Decreasing the wear of precision pairs of fuel injection equipment in diesel engines

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Abstract. Reliable operation of a diesel engine largely depends on wearing degree of joints and assemblies of fuel injection equipment. The precision components of the high-pressure fuel pump and nozzles are most susceptible to wear. The following methods are used to reduce the wear of precision pairs of a high-pressure fuel pump: increasing the hardness of the mating parts, improving the filtering quality of the fuel and increasing the fuel lubricity. From the point of implementing easiness and obtaining the greatest effect, the most promising is improving the lubricity of motor fuel. It is possible to increase the lubricity of motor fuels by using liquid fuels of biological origin or by introducing vegetable-based fluid into commercial mineral diesel fuel, for example, cameline oil. The fuel kinematic viscosity effect on the wear of the precision pairs of the fuel injection equipment in diesel engines is theoretically substantiated. The results of bench tests of a high-pressure fuel pump for wear of plunger pairs (inner cylinder - bush bearing) during operation with mixed cameline-mineral fuel are presented.

1. Introduction

Tractors of various traction classes are used in the state agricultural enterprises. They are equipped with conventional diesel engines. At the same time, diesel engines of all modifications consume fuel of petroleum (mineral) origin, the lubricating properties of which are insufficient to ensure the performance of precision pairs of fuel injection equipment.

The studies have shown [1-5] that up to 70...80% of machine failures occur due to wear of friction units, and significant material and financial resources are spent annually on their repair.

The highest failure rates affect precision parts of fuel injection equipment (up to 54%). The most common failures of high-pressure fuel pumps (HPFP) are the wear of precision pairs (inner cylinder - bush bearing).

An analysis of worn-out precision pairs fuel injection equipment in diesel engines showed [6–12] that such pairs are mainly subjected to abrasive wear by particles contained in diesel fuel and penetrating through filter elements of fuel filters.

The smallest particles, falling into the gaps of precision pairs, wear them out and increase the initial gaps. As a result, the operation processes of fuel injection, mixture formation and combustion (pressure, duration, spray quality, jet range, etc.) change. At the same time, the engine does not develop the required power, fuel consumption increases and the content of harmful substances in the
exhaust gases increases. In addition, the lack of lubricity of commercial mineral (oil) diesel fuel contributes to the wear of fuel equipment parts.

The size of the abrasive grain and the surface hardness of the material, along with pressure and friction, play an important role in the wear process.

2. Materials and methods

Abrasive wear results in volumetric wear, i.e. the volume of material removed from the surface of friction pairs during the operation of the plunger pair by abrasive particles falling into the gap between the inner cylinder and the bush bearing:

$$V = V \cdot K \cdot N \cdot t, \ m^3,$$

where $V$ is the volume of material removed from the friction surface by one abrasive particle, $m^3$; $K$ is the number of abrasive particles falling into the gap in one operating cycle of the inner cylinder; $N$ is the number of inner cylinder cycles per hour, $1/h$; $t$ is the operating time of the plunger pair, h.

At the moment of fuel injection made by the plunger of the HPFP pump section, the created pressure acts on the bush bearing wall sides of the plunger pair. This leads to an increase in the annular gap between the inner circle and the bush bearing and to the penetration of abrasive particles of a larger diameter than the initial annular gap. Let us assume that the abrasive particle has is sphere-shaped, the bush bearing and inner cylinder move relative to the abrasive particle, which is embedded simultaneously in the bush bearing and inner cylinder of the plunger pair. Then the particle fixes itself on one of the surfaces (mainly on the bush bearing) and cuts a groove on the other friction surface (Figure 1).

The material volume removed during the scratch formation on the friction surface of one of the plunger pair details during one operating cycle of the inner cylinder

$$V = S \cdot h_{ef}, \ m^3$$

where $S$ is the area of the sphere segment of the embedded particle section, $m^2$; $h_{ef}$ – effective inner cylinder stroke (slip distance), m.

Sphere segment area of an embedded particle section [8]

$$S = 2R^2 \left[ 2 + 2 \sqrt{1 - \frac{2Rh - h}{R^2}} - \left( R - h \right) \sqrt{2Rh - h^2} \right], \ m^2,$$

where $R$ is the radius of the abrasive particle, m; $h$ is the penetration depth of the abrasive particles, m.

