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Development of a mathematical model of fuel equipment and the rationale for diagnosing diesel engines by moving the injector needle

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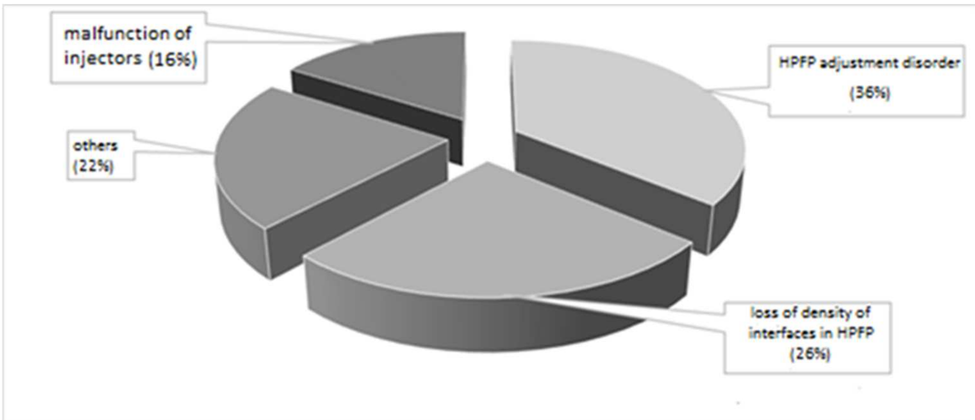
Abstract. The article discusses the problem of diagnosing diesel fuel equipment. The most common methods for diagnosing fuel equipment are presented. The rationale for diagnosing the technical condition of the plunger elements of the high pressure fuel pump by moving the injector needle is given.

1. Introduction
The analysis of the structure of the fleet of commercial vehicles showed that a significant part of it is equipped with automotive diesel engines. The most common in Russia and other countries is the KAMAZ-740 family of automotive diesel engines with split-type fuel equipment (FE) with a high-pressure multi-plunger fuel pump (HPFP) with different control systems [1]. FE of diesel engines is a complex high-tech engine assembly, the technical condition of which determines the environmental, economic and technological performance of the vehicle. Ensuring the operational state of diesel engine FE elements in operation is achieved by conducting routine diagnostics and technical inspection of the fuel system. The most important unit of diesel engine FE, which determines the operation and its characteristics as a whole, is injection pump [2]. The analysis of FE diesel engine malfunctions showed that one of the most loaded nodes is plunger elements [3] and their malfunctions (Figure 1).

Currently, the diesel FE service market offers a wide range of diagnostic equipment and techniques [4]. At the same time, the methods of diagnosing diesel engines that are widely used are performed when removed from the car and partially disassembled. The use of modern non-separable diagnostic methods for such FE will make possible to solve the problem of reducing the complexity of work and improving the quality of diagnosis [5]. Such diagnostic methods are based on the analysis of the output parameters of the diesel engine FE, functionally related to its structural parameters, however, they have not been studied enough. Available developments for the diagnosis of diesel engine FE can be divided into groups: diagnosis by the characteristics of the operation of FE; diagnosis by FE parameters; engine performance diagnostics [6].



a b c

Figure 1. Fault distribution of the diesel engine FE with a split injection system

The methods for diagnosing FE in the first group are not informative and largely depend on the experience and qualifications of the diagnostician. When applying brakeless and partial diagnostic methods, monitoring the state of FE elements is complicated by many factors that affect the reduction of power and economic parameters of a diesel engine, however, these methods allow evaluating the general technical condition of the engine. The gas-analytical method for diagnosing diesel engine FE has limited information due to the tightening of environmental standards for the content of harmful substances in exhaust gases. Diagnostic technique for FE operation parameters is an assessment of the technical condition and quality of adjustment of the main elements of the diesel fuel supply system. The most common is the method of diagnosing injection pump plunger elements of HPFP by the maximum pressure of fuel injection into a dry closed chamber or into the atmosphere, however, it is unsuitable for quantifying the hydraulic density of a plunger element due to the small range of measured pressure (up to 50 MPa), unavoidable fuel leaks through the injector needle. The most promising in our opinion is the method of diagnosing diesel engine FE at idle, without dismantling and disassembling it. That is, it is possible to refuse to use expensive and bulky stands

2. Materials and methods

To substantiate the method for diagnosing diesel engine FE by moving the injector needle, we have developed a nozzle with an optical sensor (patent No. 152362) [7] and a diagnostic device for diesel engine fuel equipment (patent application No. 2017123405). Diagnostic modes with parameters of the maximum fuel injection pressure and the injector needle movement were investigated and substantiated. The regularities of their change depending on the magnitude of the radial clearance of the pairing plunger-sleeve of HPFP are obtained. The metrological characteristics of the device made it possible to carry out measurements with an error of 0.11 %, which is sufficient for the information content of the diagnostic parameter. Testing the device on the engine confirmed the assumption about the effectiveness of this method of diagnosing diesel engine FE as a promising one.

