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RESEARCH OF RESIDUAL LIFETIME OF DIESEL ENGINES FUEL INJECTION EQUIPMENT BY MEANS OF FUEL INJECTION MATHEMATIC SIMULATION

Vasyl Muzychuk, PhD, Associate Professor
Vitaliy Komaha, PhD, Senior Lecturer
Oleksii Tokarchuk, PhD, Associate Professor
Vinnytsia National Agrarian University
Oleksandr Kukharchuk, engineer for repair of maintenance machinery
TITAN MACHINERY UKRAINE

Музичук Василь Іванович, к.т.н., доцент
Комаха Віталій Петрович, к.т.н., старший викладач
Токарчук Олексій Анатолійович, к.т.н., доцент
Вінницький національний аграрний університет
Кухарчук Олександр Олексійович, інженер з ремонту сільськогосподарської техніки
TITAN MACHINERY UKRAINE

On the basis of the engineering practices of the last 3-4 decades, it is known, tested and proven that, while mathematical models provide reliable descriptions of processes, cycles or physical phenomena, in theoretical research the theory of small deviations is comprehensively preferable to any experiment on regulating characteristics.

Considering the fact that all parameters and characteristics are sometimes interconnected by nonlinear dependences, it is impossible to distinguish the influence of a specific parameter on the sought-for value, and can be even hazardous if the determinant argument or its derivative has extreme values.

The role of mathematic simulation to determine the residual lifetime of the fuel injection equipment of diesel engines was established. It was also noted that for a reliable description of the processes going on in the fuel injection equipment it is advisable to use the method of small deviations.

The mathematical model is based on the known physical laws that describe the interdependence of the two groups of parameters: engine variables and performance parameters - both within the groups and between them. The transition of classical differential equations describing the processes of fuel supply and injection taking into account fuel leaks in precision pairs to the equations with small deviations of parameters is shown.

An analysis of correlations between the parameters of injection, fuel supply and fuel leakage was carried out and the most influential parameters were found. The influence coefficients are found and correlations between the influence coefficients and the corresponding parameters are constructed. We used the correlations found to describe the influence of the technical condition of precision pairs on the engine performance indicators. The correlations between the change in injection patterns and small deviations of the parameters describing technical condition of precision pairs are also established.

Key words: *fuel equipment, diesel engine, mathematical modeling, fuel injection,* technical condition, differential equations.

F. 56. Fig. 9. Ref. 26.

1. Introduction

Taking into account the current circumstances in the world of science, it is extremely difficult to conduct an extensive, comprehensive experiment when customers no longer formulate their additional requirements for equipment, but are content with purchase and supply the equipment available with by machine-building companies. While factory control testing (FCT) or minor control testing (MCT) are somewhat affordable for manufacturers, the major control testing (MCT) and cross sectoral control testing (CSCT) are either not carried out or are carried out in scanty fractions of the total production volume. Reliability tests are only based on the number of cycles (starts, for example), while the general performance of the products as a whole or their systems (especially transport equipment in agriculture) is not tested at all.

These circumstances speak in favor of modeling the regulating characteristics, which is less costly and allows to enjoy the existing high resolution of modern computing technology. On top of that, since over the century of a diesel engine history, a huge amount of experimental data has been accumulated for engineering new and improving the existing diesel engines, with consideration of new technologies, requirements for efficiency, toxicity, reliability and performance [1 – 10].

2. Problem formulation

1. The switch from ordinary differential equations describing the quality of dispersion to the equations in small deviations.

For the purposes of conversion into small deviations the dynamic characteristics of fuel injection, taking into account the hydraulic density of precision pairs (plunger-sleeve, valve-seat, nozzle needle) will be expressed by the following equations:

fuel pressure in the supra-plunger cavity

$$(P_H)_0 = \int_t \frac{1}{\beta V_H}\left[\left(f_H\frac{dh}{dt} - f_k\frac{dh_k}{dt} - \frac{dQ_H}{dt} - \frac{dQ_{yH}}{dt}\right)(1-\sigma)\right]dt, \qquad (1)$$

where β – is fuel compressibility factor, V_H – is fuel volume in the supra-plunger cavity, m³, f_H – cross-sectional area of the supra-plunger cavity, m², $\frac{dh}{dt}, \frac{dh_k}{dt}$ – the plunger speed and the speed of the fuel injection pump discharge valve respectively, m/s, $\frac{dQ_H}{dt}, \frac{dQ_{yH}}{dt}$ – differential characteristics of fuel delivery into supra plunger cavity and the leak through the gap in the plunger pair respectively, m³/s; σ – fuel surface tension, N/m,

fuel pressure in the volume of the discharge valve adapter

$$(P_k)_0 = \int_t \frac{1}{\beta V_k}\left[f_k\frac{dh_k}{dt}(1-\sigma) + \left(f_H\frac{dh}{dt} - \frac{dQ_H}{dt} - \frac{dQ_{yk}}{dt}\right)\sigma - \frac{f_t P_k}{Z} - 2f_t e^{-\alpha L}W_t\right]dt, \qquad (2)$$

where V_k – fuel volume in the cavity of the discharge valve adapter, m³; $f_{k,}$ f_t – cross-sectional area of the discharge valve along the unloading belt and the high-pressure pipeline respectively, m²; P_k – pressure in the cavity of the discharge valve adapter, Pa; Z – acoustic resistance, m²/s; α – damping factor of the fuel supply wave; L – the pipeline length, m; W_t – pipeline fuel delivery wave velocity, m/s.

pressure in the injector nozzle

$$\left(P_p\right)_0 = \int_t \frac{1}{\beta V_\text{ф}}\left[2f_t e^{-\alpha L}\left(\frac{P_k}{Z} + e^{-\alpha L}W_t\right)_{t-\tau} - f_H\frac{dh}{dt} - \frac{f_t P_p}{Z} - \frac{dQ}{dt} - \frac{dQ_{yH}}{dt}\right]dt, \qquad (3)$$

where $V_\text{ф}$ – fuel volume in the injector nozzle cavity, m³; τ – transport lag of the fuel supply wave, s; P_p – pressure in the cavity of the injector nozzle, Pa,

Fuel consumption characteristics, defined as the sum of the volumes that make up the volume of the supra-plunger cavity in the unsteady outflow, are determined by the Bernoulli equation

$$V_H = \left(V_\text{ц} + V_\text{п} + V_{yH} + V_{yk} + V_{yu}\right) = \int_t \sum_{i=1}^{i=5}\left(\frac{dV}{dt}\right)_i dt, \qquad (4)$$

where $V_\text{ц}$ – engine cylinder volume, m³; $V_\text{п}$ – volume of the active stroke part of the plunger, m³; V_{yH} – clearance volume in a precision plunger-sleeve pair, m³; V_{yk} – clearance volume in precision valve-seat pair, m³; V_{yu} – clearance volume in precision pair of nozzle needle, m³,

integral characteristic of injection into the cylinder

$$(Q)_0 = \int_t \left[(\mu f)_\text{ф}\sqrt{\frac{2}{\rho}}\sqrt{P_p - P_\text{ц}}\right] dt \Rightarrow V_\text{ц}, \qquad (5)$$

where $(\mu f)_\text{ф}$ – effective flow area of the spray nozzle; V_H – fuel volume in the supra-plunger cavity, m³; f_H – cross-sectional area of the supra-plunger cavity, m²,

integral fuel consumption through cut-off windows

$$\left(Q_H\right)_0 = \int_t \left[(\mu f)_{0\text{ф}}\sqrt{\frac{2}{p}}\sqrt{P_H - P_\text{вс}}\right]dt \Rightarrow V_H. \qquad (6)$$

integral fuel leakage through the plunger-sleeve leaks

$$\left(Q_{yH}\right)_0 = \int_t \left[(\mu f)_\text{щн}\sqrt{\frac{2}{p}}\sqrt{P_\text{н} - P_\text{вс}}\right]dt \Rightarrow V_\text{ун}, \qquad (7)$$

integral fuel leakage through valve-seat leaks

$$\left(Q_{yk}\right)_0 = \int_t \left[(\mu f)_{\text{щ}k}\sqrt{\frac{2}{p}}\sqrt{P_{oc} - P_o}\right]dt \Rightarrow V_{yk}, \qquad (8)$$

integral fuel leakage through leaks in the spray needle

$$\left(Q_{yH}\right)_0 = \int_t \left[(\mu f)_{\text{щ}H}\sqrt{\frac{2}{p}}\sqrt{P_p - P_o}\right]dt \Rightarrow V_{yH}. \tag{9}$$

The main feature of the transition to equations with small deviations, as recommended in the scientific papers [6, 11], is that the ratio of the change in the differential of functions to the function itself at the same moment is taken as small deviations of functions, i.e. certain values of functions (1 - 9) are taken as absolute values. These values can be conditionally called "standard" or values for systems with original design parameters, this is the first feature. The second feature of the transition should be the assumption that the value of the derivative of the function is taken by any parameter or argument as an independent variable that randomly changes during operation unequally with other changing parameters, i.e., the parameters act as random functions.

