion process in the 4-

stroke Diesel engine with the assessment of exhaust gas composition. The analysis will be carried out on the basis of the experimental results of the operating parameters and emission from the engine used in the marine applications.

2. Measurement conditions and the research object

The research object is Sulzer's 3-cylinder, 4-stroke AL25/30 Diesel engine. The engine is pulse charged by the VTR 160 Brown-Boveri turbocharger and intercooled. The fuel is diesel oil with a known specification, presented in Table 1. The fuel system consists of Bosch-type injection pumps controlled by a rotary speed control and multi-hole fuel injectors, mechanically adjustable by opening pressure. Fuel injectors are centrally located in the cylinder heads of the engine. The engine operated at a constant rotational speed of 750 rpm and was loaded by the power generator, electrically connected to the water rheostat. The basic parameters of the laboratory engine are presented in Table 2 and the schematic of the laboratory stand is presented in Fig. 1. During each observation, the engine was loaded to a value of 280 kW, determined by the electric power output from the electric power generator. Recordings of measurement results were made in quasi-steady conditions. It is assumed that mentioned conditions occur when the changes of the exhaust gas temperature measured behind the turbocharger was not greater than 1°C/min. Then the engine load was reduced by 20 kW and the measurement procedure was repeated. The measurements were conducted in the load range from 280 kW to 40 kW. Three observations were made during each engine load for variable geometry of fuel nozzles installed on all engine cylinders. Parameters of used fuel nozzles are presented in Table 3.

Table 1. Diesel oil properties

Parameter	Unit	Value
Density at 15°C	kg/m^3	823.6
Viscosity at 40°C	mm^2/s	2.57
Cetane number	–	52.9

Table 2. AL25/30 engine parameters

Parameter	Unit	Value
Rotational speed	rpm	750
Cylinder number	–	3
Cylinder diameter	mm	250
Stroke	mm	300
Compression ratio	–	12.7
Fuel nozzle opening pressure	MPa	25
Injection timing	°before TDC	18

According to Table 3 nozzles 1 and 3 differ in the fuel nozzle cone angle. The change of the diameter of holes can be considered negligible. Nozzle 2 has a reduced number of holes and an increased diameter in relation to the rest of the nozzles. The changes made resulted in a total cross section area of the nozzle 2 being 22% greater than in the nozzle 3. Combustion pressures were measured in all engine cylinders by the Kistler 6613CG1 sensors with an amplifier. Each observations consists of 7 revolutions of the engine crankshaft with sampling rate of 720 samples per shaft rotation. Fuel pressure on the front of fuel injectors was recorded by Kistler 4067E2000DS1-2.0 sensor with digital temperature compensation. In addition, temperature and pressure of the charging air, the exhaust gas, lubricating oil and cooling water of the high temperature circuit were measured at characteristic points of the engine installation. Moreover, the composition of the exhaust gas was recorded using the TESTO 350XL electrochemical gas analyzer. The air flow rate was calculated on the basis of the pressure drop measurements on the Ventouri flowmeter installed on the air intake duct. The range and accuracy of used measuring equipment are presented in Table 4.

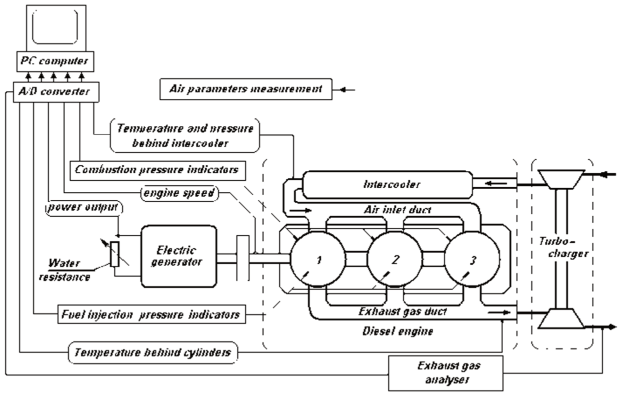


Fig. 1. The experimental setup schematic [11]

Table 3. Fuel nozzles parameters

Nozzle	Nozzle 1	Nozzle 2	Nozzle3
Holes number	9	8	9
Holes diameter [mm]	0.325	0.375	0.320
Nozzle holes angle [°]	150	158	158