When investigating the FE diagnostic mode and selecting the optimal crankshaft speed, it is necessary to determine the nature of the change in the injector needle movement and the fuel pressure in the cavity above the injection pump plunger, the fuel injection pressure and the effect of the HPFP on plunger-sleeve clearance on the values of the maximum fuel injection pressure and moving the injector needle.

The calculation was carried out for the high-pressure line of the operating diesel engine FE, in which a diagnostic nozzle, capable of detecting the movement of its valve, was installed.

3. Development of a mathematical model of the injection pump for estimating the pressure and movement of the injector needle

The mathematical model of a diesel engine FE using any method of solving the wave equation is functionally reduced to the Cauchy boundary value problem, which is based on the equation of the transient process of a working fluid in a high-pressure pipeline. The boundary conditions are made up of a system of differential equations characterizing the processes occurring in the high-pressure fuel pump — on the left and near the nozzle — on the right, for the description of which, the systems of equations of volume balance written for various FE cavities and the equations of motion of the nozzle and pump valves are used.

The process of changing the pressure in the cavity above the plunger depends on the change in the compressibility of diesel fuel, the flow rate when it expires through the gap between the plunger and the sleeve and the channel of the HPFP valve [8].

When calculating the compressibility coefficient α (m^2 / N), we used a tabular experimental dependence of the true and average values of the compressibility coefficients of diesel fuel [9] on the current pressure p (N / m^2), which is shown in Figure 2.

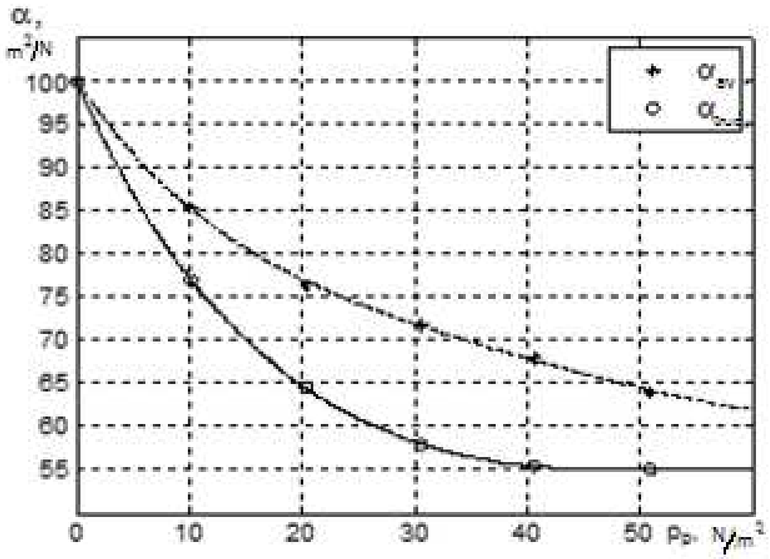


Figure 2. Dependence of the compressibility coefficient α of diesel fuel on the current pressure p: α_{tr} – is the true compressibility coefficient, α_{av} – is the average value of the compressibility coefficient

Based on tabular data, regression mathematical dependences are constructed on the current pressure over plunger p:

- values of the average compressibility coefficient of diesel fuel α_{av} in the form of a fourth-order polynomial:

$$a_{av} = a_1 + p^4 + a_2 * p^3 + a_3 * p^2 + a_4 * p + a_5 \quad (1)$$

where regression coefficients $a_1 = 5.1569 \cdot 10^{-6}$; $a_2 = -8.3799 \cdot 10^{-4}$; $a_3 = 0.0531$; $a_4 = -1.9148$; $a_5 = 100.06$;

- values of the true compressibility coefficient of diesel fuel α_{tr} p in the form of a fifth-order polynomial:

$$a_{tr} = a_1 * p^5 + a_2 * p^4 + a_3 * p^3 + a_4 * p^2 + a_5 * p + a_6 \quad (2)$$

where regression coefficients $a_1 = -7.5712 \cdot 10^{-8}$; $a_2 = 1.4734 \cdot 10^{-5}$; $a_3 = -0.0014$; $a_4 = -2.9671$; $a_5 = 0.0825$; $a_6 = 99.997$.