The dynamic characteristics of injection are expressed by the rate of pressure change (dP/dt) and the change in pressure (P (t)) during the injection time in the corresponding containers of the fuel system: the supra-plunger cavity of the discharge valve fitting and in the volume of the injector nozzle. Therefore, the initial differential equation that forms the pressure wave is the equation for the rate of pressure change in the supra-plunger cavity

$$\left(\frac{dP_H}{dt}\right)_0 = \frac{1}{\beta V_H}\left(f_H\frac{dh}{dt} - f_k\frac{dh_k}{dt} - \frac{dQ_H}{dt}\right)\times(1-\sigma) - \frac{dQ_H}{dt}(1-\sigma) \tag{10}$$

Then the pressure is $P_H(t) = \int \left({dP_H}/{dt}\right)dt$, while small changes in pressure velocity in the supra-plunger cavity are written as

$$\frac{1}{\beta V_H}\left[(\delta f_H)\frac{dh}{dt} + \left(\delta\frac{dh}{dt}\right)f_H - (\delta f_k)\frac{dh_k}{dt} - \left(\delta\frac{dh_k}{dt}\right)f_k - \left(\delta\frac{dQ_{yH}}{dt}\right) - \left(\delta\frac{dQ_{y\text{п}}}{dt}\right)\right]t \tag{11}$$

or

$$\left(\delta\frac{dP_H}{dt}\right) = K_{39}(\delta f_{\text{п}}) + K_{40}\left(\delta\frac{dh_k}{dt}\right) - K_{41}(\delta f_k) - K_{42}\left(\delta\frac{dh_k}{dt}\right) - K_{43}\left(\delta\frac{dQ_{yH}}{dt}\right) - K_{44}\left(\delta\frac{dQ_{y\text{п}}}{dt}\right). \tag{12}$$

The pressure change parameter in the supra-plunger cavity is an integral of the equation (10):

$$(P_H)_0 = \frac{1}{\beta V_H}\int\left[\left(f_H\frac{dh}{dt} - f_k\frac{dh_k}{dt} - \frac{dQ_{yH}}{dt}\right)\times(1-\sigma) - \frac{dQ_H}{dt}(1-\sigma)\right]dt, \tag{13}$$

then

$$(\delta P_H) = \frac{1}{\beta V_H}\left[(\delta f_{\text{п}})\frac{dh}{dt} + \left(\delta\frac{dh}{dt}\right)f_H - (\delta f_k)\frac{dh_k}{dt} - \left(\delta\frac{dQ_{yH}}{dt}\right) - \left(\delta\frac{dh_k}{dt}\right)f_k\left(\delta\frac{dQ_H}{dt}\right)\right]t, \tag{14}$$

or

$$(\delta P_H) = K_{45}(\delta f_{\text{п}}) + K_{46}\left(\delta\frac{dh}{dt}\right) - K_{47}(\delta f_k) - K_{48}\left(\delta\frac{dQ_{y\text{п}}}{dt}\right) - K_{49}\left(\delta\frac{dh_k}{dt}\right) - K_{50}\left(\delta\frac{dQ_{\text{п}}}{dt}\right). \tag{15}$$

The rate of pressure change in the cavity of the discharge valve adapter is described by the equation:

$$\left(\frac{dP_k}{dt}\right)_0 = \frac{1}{\beta V_k}\left[f_k\left(\frac{dh_k}{dt}\right)(1-\sigma) + \left(f_H\frac{dh}{dt} - \frac{dQ_H}{dt} - \frac{dQ_{yk}}{dt}\right)\sigma - \frac{f_t}{Z}P_k - 2f_t e^{-\alpha L}W_t\right]. \tag{16}$$

In small deviations, is written as:

$$\left(\delta\frac{dP_K}{dt}\right) = \frac{1}{\beta l_K\left(\frac{dP_K}{dt}\right)}\left[(\delta\int K)\frac{dh_K}{dt} + \left(\delta\frac{dh_K}{dt}\right)\int K + (\delta\int II)\frac{dh}{dt} + \left(\delta\frac{dh}{dt}\right)(\int II) - \left(\delta\frac{dQ_{II}}{dt}\right) - \left(\delta\frac{dQ_{YK}}{dt}\right) - \frac{\int T}{Z}(\delta P_K) - 2\int Te^{-01}W\right]. \tag{17}$$

or

$$\left(\delta\frac{dP_K}{dt}\right) = K_{51}(\delta\int K) + K_{52}\left(\delta\frac{dh_K}{dt}\right) + K_{53}(\delta\int II) + K_{54}\left(\delta\frac{dh}{dt}\right) - K_{55}\left(\delta\frac{dQ_{II}}{dt}\right) - K_{56}\left(\delta\frac{dQ_{YK}}{dt}\right) - K_{57}(\delta P_K) - K_{58}(\delta P_K). \tag{18}$$

The pressure in the adapter cavity during the injection period is determined by integrating the equation (16):

$$(P_K)_0 = \frac{1}{\beta V_K}\int\left[\int K\left(\frac{dh_K}{dt}\right)(1-\sigma) + \left(\int II\frac{dh}{dt} - \frac{dQ_{II}}{dt} - \frac{dQ_{YK}}{dt}\right)\sigma - \frac{\int T}{Z}P_K - 2f_t e^{-\alpha L}W_t\right]dt \tag{19}$$

Then, in small deviations (19) shall be written as:

$$(\delta P_K) = \frac{1}{\beta V_K}\left[(\delta\int K)\frac{dh_K}{dt} + \left(\delta\frac{dh_K}{dt}\right)\int K + (\delta\int II)\frac{dh}{dt} + \left(\delta\frac{dh}{dt}\right)\int II - \left(\delta\frac{dQ_{II}}{dt}\right) - \left(\delta\frac{dQ_{YK}}{dt}\right) - -(\delta P_K)\frac{\int II}{Z} - 2\int IIe^{-01}W_t\right], \tag{20}$$

or

$$\left(\delta\frac{dP_K}{dt}\right) = K_{59}(\delta\int K) + K_{60}\left(\delta\frac{dh_K}{dt}\right) + K_{61}(\delta\int II) + K_{62}\left(\delta\frac{dh}{dt}\right) - K_{63}\left(\delta\frac{dQ_{II}}{dt}\right) - K_{64}\left(\delta\frac{dQ_{YK}}{dt}\right) - K_{65}(\delta P_K) - K_{66}. \tag{21}$$

The rate of pressure change and the pressure in the injector nozzle are represented similarly in small deviations. Deviations of the parameters that form the injection characteristics are considered.

The rate of fuel pressure change in the nozzle volume is described by the differential equation:

$$\left(\frac{dP_p}{dt}\right)_0 = \frac{1}{V_\phi \beta}\left[2\int Te^{-01}\left(\frac{P_K}{z} + e^{-01}W\right)_{t-\tau} - \int II\frac{dh_{II}}{dt} - \int T\frac{P_p}{z} - \frac{dQ}{dt} - \frac{dQ_{yII}}{dt}\right]. \quad (22)$$

Then

$$\left(\delta\frac{dP_p}{dt}\right) = \frac{1}{\beta V_\phi}\left[2\int Te^{-01}\frac{(\delta P_K)}{z} + 2\int Te^{-201}W_{t-\tau}(\delta\int II)\frac{dh_{II}}{dt} - \left(\delta\frac{dh_{II}}{dt}\right)\int II - (\delta P_p)\frac{\int T}{z} - \left(\delta\frac{dQ}{dt}\right) - \frac{dQ_{yII}}{dt}\right], \quad (23)$$

or

$$\left(\delta\frac{dP_p}{dt}\right) = K_{67}(\delta P_k) + K_{68} - K_{69}(\delta f_п) - K_{70}\left(\delta\frac{dh_п}{dt}\right) - K_{71}(\delta P_p) - K_{72}\left(\delta\frac{dQ}{dt}\right) - K_{73}\left(\delta\frac{dQ_H}{dt}\right). \quad (24)$$

Taking into account the additional conditions

$$\left.\begin{array}{c} \sigma = \begin{cases} 0 & h_k < h_{k_0} \\ 1 & h_k \geq h_{k_0} \end{cases} \\ V_k = \begin{cases} V_k & h_k < h_{k_0} \\ V_k + V_H & h_k \geq h_{k_0} \end{cases} \\ Z = \alpha\,\rho_t \\ W_{t-\tau} = e^{-\alpha L}\frac{P_k}{z} + e^{-2\alpha L}W_{(t-\tau)} - \frac{1}{z}P_p \end{array}\right\}, \quad (25)$$

The latter of (25) is expressed as follows in small deviations:

$$(\delta W_{t-\tau})_0 = \delta(P_k)\frac{e^{-\alpha L}}{z} + \frac{e^{-\alpha L}\delta(P_k)}{W_{t-\tau}} - (\delta P_p)\frac{1}{z}, \quad (26)$$

$$(\delta W_{t-\tau}) = K_{74}\delta(P_k) + K_{75}\delta(P_k) - K_{76}(\delta P_p). \quad (27)$$

2. Fuel consumption characteristics in small deviations during injection

To solve the given system of equations in small deviations we have to translate the input-output characteristics (5 - 9) and the atomization quality equations, we represent this.