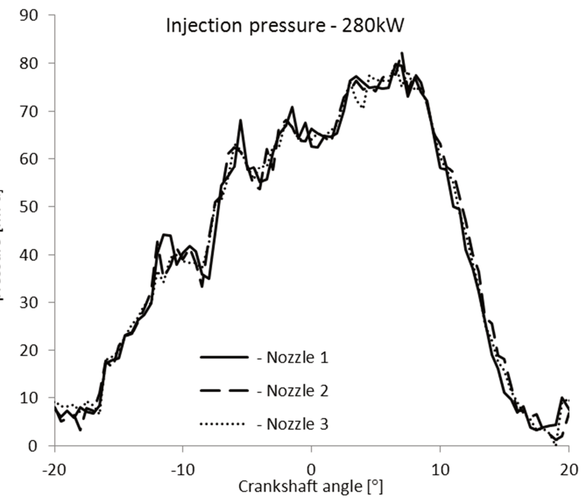
Table 4. Ranges and precisions of measurement equipment

Parameter	Range	Precision values
Environment air temperature	0–60°C	±0.5°C
Environment air humidity	0–90%	±2.0%
Exhaust gas temperature	0–650°C	±1.35%
Carbon oxide	0–10000 ppm	±5.0%
Sulfur dioxide	0–5000 ppm	±5.0%
Nitric oxide	0–3000 ppm	±5.0%
Nitric dioxide	0–500 ppm	±5.0%
Carbon dioxide	0–50%	±0.5%
Oxygen	0–25%	0.8%
Temperatures	0–100°C	±0.35%
Pressures	0–4 bar	±0.3%
Fuel injection pressure	0–2000 bar	±0.8%
Combustion pressure	0– 50 bar	±2.0%
Fuel consumption [kg/h]	–	±3.0%
Electric power [kW]	–	±0.5%

3. Results and discussion

According to discussed considerations, the increase in the total cross-sectional area of the injector holes (nozzle 2) in the classical construction of the fuel system results in the reduction of fuel injection pressure. Figure 2a shows an example fuel pressure measurement on the front of the injector during operation of the engine with a load of 280 kW. According to the presented results, fuel pressure characteristics are not significantly different for different spray geometries. However, it should be noted that the sampling frequency of the pressure measurement is 720 measurements per crankshaft rotation. According to [9] the increase of the diameter and the number of fuel nozzle causes the reduction of the maximum combustion pressure. Figure 2b shows an example of the pressure characteristics measured on the indicator gap during operation of the engine with a load of 280 kW. According to presented results, the use of nozzles 2 and 3 results in the reduction of the maximum combustion pressure by about 0.5 MPa. Figure 3 presents the calculated values of specific fuel consumption (SFC) and air/fuel excess ratio. According to the presented results, the use of nozzle 3 reduces the fuel consumption by average 2.8% for all considered loads of the engine. This means that the fuel injection cone angle of 158° is the preferred solution to reduce the fuel consumption for this combustion chamber design. The worst solution in the point of fuel consumption is the nozzle 2. The increase of the diameter of the fuel nozzle holes, and limiting theirs numbers probably increases the intensity of the evaporation and the combustion process. This leads to the reduction of maximum pressure and temperature of the combustion process. It should be noted that the value of the air/fuel excess ratio, determined by the measuring the oxygen content in the exhaust gas, does not change significantly. Certain small changes are only visible when the engine operates at relatively low load.

a)



