Investigation upon the operational factors influencing the smoke emissions from diesel engine

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Abstract. The results of laboratory examinations of the ecological characteristics of diesel internal-combustion engine are represented in this article. The powerful, economical and ecological parameters of a diesel engine investigated in this article are presented by the speed characteristics of effective torque, injected fuel per cycle, and fuel and air consumption per hour, temperature, and smoke emissions of exhaust gases. On the basis of the experimental dependences a mathematical model is designed. Parameters influencing significantly the smoke emissions of exhaust gases are investigated. The factors, injected fuel per cycle and excess-air factor, are examined at different operation regimes of the engine.

1. Introduction
The solution of the ecological problems of our planet is unthinkable without the circumscription of harmful emissions in the environment. Ecological emission standards lay down higher regulations for the concentration of harmful substances in the cars exhaust gases. Therefore, the development of automobile drives and internal-combustion engines is associated significantly with the adherence of the ecological standards during the exploitation of vehicles. The engines working with ignition from compression and using diesel fuel show a considerable inclination in the generation of hard particles due to the character of the running combustible process. Constructions using alternative and combined fuels are developed as well as the optimal concentrations of the substances are investigated by [1] with the goal of minimizing these emissions through the increase of the homogeneous part of diesel combustion process.

Mathematical models for the analytical determination of internal-combustion engines characteristics are designed on the basis of the experimental investigations. The ecological behaviour of the engine at predominant influence of either one or a combination of some factors can be prognosticated by means of such models. Because of the great variety of the constructions the universal models give considerable derivations when they are applied to specific engines. This is the reason why mathematical models, worked out for particular operating processes and applicable for a certain type, to be used more frequently.

The goal of this investigation is the determination of the analytical dependences of diesel engines and the parameters exerting considerable influence on the generation of harmful emissions in the exhaust gases (EG).
2. Presentation

In a diesel engine diffusion combustion of non-homogeneous fuel mixture is taking place. In the combustion zone the composition of fuel mixture is close to the stoichiometric one, \( \lambda \ll 0 \), and the temperature is high. A section of much enriched mixture is touching to this zone, which creates conditions for the running of fuel pyrolysis with a small quantity of oxygen. Here the generated soot begins immediately to burn at oxygen entrance. The speed of the soot generation exceeds their speed of burning at the beginning of the process, while at the end of the process the speed of soot burning is higher than the speed of soot generation. The soot burning continues through the process of expansion, but around the combustion chamber walls the temperature of the gases is tempered through the whole cycle and due to this the soot fell there is not taking place in the combustion. Soot can be also generated when the fuel flow fall upon considerably cool walls of the cylinder.

The raised amount of soot and smoke emissions of EG sharply increase if the fuel mixture is enriched. This circumstance has a great significance in the conditions of diesel engines operation, as the reasons for this are different. One reason can be the irregular control of the fuel apparatus influencing the amount and the quality of the fuel injection, while another reason can be the irregular choice of the characteristic of the injected fuel per cycle. Moreover, increased smoke emissions can be read not at all but at merely some regimes of operation of the diesel engine. Some investigations in this respect present, that during the vehicles movement in urbanized territories, the engines operate mainly at transitional operating modes.

In such circumstances the working process takes place at different from the optimal thermodynamic parameters of the environment in the combustion space, and as a result the ecological characteristics are made strongly worse [2]. That characteristics of the EG also depend on the type of fuel used - vegetable oils, biodiesel or diesel [3].

Together with the raised smoke emissions due to the enriched fuel mixture the fuel consumption also increases. In order the desired characteristic of the effective torque, \( M_e \), the decreased smoke emissions of exhaust gases, and the decreased fuel consumption to be reached, it is necessary the characteristics of injected fuel per cycle, \( q_c \), and the excess-air factor to be optimized at all speed regimes. The combustion temperature in the combustion chamber, where highly cancerogen and polycyclic aromatic hydrocarbons are not generated, needs to be controlled. Soot contains highly cancerogen and polycyclic aromatic hydrocarbons, which fall in human body by the respiratory system, and are held back in lungs alveoli [4, 5].

An experimental examination for the determination of the smoke emissions of EG, the characteristic of injected fuel per cycle, and the powerful and ecological characteristics of EG is carried out without catalytic treatment by means of test stand. The test stand is equipped with diesel engine, dynamometer, smoke-emissions-meter Hartridge, fuel and air flow-consumption-meter, sensors for measuring the EG-temperature in the release collector, and etc.