Fuel input-output characteristics in small deviations.

Injection differential characteristics:

$$\left(\delta\frac{dQ}{dt}\right) = (\delta\mu f)_\phi\sqrt{\frac{2}{p}}\sqrt{P_p - P_ц} + \sqrt{\left(\frac{2}{\delta\rho}\right)}(\mu f_\phi)\sqrt{P_p - P_ц} + \mu f_\phi\sqrt{\frac{2}{p}}\sqrt{(\delta P_p) - P_ц} + \mu f_\phi\sqrt{\frac{2}{p}}\sqrt{P_p - (\delta P_ц)}, \quad (28)$$

$$\left(\delta\frac{dQ}{dt}\right) = K_{77}(\delta\mu f_\phi) + K_{78}(\delta\rho) + K_{79}(\delta P_p) + K_{80}\delta(P_ц). \quad (29)$$

Injection integral characteristics in small deviations according to the equation (5):

$$\left(\delta\frac{dQ}{dt}\right) = (\delta\mu f)_\phi\sqrt{\frac{2}{p}}\sqrt{P_p - P_ц} + \sqrt{\left(\frac{2}{\delta\rho}\right)}(\mu f_\phi)\sqrt{P_p - P_ц} + \mu f_\phi\sqrt{\frac{2}{p}}\sqrt{(\delta P_p) - P_ц} + \mu f_\phi\sqrt{\frac{2}{p}}\sqrt{P_p - (\delta P_ц)} \quad (30)$$

or

$$\left(\delta\frac{dQ}{dt}\right) = K_{77}(\delta\mu f_\phi) + K_{78}(\delta\rho) + K_{79}(\delta P_p) + K_{80}\delta(P_ц). \quad (31)$$

or

$$(\delta Q) = K_{81}(\delta\mu f_\phi) + K_{82}\sqrt{\frac{2}{(\delta\rho)}} + K_{83}\sqrt{(\delta P_p) - P_ц} + K_{84}\sqrt{P_p - \delta(P_ц)}. \quad (32)$$

The differential characteristic of the fuel consumption through the cut-off windows in small deviations is represented from the equation (16). In this equation the value of $P_{вс} \ll P_H$ by 2 degrees, therefore the deviation $(\delta P_{вс})$ by 2-3 times doesn't introduce any substantial change and can be neglected in small deviations, then (6) can be rewritten in differential form as follows:

$$\left(\frac{dQ}{dt}\right)_0 = \mu f_0\sqrt{\frac{2}{p}}. \quad (33)$$

In small deviations, omitting transformations

$$\left(\delta\frac{dQ_H}{dt}\right) = (\delta\mu f_0)\sqrt{\frac{2}{p}}\sqrt{P_H} + \sqrt{\left(\frac{2}{\delta\rho}\right)}\sqrt{P_H}\,\mu f_0 + \sqrt{\delta P_H}(\mu f_0)\sqrt{\frac{2}{p}} \quad (34)$$

or

$$\left(\delta\frac{dQ_H}{dt}\right) = K_{85}(\delta\mu f_0) + K_{86}(\delta\rho) + K_{87}(\delta P_H), \quad (35)$$

While the integral equations through the above equations in small deviations

$$(dQ_H) = \left[(\delta\mu f_0)\sqrt{\frac{2P_H}{p}} + \sqrt{\left(\frac{2P_H}{\delta p}\right)}(\mu f_0) + \sqrt{(\delta P_H)}(\mu f_0)\sqrt{\frac{2}{p}}\right]t \quad (36)$$

or

$$(dQ_H) = K_{88}(\delta\mu f_0) + K_{89}(\delta p) + K_{90}(\delta P_H). \quad (37)$$

Quite similarly, the differential and integral characteristics of the fuel through the plunger-sleeve gap will be presented in small deviations ($P_{вс} \ll P_H$) the equation (7):

$$\left(\frac{dQ_{yH}}{dt}\right)_0 = (\mu f)_{шн}\sqrt{\frac{2P_H}{p}},$$

Then the differential leak (36) through clearances in small deviations will be:

$$\delta\frac{dQ_{yH}}{dt}_0 = (\delta\mu f)_{щ}\sqrt{\frac{2P_H}{p}} + \sqrt{\frac{2(\delta P_H)}{p}}(\mu f)_{щ}+(\mu f)_{щ}\sqrt{\frac{2P_H}{(\delta p)}}, \tag{38}$$

or

$$\left(\delta\frac{dQ_{yH}}{dt}\right) = K_{91}(\delta\mu f)_{щ} + K_{92}(\delta P_H) + K_{93}(\delta p). \tag{39}$$

While the integral leak through the plunder pair clearances will be from the integral (7):

$$(Q_{yH})_0 = \int\left[(\mu f)_{щ}\sqrt{\frac{2P_H}{p}}\right]dt, \tag{40}$$

$$(\delta Q_{yH}) = (\delta\mu f_{шн})\sqrt{\frac{2P_H}{p}}\,t + \sqrt{\frac{2(\delta P_H)}{p}}(\mu f_{шн})t + (\mu f_{шн})t\sqrt{\frac{2P_H}{(\delta p)}}, \tag{41}$$

or

$$(\delta Q_{yH}) = K_{94}\left(\delta\mu f_{щ}\right) + K_{95}(\delta P_H) + K_{96}(\delta p). \tag{42}$$

The differential and integral leaks in a precision valve-seat pair determine the value of the residual pressure in the high-pressure pipeline (P_{res}), which determines the beginning of the discharge valve start-up at $t > t = 0, P_H = P_k$. That's why this leak has crucial significance, because if $(\delta P_H = \delta P_k)\,|_{t=0} \leq (\delta P_H) - (\delta P_k)_{t=0}$,

then

$$\left(\frac{d^2h_k}{dt^2}\right)_{t=0} \lessgtr \left(\frac{d^2h_k}{dt^2}\right)_{h_k=h_{k_0}}. \tag{43}$$

Of the equation (8) we can see that $P_{res} \gg P_o$ then the differential leakage through the clearance of the discharge valve is represented as:

$$\left(\delta\frac{dQ_{yk}}{dt}\right)_0 = \left(\mu f_{щk}\right)\sqrt{\frac{2P_{res}}{p}}, \tag{44}$$

While in small deviations

$$\left(\delta\frac{dQ_{yk}}{dt}\right)_0 = \left(\delta\mu f_{щk}\right)\sqrt{\frac{2P_{res}}{p}} + \left(\mu f_{щk}\right)\sqrt{\frac{2}{p}}\sqrt{(\delta P_{res})} + \left(\mu f_{щk}\right)\sqrt{2P_{res}}\sqrt{\frac{1}{\delta\rho}}, \tag{45}$$

or

$$\left(\delta\frac{dQ_{yk}}{dt}\right)_0 = K_{97}\left(\delta\mu f_{щk}\right) + K_{98}(\delta P_{res}) + K_{99}(\delta\rho). \tag{46}$$

The integral leakage through the clearances of the discharge valve will be equal in small deviations from (44):

$$(Q_{yk})_0 = \left[\left(\mu f_{щk}\right)\sqrt{\frac{2P_{res}}{p}}\right]t, \tag{47}$$

$$(\delta Q_{yk}) = \left[\left(\delta\mu f_{щk}\right)t\sqrt{\frac{2P_{res}}{p}}\sqrt{(\delta P_{res})} \times \sqrt{\frac{2}{p}}\left(\mu f_{щk}\right) \times t\sqrt{\frac{1}{(\delta\rho)}}\sqrt{2P_{res}}\left(\mu f_{щk}\right)\right], \tag{48}$$

$$(\delta Q_{yk})_0 = K_{100}\left(\delta\mu f_{щk}\right) + K_{101}(\delta P_{res}) + K_{102}(\delta\rho). \tag{49}$$