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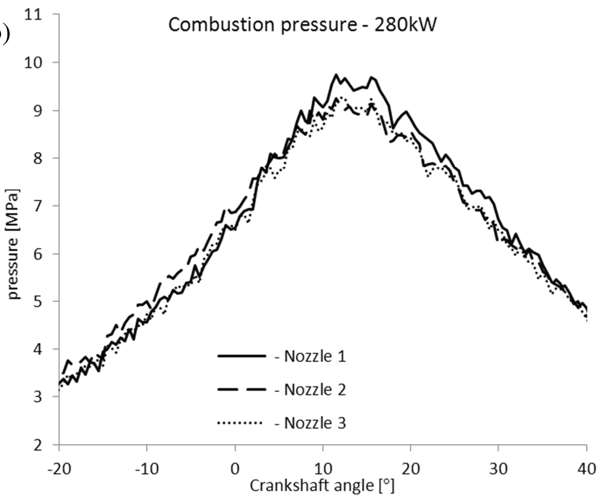
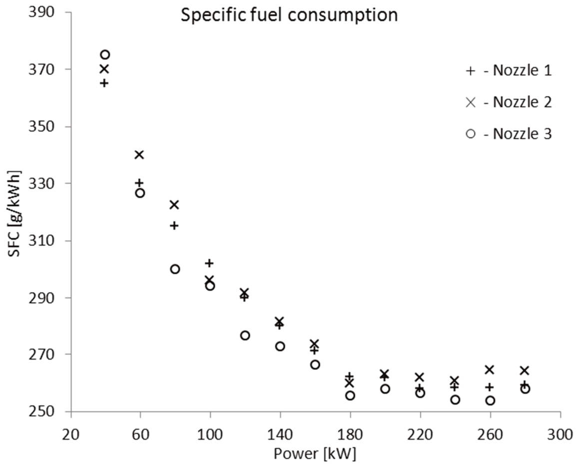


Fig. 2. The example of a) fuel injection pressure and b)combustion pressure

a)



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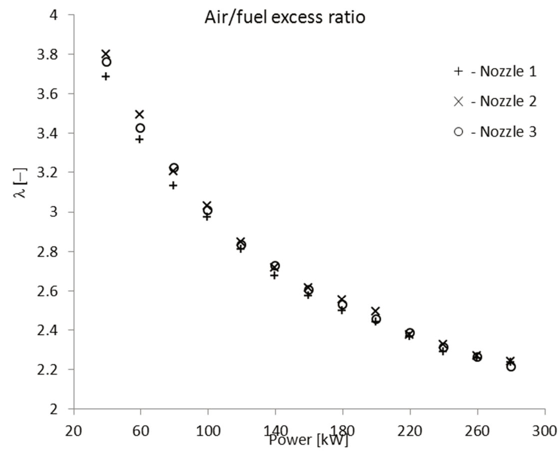

Fig. 3. Specific fuel consumption and the air/fuel excess ratio

Interesting is fact, that the use of nozzles with the geometry of Table 3 does not cause significant changes in the exhaust gas temperature measured behind the cylinders and in the front and behind the turbocharger. The use of the nozzle 2 causes the rise of exhaust gas temperature by a value not higher than 5°C and only when the engine operates with the maximum considered load. This is the evidence of the theory of the combustion process extension. For a relatively heavy load and the nozzle 2, the 2% increase of the airflow was also observed. No turbocharger speed changes and the charging air pressure changes were observed. However, the changing of the geometry of the fuel nozzles results in significant change in the composition of the exhaust gas. Figure 4 shows results of the measurement of the exhaust gas composition. The small change in the fuel consumption entails changes in the carbon dioxide (CO_2) content in the exhaust gas. Despite the smallest SFC was observed for nozzle 3, the smallest CO_2 fraction in the exhaust gas was measured for the operation of the nozzle 2. The carbon monoxide (CO) content was also higher for the nozzle 3, especially when the engine operates at a relatively high load. According to the presented results, the use of nozzle 1 results in the highest CO content in the exhaust gas. The difference in CO fraction during engine operation with a relatively high load was even 50%. Based on the results from [9] it can be stated that the reason for this is the smaller value of the fuel injection cone angle in nozzle 1.

