Metrological and methodological support for bench studies of diesel engines

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Abstract. The presence of exhaust gas emissions into the environment during the operation of diesel engines is an urgent environmental problem both for the state of the atmospheric air and for agricultural crops, since the toxic substances contained in the exhaust gas are absorbed by plants and accumulated in crop production. Thus, the environmental characteristics of diesel engines are now becoming the most important indicators in their operation. The problem of improving the performance of diesel engines is solving at the present stage in the process of their production by improving and carefully working out the design schemes and technological processes associated with the manufacture of their most important parts, and in the process of operation – by improving the methods and tools used for maintenance and repair. This approach can be considered justified due to the influence of a significant number of factors (design and operational) on the performance of diesel engines. As is known, the quality characteristics of the mixture formation and the nature of the fuel combustion process depend on the fuel supply parameters. Analysing the state of the problem of improving the performance of diesel engines, we can draw the following conclusion: the main reason for the deviation of the working cycle of engines from the set values is the deviation of the fuel supply parameters from the optimal values.

1. Introduction

Diesel engines failures are mainly caused normal wear (40%) and untimely and low-quality maintenance (36%). Considering the scientific opinion that the largest number of engine failures is associated with deviations in the operation of the fuel system (up to 70%), we can reasonably conclude that the fuel losses occupy a decisive place in the total losses [1-3].

The task of fuel equipment (FE) is to ensure the same conditions for the operation of all cylinders of a diesel engine. In that way, the FE must provide that the fuel supply to all cylinders is identical in terms of parameters such as: cycle flow, fuel injection start angle, injection characteristics [4-6]. The non-identify of the specified parameters for all cylinders of the diesel engine is the reason for the different types of the leakage and naturally for the differences in the operation indicators for various diesel engine cylinders. For example, during the operation of diesel engines, which are forced by the excess-air coefficient, this circumstance is the reason for the deterioration of fuel economy and intensive wear of the cylinder-piston group (CPG) [7, 8].

In connection with the above, the establishment of optimal parameters of the fuel system of low-pressure diesel engines on the process of repair and maintenance work is an urgent task.
2. Research results

Bench non-motor tests. Research to determine the effect of fuel temperature on the performance of the fuel pump (FP) was carried out using an installation developed on the basis of the KI-921M stand. The fuel temperature in the stand tank was maintained by means of a closed type electric heater and a potentiometer. Thermocouples of the CCT type were mounted in the FP cylinder [9-11].

The rotation speed of the high-pressure fuel pump (HPFP) shaft was set using a tachometer. Control – using a manual attached tachometer type SK-571. To control the backpressure in the fuel injection pipe, a special valve and a stand pressure gauge were used. The mass method (accuracy up to 0.5 g) was used to measure the performance of the FP.

When conducting studies to establish the impact of operational and design parameters on the performance of the FP, we were based on a plan that allowed us to create a multifactor model of productivity based on the nodal point method.

Experimental studies to determine the effect of pressure in the cavity filling of the HPFP to the fuel parameters were conducted in the nominal mode (main mode, in which the estimated stability parameters of the process fuel). Subject to the condition that components are unchanged, the value of the cyclic fuel supply is decisive for all other parameters of the injection process, which are its derivatives.

The amount of fuel supply is formed by the HPFP together with high-pressure fuel lines (HPFL) and nozzles. Taking this into account, before conducting the research, we pre-selected control, bench and working HPFL and injectors that would meet the requirements established by the TGD «System for standardization of diesel fuel equipment in repair and maintenance production» were pre-selected.

The values of the hydraulic characteristics (in particular, the effective flow area $\mu f$) of the discharge valves, HPFL valves were determined by spilling using diesel fuel having a temperature of 28...30°C at a pressure of 0.1 MPa. The control of the pressure was monitored using a pressure gauge (scale interval is 0.02 MPa). The spray nozzles were spilled at a pressure of 5.0 MPa.

The amount of fuel consumption during the pouring process was determined using a weight method with the weight of the sample equal to 500 g. The duration of filling the sample was determined using a stopwatch (scale interval is 0.20).

The value of the effective cross section was calculated using the expression:

$$\mu f = \frac{100G}{t\sqrt{2g\gamma \Delta P}}$$  

(1)

where $G$ - the weight of the sample, g; $t$ - the duration of the filling of the sample, s; $g$ - acceleration of gravity, cm/s²; $\gamma$ - specific fuel mass, g/cm³; $\Delta P$ - pressure drop, g/cm².