Closing leaks are the leaks through clearances in a precision pair of a nozzle-needle. Since these leaks are diverted to the fuel tank without backpressure, the differential leak is written as follows, instead of the equation (9).

$$\left(\delta\frac{dQ_{yH}}{dt}\right)_0 = \mu f_{щн}\sqrt{\frac{2}{p}}\sqrt{P_p} \quad \text{т.к.} \quad P_p \gg P_0, \tag{50}$$

In small deviations

$$\left(\delta\frac{dQ_{yH}}{dt}\right) = \left(\delta\mu f_{щн}\right)\sqrt{\frac{2P_p}{p}} + \sqrt{\frac{2(\delta P_p)}{p}}\left(\mu f_{щн}\right) + \left(\mu f_{щн}\right)\sqrt{\frac{2P_p}{(\delta p)}}, \tag{51}$$

or

$$\left(\delta\frac{dQ_{yH}}{dt}\right) = K_{103}\left(\delta\mu f_{щн}\right) + K_{104}(\delta P_p) + K_{105}(\delta p) \tag{52}$$

While the integral leakage over the injection period through the clearances of the nozzle-needle in small deviations from (50) will be expressed by the following integral:

$$(\delta Q_{yH})_0 = \int\left[\mu f_{щн}\sqrt{\frac{2}{p}}\sqrt{P_p}\right]dt, \tag{53}$$

In small deviations through the divergences of parameters:

$$(\delta Q_{yH}) = \left(\delta \mu f_{\text{щн}}\right)\sqrt{\frac{2P_p}{p}}\,t + (\delta P_p)\sqrt{\frac{2}{p}}\left(\mu f_{\text{щн}}\right)t + \sqrt{\frac{1}{(\delta\rho)}}\sqrt{2P_p}\left(\mu f_{\text{щн}}\right)t, \tag{54}$$

or

$$\delta Q_{yH} = K_{106}\left(\delta \mu f_{\text{щн}}\right) + K_{107}(\delta P_p) + K_{108}(\delta \rho). \tag{55}$$

The obtained equations in small deviations enable to differentiate the influence of any parameter affecting the injection characteristics, which is extremely important for analysis when designing, adjusting for a selected diesel and introducing operational changes.

3. Results of research

We consider the brands of the KamAZ, MAZ etc. as the most commonly used vehicles in the agricultural industry, while the most widely used diesel engines are KamAZ-740, YaMZ-238. Their reliability guarantees their ability to perform transport operations. The higher the guaranteed life, the lower the likelihood of interruption of the technological process of agricultural production; the lower the failure rate, the higher the availability. Therefore, the less the influence of a parameter deviation on the fuel equipment and diesel engine operations, the longer the diesel engine operating time (in engine hours), the fewer failures, and the probability of failures during the residual life period will be zero or close to it. Hence, the residual life of a diesel engine are engine hours remaining until the end of the guaranteed resource. They are defined as

$$T_0 = T_г - T_н, \tag{56}$$

where $T_г$ – guaranteed engine hours; $T_н$ – fixed running engine hours.

The latter is normally defined by kilometerage (by speedometer) for trucks and by running hours (service hour meter) for tractors and harvester vehicles.

It is not possible to cover all diesel systems that serve to obtain mechanical energy and group them under one indicator (running hours or 1 mileage) in such complex machines as KamAZ or MAZ for the following reasons:

- unequal wear resistance of friction pairs, systems and units;
- different mechanical and thermal loads;
- different restoration costs;
- different aging rates during storage.

Based on this, the warranty life of a vehicle always differs from the warranty life of the power plant. We will consider the data (we acknowledge the advertising purposes of the data) of manufacturers for YaMZ - 10-12 thousand hours, KamAZ - 8-10 thousand hours, mileage of the vehicles (on paved roads), for average hours (11 103 and 9 103) 660,000 and 540,000 km before overhaul (average operating speed is assumed to be 60 km/h) respectively.

The actual operating conditions of the vehicles in the national economy, in special situations, are generally incomparable both in terms of load and mileage. This enables us to suggest (and this is close to the real average statistical data) that diesel engines have a guaranteed operating time within 6000 hours, especially since they have been undergoing the reliability tests, that are assigned as large control tests, are carried out in Ukraine and in the post-USSR countries [12]. Based on this, we will determine the residual life of the fuel equipment and diesel engine by small deviations of their parameters, taking into account the rate of their change in operating time.

It should be especially noted that changes in the effective indicators from small deviations of the parameters of the fuel equipment will be presented separately, and then we represent their combined effect and deviations of the parameters of the fuel equipment, and the deviations of the parameters of the diesel engine.

1. The influence of the technical condition of precision pairs on the effective performance of a diesel engine

Here, the technical condition of precision pairs of fuel equipment is understood as: diametrical clearance, wear of the cut-off edge of the plunger and bypass (cut-off) holes for the plunger-sleeve pair and changes in the plunger stroke from wear of the cam, pusher and plunger shank [13].

For the discharge valve, the arguments remain valid: diametral clearance and change in the discharge stroke due to wear of the tapered sealing lips. For the nozzle needle, the diametral clearance, the minimum movement of the needle are taken into account, which changes both the kinematic and dynamic parameters of the injection due to wear of the seal cone. It is also necessary to take into account changes in the effective flow area either due to erosive wear, or due to coking of the spray holes. For the specified technical condition of precision pairs, the output parameters of the diesel engine are considered equal to the initial state.

It should be specially noted that the values at operating hours of more than 6000 hours are taken as the limiting values of deviations of small parameters, and the rates of change of small deviations are taken as arithmetic mean according to the data given in [13].

The below analysis is not absolutely new, similar data are available in [3, 14 - 16] and other non-life tests, but our results relate to the basic principle, discrete changes in these parameters and the determination of the residual life of fuel equipment and diesel engine.

Fig. 1 - 4 shows the calculated by the mathematical model and the experimental values of the deviations of the diesel indicators from the deviations of the parameters of the fuel equipment in terms of operating time (T, hour). The calculated values were obtained according to the mathematical model of the injection process in small deviations, which were obtained as a result of operating time every 1500 hours. The output parameters of the diesel engine were compared with the experimental indicators, and the calculated data were obtained according to the mathematical model and the calculation method of Professor PhD in Technical Sciences V.P. Muravyova and PhD in Technical Sciences V. A. Kulakova [20].

In this case, the change in the parameters of the diesel engine (Fig. 1 - 4) were taken to be zero, the changes in the N_e, g_e of the diesel engine obtained experimentally, are the maximum possible from the simultaneous (total) effect of the change in the parameters of the fuel equipment and the diesel engine, while the calculated data are differentiated, that is, only from those that are applied to a specific drawing, as averages of the action and parameters.

The established mutual influences do not contradict to the scientific works [15, 16, 19] and others, but the fact of practical coincidence of the experiment and calculation allows us to keep hoping that it is possible to find the criteria for the limiting state of a diesel engine, as suggested in [8].