**Figure 1.** For determining the volume of the removed material during the formation of scratch on the friction surface: 1 – bush bearing; 2 – inner cylinder; 3 – abrasive particle; $h_{ef}$ – the path of the particle; $\delta$ is the initial annular gap between the bush bearing and the inner cylinder; $a$ is the diameter of the abrasive particle; $h$ is the penetration depth of the abrasive particles; $F$ is the specific pressure per unit area of the abrasive particles during compression of the bush bearing; $N$ is the specific pressure acting on the abrasive particle during the inner cylinder movement; $S$ – segment area of the embedded particle section.
Abrasive particle penetration depth

\[
h = \frac{\sqrt{F^2 + \left(\frac{F \cdot \nu_{\text{av}} + f \cdot F}{2 \cdot g}\right)^2}}{\pi \cdot R \cdot \sigma_{0.2}}, \text{ m}
\]  

where \( F \) is the specific pressure per unit area of an abrasive particle, N/m\(^2\); \( f \) is the friction coefficient; \( g \) is the gravity acceleration, m/s\(^2\); \( \nu_{\text{av}} \) – the average speed of the inner cylinder, m/s; \( \sigma_{0.2} \) – proof stress at 0.2 percent set, Pa.

The friction coefficient depends on the viscosity of the fuel and the load per unit abrasive particle:

\[
f = \eta \cdot \frac{\nu_{\text{av}}}{\delta_b \cdot F},
\]

where \( \eta \) is the dynamic viscosity coefficient, N·s/m\(^2\); \( \delta \) is the initial annular gap between the bush bearing and the inner cylinder, m.

With a decrease or increase in the viscosity of the fuel, its flow rate through the annular gap between the bush bearing and the inner cylinder changes, as well as the number of abrasive particles that fall into the gap with the fuel. Fuel consumption, m\(^3\)/s, through the annular gap between the bush bearing and the inner cylinder is determined using the Hagen-Poiseuille equation

\[
Q = \frac{\pi \cdot d_{bb} \cdot \delta^3 \cdot (P_a - P_c)}{12 \cdot \nu \cdot \rho_f \cdot l_{bb}}, \text{ m}^3/\text{s}
\]

The number of abrasive particles falling into the gap between the inner cylinder and the bush bearing in one operating cycle of the plunger

\[
K = \frac{\pi \cdot d_{bb} \cdot \delta^3 \cdot (P_a - P_c)}{12 \cdot \nu \cdot \rho_f \cdot l_{bb}} \cdot k \cdot \tau_{ef},
\]

where \( d_{bb} \) is the bearing bush diameter of the plunger pair, m; \( \delta \) is the annular gap between the inner cylinder and the bush bearing, m; \( P_a \) is the fuel pressure in the above-plunger space of the pump sections, Pa; \( P_c \) is the fuel pressure in the filling cavity of the HPFP, Pa; \( \nu \) is the kinematic viscosity of the fuel, m\(^2\)/s; \( \rho_f \) is fuel density, kg/m\(^3\); \( l_{bb} \) – gap length in the direction of leaks (bush bearing length of the plunger pair), m; \( k \) – the number of abrasive particles in 1 m\(^3\) of fuel; \( \tau_{ef} \) is effective stroke period of the plunger, s.

The length of the effective stroke of the plunger depends on its average speed and active time, i.e.

\[
h_{ef} = \frac{\nu_{\text{av}} \cdot \tau_{ef}}{\nu_{\text{av}} \cdot \tau_{ef}}, \text{ m}
\]

The number of strokes (operating cycles) of the plunger per hour depends on the hourly fuel consumption:

\[
N_s = \frac{4 \cdot G_h}{\pi \cdot d_{bb} \cdot h_{ef} \cdot \rho_{f} \cdot b}, \text{ s}
\]

where \( G_h \) is hourly fuel consumption, kg/h; \( b \) is the number of pump sections.

The formula for calculating volumetric wear as a result of abrasive wear during the plunger pair operation is obtained by substituting expressions (2), (7) and (9) into formula (1):
Analyzing the formula (10) shows that the volume wear depends on the abrasive particles that fall into the inner cylinder - bush bearing gap, the gap in the inner cylinder - bush bearing interface, the hourly fuel consumption, the number of abrasive particles, the viscosity and density of the fuel, the average speed of the inner cylinder, the effective stroke and HPFP operating time.

Thus, an increase in the kinematic viscosity of the fuel will reduce the wear of precision pairs of diesel engine fuel equipment under otherwise equal conditions.

It is possible to increase the lubricity of motor fuels by using vegetable-based liquid fuels or by introducing vegetable oil into commercial mineral diesel fuel (DF), for example, cameline oil [13-20].

To confirm the above, comparative accelerated tests of the high-pressure fuel pump for wear of plunger pairs (inner cylinder - bush bearing) were carried out while working on mixed cameline-mineral fuel with a ratio of components of 10% CamM + 90% DF; 20% CamM + 80% DF; 30% CamM + 70% DF; 40% CamM + 60% DF; 50% CamM + 50% DF, in laboratory conditions on a modernized bench for adjusting and testing fuel equipment KI-15711M-01-GOSNITI (Figure 2).