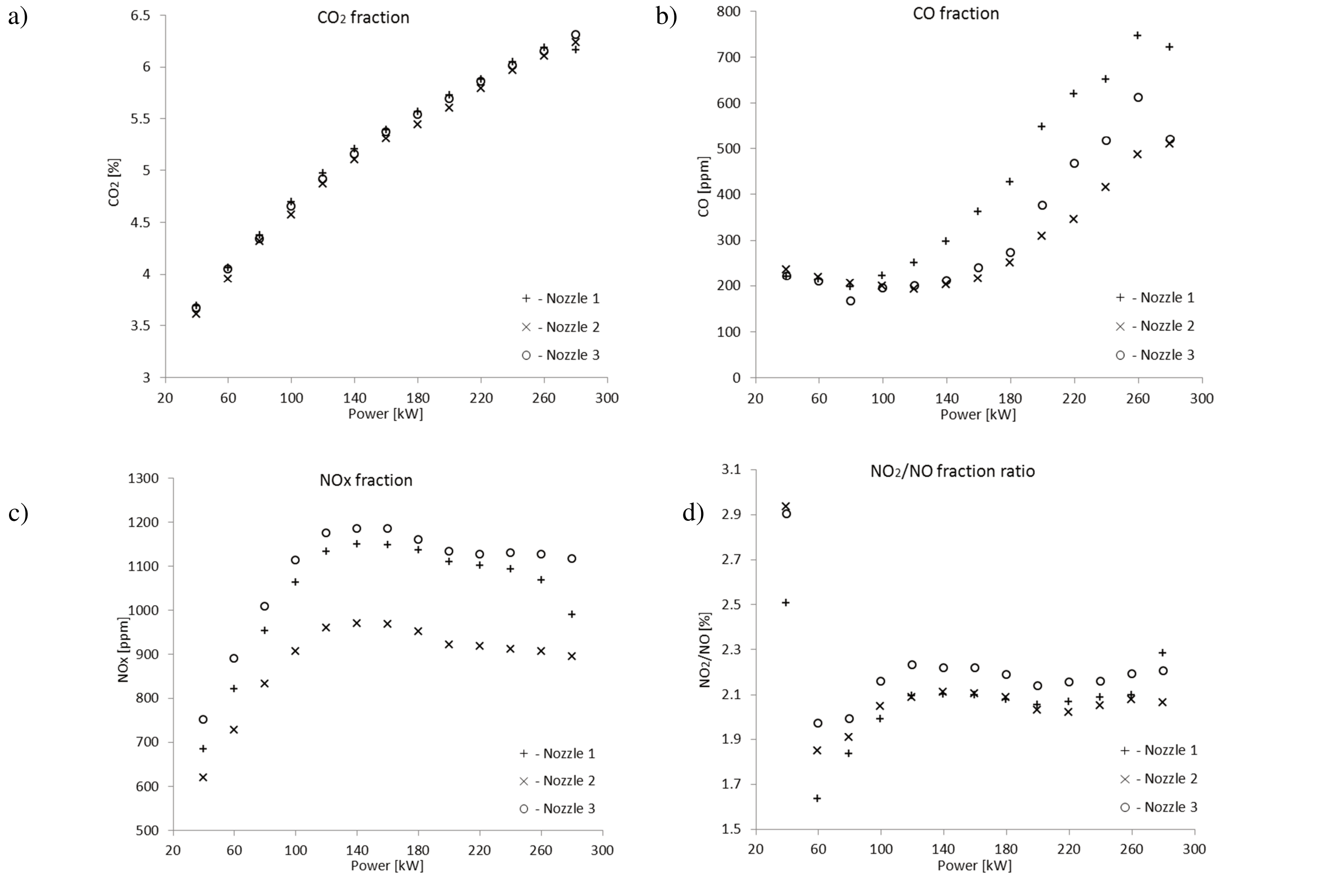

Fig. 4. Fractions of chosen chemical compounds in the exhaust gas

Figure 4c and Figure 4d show the nitric oxides fractions in the exhaust gas as a sum of nitric dioxide (NO_2) and nitric oxide (NO) and the ratio of NO to NO_2. According to the presented results, the use of the best nozzle from the fuel consumption point of view (nozzle 3) increases the NO_x fraction in the exhaust gas. This may be due to the increase of combustion temperature, but this conclusion must be taken with the great care due to the lack of appropriate measurement data. The smallest NO_x fraction was observed for nozzles 2. This result is consistent with the results of the CFD calculations for the same research object [9]. The increase of the diameter of the fuel nozzle holes and the reduction of the number of holes helps to reduce the NO fraction in the exhaust gas. It should also be noted that the use of nozzle 3 results in the increase of the NO_2 fraction in the exhaust gas. During the study, NO_x and CO emissions were also calculated. Figure 5 shows the CO and the NO_x emission. The NO_x emission value was corrected to the standard pressure, humidity and ambient conditions in accordance with the ISO 8178 standard regulation.