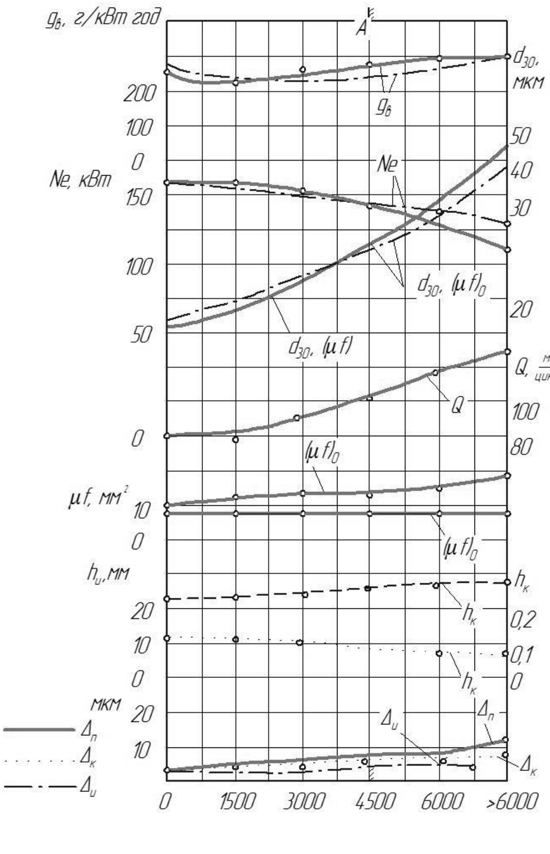

Fig. 1. Changes in diesel indicators from small deviations of operating time parameters (diesel KamAZ-740) $(N_e)_0$ = 160 kW; n = 2600 1/min.
experiment —— calculation -·-·-·-

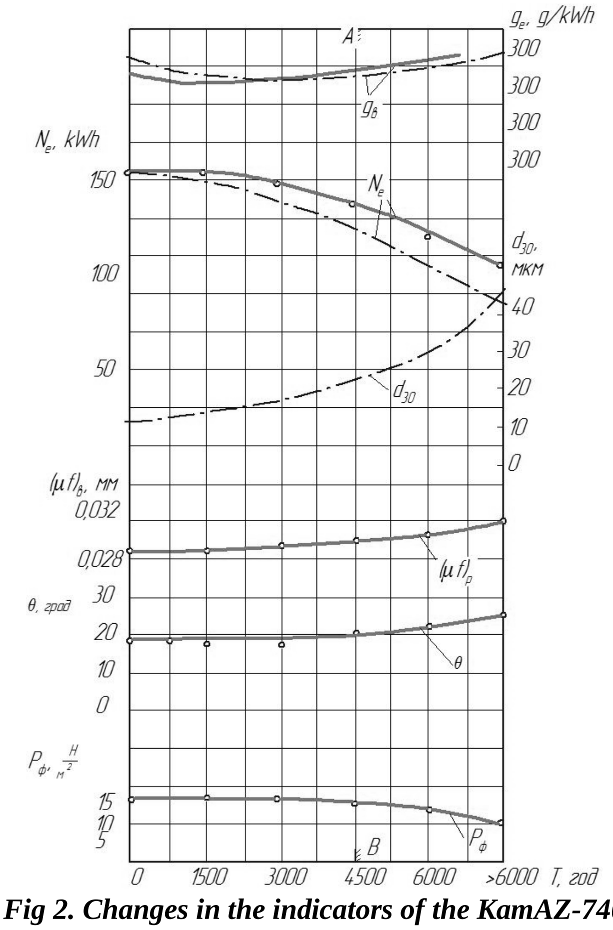

Fig 2. Changes in the indicators of the KamAZ-740 diesel engine from small deviations of parameters in terms of operating time $(N_e)_0$ = 160 kW; n = 2600 rpm.
experiment —— calculation -·-·-·-

The analysis of the data (Fig. 1) shows that the increase in the wear of precision pairs (Δ*n*, *к*, *u*) requires an increase in the cyclic feed ($q_ц$), but its increase to 128 mm3 does not improve the quality of atomization (d_{30}), when operating in within 6000 hours (d_{30}) lies within 50 μm, while during the operating time from 0 to 2000 hours it lies within 15 - 26 μm, which are considered optimal [6, 20].

Diesel power N_e is reduced by almost a third, and the specific effective fuel consumption g_e increases by the same virtually relative value.

A tangible increase in the effective flow area of the cut-off window $(\mu f)_o$, due to cavitation wear of the cut-off adjustable edge, requires an increase in the active stroke of the plunger for growth ($q_ц$) and decrease (d_{30}), while slightly decreasing only up to an operating time of 3700 hours [21].

The relatively intensive wear of precision pairs virtually ends with the same operating time (Δn, k, u), and the increase in the cycle feed with this operating time is only 25%, the engine loses almost no power. Specific effective fuel consumption remains in the range of 210 - 217 g / kWh

Based on the data obtained, according to these small deviations (Fig. 1), it would be possible (tentatively, but not finally) to establish the maximum operating time of 4500 hours, because it is necessary to analyze the remaining small deviations in the first place (Fig. 2 - 4), and then analyze changes in work processes for individual cycles of fuel equipment and diesel at the end of the operating time corresponding to 4500 hours.

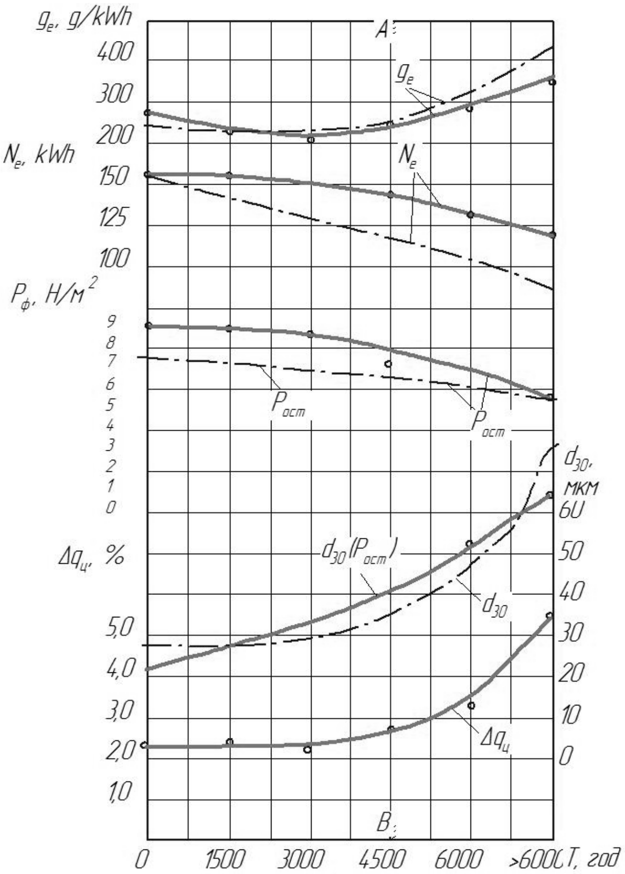

Fig 3. Changes in the indicators of the KamAZ-740 diesel engine from small deviations of parameters in terms of operating time $(N_e)_0$= 160 kW; n = 2600rpm.
experiment—— calculation -·-·-·-·-

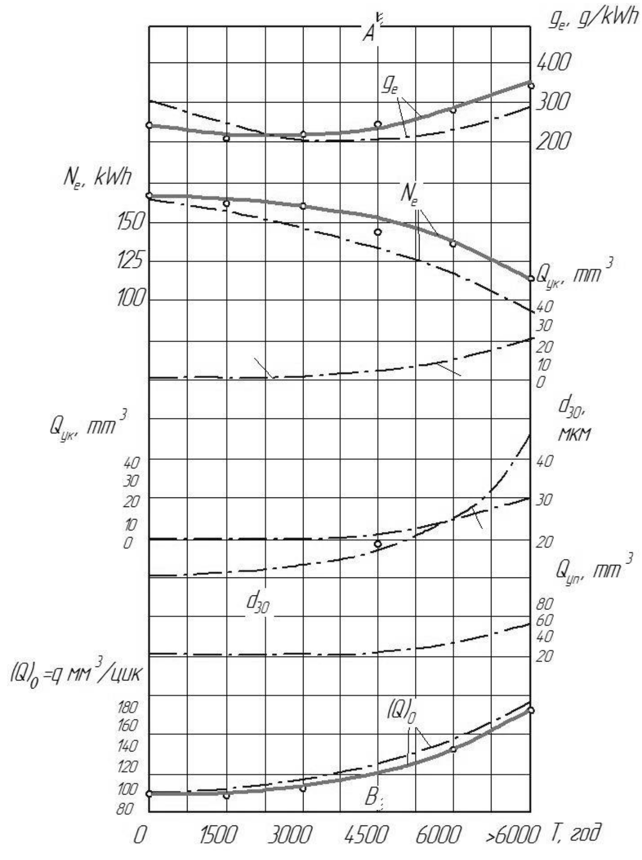

Fig 4. Changes in the indicators of the KamAZ-740 diesel engine from small deviations of parameters in terms of operating time $(N_e)_0$ = 160 kW; n = 2600 rpm.
experiment—— calculation -·-·-·-·-

A noticeable loss of power (Fig. 2-4) occurs after 4500 hours due to a sharp increase in the advance of the feed (Θ) and the effective flow area of the atomizer $(\mu f)_p$, and a decrease in the stiffness of the needle spring $(\mu f)_p$, leads to an increase in the diameter of the drop (d_{30}), which results in the decrease of power (N_e) and the increase of the specific fuel consumption (g_e). Note that this is close to 4500 hours again, i.e. we can again, yet tentatively, set the limit operating time of 4500 hours for these small deviations as well.