The frequency of rotation of the camshaft of the fuel pump and the cyclic flow rate were changed in accordance with the range that is characteristic of high-speed and load operation modes of the fuel system of the D-240 engine. Experimental studies were carried out using diesel fuel with a density of 0.822 g/cm³ and a kinematic viscosity of 1.9 cSt at 20°C and atmospheric pressure. At the entrance to the fuel pump, the fuel had a temperature of 40 °C. This temperature was kept constant throughout the experiment and was monitored using a mercury thermometer.

During the experiments, the value of the pumping pressure ($P_s$) was changed within the range of 0...1.1 MPa by a corresponding change in the tightening force and selection of the springs of the booster pump, drain valve and a change in pressure at the FP. The pressure at the inlet to the fuel pump was controlled using a laboratory pressure gauge (accuracy class 0.5 and measurement limits 0...2.5 MPa).

The study of the cumulative effect of the technical condition of the HPFP pump precision elements and pressure in the injection cavity on the main fuel supply parameters (cycle feed, feed coefficient, fuel injection lag angle and feed irregularity) performed the selection of
precision pairs (plunger pair and pressure valve). Plunger pairs and pressure valves were installed in the first section of the HPFP in combination, which is shown in table 1 below.

| № experiments | Plunger pair | Delivery valve |
|---------------|--------------|----------------|
|               | marking      | gap, \( S_n \), \( \mu m \) | marking      | gap, \( S_k \), \( \mu m \) |
| 1             | G – 10       | 2,0            | G – 07       | 20,            |
| 2             | G – 35       | 6,0            | G – 05       | 20,0           |
| 3             | G – 20       | 8,0            | G – 06       | 6,0            |
| 4             | G – 36       | 10,0           | G – 06       | 6,0            |

Adjustment to the nominal cyclic feed was carried out after each replacement of the precision elements of the HPFP. After that, the value of the geometric active stroke of the plunger was determined by pouring using a bench pump. The amount of pressure in the filling cavity of the HPFP was controlled using two pressure gauges: the first was installed at the inlet to the pump, the second was directly connected to the U-shaped channel.

The value of the feed coefficient was established by finding the ratio of the volumetric amount of fuel supplied by the nozzle during the injection process \( q_c \) to the volume described by the plunger during the active stroke \( q_{c,A} \):

\[
\eta_n = \frac{q_c}{q_{c,A}}. \tag{2}
\]

The value of the geometrical active stroke of the plunger was set by pouring the HPFP under a pressure that exceeded the opening pressure of the discharge valve. It should be noted that the geometrically active stroke is equal to the distance traveled by the plunger from the beginning of overlapping with the upper end of the inlet of the sleeve to the beginning of the opening of the bypass with the screw edge of the plunger.

Values of the actual and geometric active strokes of the plunger during the injection pump operation differ from each other, since the geometric active stroke is determined in the process of pouring on an idle pump. It is important that as a result of the fact that the plunger is moving at a significant speed, the process of compressing the fuel in the over-plunger space begins before the plunger will overlap the inlet of the sleeve.

Changing the technical condition of the injection line elements leads to a change in the amount of fuel supplied by the injector. Thus, the feed coefficient by sections characterizes the identity of their work.

The value of the injection delay angle is determined by the difference between the angles of the beginning of the feed \( \varphi_{n,p} \) and the injection \( \varphi_{n,v} \) of fuel:

\[
\varphi_z = \varphi_{n,p} - \varphi_{n,v}. \tag{3}
\]

Bench motor tests. Experimental studies were carried out using a control engine 4CH11/12.5, which had been run-in for 60 hours and providing stable indicators of torque, rotation speed and specific fuel consumption. The experiments were carried out using diesel fuel and diesel oil from the single delivery. The temperature of the coolant and oil was maintained in the range of 80...85°C.

The value of the injection advance angle was set in accordance with the operating instructions, according to which this angle for the 4CH11/12.5 engine is 24...9 in the angle of rotation of the crankshaft.
Installing HPFP with working fuel lines and injectors, the engine warmed up to stabilization time of consumption and full warm-up of the fuel pump. After that, we started to write down the speed characteristics of the 4CH11/12.5 engine with an interval of rotation frequencies of 600...2200 min⁻¹. In the course of the experiments, the indicators of effective power and torque, effective specific and hourly fuel consumption were determined.

Used equipment, instruments and measuring equipment. Hydraulic characteristics (in particular, the value of the effective cross-section) of the HPFP elements were determined using the stand KI-22201 and the prefix KI-15713.

Specially manufactured devices were used to determine the hydraulic characteristics of injection valves and spray nozzles. The effective flow section of the sprayer was set using a pressure gauge of 0...25 MPa (scale interval is 0.2 MPa), and pressure valves-a pressure gauge of 0...1.6 MPa (scale interval is 0.02 MPa).