When calculating the fuel supply process in the diesel engine FE elements, the dependence of the diesel compressibility coefficient was used. The leakage from the cavity above the plunger between the sleeve and the plunger was considered and the volumetric flow equation of it had the following form [8]:

$$Q_{pH} = \frac{\pi \cdot \beta_e \cdot \Delta p^2 \cdot \delta_p^3 \cdot d_p \cdot \ln(c_\mu)}{12 p_0 \cdot \delta_p \cdot \mu_{t(p0)} l_p \cdot c_\mu^{\Delta p / p_0 - 1}} \mp \pi \cdot c_p \cdot \frac{\delta_p}{2}, \quad (3)$$

where β_e – correction factor for eccentricity of coupling (from 1.15 to 1.4); $\Delta p = p_p - p_0$ – differential pressure in the gap, (N / m^2); δ_p – annular clearance, (m); c_μ = 1.0025 – constant coefficient; $\mu_{t(p0)}$ – dynamic viscosity of fuel at atmospheric pressure p_0 = 0.1 MPa (kg / (s · m)); d_p, l_p – accordingly, the diameter and length of the sleeve, (m); c_p – HPFP plunger speed, (m / s).

Formula (3) takes into account the change in the coefficient of dynamic viscosity μ_t from pressure p_p over the plunger of HPFP:

$$\mu_t = \mu_{t(p0)} \cdot c_\mu^{\,p / p_0}, \quad (4)$$

The coefficients of dynamic μ and kinematic ν viscosity of diesel fuel are interconnected:

$$\nu = \frac{\mu_t}{\rho_t}, \quad (5)$$

where ρ_t – the density of diesel fuel, (kg / m^3).

The values of dynamic viscosity μ are usually calculated from the values of kinematic viscosity ν, obtained experimentally.

According to the tabular data of the experimental dependence of the kinematic viscosity of the fuel on temperature T_t, [9] a regression dependence having the following form was constructed:

$$\nu = a_1 + a_2 \cdot \log(T_T) + a_3 \cdot \log(T_T)^2, \quad (6)$$

where regression coefficients, a_1 = 0.76149543; a_2 = -0.36487040; a_3 = 0.043988593; T_T – the temperature of the diesel fuel, (^{0}C).

The regression dependence (6) of kinematic viscosity ν of the diesel fuel on its temperature T_T in the form of a graph is shown in Figure 3.

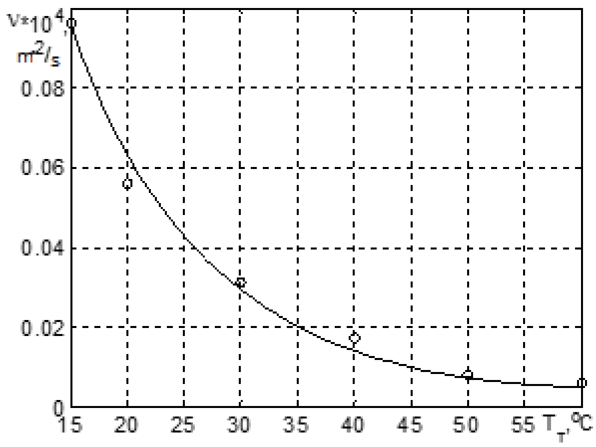


Figure 3. Dependence of kinematic viscosity ν of the diesel fuel on its temperature T_T: o - experimental data; ─ - simulation results

The volumetric flow rate of the fuel entering through the space of the injection valve through the fuel line to the nozzle was determined by the following formula:

$$Q_o = \mu_i \cdot f_i \cdot \sqrt{2\frac{|p_{i1}-p_{i2}|}{\rho_t}} \cdot sign(p_{i1} - p_{i2}), \tag{7}$$

where μ_i – injection valve space flow coefficient; f_i – injection valve space area, (m^2); p_{i1} – pressure of the flow entering the space, (N / m^2); p_{f2} – flow pressure at the outlet of the injection valve space, (N / m^2); ρ_t – diesel fuel density, (kg / m^3).

The dependence of the flow coefficient of the injection valve space in equation (7) was calculated by the regression dependence:

$$\mu_i = \sqrt{\frac{156}{Re_i^2} + 1} - \frac{12{,}5}{Re_i}, \tag{8}$$

$$Re_i = \frac{\upsilon_i \cdot d}{\nu}$$

where Re_i – Reynolds number, υ_i – fluid flow rate, d – hole diameter.

The equation for changing the flow coefficient of the shut-off hole of the plunger sleeve was adopted similar to equation (8), and the regression coefficients were taken in accordance with the literature.