We mentioned the unequal wear resistance of parts and assemblies, and given the tolerances for manufacturing, heat treatment, assembly of the fuel system and adjustment, we are convinced that the unevenness ($\Delta q_{ц}$) of the fuel supply through the cylinders is more fraught than anything. Therefore, an equally important parameter should be the value of the residual pressure in the pipeline (P_{res}) (Fig. 3). A decrease in the residual pressure delays the formation of a pressure impulse and the moment when the needle starts to move. This leads to deterioration in the quality of atomization (d_{30}), decrease in power (N_e) and the increase in fuel consumption (g_e).

We shall note that this is felt at 4500 hours of operation, when the unevenness of the feed on the cylinders increases sharply. Such a phenomenon was established for automotive diesel engines by prof. V.F. Anisimov at the level of 1500 - 1800 hours of operation [24, 26].

The clearances of precision pairs increase with operating time (Fig. 4). It should be noted that an increased number of leakages in atomizer ($Q_{уи}$) and ($Q_{ук}$) leads to a sharp drop in residual pressure (P_{res}), which results in the increase of average volumetric diameter of the droplet, delay in the combustion process far beyond tc, which leads to a sharp decrease in power and increased fuel consumption (g_e). The presented data testify that this occurs within the total operating time of 4500 hours, when the droplet diameter increases from 20 microns to 48 microns, which is unacceptable for high-speed diesel engines [24].

We shall tentatively conclude that 4500 engine hours can be considered a guaranteed resource. According to the above analysis, we outline this in Fig. 1 - 4 by the vertical line A - B, dividing the operating time into permissible (left) and dangerous (right).

2. Changes in the parameters of injection characteristics from small deviations of precision pairs

Integral (Q) and differential (ΔQ) injection characteristics along with other optimal parameters of the fuel equipment and diesel engine determine the power and efficiency indicators. In this case, the quality of spraying according to the average volumetric diameter of the droplet should be in the range of 20 - 30 μm for undivided combustion chambers [24, 25].

Figures 5 - 9 show the results of processing oscillograms of non-motorized tests and calculation results using a mathematical model of injection in small deviations for the studied diesel engine KamAZ-740

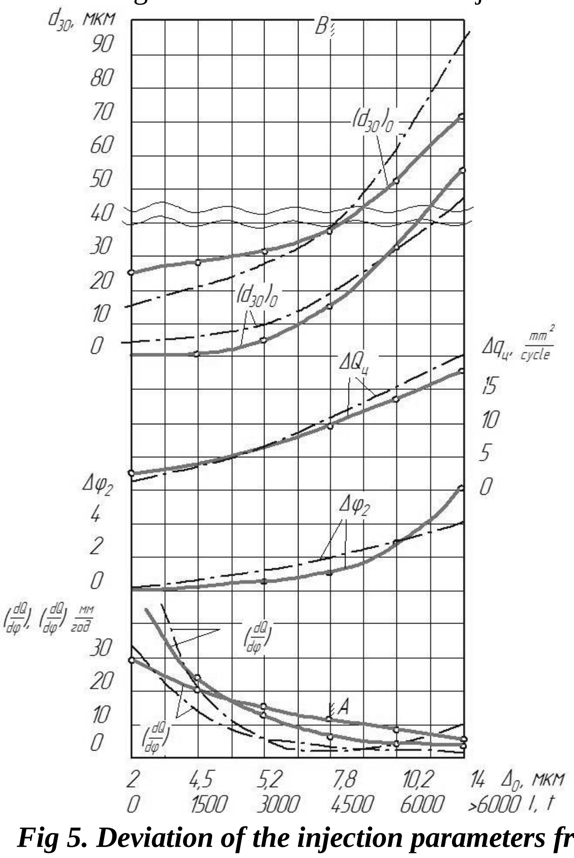

Fig 5. Deviation of the injection parameters from the change in the parameters of the "Plunger-sleeve" $(N_e)_0$ = 160 kW; n = 2600 rpm; φ_2=26 deg.n.c. shaft.
experiment—— calculation -·-·-·-

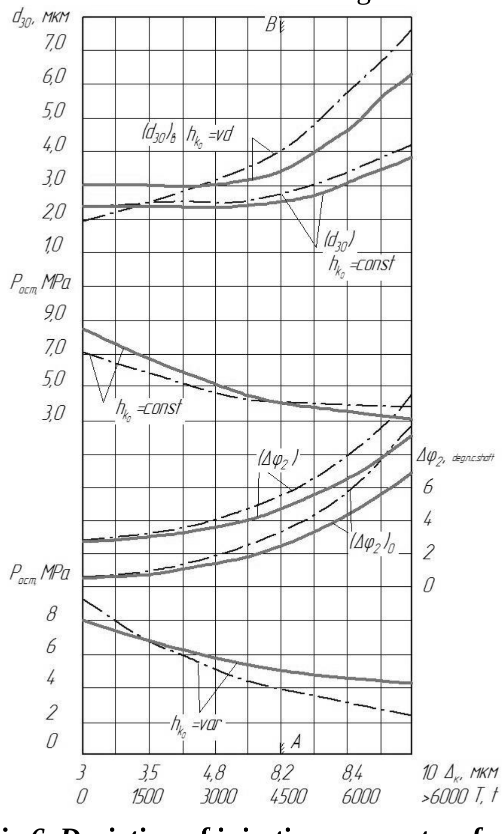

Fig 6. Deviation of injection parameters from changing the "valve-seat" parameters $(N_e)_0$ = 160kW; n = 2600 rpm; φ_2 = 26 deg.n.c.shaft.
experiment—— calculation -·-·-·-

Let us analyze the changes in the parameters of the injection characteristics, the results of which will be used to analyze the combustion processes from two pursued positions: firstly, this will prove the validity of the mathematical model of injection in small deviations, secondly, it will enable us to scientifically substantiate the correction of injection processes in order to maintain acceptable parameters of fuel burnup and increase the duration of diesel engine operation, and the most desirable argument (consequently) will be the exact boundary of the warranty period, on the basis of which the residual resource will also be precisely determined. This is currently the most important for the machinery that is operated in agricultural production.

Since the main defining parameters of the injection duration (φ2) are Q and $^{dQ}/_{d\varphi} = f(P_p, h_u)$, the atomization quality is $d_{30} = f(P_p(h_u), \mu f_p)$, therefore, the parameters of the deviation of the injection characteristic are the change in $Q \times {}^{dQ}/_{d\varphi}$, $\Delta\varphi_2$, $\Delta Q_з$, ΔQ_n as parameters affecting the quality of spraying at the beginning of the supply and at the end of the supply (Fig. 5).

The data (Figure 5) show and to some extent warn that the change in the injection speed along the leading and trailing edges $\left(\frac{dQ}{d\varphi}\right)_{n,з}$ are most significant within up to 3000 hours of operation, with the wear of Δp = 5.2 µm, and then settle back approaching a constant value within 5 - 7 mm3 per cycle. Although it is known that a decrease in the hydraulic density of a precision pair “sleeve - plunger” significantly affects the forming pressure pulse $P_н$ and its rate of change ($dP_н/d\varphi$), according to which the pulse is formed ($dP_к/d\varphi$), and the injection characteristic is determined by ($dP_p/d\varphi$) (the equations are given in [13]).

The emerging “d'Alembert’s paradox” is solvable, since leaks through the clearances in the “sleeve - plunger” pair should be considered as two processes: the formation of a pressure wave at the pipeline inlet and a reflected wave ($W_{t=\tau}$) (the equation is given in [13]), which both in phase and in amplitude can distort the impulse entering the high-pressure pipeline, as shown in the work [11].

