REGULARITIES OF CHANGES IN SPECIFIC CARBON AND NITROGEN OXIDE EMISSIONS OF MARINE AND TRANSPORT DIESEL ENGINES

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Abstract. The analysis of operating conditions of vessels of a fishing fleet is carried out. Features of the processes of injection and mixture formation in diesel engines with a volumetric method of mixture formation are noted, for the estimation of which original authorial indices and the combustion process equation that explicitly takes into account the influence of operational and constructive factors are used. On its basis, equations for the calculation of carbon and nitrogen oxides in exhaust gases have been developed.

Keywords: marine diesels, original parameters of injection and mixing processes, original combustion equation, formulas for calculating the nitrogen oxides and solid carbon particles in the exhaust gases

Introduction

Vessels of fishing industry fleet operate in different regions of the World Ocean. The external working conditions differ in the barometric pressure, temperature and humidity of the air due to the presence of different intensity and directions of wind and currents. The factors influence ships technical condition.

Experimental research of operating factors is often accompanied by great material costs and is not always possible. Therefore, it is necessary to apply the computer simulation method which will allows to study the influence of each of the acting factors and develop recommendations for limiting emissions of harmful substances with the exhaust gases which is especially important at the present time. In addition, with the products of incomplete combustion, their chemical energy is lost. The cylinders of the diesel don't receive it completely. Thereby, the operational costs and probability of fires in the exhaust system increase. The purpose of the paper is to describe the method for calculating the content of solid carbon particles in the exhaust gases.

Theoretical study

The predominant method of mixture formation in engines used in fishing fleet vessels is jet (volumetric) one, in which the volume of the combustion chamber is divided into a series of macro-volumes proportional to the number of nozzles of the atomizer. A fuel jet flows from each nozzle opening containing [1] a set of drops with diameters from 10-6 to 10-3 m. The smallest drops are located on the surface of the fuel jets and the larger ones are closer to the axis and in the frontal part

Investigations of the process of atomization of fuel by injectors of diesel engines [1] have shown that the air content in the jet of fuel for the normal course of the combustion process is not enough. The maximum value of the air-fuel ratio is on the axis of the jet (18 - 25) % of the theoretically necessary. On the parts of the jet located closer to the nozzle openings of the nozzle the amount of air is even smaller. At the same time, on the surface of the fuel jet the excess air coefficient is about 0.9 [2]. As a result, there is a significant temperature inhomogeneity in the jet of fuel. As shown by experiments conducted at the Central Research Diesel Institute [3] and other research organizations the fuel ignites on the surface of the jets near the nozzle apertures and covers the entire surface in 0.5-1 ms.

Thus, the main combustion process in the fuel supply period occurs on the surface of the fuel jets which are the boundary of the heat and mass exchange of the air charge with the fuel. Obviously, the greater the surface area of the fuel jets attributed to one mass unit of fuel the higher the speed and completeness of combustion. Therefore, the ratio of the total surface area of fuel jets F to the value of the cyclic fuel supply is proposed as one of the indicators characterizing the quality of the sputtering and mixture formation processes [4].

$$K_{p\Sigma} = F_{\phi\Sigma} / g_{\text{ц}} \ , \qquad (1)$$

where $F_{\phi\Sigma}$ – total surface area of fuel jets at the end of the fuel supply process;

$g_{\text{ц}}$ – cyclic fuel supply

In high-speed diesel engines the duration of the fuel supply process is comparable to the period of ignition delay. In low- and medium-speed marine diesel engines, the ignition delay period is much shorter than the fuel delivery time.

To calculate the surface of fuel jets it is necessary to have dependencies that take into account the influence of fuel pressure, the diameter and number of nozzle openings, the average parameters of the state of air charge during the combustion period and the characteristics of the fuel.

So far a large number of experimental studies have been carried out to identify the effect of individual factors on the fuel atomization process. The experiments were carried out both on models and when fuel was injected directly into ICE cylinders. In the first case, the units are made in the form of cylindrical "bombs" simulating a certain type of combustion chamber of an internal combustion engine. As a working medium, cold air or a hot inert gas was used [3]. The advantage of this method of investigation is that it is possible to eliminate the influence of the turbulence of the air charge in the cylinder and the combustion process. The process of combustion in marine ICEs begins long before the end of the injection of fuel and does not provide an opportunity to fully determine the parameters characterizing the quality of spraying of fuel. The advantage of research in ICE is that it is possible to estimate the length of the fuel jet and the distribution of fuel in it under the conditions of the combustion process in a specific type of ICE. While studies on models are conducted at constant charge pressures, usually lower than in real ICE.