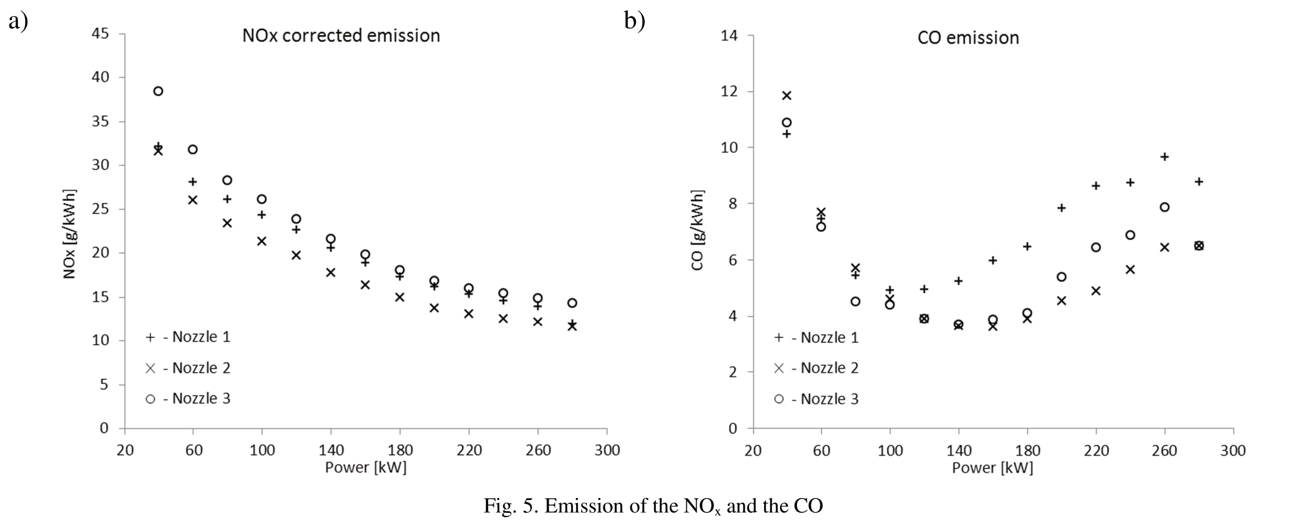

Fig. 5. Emission of the NO_x and the CO

According to the results presented in Fig. 5, the highest NO_x emission is observed during the operation of the nozzle 3 and the smallest for the nozzle 2. The reduction of NOx emission for nozzle 2 is from 21.3% to 24% for all considered loads of the engine in relation to results for nozzle 3. It should be noted that fuel nozzle 2 has the smallest number of holes with the largest diameters. As a result, the process of the evaporation and the combustion of the fuel are extended relative to the remaining geometry of nozzles. As a result, temperature and pressure of the combustion process are reduced, thus reducing NO_x emission and increasing of the fuel consumption. Obtained results of the NO_x emission are qualitatively convergent for experimental results with the use of diesel oil [12]. The increase of the diameter of the holes and the reduction the holes number causes the reduction of NO_x emission and the increase of SFC. According to results of Fig. 5b, the highest CO emission is obtained for the nozzle 1. The visible increase of emissions is evident especially during high load operation. As a result of this situation, may be fuel combustion near relatively cold cylinder walls due to too low fuel injection cone angle.

Conclusions

The paper presents the results of experimental research on the effect of changes of fuel nozzle geometry on performance parameters and emission from marine 4-stroke Diesel engine. Obtained results show that, from the engine efficiency point of view, the best solution is to use lower diameter of nozzles holes with larger numbers of holes, as is the case in the nozzle 3. The use of this solution intensifies the atomization and evaporation process. The combustion process in this case is faster, which in consequence leads to higher combustion temperature and pressure and reduced fuel consumption. However, this regulation of the combustion process leads to an increase in the NO_x emission. Reduction of the NO_x emission is possible by slowing down the combustion process. Therefore, from the point of view of the reduction of NO_x emission, it is preferable to use fuel nozzles with larger holes diameter with a limited number of holes. Such a configuration occurs in the nozzle 2. The analysis of results showed that for the shape of the combustion chamber of the test engine, a better solution is to use a higher fuel injection cone angle. In this case, the combustion takes place away from the cylinder walls, resulting in lower the CO emission.

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