These leaks characterize the volumetric flow rate of total fuel leaks from the cavity above the plunger and were taken into account in the mathematical model [10]:

$$\sum Q(p_p) = Q_{pH} + Q_f + Q_o(h_p), \tag{9}$$

where Q_{pH} – the volumetric flow rate of fuel leaks along the space between the sleeve and the plunger, (m^3 / s); Q_f – the volumetric flow rate of the fuel entering the nozzle, (m^3 / s); Q_o – volumetric fuel consumption through the shut-off hole in the plunger sleeve, (m^3 / s).

The mathematical model of the dynamics of pressure changes in the cavity above the plunger depending on the displacement of the plunger of the injection pump is presented in the form of a nonlinear differential equation of the first order with ordinary derivatives and has the following form [11]:

$$\alpha_t p_p \cdot V h_p \cdot \frac{dp}{dt} = f_p \cdot c_p(\varphi) - \sum Q(p_p), \tag{10}$$

$$c_p(\varphi) = 0, if\ 0 \geq h_p \geq h_{p(max)}$$

where $\alpha_t(p_p)$ – fuel compressibility factor, (m^2 / N); $V(h_p)$ – cavity volume above the plunger, (m^3); f_p – plunger area, (m^2); φ – angle of rotation of a camshaft of HPFP; $c_p(\varphi)$ – plunger speed, (m / s); ΣQ – volumetric flow rate of total leakage from the cavity above the plunger, (m^3 / s); $h_p = \int c_p(\varphi(t))dt$ – plunger movement, (m); p_p – pressure in the plunger pair, (N / m^2); t – time, (s); $h_{p(max)}$ – maximum displacement of the fuel injection plunger of HPFP, (m).

To calculate the displacement of the injector needle, we used a model of the dynamics of pressure in the cavity under the injector needle, depending on its displacement in the form of a first-order nonlinear differential equation with ordinary derivatives, which has the following form:

$$\alpha_t p_p \cdot V(x_i) \cdot \frac{dp_f}{dt} = Q_f - \sum Q(p_f), \tag{11}$$

where $\alpha_t(p_p)$ – fuel compressibility factor, (m^2 / N); p_f – pressure in the cavity under the injector needle, (N / m^2); $V(x_i)$ – cavity volume under the valve, (m^3); Q_f – fuel flow into the cavity under the injector needle, (m^3 / s); $\Sigma Q(p_f)$ – total leakage from the cavity under the injector needle, (m^3 / s); x_i – movement of the injector needle, (m); t – time, (s).

The total fuel leakage from the cavity under the injector needle is described by the volumetric flow rate formula [12]:

$$\sum Q(p_f) = Q_f + q_{i2}, \tag{12}$$

where Q_f – volumetric fuel consumption in the space between the guide sleeve and the injector needle, (m^3 / s); q_{i2} – volumetric fuel consumption through the nozzle openings due to pressure at the inlet and outlet of the nozzle openings, (m^3 / s).

The formula for leaks from the cavity under the injector needle between the guide sleeve and the valve is as follows:

$$Q_f = \frac{\pi \cdot \beta_e \cdot \Delta p^2 \cdot \delta^2 \cdot d_i \cdot \ln(c_\mu)}{12 p_0 \cdot \delta \cdot \mu_{t(p0)} \cdot l_i \cdot c_\mu{}^{\Delta p / p_0 - 1}} \mp \pi \cdot c_i \cdot d_i \cdot \frac{\delta_p}{2} \tag{13}$$

where β_e – eccentricity correction factor (from 1.15 to 1.4); $\Delta p = p_f - p_0$ – differential pressure in the space, (N / m^2); δ_i – annular clearance, (m); $c_\mu = 1.0025$ – constant coefficient; $\mu_{t(p0)}$ – dynamic viscosity of fuel at atmospheric pressure, (kg / (s·m); d_i, l_i – accordingly, the diameter and length of the space between the guide sleeve and the injector needle, (m); c_i – injector needle speed, (m / s).

The equation for the dependence of the volumetric flow rate q_{l2} of the fuel through the nozzle openings on the pressure change is described similarly to equation (7), and the dependence of the flow coefficient of the conical space between the seat and the valve is regressed (8).