It is established that the quality of spraying of fuel depends on the parameters of the fuel supply process, the geometric characteristics and technical condition of the elements of the HP fuel system, the pressure and temperature of the air charge; Density, viscosity and surface tension coefficient of injected fuel; Turbulence charge; Initial disturbances arising in the fuel as a result of the action of direct and reverse pressure waves in the HP fuel system high pressure line and during the flow period through the atomizer; Presence of dissolved air in the fuel, etc

At present, there are various methods for modeling the parameters of fuel jets. So, in the method of Professor A.S. Lyshevsky [1] based on the processing of experimental data of Russian and foreign authors, dependencies were obtained for the calculation of the length of the fuel jet, the angle of its cone, the distribution of fuel droplets along the diameters, and the excess air coefficient in the sections of the jet. The diameter of the droplets and their distribution were found by processing the prints left by the drops on the smoky plate.

Professor R.Z. Kavtaradze [2] developed the method of A.S. Lyshevsky adding to it the effect on the geometric characteristics of the fuel jet vortex motion of the air charge in the cylinder.

In the future, optical, holographic and electrical methods were used.

Holographic methods make it possible to obtain a three-dimensional image of the fuel jet and a more complete estimate of the diameters and distribution of droplets.

In the studies of Professor Yu.B. Sviridov [3] atomizers with hole diameters of 0.35 and 0.56 mm were used . The injection was carried out in a medium at normal temperature in order to avoid evaporation of the fuel. The study of the holograms obtained showed the existence of two regions in the stream: a luminous shell and a dark core.

The presence of a luminous shell and a dark nucleus was confirmed in [5]. The values of the fuel concentration, average relative distance between the drops and the coefficient of excess air along the length and radius of the jet were obtained:

$$d = 0.806 / \sqrt[3]{C_{жс}}\ ; \tag{2}$$

$$a = \left(\rho_{г}\left(1 - C_{жс}\right)\right) / \left(\rho_{жс} C_{жс} K_{cm}\right), \tag{3}$$

where d is the distance between the droplet centers; Where is the distance between the droplet centers

$C_{жс}$ - concentration of fuel;

$\rho_{г}$ and $\rho_{жс}$ - density of air and fuel, respectively;

d is the stoichiometric coefficient.

Professors R.A. Gafurov and G.A. Glebov from the Kazan State Technical University continued the experiments described above [6] under the conditions of multiple injection of fuel. The fact is that in the real fuel system of a diesel engine with periodic fuel injection, the walls of the fuel pipe oscillate, as a result of which the jet of fuel breaks, with the release of vapors and dissolved gases [7].

The two-layer structure of the fuel flare is confirmed. A spectrum of droplets in its central part is obtained. It is established that before the fuel cutoff, 2 peaks are detected in the spectrum: 90-100 and 180-200 μm, and after the cutoff a third peak is observed at a level of about 300 μm, which corresponds approximately to the diameter of the nozzle opening (0.32 mm).

The data obtained in this paper allow us to distinguish two phases of the jet disintegration: intracannel decay due to cavitation and aerodynamic crushing when the fuel flare interacts with the air charge inside the cylinder.

In [8] and [9], dependences were obtained that relate the average mass temperature of the air charge in the engine cylinder and the average fuel delivery pressure with the geometric parameters of the fuel jet and the concentration of fuel in it, while a number of coefficients were applied.

Taking into account the conducted experiments and the geometric characteristics of the fuel jet, calculated by the method of Professor A.S. Lyshevsky, a model and method for calculating the combustion process has been developed [4]. According to which the relative amount of burnt fuel is calculated:

$$x_i = 1 - \exp\left[-B \cdot C \cdot Д \cdot \mathrm{E} \cdot \mathrm{K}^{0,2} (\tau_i / \tau_Z)^{m+1} \right] \tag{4}$$

where τ_i is the length of time from the start of the combustion process to the *i*-th moment, s;

τ_Z - duration of the combustion process, c;

m = 1.88 - an indicator determined experimentally for a number of marine diesels.