Injection characteristics were established using a strain gauge sensor of the supply law TZP-1, calibration of which was carried out using a hydraulic press equipped with an exemplary pressure gauge, with increasing and decreasing pressure. In each case, 6...8 points were recorded. Ray deflection was recorded according to the indications of the oscilloscope screen and on the film. Each sensor was calibrated at 3 times the load and unload.

The fuel supply oscillography process was performed using the MPO-2 oscilloscope together with the UTS-1-VT-12 strain gauge station. Oscillograms were processed at a 10-fold increase.

To conduct non-motor bench tests, the KI-921M control bench was used. In the course of the experiments, the values of the productivity of the HPFP section were determined when working with nozzles, experimental sprayers were installed, followed by the determination of the central heating tube at 3 mm / cycle in the nominal mode (1100 min⁻¹) and the overload mode (850 min⁻¹). In addition, the stability of the parameters of the control HPFP was tested using an exemplary (model) nozzle complete with an exemplary (model) HPFP.

The beginning of the experiments was preceded by a thorough check of the completeness of the stand and the technical condition of the units, assemblies and devices. Pressure gauges, tachometers, and cycle counters were also checked.

It should be noted that in order to avoid errors in the process of determining the performance of the HPFP sections as a result of possible accumulation of fuel in the fuel collection chambers of the sensors, longitudinal grooves are cut in the guide bushings of the sensor cases of the stand.

The measuring range of the measuring beakers is 100 cm³, the scale interval is 1 cm³.

In addition, the rotational speed of the camshaft of the HPFP was controlled by the tachometer SC TIC-751.

Measures to test and adjust the control nozzles were performed using the KI-3333 device.

Power and economic indicators for the cylinders of the 4CH11/12.5 engine were installed using the KI-1363B electric brake stand, and the cylinders were turned off using a special device.

Methodology for assessing the error in establishing the value of the effective flow area. The value of the limiting value of the relative error of the experiment was set according to the expression:

\[
\frac{\Delta_f(a)}{a} = \pm d[f(u, v, w, ...)],
\]

where the set value \( a \) - a function of variables \( u, v, w \), that is, the limiting relative error of the value is the differential of its natural logarithm.

The values of the effective flow area will be equal to:

\[
\mu_f = \frac{c}{t\sqrt{2g\gamma\Delta P}},
\]

or
By differentiating expression (6) with respect to each variable \((G, \gamma, P, t)\), it is possible to establish the value of the relative error introduced specifically by each variable by the quantity:

\[
\delta_i = \frac{\Delta t}{t}; \quad \delta_\gamma = \frac{\Delta \gamma}{2\gamma}; \quad \delta_G = \frac{\Delta G}{G}; \quad \delta_P = \frac{\Delta P}{2P},
\]

where \(\delta_i\) - the maximum relative error of setting the duration of filling the sample (taking the value of the absolute error when measuring using a stopwatch equal to \(0.4\) s, with a measurement duration of \(115\) s we get \(0.35\)); \(\delta_\gamma\) – the maximum relative error that is introduced as a result of instability of the specific mass of the pouring liquid (taking the value of the specific mass when it is established using the hydrometer, and also taking into account possible fluctuations in temperature equal to \(\Delta \gamma = 0.0033\) g/cm\(^3\), we get \(\delta_\gamma = 0.25\%\)); \(\delta_G\) – maximum relative error which is introduced as a result of inaccurate setting of the mass of sample (taking the weight equal to \(500\) g, and the value of the absolute error =\(5\) g, get =\(1.0\%\)); - \(\delta_P\) - the maximum relative error which is introduced as a result of inaccuracy of the pressure measurement (when using a model pressure gauge (measurement limit up to \(250\) kg/cm\(^2\) (25 MPa)) the value of the absolute error =\((250-0.5)/100=1.25\) kg/cm\(^2\) (0.125 MPa), and when measuring the pressure of \(50\) kg/cm\(^2\) (5 MPa), we get =\(1.25\%\)).

Thus, based on the rule of addition of errors, the error in establishing the value of the effective cross section will be equal to:

\[
\delta_{\mu f} = 0.35\%+0.25\%+1.0\%+1.25\%=2.85\%.
\]

3. Conclusion

The proposed metrological and methodological support will ensure the establishment of the influence of: fuel temperature on the capacity of the FHP; operational and design parameters for the capacity of the FHP; pressure in the filling cavity to the fuel supply parameters for various technical conditions of the precision parts of the HPFP; uneven fuel supply for diesel performance. In addition, it is possible to evaluate the reliability of the FHP.

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