The smoke emissions of EG is determined with dynamometer Hartridge by measuring the opacity (optical density) of EG in relative units for transparency from 0 to 100. The relation between the smoke emissions according Hartridge, \( K \), and the coefficient of light absorption by smoke is represented in [6], and is of the form

\[
k = -\frac{1}{L} \cdot ln \left( 1 - \frac{K}{100} \right),
\]

where \( k \) is the coefficient of light absorption, \( m^{-1} \); \( L \) is the measuring length of dynamometer chamber, \( m \); \( K \) is the smoke emissions according Hartridge, \%. 

The diesel engine investigated has a power \( N_p = 142 kW \) at \( n_{cs} = 2100 \, min^{-1} \); maximal torque \( M_{max} = 697 \, Nm \) at \( n_{cs} = 1300 \, min^{-1} \), with direct injection.

The external and partial speed characteristics are taken for the determining the analytical dependences of the effective torque. The changes in the external (curve 1) and the partial characteristics (curves 2, 3, and 4) of the effective torque as a function of the crankshaft speed are represented in Figure 1, while the corresponding characteristics changes of the injected fuel per cycle are shown in Figure 2.
During the process of operation the automobiles work with variable load and variable speed regimes. For each speed regime the engine can undergo changeable load and develop the adequate torque. In the diesel engines the load change is achieved by change of the injected fuel per cycle. We assume boundaries of the investigated field, which includes the speed regimes at crankshaft speed from 800 min$^{-1}$ to 2000 min$^{-1}$, and load regimes at injected fuel per cycle from 40 mg/cycle to 100 mg/cycle.

The approximation equations of the mathematical model are valid for these boundaries.

The experimental investigations are carried out by repeated measurements of the parameters (the investigated characteristics) of the multi-factorial experiment. A mathematical model of two-factorial experiment is chosen during the experiment planning. The uniform plan of the two-factorial experiment is used for the determining the mathematical dependence of the effective torque on both injected fuel per cycle and the crankshaft speed. After the design of the planning matrix the regression coefficients are defined, and the following formula is got

$$M_e = a_{e_1} + a_{e_2} \cdot q_c + a_{e_3} \cdot n_{cs} + a_{e_4} \cdot q_c^2 + a_{e_5} \cdot n_{cs}^2 + a_{e_6} \cdot q_c \cdot n_{cs} - (a_{n_1} + a_{n_2} \cdot n_{cs} + a_{n_3} \cdot n_{cs}^2),$$

(1)

where $M_e$ is the effective torque; $a_{e_1} \div a_{e_6}$ and $a_{n_1} \div a_{n_3}$ are the regression coefficients; $q_c$ is the injected fuel per cycle, and $n_{cs}$ is the crankshaft speed.

The matrix, containing experimental data is supplemented with the corresponding analytical magnitudes of the effective torque.

The external and the partial characteristics of the injected fuel per cycle as a function of the rotation frequency of fuel injection pump (FIP) and the rack position are presented with the polynomial

$$q_c = b_1 + b_2 \cdot n_p + b_3 \cdot h_p + b_4 \cdot n_p^2 + b_5 \cdot h_p^2 + b_6 \cdot n_p \cdot h_p,$$

(2)

where $b_1 \div b_6$ are the regression coefficients; $n_p$ is the rotation frequency of FIP, and $h_p$ is the rack position of FIP.

The results from the analytical formula (2) are presented in Figure 2 as thick curves.

For the determination of the analytical dependence of smoke emissions of engine EG are constructed the experimental characteristics, $K(n_{cs}, q_c)$, presented in Figure 3, where curve 1 is the external characteristic, while curves 2, 3, and 4 are partial (at different position of the accelerator pedal).
The adequacy of the mathematical model and the magnitudes of the coefficients for all investigated parameters are defined by using mathematical statistics methods and criteria. For this purpose, the sums of the numbers squared of the derivations of the experimental values from the calculated ones of the investigated parameters, the degree of freedom of the experiment, the remaining dispersion, and the dispersion of reproduction are determined.

The adequacy is checked and proved by F-test.

A commonly accepted theory for the soot formation is not available, but the numerous experimental data, found in many references, allow the more important factors, represented as decisive, one of which is the excess-air factor, $\lambda$, to be examined.

The excess-air factor in this article is determined by indirect measurement, by experimental determining the fuel consumption, $G_f$, and the air consumption according the following formula

$$\lambda = \frac{G_a}{G_f \cdot L_0},$$

where $\lambda$ is the excess-air factor; $G_a$ is the air consumption, kg/h; $G_f$ is the fuel consumption, kg/h; $L_0$ is the theoretically necessary air amount for the combustion of one unit quantity of fuel.