The model of the dynamics of the movement of the injector needle in the form of a differential equation has the following form [13]:

$$m_i \cdot \frac{d^2 x_i}{dt^2} = F_{dri}(t) - F_{Ti} - F_{Cpi} = A_i \cdot p_t(t) - k_{Ti} \cdot \frac{dx_i}{dt} - c_{pi} \cdot (x_i + x_{0i}), \tag{14}$$

$$\frac{x_i}{dt} = 0, \quad \text{if} \quad 0 \geq X_i \geq h_{i(max)}$$

where F_{dri} – the force acting on the valve from pressure under the injector needle, (N); F_{Ti} – friction force of viscous resistance to the injector needle movement, (N); A_i – the injector needle work face, (m^2); $p_t(t)$ – pressure under the injector needle, (N / m^2); F_{Cpi} – the injector needle return force, (N); m_i – the mass of moving parts of the valve, (kg); k_{Ti} – coefficient of friction of the sealing part of the injector needle, (N · s / m); c_{pi} – the injector needle spring rate, (N / m); x_i – the injector needle moving, (m); x_{0i} – pre-deformation of the injector needle spring, (m); t – time, $h_{i(max)}$ – the injector needle maximum stroke, (m).

The inequality describes restrictions on the movement of the injector needle when it is planted on block stops. To calculate the fuel supply pressure values by the injection pump plunger pair when the injector needle moving, the inverse modeling problem was used, which consisted in determining the time-varying values of driving force F_{mov}, caused by the pressure under injector needle p_t for known time-varying movements of the injector needle mass m_i. The inverse solution of the original differential equation (14) has the following form:

$$F_{mov}(t) = A_i \cdot p_t(t) = m_i \cdot \frac{d^2 \cdot x_i}{dt^2} + k_{Ti} \cdot \frac{dx_i}{dt} + c_{pi} \cdot (x_i + x_{0i}), \tag{15}$$

then

$$p_t(t) = \frac{F_{dri}(t)}{A_i},$$

Driving force F_{mov} (t) is nothing more than the fuel supply pressure per unit area of the plunger of HPFP p_p without leakage Q_f through the plunger-sleeve coupling, which cause a change in the technical condition of the plunger pair of the HPFP. To create a mathematical model, leakages Q_f (3) were described by changing the fuel supply pressure over Δp_p plunger in empirical form:

$$\boldsymbol{\Delta p_p} = \frac{\sqrt[3]{\boldsymbol{d_p\,(q_p - k_p) a_p^2\; ln\, b_p^2}}}{\boldsymbol{a_p\;\, ln\, b_p}}, \tag{16}$$

where variables $a_p = \frac{\pi \cdot \beta_e \cdot \Delta p^2 \cdot \delta_p{}^3 \cdot d_p \cdot \ln(c_\mu)}{12 p_0 \cdot \delta_p \cdot l_p}$; $b_p = c_\mu$; $d_p = p_0$; $k_p = \pi \cdot c_p \cdot d_p \cdot \frac{\delta_p}{2}$; $q_p = Q_{pH}$.

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Then the mathematical model of the change in the pressure of the fuel supply when moving the injector needle:

$$p_p = \frac{m_i \cdot \frac{d^2 x_i}{dt^2} + k_{tri} \cdot \frac{dx_i}{dt} + c_{pi} \cdot (x_i + x_{0i})}{A_i} + \Delta p_p, \tag{17}$$

$$p_p = p_t + \Delta p_p$$

The analysis of the structure of differential equation (14) shows that the state of movement of the injector needle depends on the following factors: changes in diesel pressure under the working belt of the injector needle; fuel viscosity; the stiffness coefficient of the injector needle spring and the preliminary tightening (deformation) of the valve spring, limiting the movement of the valve and the radial clearance of the HPFP plunger-sleeve clearance. Therefore, to solve equation (17) of the inverse modeling problem and determine the dependence of the pressure change under the injector needle on the movement of the valve, it is necessary to control the diesel fuel viscosity, for example, by temperature.

The solution to this problem made it possible to confirm the possibility of applying the method of monitoring the technical condition of diesel engine FE elements for moving the injector needle and to compare with the control method for the maximum value of the injection pressure. For this, a mathematical fuel supply model was used in the Simulink visual graphical programming environment (Matlab) based on equations (1-14) with a diagnostic unit based on equation (17). Due to the fact that the pressure wave of the fuel under the injector needle is offset in time relative to the pressure wave in the HPFP, a time offset unit by $\boldsymbol{t} = \boldsymbol{\tau}$ was introduced for the restored pressure value under the injector needle to compare the values.

Calculation of delay time $\boldsymbol{\tau}$ of the pressure wave under the injector needle is:

$$\boldsymbol{\tau} = \boldsymbol{k_d} \cdot \frac{\boldsymbol{L_{tr}}}{\boldsymbol{a_v}}, \tag{18}$$

where $\boldsymbol{L_{tr}}$ **–** the length of the fuel line from the HPFP to the nozzle, (m); $\boldsymbol{a_v}$ - the speed of the fuel pressure wave in the pipeline, (m / s); $\boldsymbol{k_d}$ - wave deformation coefficient.