In order to create identical conditions for comparative testing of plunger pairs of pump sections when operating on various diesel fuel types, an experimental high-pressure fuel pump (hereinafter HPFP) was developed and manufactured. The experimental HPFP is based on a high-pressure fuel pump of the brand 902.1111008-20, in which the 4UTNM pump HPFP sections are installed instead of the standard pump sections. Moreover, individual filling cavities for each section are created to ensure the stroke of the plunger in accordance with the stroke of the 4UTNM pump plunger, cam profiles of the wobbler shaft are treated according to the cam profiles of the 4UTNM pump brand.

\[
I_p = S \cdot \frac{G_h \cdot \delta^3 \cdot (P_a - P_p) \cdot \nu_{av} \cdot k \cdot t \cdot m^3}{3 \cdot d_{h0} \cdot \rho \cdot \rho_f^2 \cdot I_{h0} \cdot b \cdot h_{ef}}
\]  

Figure 2. The modernized stand for testing and adjusting the fuel equipment KI-15711M-01: 1 – KI-15711M-01 stand; 2 – experimental high-pressure fuel pump for comparative testing of pump sections when working on various types of diesel fuel; 3 – electric pump 2112 ATS 453-453 (6 pieces); 4 – containers for DF (6 pieces); 5 – battery; 6 – personal computer; 7 – ACP ZET 210; 8 – fuel pressure sensors (12 pieces); 9 – synchronization sensor (standard position sensor of the drive shaft of the stand KI-15711M-01).
Comparative accelerated wear tests of the U16s15 plunger pairs were checked during operation (120 h) of the experimental HPFP using mineral and mixed camelina-mineral fuel. Mineral fuel was supplied to the first and second sections of the HPFP, mixed camelina-mineral fuel with different percentage of camelina oil was supplied to the remaining pairs of the pump sections in pairs (10% of CamM to the third and fourth sections, 20% of CamM to the fifth and sixth ones, 30% of the CamM to the seventh and the eighth, 40% of the CamM to the ninth and tenth, 50% of the CamM to the eleventh and twelfth). In order to reduce the test duration by 20 times in accordance with the OST 23.1.364-81 industry standard, an abrasive (specific surface area of 10,500 cm$^2$/g, particle size of 3...6 μm) in the amount of 150 g per 1 ton of fuel was introduced into the tested fuels, which corresponds to 127 g/m$^3$.

The studied U16s15 brand (4UTNM 1111410-01) plunger pairs were selected from the same batch manufactured at the Altaiskii zavod toplivnoi apparatу when completing the experimental HPFP. Plunger pairs were selected according to the same water density, which was determined on a KI-759 device.

The pump sections of the experimental HPFP are adjusted according to the normative and technical documentation for the D-243 diesel engine (4CH 11/12.5) for a nominal volumetric cyclic fuel supply of 72.5±0.5 cm$^3$ for 1000 cycles with a camshaft speed of 1100 rpm. Tests of the experimental HPFP were carried out at a nominal rotation frequency of the cam shaft of 1100 rpm for 120 hours, which corresponds to a tractor operating time of 2,400 hours.

3. Results and discussion
As a result of measuring the details of the plunger pairs of the HPFP pump sections, it was found that the greatest wear on all tested samples was observed from the upper end to the inlet. The inner cylinders wear out more than the bush bearing. This is due to the fact that the abrasive particles falling with the fuel into the gap of the plunger pair are mainly fixed on the bush bearing and cause scratches on the inner cylinder surface.

Analyzing the measurement results (Figure 3) of the details in the plunger pairs of the HPFP pump sections has shown that the greatest wear in precision pairs is observed when using commercial mineral diesel fuel and is 5 μm. The wear in precision pairs with an increased to 50% camelina oil content in mixed camelina-mineral fuel is 2.7 μm, i.e. reduced by 1.8 times.

The conducted research made it possible to establish theoretical correlations between the effect of the parameters of the HPFP working process, the geometric parameters of the precision pairs of fuel equipment and the physical and mechanical properties of the fuel on the wear of the inner cylinder - bush bearing unit.

![Figure 3](image)

Figure 3. The wear change in the inner cylinder - bush bearing unit depending on the content of camelina oil in mixed camelina-mineral fuel.
4. Conclusions
The presented calculations are confirmed by bench tests. As a result of carrying out the tests, it was established that the smallest wear of the inner cylinder-bush bearing unit is observed when using mixed cameline-mineral fuel with a ratio of components of 50% CamM + 50% DF.

The research results confirm the possibility of using cameline oil in mixed cameline-mineral fuel in order to reduce the wear of precision pairs of fuel injection equipment in diesel engines.

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