In its turn:

$$B = \left(\frac{\mu_{сп}}{\mu_{сэ}}\right)^{1,42} \cdot \left(\frac{d_{сп}}{d_{сэ}}\right)^{1,05} \cdot \left(\frac{P_{fп} - P_{сп}}{P_{fэ} - P_{сэ}}\right)^{0,71} \cdot \left(\frac{\rho_{Тп}}{\rho_{Тэ}}\right)^{1,05} \cdot \left(\frac{\sigma_э}{\sigma_п}\right)^{0,37} \left(\frac{\mu_п}{\mu_э}\right)^{0,32} \frac{P_{цэ}}{P_{цп}} \frac{T_{цп}}{T_{цэ}} \frac{J_{сп}}{J_{сэ}} \frac{g_{цэ}}{g_{цп}}; \tag{5}$$

$$C = \frac{tg\gamma_п (1/\cos\gamma_п + tg\gamma_п)}{tg\gamma_э (1/\cos\gamma_э + tg\gamma_э)}; \tag{6}$$

$$Д = \frac{\tau_{инд.п}}{\tau_{инд.э}} \frac{\tau_{впр.э}}{\tau_{впр.п}} \left(\frac{\tau_{Zп} - 0,5\tau_{впр.п}}{\tau_{Zэ} - 0,5\tau_{впр.э}}\right)^{1,6}; \tag{7}$$

$$E = 6,908 \frac{\alpha_{1п}}{\alpha_{1э}} \left(\frac{P_{сп} + P_{\max п}}{P_{сэ} + P_{\max э}}\right)^{0,5} \tag{8}$$

$$K_V = \left(\frac{\mu_{сп}}{\mu_{сэ}}\right)^{2,13} \left(\frac{d_{сп}}{d_{сэ}}\right)^{1,575} \left(\frac{P_{fп} - P_{цп}}{P_{fэ} - P_{цэ}}\right)^{1,065} \left(\frac{\tau_{впр.п}}{\tau_{впр.э}}\right)^{1,5} \cdot \left(\frac{\rho_{Тп}}{\rho_{Тэ}}\right)^{1,575} \cdot \left(\frac{\sigma_э}{\sigma_п}\right)^{0,555} \left(\frac{\mu_{1п}}{\mu_{1э}}\right)^{0,48} \left(\frac{P_{цэ}}{P_{цп}}\right) \cdot \left(\frac{T_{цп}}{T_{цэ}}\right) \cdot \frac{tg^2\gamma_п}{tg^2\gamma_э} \cdot \frac{V_{еэ}}{V_{еп}} \tag{9}$$

In equations (4) - (10), a high-speed engine with a cylinder diameter of 0.24 m, a piston stroke of 0.27 m and a rotation speed of 1500 rpm (four stroke turbo charged 24 / 27 Russian Standard) was adopted as an engine with an "e" index.

In the equations (2) - (9) the following notations are adopted:

μc	- the nozzle flow rate;
d_c	- diameter of nozzle openings in injectors of one cylinder of internal combustion engines, mm;
P_f	- average fuel pressure in the injector during injection, kPa;
$P_ц, T_ц$	- average pressure and temperature of the working fluid in the combustion engine cylinder during the combustion period, MPa;
J_c	- number of nozzle openings in injectors of one cylinder of internal combustion engine, units;
$\rho_т$	- density of fuel, kg / m3;
σ	- coefficient of surface tension of fuel, N / m;
μ	- dynamic viscosity of fuel, Pa. from;
$g_ц$	- cyclic fuel supply, g;
γ	- the angle of the cone of the fuel jet, deg .;
$\tau_{инд}$	- the period of fuel ignition delay, s;
$\tau_{впр}$	- duration of fuel injection, s;
α_1	- average air-fuel ratio during combustion;
τ_z	- duration of the combustion process, s;
τ_i	- current time, counted from the beginning of ignition of fuel, with;
V_e	- volume of the combustion chamber;

Parameters with the index "n" refer to the diesel engine being designed (researched), and with the index "e" to the engine adopted as the standard. The exponents are obtained from the equations of Professor A.S. Lyshevsky, who processed experimental data on fuel dispersion.

The exponents in expressions (5) - (9) are obtained as a result of processing the dependencies for the sputtering process established by Professor A.S. Lyshevsky on the basis of processing and generalization of the results of experiments conducted in Russia and abroad.