The experimental curves of the changes in air consumption, $G_a$, and fuel consumption, $G_f$, as a function of the crankshaft speed, $n_{cs}$, are shown respectively, where curve 1 is the external characteristic, and curves 2, 3, and 4 are the partial characteristic of $G_f$. 

**Figure 3.** A change of smoke emissions of EG of diesel engine, $K$, at different crankshaft speed $n_{cs}$: curve 1 is the external characteristic; curves 2, 3, and 4 are the partial characteristics. Experimental – - - -; Analytical – ———.

**Figure 4.** A change of air consumption, $G_a$, at different crankshaft speed $n_{cs}$. Experimental – - - -; Analytical – ———.

**Figure 5.** A change of fuel consumption, $G_f$, at different crankshaft speed $n_{cs}$. Experimental – - - -; Analytical – ———.
The approximation equation describing the change of fuel consumption is of the form

\[ G_f = n_{CS} \cdot i \cdot i_{dp} \cdot q_c, \] (4)

where \( G_f \) is the fuel consumption, kg/h; \( n_{CS} \) is the crankshaft speed; \( i \) is the number of cylinders; \( i_{dp} \) is the driving ratio between the crankshaft and the shaft of fuel injection pump; \( q_c \) is the amount of injected fuel per cycle. After the mathematical processing of equations (2) and (3), the well-known formula from [7] \( \lambda = \frac{V_{en} \cdot \eta_f \cdot \rho_a}{q_c \cdot L_0} \), and the experimental data presented in Figure 4, are obtained the following formulae for

- air consumption \( G_a \)

\[ G_a = V_{en} \cdot i \cdot \eta_f \cdot \rho_a \cdot \frac{n_{CS}}{\tau}, \] (5)

where \( G_a \) is the air consumption, kg/h; \( V_{en} \) is the engine working volume, m³; \( i \) is the number of cylinders; \( \rho_a \) is the air density and \( \eta_f \) is the volumetric efficiency; \( \tau \) is the engine stroke.

- volumetric efficiency \( \eta_f \)

\[ \eta_f = a_1 \eta_2 + a_2 r_3 \cdot n_{CS} + a_3 \cdot n_{CS}^3, \] (6)

where \( \eta_f \) is the volumetric efficiency; \( a_1, a_2 \) and \( a_3 \) are regression coefficients; \( n_{CS} \) is the crankshaft speed.

![Figure 6. An analytical curve of change of the volumetric efficiency \( \eta_f \) at different crankshaft speed \( n_{CS} \).](image)

Then the change of the excess-air factor \( \lambda \) as a function of crankshaft speed \( n_{CS} \) acquires the form presented in Figure 7.

![Figure 7. A change of the excess-air factor \( \lambda \) at different crankshaft speed \( n_{CS} \).](image)

One of the most widespread methods for the decrease of smoke emissions and improvement of fuel economy is the recirculation of cooled exhaust gases [8, 9]. The degree of cooling of EG influences the smoke emissions and the wearing of engine details. The higher EG temperature leads to higher smoke emissions and greater wearing of engine details. This makes the experimental investigation of
the temperature of exhaust gases $T_{eg}$ actual. The experiments carried out prove the dependence of EG temperature on the crankshaft speed and the injected fuel per cycle. The experimental characteristics are presented in Figure 8.

The regression coefficients of the approximation polynomial are got by using the uniform plan of the two-factorial experiment. The obtained polynomial of second degree is of the form

$$T_{eg} = a_{11} + a_{12} \cdot n_{cs} + a_{13} \cdot q_c + a_{21} \cdot n_{cs}^2 + a_{22} \cdot q_c^2 + a_{31} \cdot n_{cs} \cdot q_c,$$

where $T_{eg}$ is the temperature of exhaust gases, $K$; $a_{11}$ to $a_{31}$ are regression coefficients; $n_{cs}$ is the crankshaft speed; $q_c$ is the amount of injected fuel per cycle.

During the experimental examination is established that the smoke emissions of EG vastly depends on two parameters – excess-air factor $\lambda$, and crankshaft speed $n_{cs}$.

The experimental curves of smoke emissions, $K$, depending on the excess-air factor $\lambda$, and the crankshaft speed, $n_{cs}$, are constructed at constant values of crankshaft speed, and are presented in Figure 9. For the values $n_{cs} = 600 \text{ min}^{-1}$ and $n_{cs} = 800 \text{ min}^{-1}$ the curves are constructed by the method of extrapolation. The change of smoke emissions $K$ at constant magnitudes of excess-air factor $\lambda = 1.5; 2.0; 2.5; 3.0$ is shown in Figure 10. These dependences are used for the extrapolation of curves at $n_{cs} < 1000 \text{ min}^{-1}$.