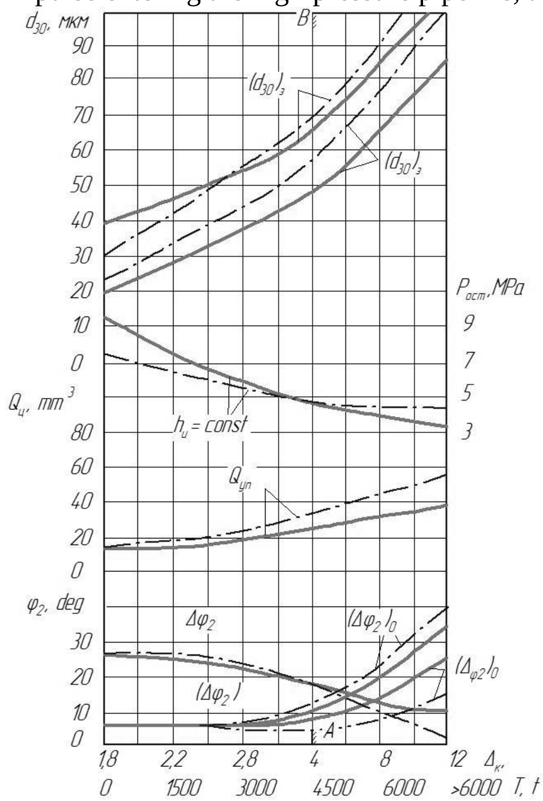

Fig 7. Deviation of the injection parameters from the change in the "needle-spray" parameters: $(N_e)_0$ = 160 kW; n = 2600 rpm; φ_2 = 26 deg. n. count shaft. experiment —— calculation -·-·-·-

It[11] confirms that the increase in the duration of the injection is determined by the fact that the increased cycle feed ($\Delta q_ц$) leads to an increase in the duration of the injection ($\Delta\varphi_2$) (Fig. 5). Hence all the consequences, in particular, the deterioration in the quality of atomization (d_{30}), both on the leading and trailing edges. This can additionally be explained by the following fact: the increased cycle feed at (μf) = const increases (φ_2) the duration of the feed. Overlapping a direct pressure wave ($dP_н/d\varphi$) on a reflected one (W_t/z) gives a noticeable negative effect - the pressure at the end of the injection drops sharply, the quality of atomization (d_{30}) deteriorates along the leading and trailing edges, which is confirmed by both the calculation and the experiment, however the operating time is near 4500 hours again.

We will not pay attention to this fact until we analyze the change in the input pressure impulse into the pipeline after the discharge valve. If this value for the operating time remains by the end of the injection (for the analysis of the needle-sprayer) with a two-fold change $(\mu f)_p$ according to non-motorized and motor tests, then our expectations and hopes will come true.

Let us analyze the deviations of the injection parameters from the change in the “valve-seat” parameters (Fig. 6). The “valve-seat” assembly can be considered as a plunger with a smaller diameter, in which the active disturbance of the input pulse is carried out before the release of the unloading band from the seat channel ($h_к$ = h_{K_0}). If $h_к$ > h_{K_0}, then the perturbation is carried out by the plunger, and the compressing volume is equal to the sum ($V_н + V_к$),, the damping of the disturbance at the beginning of the high-pressure pipeline is noticeable, although at the end of the pipeline it is not so obvious, since the movement of the needle and its diameter is significantly less than the stroke and diameter of the plunger.

On the basis of the above it is also obvious that there are changes in the parameters that determine the changes in the characteristics of the injection and the quality of atomization from the "valve-seat" parameters (Fig. 6).

From the data obtained, when the conical landing surface was brought up ($h_к$ = var) even by a very small amount $\left(\delta h_{K_0}\right)$, [13] the rate of pressure change in small deviations (equation 15) significantly decreases, and after closing the valve ($h_к$ = 0), the residual pressure decreases sharply, to significantly lower limits, than established for the case of new apparatus (equations 16 - 18).

Hence the increase in the duration of the injection both along the front $(\Delta\varphi_2)_п$ and along the rear $(\Delta\varphi_2)_з$ injection fronts. While maintaining the unloading stroke of the discharge valve (h_{K_0} = *const*), the residual pressure decreases without going beyond the reduction limit of the order of 3 MPa, and this determines that the discharge valve is more fraught not by wear of the diameter of the discharge belt, but by the density of the shut-off cone. This argument is also confirmed by the fact that the retention of residual pressure contributes to a clearer onset of the formation of a pressure wave at the pipeline inlet ($dP_к/dt$), (equation 14), while the average volumetric diameter of the droplet in operating time varies slightly over time. An increase in (h_{K_0}) = var in operating time leads to a sharp deterioration in the quality of spraying (Fig. 6) ($d_{30} = f\left(h_{K_0}\right)$ = var).

The analysis of the above data inspires once again and brings hope that the valve operating time equal to 4500 hours, sort of retains its critical merit. Below we shall give this the final status as a parameter.

Changes in the characteristics of the injection depending on the parameters of the spray needle are shown in Fig. 7. Analysis of these data shows that as the hydraulic density (Δu) decreases, the duration of the injection decreases by almost 3 times, and the duration of the front $(\Delta\varphi_2)_п$ and rear $(\Delta\varphi_2)_з$ fronts does so up to 40 degrees of crankshaft rotation. Leaks through the clearances reach up to 40 mm3/cycle (Q_{yu}), as a result of which, even with a constant value of the needle displacement, the residual pressure (P_{res}) decreases from 9 to 3 MPa. All these influences result in a sharp deterioration in the quality of atomization in general, and along the leading and trailing edges (d_{30}) reaches 100 or more micrometers. In this case, the burnout process is significantly shifted beyond tc, which reduces the power and torque of the diesel engine.

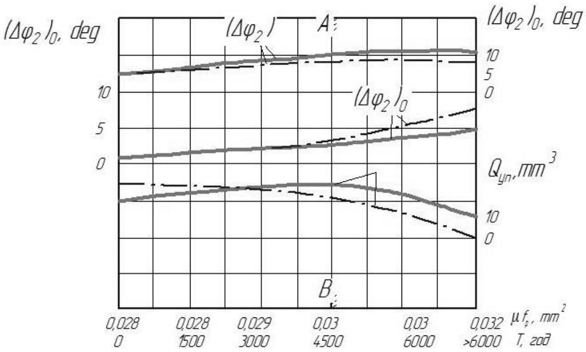

Fig 8. Deviation of the injection parameters by changing the atomizer (non-motorized tests):
$(N_e)_0$ = 160 kW; n = 2600 rpm;
φ_2 = 26 deg. n. count shaft.
experiment —— calculation -·-·-·-

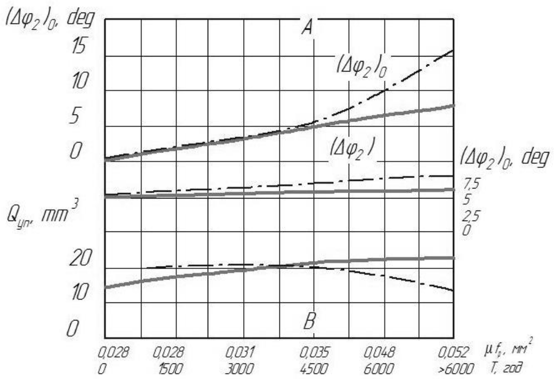

Fig 9. Deviation of injection parameters by changing the atomizer (non-motorized tests):
$(N_e)_0$ = 160 kW; n = 2600 rpm;
φ_2 = 26 deg. n.c. shaft.
experiment calculation -·-·-·-

Here we especially note that the maximum permissible change should be considered the duration of injection (φ_2) - which decreased from 260 to 80, and the average volumetric diameter of the droplet increased along the fronts to 60 μm. Taking into account the increase in the duration of the injection to 100, then the total φ_2 is equal to 28 degrees of crankshaft rotation. If the average droplet diameter in the main injection was 18 - 25 microns, then the deterioration generally does not exceed 12 - 14 microns to the operating time of

4500 hours. Thus, the gap of 4 microns in the spray needle and the recoverable leakage within 40 - 45 mm3 will not lead to disruption of the diesel engine operation. This circumstance should be taken into account when analyzing the operation process of a diesel engine.

The most positive fact remains that if the effective flow area of the nozzle $(\mu f)_p$ does not change during the operating time (Fig. 8), then the working sections of the characteristic both along the fronts and during the main injection remain constant, changing just by an imperceptible amount. In contrast, the parameter (Q_{yu}) leads in comparison with motor tests (Fig. 4), this is probably due to the fact that if the kinematic viscosity increases with increasing temperature and leaks increase due to the increase in the integral characteristic of the injection.

Motor tests and oscillography of the parameters of the differential characteristics of the injection (Fig. 9) give grounds to conclude that an increase in the flow area of the nozzle, due to erosive wear of the spray holes from 0.028 to 0.052 mm2, leads to a significant increase in the duration of the trailing edge. This can be explained by the need to squeeze out the fuel with the needle after the end of the cutoff. An increase in $(\mu f)_p$ leads to a decrease in fuel leakage. The change in fuel leakage towards a decrease starts from μf_p = 0.035 mm2 to 0.052, with operating time from 4500 to 6000 hours.