As can be seen from the presented data, the complexes of the parameters "B", "C" and "D" in an explicit relative form characterize the effect of design parameters of fuel equipment, fuel characteristics and parameters of the working fluid in the diesel cylinder on the development of fuel jets, the duration of fuel injection and combustion. The duration of the combustion process is also calculated taking into account the above factors according to the following expression:

$$\tau_{Zn} = \tau_{Zэ}\left(\frac{\mu_{сэ}}{\mu_{cn}}\right)^{1,42} \cdot \left(\frac{d_{сэ}}{d_{сп}}\right)^{1,05} \cdot \left(\frac{P_{fэ} - P_{цэ}}{P_{fп} - P_{цп}}\right)^{0,71} \cdot \left(\frac{\rho_{Тэ}}{\rho_{Тп}}\right)^{1,05} \cdot \left(\frac{\sigma_{п}}{\sigma_{э}}\right)^{0,37} \cdot \left(\frac{\mu_{э}}{\mu_{n}}\right)^{0,32}$$
$$\cdot \frac{J_{сэ}}{J_{сп}} \frac{g_{цп}}{g_{цэ}} \sqrt{\frac{(P_{cc} + P_{max})_{э}}{(P_{cc} + P_{max})_{п}}} \frac{T_{цэ}}{T_{цп}} \cdot \frac{tg\gamma_{э}}{tg\gamma_{\bullet}} \frac{(1/\cos\gamma_{э} + tg\gamma_{э})}{(1/\cos\gamma_{п} + tg\gamma_{n})} \frac{\tau_{инд.э}}{\tau_{инд.п}} \frac{\alpha_{1э}}{\alpha_{1п}} \left(\frac{n_{э}}{n_{п}}\right)^{0,2} K_{v}^{-0,2} \qquad (10)$$

The more nozzle openings in the atomizer (nozzles with several nozzles in the cylinder), the higher the injection pressure of the fuel and the smaller the diameter of the nozzle openings (all other conditions being equal and constant cyclic delivery), the greater $F_{\phi\Sigma}$ and Kp2. The deterioration of the details of the cylinder-piston group, the decrease in the barometric pressure, the increase in the temperature of the air entering the cylinders, the deterioration of the precision elements of the fuel equipment, the deterioration of the technical state of the pressurization and gas exchange systems violate the quality of the processes of mixture formation and combustion, that is, the values of the indices B, C, D, E and The combustion process is prolonged. In this case, a relatively larger amount of fuel will be located near the surface layer, the combustion process will proceed faster. At the same time, the relative amount of fuel located in the zones with the minimum coefficients of excess air in combustion will decrease, and its residence time will decrease there.

The use of heavy fuels containing a large amount of aromatic and high-molecular hydrocarbons is accompanied by the enlargement of the droplets during atomization and, consequently, an increase in the relative amount of fuel in zones with a lack of oxygen.

Temperatures in the combustion zones reach 2600-2900 K, the farther from the top dead center combustion occurs, the less temperature and more carbon can burn by the time of the opening of the gas distribution valves [2,3]. At such temperatures, intense heating of the supplied fuel occurs, and the lack of air inside the fuel jets causes pyrolysis of the fuel molecules with the elimination of hydrogen and the formation of carbon.

High temperatures in the combustion zone warm up the fuel layers, which are located closer to the axis of the jet, and due to lack of oxygen, soot forms in them. The process of its formation is conventionally divided [10] into three main phases: the formation of an embryo, growth of embryos into soot particles, coagulation of primary soot particles, burnout. The rate of soot formation is determined by the rate of chemical processes, leading to the appearance of an embryo (ie, the kinetics of the process).

Experiments conducted at the Department of ship power plants on the engine compartment 1NVD 24 are shown in Figure 1.

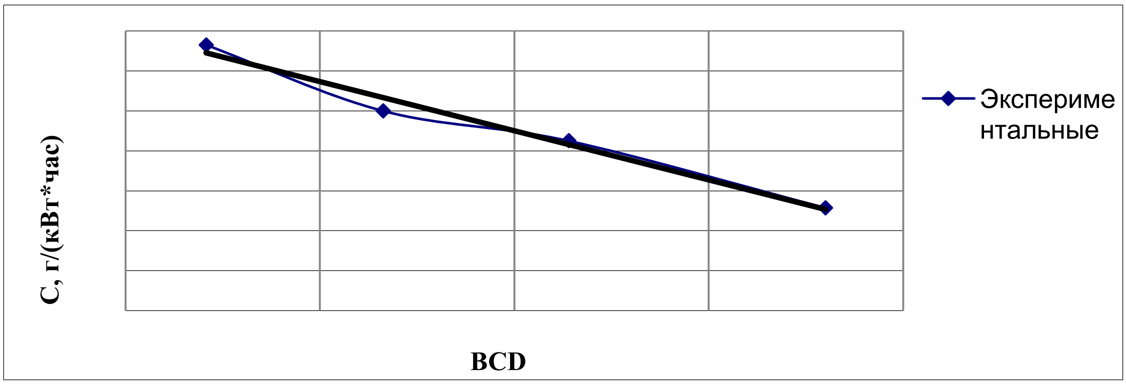

Figure 1. Results of experiments on the engine 1NVD 24 [10]