The parameters of the differential equations of mathematical models are selected according to the technical characteristics of KAMAZ-740.11 diesel engine [8]. The integration of the process model was carried out by the numerical method of computational mathematics - Runge-Kutta 4.5 orders of magnitude with a variable integration step and accuracy of 0.001, in the time interval 0 ... 0.04 s. The dependences of the change in the instantaneous values of pressure $\boldsymbol{p_p}$ in the HPFP, and pressure $\boldsymbol{p_f}$ restored by the simulation in time $\boldsymbol{t}$ are shown in Figure 4.

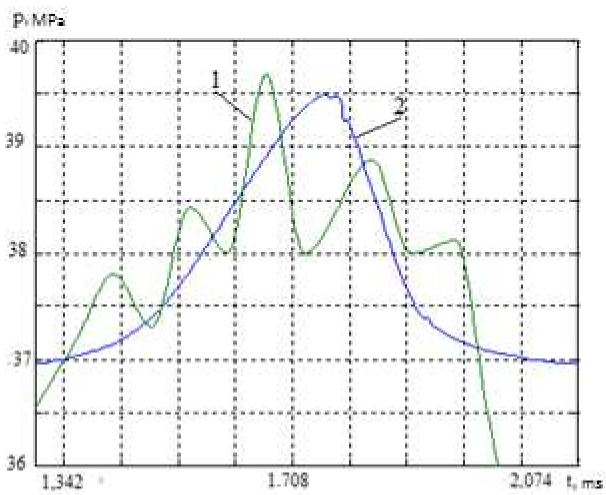

Figure 4. Dependences of the pressure change in the HPFP and the pressure restored by

simulation over time for a KAMAZ-740.11-240 diesel engine operating at rotation of $\boldsymbol{n}$ = 600 min^{-1} without load: (1) - data of mathematical interpolation; (2) - simulation results.

A comparative analysis of the dependences of the time variation of the fuel supply pressure by the HPFP plunger pair (1) and the pressure restored by simulation (2) (Figure 4) showed that the restoration of the fuel supply pressure by the plunger pair by moving the injector needle cannot record high-frequency pressure fluctuations due to the high mass of the last one. The difference in maximum pressure values is not more than 5 %.

Thus, the conducted analytical studies show that the use of moving the valve of the diesel nozzle as a diagnostic parameter will make possible to evaluate the technical condition of the diesel fuel equipment.

4. Effect of the radial clearance "plunger-sleeve" of the HPFP on the values of the maximum pressure of the fuel injection and the movement of the injector needle during diagnosis

To evaluate the effect of the radial clearance of the HPFP plunger pair on the movement of the injector needle and the maximum fuel supply pressure, we conducted laboratory studies of the KAMAZ-740.11-240 diesel engine operating at idle speed (n = 600 min-1), the results of which are given in the form of correspondences in Figure 5.

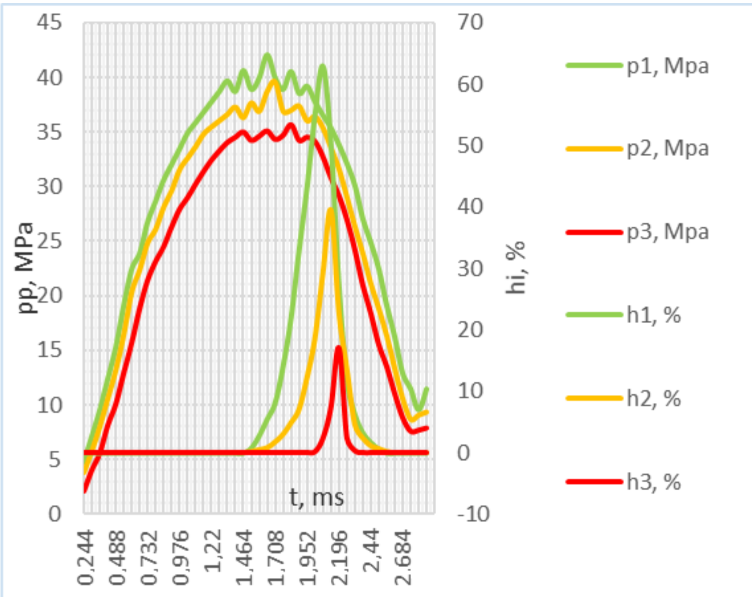

Figure 5. Time dependence of the injector needle movement, the fuel supply pressure of the plunger pair in time on the radial HPFP plunger-sleeve clearance of the KAMAZ-740.11-240 diesel engine: p1, h1 – respectively the time dependence of the injector needle movement, the fuel supply pressure of the plunger pair in time on the radial HPFP plunger-sleeve clearance of the KAMAZ-740.11-240 diesel engine ΔS = 4 μm; p2, h2 – respectively the time dependence of the change in pressure of the fuel supply by the plunger pair of the HPFP when the radial clearance of the plunger-sleeve ΔS = 7 μm; p3, h3 – respectively the time dependence of the change in pressure of the fuel supply by the plunger pair of the HPFP when the radial clearance of the plunger-sleeve ΔS = 10 μm.

