Prospects for the use of high-speed ESP for oil production

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Abstract. Increasing the ESP speed allows for a reduction in the linear dimensions of the units and the number of standard sizes. By reducing the length, wear resistant materials can be used in each pump stage. However, with an increase in the rotational speed to 10000 rpm, the head of the pumping stage increases significantly (> 35m) and the processes of friction and wear intensify. Test benches for wear testing of pump stages and thrust bearings of hydraulic protectors and submersible electric motors with a rotation speed of up to 12000 rpm have been developed. It was revealed that the greatest change in mass is observed in the guide vane due to erosive wear, which varies depending on the rotational speed in a power-law dependence with an exponent of ~ 3.4. When testing a thrust bearing made of SiC with a rotational speed of 10800 rpm and an axial force of 6000N, the friction coefficient varied in the range 0.028 ... 0.008. Bearing made of hard alloy WC-8Co scored at 4.92MPa load. It is necessary to determine the optimal area of application of high-speed ESPs and the allowable wear values of the pump stages.

1. Introduction
Increasing the rotational speed of electric submersible pump (ESP) installations is a promising direction in the development of mechanized oil production ESP [1]. However, along with the obvious advantages, one should expect problems with reliability due to the intensification of tribological processes and wear of parts. The pump stage is an important component of the ESP pump, the reliable operation of which mainly determines the prospects for the use of high-speed installations. The existing calculation methods [2, 3] are based on the kinetic model of wear and are used only to assess the nature of the flow of liquid with abrasive in the channels of pump stages (PS). Therefore, comprehensive studies of tribological processes in the main tribo-couplings are necessary: thrust bearings, pump stages, radial couplings and seals.

2. General provisions
The head of the pumping stage with an increase in the speed of rotation increases in accordance with the dependence \( H_1=H_2\left(\frac{n_1}{n_2}\right)^n \), \( H_1, H_2 \) – head pressure, m; \( n_1, n_2 \) – rotation frequency, rpm.

Accordingly, the length of the pump section \( L_1=H_2\left(\frac{n_1}{n_2}\right)^2 \). The linear dimensions are reduced mainly for the pump rotor, table 1. For example, the impeller diameter is reduced by ~ 1.5 times. The guide device practically does not change the diametrical dimensions. The diametrical dimensions of radial seals and bearings are reduced in approximately the same proportion.
Table 1. Comparative dimensions of the pump stage of conventional and high-speed pumps.

| Main characteristic        | Rotation frequency, rpm |
|----------------------------|-------------------------|
|                            | 3000       | 10000       |
| The weight impeller, kg    | 0.154      | 0.064       |
| The weight GD, kg          | 0.275      | 0.442       |
| Shaft diameter, m          | 0.017      | 0.012       |
| $D_{\text{MAX}}$ impeller, m | 0.071    | 0.055       |
| The height PS, m           | 0.025      | 0.023       |
| The distance between the bearings of the module, m | 0.5…1.5 | 0.230 |

Changing the size and reducing the number of parts allows you to use highly wear-resistant materials in friction pairs.

On the other hand, a significant increase in the rotation frequency and head pressure of the pump stage increases the requirements for the imbalance of rotating parts and gaps due to wear of the radial and axial seals. Increasing the seal gaps increases the flow of fluid through them, which can lead to parametric failure.

Differences are also observed in the dynamics of the compared installations. If a conventional pump, for example, size 5-50, has 10-11 natural oscillation frequencies in the range of 10-13 Hz, then a high-speed pump operates in the «subcritical» region. Its natural frequency corresponds to a rotation frequency of $\sim 208...218$ Hz. The nominal rotation frequency of the pump shaft is 10000 rpm ($\sim 167$Hz).

The main types of wear that occur in high-speed pumps are abrasive wear of seal, erosion wear of the flow part, fretting and shaft corrosion. High load thrust bearing parts are prone to cracking. In the flow path cavitation is observed. This is a distinctive feature of high-speed pumps.

3. The results of the tests of pumping stages

To investigate the effect of the rotation frequency on the wear of the pump stages tested serial pump stages of sizes 5-50 and 5-80 made of powder materials in the frequency range 2970…6000 rpm on a stand with a closed loop of liquid movement and with a one-time filling of abrasive. Quartz sand of the F100 dimension with a concentration of 18 г·л$^{-1}$ was used as an abrasive. The liquid flow rate was 2.0 м$^3$·д$ay^{-1}$. Experience time – 360 min.

The change in the mass of pump stage parts during the experiment illustrates a characteristic feature, figure 1. Thus, the wear rate of the impeller is significantly less than the wear rate of the guide device. In the first case, we observe a linear dependence of the wear rate on the rotation frequency. In the second case, there is a power dependence with an exponent of $\sim 3.4$. When the rotation frequency is doubled, the wear rate of the impellers increased by 2.6 times, and the guide devices increased by 11 times for the studied speed range. When considering the influence of the speed of rotation on wear, it is necessary to keep in mind the flow rate of the liquid, which is functionally dependent on the speed of rotation.
Based on the analysis of worn surfaces, of measurement of the wear value, it can be concluded that the prevailing share in the total change in the mass of erosive wear is the flow part. Moreover, the guide device is subject to erosion wear to a greater extent than the impeller. It relates to the zones of inhibition of flow in which there are whirls of fluid with abrasive particles at high speed. The impeller has a flow inhibition zone on the input edge of the blade. In the channels of the impeller, the flow of liquid with abrasive particles is directed along the walls. Abrasive particles interact with the surface of the channels at small angles, which is accompanied by a low intensity of erosion wear. It should be borne in mind that the angles of collision of particles with the surface of the flow part change with increasing mass of the abrasive particle.