The complex of indicators «ВСД» with increasing load on the engine increases, by increasing the differential pressure on the injection, expanding the cone of the fuel jet and increasing the cyclic supply of fuel. As can be seen from Fig. 1, the specific carbon emission decreases inversely proportional to the growth of the product of the IRR.

The following equation is obtained

$$C = -0.123 \cdot ВСД + 0{,}1884 \qquad (11)$$

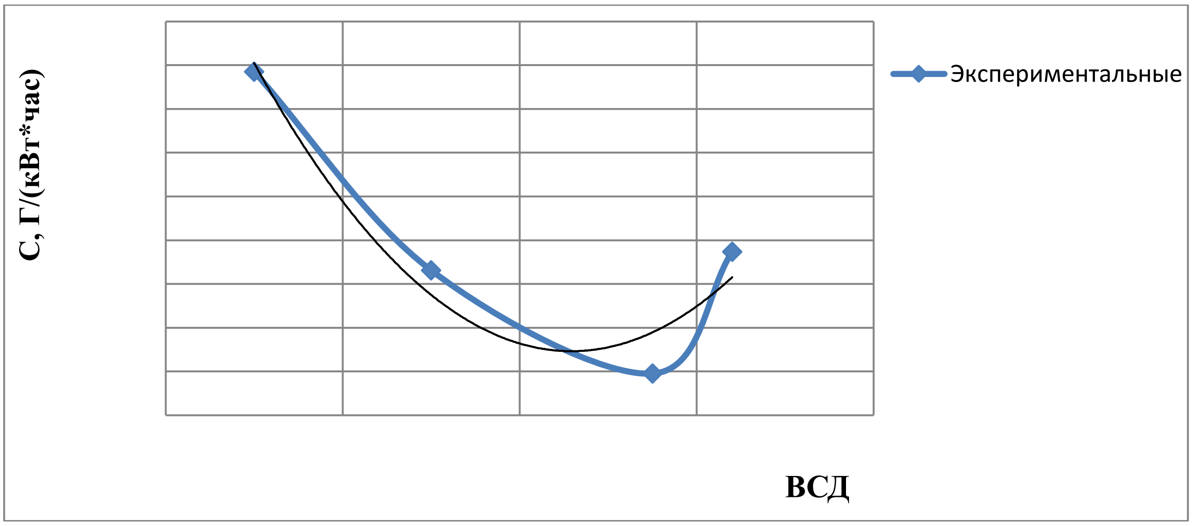

Figure 2. Results of experiments on the KAMAZ - 740.10 engine [10,11]

As can be seen from Figures 1 and 2, the mixture formation in the engines proceeds identically. At the same time, in the KAMAZ-740.10 engine, the range of VSD indicators increases with the load increase. At full load (2200 rpm, Me = 380 Nm), part of the fuel injected on the walls of the combustion chamber and evaporates from them, which causes a certain increase in specific carbon emissions.