To determine the movement of the injector needle, a device of our own design was used (Figure 6). The nozzle design assumes that the movement of valve 12 is rigidly connected with the movement of the nozzle rod with bar 5. The bar of the nozzle rod, when the injector needle is raised, narrows the cross section of the emitter beam, proportionally reducing the amount of light

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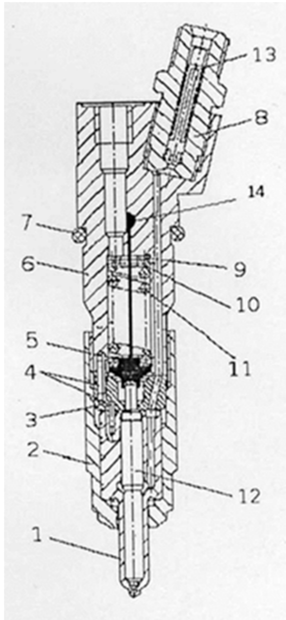
flux to the receiver, which causes a decrease in the voltage across the resistor of the electric circuit of the displacement sensor valve, the value of which is measured by an oscilloscope.

a)



b)

Figure 6. Diagnostic device for diesel fuel equipment: a - general view of the device; b – the nozzle of a diesel internal combustion engine (RF patent No. 152362); 1 – nozzle atomizer body; 2 - sprayer nut; 3 - spacer; 4 - guide pins; 5 - nozzle rod with a bar; 6 - nozzle body; 7 - sealing ring; 8 - nozzle nipple; 9, 10 - adjusting disks; 11 - nozzle spring; 12 - spray valve;13 - slotted filter; 14 - valve movement sensor

The sensor for moving the valve of nozzle 12 is made in the form of an optical pair. A BPW41N photodiode and a BIR-BM1331 infrared LED, respectively, were used as a receiver and an emitter. The electrical circuit of the injector needle displacement sensor contains resistors R1 to limit the maximum current of the LED and R2 to match the current of the oscilloscope channel.

The fuel supply pressure by the plunger pair was determined at the stand. The injector needle movement was evaluated by measuring the voltage across the photodiode resistance, which varied in proportion to the valve movement. For the convenience of the research, the movement was presented as a percentage, 100 % corresponded to a maximum stroke of 0.25 mm of the valve movement "with a stop at the stop block".

It turned out that an increase in the radial clearance of the HPFP plunger-sleeve ΔS to 10 μm leads to a decrease in the maximum value of the fuel supply pressure by 6.52 MPa (15.5 %) and a decrease in the maximum movement of the injector needle by 43%, which indicates high information content diagnostic parameter. An increase in the radial clearance of the plunger-sleeve of the HPFP also leads to a decrease in the speed of movement of the injector needle and the delay in its full opening by 0.122 ms.

On the curves of the change in the fuel supply pressure (Fig. 5), there are high-frequency harmonic oscillations occurring due to throttling of the fuel through the nozzle nipple openings during the movement of the injector needle. Their amplitude and oscillation period changed insignificantly when the size of the "plunger-sleeve" space is not higher than 7-8 μm. However, with an increase of the space to 10 μm, pressure fluctuations are significantly reduced, also showing a sharp decrease in the mobility of the injector needle. The movement of the injector needle also reflects the duration of the fuel injection. The longer the valve moves, the more fuel falls into the cylinder. The increase in the radial clearance of the HPFP plunger pair leads to a significant reduction in the cyclic fuel supply, which causes uneven operation of the engine cylinders and an increase in dynamic loads on engine parts.

The dependence of the movement of the injector needle on the radial clearance of the plunger pair of the HPFP is linear (Fig. 7), which is characterized by a high correlation coefficient (0.9) of analytical and experimental studies.