As it is seen in Figure 9 the curves of smoke emissions $K$ are parabolas originated from one and the same point. A new coordinative system „$K - x$“, which origin of coordinates coincides with the beginning of parabolas, and axis „$x$“ coinciding with $\lambda$ and a connection between „$x$“ and „$\lambda$“, as $x =$
is introduced for determining the equation of these curves. It is settled that the parabolas can be described by the following equation

$$K = A \cdot x^2,$$  \hspace{1cm} (8)

where $K$ is the smoke emissions, %; $A$ a coefficient depending on the crankshaft speed, $n_{cs}$.

The dependence of coefficient $A$ on crankshaft speed, obtained by the method of least squares, is depicted with the following polynomial

$$A = (b_{K_4} \cdot n_{cs}^2 + b_{K_2} \cdot n_{cs} + b_{K_3}).$$  \hspace{1cm} (9)

After a mathematical processing of equations (8) and (9), and a return to the coordinative system “$K - \lambda$”, the following equation is got

$$K = (b_{K_4} \cdot n_{cs}^2 + b_{K_2} \cdot n_{cs} + b_{K_3}) \cdot (b_{K_4} - b_{K_5} \cdot \lambda)^2 + b_{K_6},$$  \hspace{1cm} (10)

where $b_{K_4} = b_{K_5}$ are approximation coefficients for both equations (9) and (10); $b_{K_4}$ and $b_{K_5}$ are coefficients of the equation of the connection between both coordinate systems; $b_{K_6}$ is the magnitude of smoke emissions independent on $\lambda$; $n_{cs}$ is the crankshaft speed.

This equation is valid for $\lambda$ quantities in the limits $\lambda_{\min} \leq \lambda \leq \lambda_{\max}$, (in this case $1.3 \leq \lambda \leq 4.5$). In Figure 9 is clear that the smoke emissions of EG at $\lambda > \lambda_{\max}$ ($\lambda > 4.5$) in practice does not depend on $\lambda$, and gets magnitude $K = b_{K_6}$. Experimental data for the smoke emissions behavior $K$ at $\lambda < \lambda_{\min}$ ($\lambda < 1.3$) are not generated, but presumably with the decrease of $\lambda$, the smoke emissions of EG is sharply raised.

The investigation carried out and the comparative analysis between the theoretical and experimental dependences for the smoke emissions of EG show that with the slowing down of crankshaft speed the smoke of EG is increased, and becomes greater than the allowed practical standard. The most probable reason for the increased smoke emissions of EG at low values of crankshaft speed could be the raised injected fuel per cycle $q_c$ and the reduced excess-air factor $\lambda$.

In order the influence of both parameters $q_c$ and $\lambda$ to be determined, theoretical investigations with the analytical dependences obtained at conditionally assumed maximal quantity of smoke emissions $K = 45 \%$ ($k = 1.2 \text{ m}^{-1}$) are carried out. The investigations pointed out in references [2, 3] have proved that the simultaneous reduction of the basic harmful emissions from diesel engines, i. e. $NO_X$ and soot, is impossible. During the present investigations it is assumed the smoke emissions to be limited up to $K = 45 \%$, which does not lead to an increased amount of $NO_X$.

The results from the theoretical investigation show, that in order the smoke emissions of EG to be reduced at low values of crankshaft speed, it is necessary the injected fuel per cycle, $q_c$, to be decreased and the excess-air factor to be increased, as is shown in Figure 11 and Figure 12, respectively (see in both cases dot lines).

![Figure 11. A change of injected fuel per cycle $q_c$ at different crankshaft speed $n_{cs}$.](image1.png)

![Figure 12. A change of excess-air factor $\lambda$ at different crankshaft speed $n_{cs}$.](image2.png)
3. Conclusion
During the investigation is settled that the smoke emissions of EG is raised at low crankshaft speed due to worse quality of fuel injection and slow down of incoming air, which leads to oxygen shortage. At higher crankshaft speeds the diffuse combustion is disturbed because of the greater fuel amount.

The developed mathematical model refers to trucks, building machines and buses driven by diesel engines. The model points out the dependence of powerful and ecological parameters on crankshaft speed and engine loading. The factors injected fuel per cycle and excess-air factor influencing the smoke emissions of exhaust gases are examined at different operation regimes of the engine.

For the engine investigated the increase of smoke emissions of exhaust gases leaving directly the engine is greatest at crankshaft speed below 1200 \( \text{min}^{-1} \) and excess-air factor less than 2.5. The characteristics pronounced to exert influence on the excess-air factor by changing the injected fuel per cycle could be applied in the systems for control of such a type of engines with the aim of optimizing the regeneration processes of filters for hard substances.

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