The following equation is obtained

$$C = 10^{-5}(ВСД)^2 - 0{,}0022 \cdot ВСД + 0{,}1535 \qquad (12)$$

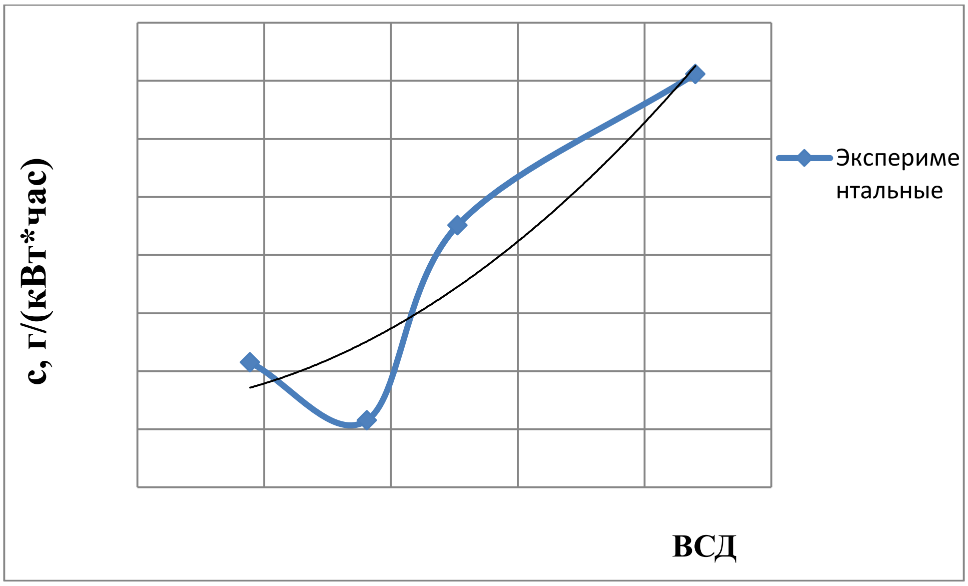

Figure 3 Results of experiments on the engine MAN D 2866LXE

A similar picture for emissions can be observed on the engine MAN D 2866LXE (1500 rpm, 179 kW). In this engine, as can be seen from figure 3, film-based mixture formation starts with a load of 50%. In connection with the connection of the film mechanism of the mixture formation, the results are approximated by square polynomials with an error not exceeding 10%. The carbon emission minimums correspond to the time of film-based mixture formation.

The following equation is obtained

$$C = 47{,}282(ВСД)^2 - 4{,}7344 \cdot ВСД + 0{,}1564 \qquad (13)$$

Therefore, based on the developed model, the following dependence was obtained for the calculation of carbon emissions:

$$c_п = c_э\left(\frac{J_{сэ}}{J_{сп}}\right)^{0,4}\left(\frac{d_{сэ}}{d_{сп}}\right)^{0,8}\left(\frac{g_{цэ}}{g_{цп}}\right)^{-0,4}\left(\frac{p_{тэ}}{p_{тп}}\right)^{0,3}\left(\frac{C_э}{C_п}\right)^{0,08}\left(\frac{\tau_{zэ}}{\tau_{zп}}\right)^{-0,12}\left(\frac{n_э}{n_п}\right)^{-0,12}\left(\frac{z_э}{z_п}\right)^{0,12}\left(\frac{\alpha_э}{\alpha_п}\right)^{-0,08} \qquad (14)$$

Parameters with the index "e" refer to the engine adopted for the standard, with the index "n" - to the engine being researched (projected).

As parameters in the expression (14), the parameters of the engine 1NVD 24 are applied. In calculating (14) for the KAMAZ-740.10 engine, a carbon emission of 0.033 g / (kWh) was obtained at the minimum carbon emission mode. The experimental value for this regime is 0.0319 g / (kWh). Thus, the discrepancy between the experimental and calculated data is 9-10%. So, with a load of 50% for the KAMAZ engine (2200 rpm Me = 200 Nm), the experimental emission values are 0.037 g / (kWh), and the calculated value is 0.041 g / (kWh).

Method of application

When designing the number of parameters included in the equation is specified by the first method. For example, the diameter and number of nozzle holes, the fuel pressure in the nozzle and the other ones are used to achieve expected economy of mechanical and thermal loads. In this case, the calculation of the indicator process is carried out and the content of solid carbon particles in the exhaust gases is calculated simultaneously.

In the second method, when the permissible level of increase in the carbon solids content in the exhaust gas is determined, calculations can be made without modeling the indicator process by making a reduction in the number of nozzle holes due to their contamination or reduction in fuel supply due to wear of the precision elements, reducing the air pressure in the diesel cylinders in end of scavenging process due to the deterioration of the technical state of the supercharge and gas exchange systems or the reduction of barometric pressure or gas transfer system contamination .

Conclusion

Thus, on the basis of theoretical and experimental studies, an equation is proposed for calculating the content of solid carbon particles in exhaust gases of diesel engines. The peculiarity of the equation is the explicit consideration of the influence of a number of structural and operational factors. These include: the diameter and number of nozzle openings, the average fuel pressure, the cyclic fuel supply and its physical properties (fuel density, surface tension coefficient, viscosity), the speed of the crankshaft and the diesel engine speed. The discrepancy between the experimental and calculated values is up to 10%.

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