4. Conclusions

1. The parameters of the differential characteristics of the injection remain within the permissible limits in operation both in non-motorized and motor tests up to 4500 hours of operation. In addition, the coincidence of small deviations obtained in the course of the experiment and by calculation through the mathematical model of injection in small deviations is quite satisfactory for such multiparameter relationships. Large percentages of the difference with operating time in the area of 6000 hours and more are obtained for the sole reason of the maximum deviations being taken from the data on operational tests, i.e. according to statistical data, the processing of which, in our opinion, is not quite correct. But, nevertheless, our calculations mostly confirm these data.

2. Having the results of the analysis of changes in the parameters of the differential characteristics of the injection, it is a fair assumption to say that the operating time of the fuel equipment up to 4500 hours is a guaranteed (flawless) operating time (Fig. 9). We will consider this as a starting point for further research to determine the residual life, but first it is necessary to establish the maximum permissible small deviations of the technical state of precision pairs and adjustments of fuel equipment.

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ДОСЛІДЖЕННЯ ЗАЛИШКОВОГО РЕСУРСУ ПАЛИВНОЇ АПАРАТУРИ ДИЗЕЛЬНИХ ДВИГУНІВ МАТЕМАТИЧНИМ МОДЕЛЮВАННЯМ ВПРИСКУ ПАЛИВА

На основі інженерної практики останніх 3-4 десятиліть відомо, перевірено і доведено, що, в той час як математичні моделі забезпечують надійне опис процесів, циклів або фізичних явищ, в теоретичних дослідженнях теорія малих відхилень в цілому краще, ніж теорія малих відхилень. будь-який експеримент по регулюючим характеристикам.

З огляду на той факт, що всі параметри і характеристики іноді пов'язані між собою нелінійними залежностями, неможливо виділити вплив того чи іншого параметра на шукане значення, а може бути навіть небезпечно, якщо аргумент детермінанта або його похідна має екстремальні значення.

Встановлено роль математичного моделювання для визначення залишкового ресурсу паливної апаратури дизельних двигунів. Також було відзначено, що для достовірного опису процесів, що відбуваються в апаратурі вприскування палива, доцільно використовувати метод малих відхилень.

Математична модель заснована на відомих фізичних законах, які описують взаємозалежність двох груп параметрів: змінних двигуна і параметрів продуктивності - як всередині груп, так і між ними. Показаний перехід класичних диференціальних рівнянь, що описують процеси подачі і впорскування палива з урахуванням витоків палива в точних парах, до рівнянь з малими відхиленнями параметрів.

Проведено аналіз кореляцій між параметрами уприскування, подачі палива і витоку палива і визначені найбільш важливі параметри. Знайдено коефіцієнти впливу і побудовані кореляції між коефіцієнтами впливу і відповідними параметрами. Знайдені кореляції використовувалися для опису впливу технічного стану прецизійних пар на показники роботи двигуна. Встановлено кореляції між зміною схем уприскування і невеликими відхиленнями параметрів, що описують технічний стан прецизійних пар.

Ключові слова: *паливна апаратура, дизельний двигун, математичне моделювання, впорскування палива, технічний стан, диференціальні рівняння.*

Ф. 56. Рис. 9. Літ. 26.

ИССЛЕДОВАНИЕ ОСТАТОЧНОГО РЕСУРСА ТОПЛИВНОЙ АППАРАТУРЫ ДИЗЕЛЬНЫХ ДВИГАТЕЛЕЙ МАТЕМАТИЧЕСКИМ МОДЕЛИРОВАНИЕМ ВПРЫСКА ТОПЛИВА

На основе инженерной практики последних 3-4 десятилетий известно, проверено и доказано, что, в то время как математические модели обеспечивают надежное описание процессов, циклов или физических явлений, в теоретических исследованиях теория малых отклонений в целом предпочтительнее, чем теория малых отклонений. любой эксперимент по регулирующим характеристикам.

Учитывая тот факт, что все параметры и характеристики иногда связаны между собой нелинейными зависимостями, невозможно выделить влияние того или иного параметра на искомое значение, а может быть даже опасно, если аргумент детерминанта или его производная имеет экстремальные значения.

Установлена роль математического моделирования для определения остаточного ресурса топливной аппаратуры дизельных двигателей. Также было отмечено, что для достоверного описания процессов, происходящих в аппаратуре впрыска топлива, целесообразно использовать метод малых отклонений.

Математическая модель основана на известных физических законах, которые описывают взаимозависимость двух групп параметров: переменных двигателя и параметров производительности - как внутри групп, так и между ними.

Показан переход классических дифференциальных уравнений, описывающих процессы подачи и впрыска топлива с учетом утечек топлива в точных парах, к уравнениям с малыми отклонениями параметров.

Проведен анализ корреляций между параметрами впрыска, подачи топлива и утечки топлива и определены наиболее важные параметры. Найдены коэффициенты влияния и построены корреляции между коэффициентами влияния и соответствующими параметрами. Найденные корреляции использовались для описания влияния технического состояния прецизионных пар на показатели работы двигателя. Установлены корреляции между изменением схем впрыска и небольшими отклонениями параметров, описывающих техническое состояние прецизионных пар.

Ключевые слова: *топливная аппаратура, дизельный двигатель, математическое моделирование, впрыск топлива, техническое состояние, дифференциальные уравнения.*

Ф. 56. Рис. 9. Лит. 26.

ВІДОМОСТІ ПРО АВТОРІВ

Музичук Василь Іванович – кандидат технічних наук, доцент кафедри «Технологічних процесів та обладнання переробних і харчових виробництв» Вінницького національного аграрного університету (вул. Сонячна, 3, м. Вінниця, 21008, Україна, e-mail: wasil@vsau.vin.ua).

Комаха Віталій Петрович – кандидат технічних наук, старший викладач кафедри агроінженерії та технічного сервісу Вінницького національного аграрного університету (вул. Сонячна, 3, м. Вінниця, Україна, 21008, e-mail: komacha@vsau.vin.ua).

Токарчук Олексій Анатолійович – кандидат технічних наук, доцент кафедри «Технологічні процеси та обладнання переробних і харчових виробництв» Вінницького національного аграрного університету (вул. Сонячна, 3, м. Вінниця, 21008, Україна, e-mail: tokarchuk@vsau.vin.ua).

Кухарчук Олександр Олексійович – інженер з ремонту технічного облуговування та діагностики сільськогосподарської техніки TITAN MACHINERY UKRAINE (вулиця Ватутіна, 172, Вінниця, 21000, +380682134829)

Музычук Василий Иванович – кандидат технических наук, доцент кафедры «Технологических процессов та оборудования перерабатывающих та пищевых производств» Винницкого национального аграрного университета (ул. Солнечная, 3, г. Винница, 21008, Украина, e-mail: wasil@vsau.vin.ua).
Комаха Виталий Петрович - кандидат технических наук, старший преподаватель кафедры Агроинженерия и технического сервиса Винницкого национального аграрного университета (ул. Солнечная, 3, г.. Винница, Украина, 21008, e-mail: komacha@vsau.vin.ua).
Токарчук Алексей Анатольевич – кандидат технических наук, доцент кафедры «Технологические процессы и оборудование перерабатывающих и пищевых производств» Винницкого национального аграрного университета (ул. Солнечная, 3, г.. Винница, 21008, Украина, e-mail: tokarchuk@vsau.vin.ua).
Кухарчук Александр Алексеевич - инженер по ремонту и техническому обслуживанию и диагностики сельскохозяйственной техники TITAN MACHINERY UKRAINE (улица Ватутина, 172, Винница, 21000, +380682134829)

Muzychuk Vasyl – PhD, Associate Professor of the Department of “Technological Processes and Equipment of Processing and Food Productions” of the Vinnytsia National Agrarian University (3 Solnechnaya St, Vinnytsia, 21008, Ukraine, e-mail: wasil@vsau.vin.ua).
Vitaliy Komaha - Candidate of Technical Sciences, Senior Lecturer of the Department of Agroengineering and Technical Services of the Vinnytsia National Agrarian University (3, Solnyshchaya St., Vinnitsa, Ukraine, 21008, e-mail: komacha@vsau.vin.ua).
Oleksii Tokarchuk – PhD, Associate Professor of the Department of “Technological Processes and Equipment of Processing and Food Productions” of the Vinnytsia National Agrarian University (3, Solnyshchaya St., Vinnytsia, 21008, Ukraine, e-mail: tokarchuk@vsau.vin.ua).
Oleksandr Kukharchuk - engineer for repair of maintenance and diagnostics of agricultural machinery TITAN MACHINERY UKRAINE (Vatutina, 172, Vinnytsia, 21000, +380682134829)