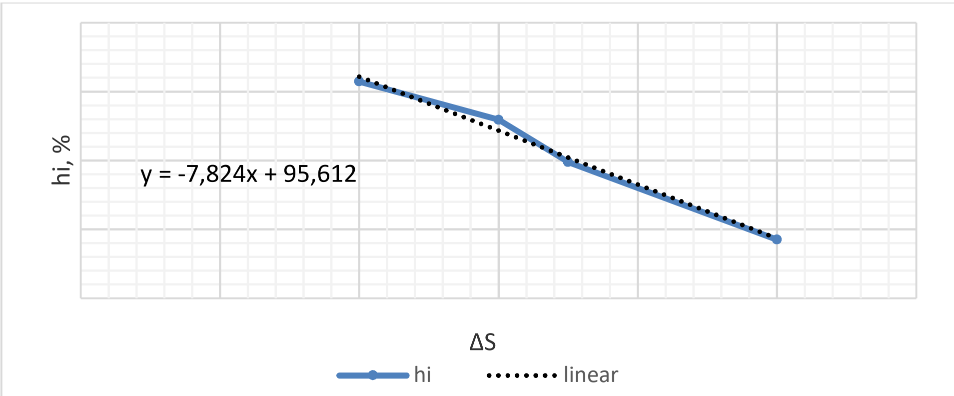


Figure 7. Dependence of the diagnostic parameter hi on structural ΔS: hi – experimental data, linear [hi] – linear regression model results

Operational studies of the injector needle displacement of the KAMAZ-740-11-240 diesel with a change in the fuel pressure and an increase in the radial clearance of the plunger pair of the HPFP are shown in Figure 8.

That is, disturbance of motor performance occurs when the value of the displacement of the injector needle is 32%, which corresponds to the maximum fuel-injection pressure of 38.1 MPa and maximum radial clearance of the plunger-sleeve of the HPFP of 8 μm when operating at idling speed.

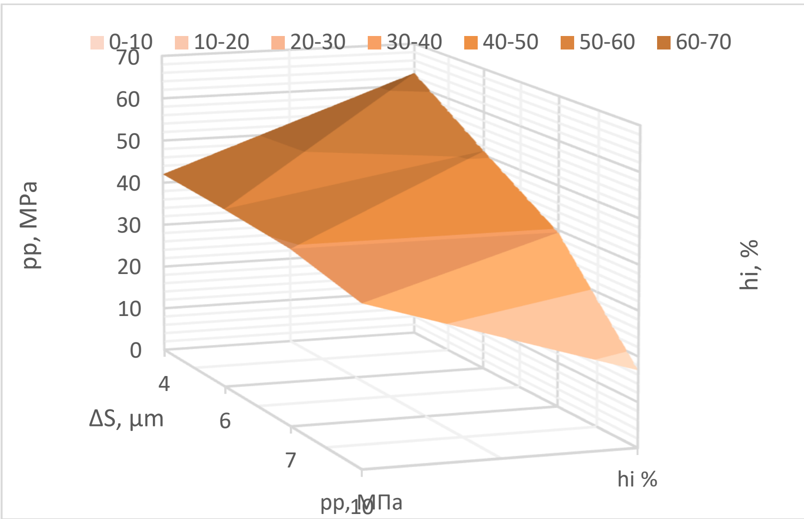


Figure 8. Dependence of the maximum values of the injector needle displacement (hi) and the fuel supply pressure (pp) on the radial clearance of the plunger-sleeve of the HPFP (ΔS)

5. Conclusion

Based on the analytical dependencies of the mathematical model (10), (14), (17) and the resultant surface of interference of parameters (Fig.7) as an example of the fuel equipment automotive diesel family KAMAZ 740.11-240 the following can be stated:

1. Increasing the radial clearance «plunger-bushing» of the HPFP of the KAMAZ ICE to 10 µm reduces the maximum fuel injection pressure by 6.52 MPa (15.5%), leads to a decrease in the maximum of injector needle displacement by 0.11 mm (43%) and decrease in the speed of injector needle displacement and delay of its full opening by 0.122 ms.

2. The error in the value of the fuel supply by the plunger pair of the HPFP for the injector needle displacement at idling speed does not exceed 5%.

3. Improper operation of the diesel KAMAZ 740.11-240 comes at a value of maximum injector needle displacement of 0.08 mm (32%), which corresponds to the maximum fuel injection pressure of 38.1 MPa and a maximum radial clearance plunger-bushing of the HPFP of 8 µm when operating at minimum idling speed.

Thus, diagnosing auto-tractor diesel fuel equipment with multi-plunger HPFP by injector needle displacement is informative and can be used in the diagnostics, as well as the method of assessment of technical condition of the elements of diesel fuel equipment at the maximum injection pressure. Timely detection of engine failures will reduce operating costs.

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