Tests of the high-speed pumping stage of the AKM pump at a speed of 9500 rpm were carried out on the developed test bench, figure 2, under similar conditions. Test time 5070 min with filling of new portions of abrasive after 600 min. It was found that the impeller also wears out less than the guide device. However, the wear rate of the guide device — $1.4 \times 10^{-4}$ g·min$^{-1}$ is significantly lower than that of serial PS when tested with a lower speed. This effect can be partially explained by more wear-resistant materials used in high-speed PS (alloy steels, titanium). In addition, the AKM pumping stage was tested at the right boundary of the pressure-flow characteristic.

4. Testing of movable joints PS

One of the main criteria for the performance of pump stages is the amount of wear of movable joints: radial and axial seals. This is especially important for high-pressure pumping stages.

According to the above method, pumping stages made of powder materials of sizes 5-50, 5-80 were tested under the same conditions, but with a changing Pump flow. Analyzing the results, table 2, you can see an increase in the wear rate of radial interfaces $V_{rad}$ and axial interfaces $V_{ax}$ with increasing rotation frequency.

| Frequency, rpm | Pump flow, m$^3$·hour$^{-1}$ | $V_{rad}$, $10^4$, mm·min$^{-1}$ | $V_{ax}$, $10^4$, mm·min$^{-1}$ | $V_{b}$, $10^5$, mm·min$^{-1}$ |
|---------------|-------------------------------|-------------------------------|-------------------------------|-------------------------------|
| 2950          | 1.38                          | 2.6                           | 5.6                           | 1.94                          |
| 2950          | 1.58                          | 5.6                           | 4.6                           | 2.9                           |
| 5705          | 1.92                          | 15                            | 14.3                          | 3.1                           |
| 9500$^a$      | 2.2                           | 0.31$^a$                      | -                             | 1.16                          |

$^a$ PS high speed pump
It is obvious that an increase in feed leads to an increase in the number of abrasive particles per unit time in the wear zone.

The change in the wear rate of pump stage parts is non-linear. The wear rate of the intermediate bearing sleeve $V_b$ on which the shaft end rests is practically proportional to the rotational speed.

Tests of the pump stage of the AKM pump on a high-speed stand, figure 2, showed higher wear resistance values of the intermediate bearing sleeve, table 2, than that of a standard pump stage.

At the same time, the wear resistance of the protective shaft bushings made of hard alloy WC-8Co of this pump stage is slightly lower $-3.1 \times 10^{-5}$ mm·min$^{-1}$ than that of the intermediate bearing bushings, but significantly higher than the wear resistance of the serial pump stage. This is due to the bushing’s materials - the pump stage AKM they are WC-8Co of solid alloy. An unambiguous conclusion in favor of the higher wear resistance of the moving couplings of a high-speed pump cannot be made only based on these results. The permissible wear value of the mates is of fundamental importance. Both the degradation of the pressure-flow characteristic and the dynamics of the installation depend on it. Apparently, for high-speed installations, this value does not exceed a few tenths of a millimeter. Serial installations can be operated with a wear value of several millimeters without the occurrence of parametric failure.

At the present stage of application of high-speed pumps AKM or other models, the limiting values of the wear of radial seals are unknown, at which a parametric failure of the installation occurs due to a decrease in pressure due to wear.

Figure 2. General view of the test bench for pumping stages:
1 – body, 2 – high-speed spindle, 3 – test chamber, 4 – sensors, 5 – mixer, (without a computer control and measurement system).

5. Thrust bearing testing
Structurally, each high-speed pump module has a thrust bearing and two radial bearings. Moreover, due to the reduction in radial dimensions, the thrust bearing has a small contact area compared to bearings of serial pumps, within $-5 \times 10^{-4}$ m$^2$. Hard alloy and silicon carbide are used as materials.

We tested a thrust bearing SiC - SiC with diametric dimensions: $D = 31$ mm, $d = 18$ mm on the developed stand [4], figure 3, under the conditions: MDPS oil, axial force up to 6000 N, rotation frequency 10800 rpm, temperature 80...85 °C.
As a result, the following results were obtained, figure 4. The friction coefficient varies from 0.028 to 0.008 when the axial force changes from 1000 to 6000 N. The maximum pressure value is 12 MPa. Radial cracks have formed on the surface of the silicon carbide rings. The test process at high values of axial force was accompanied by intense heat release. A bearing with similar dimensions made of WC-8Co hard alloy at a pressure of 4.92 MPa suffered a scoring, figure 5.

![Figure 3. General view of the stand for testing thrust bearings of high-speed pumps: 1 – body, 2 – computer measurement system, 3 – torque generator, 4 – test chamber, 5 – loading system, 6 – axial force sensor.](image)

![Figure 4. Graph of the coefficient of friction and temperature, • – coefficient of friction, □ – temperature.](image)

Apparently, the ultimate pressure that a thrust bearing of this design can reliably withstand is <4 MPa.
Figure 5. Scoring of the bearing friction surface made of WC-8Co.

The conditions for using thrust bearings must be considered. Their operation takes place in a formation fluid with an abrasive. Therefore, the results obtained can be used for thrusting bearings of submersible electric motors. The thrust bearings of the pump modules require abrasive testing. It can be assumed that the obtained values of the axial force in this case will be even less.

6. Conclusions
1. The prevailing wear mechanisms of high-speed installations of ESP are: abrasive wear of movable joints and erosion (cavitation) wear of the flow path of the pump stage, as well as seizure and thermal scoring of thrust bearings.
2. There is no dependence of the change in pressure on the degree of wear of radial and axial seals, which does not allow determining the optimal field of application of high-speed pumps in terms of complicating factors.
3. Prospects for the use of high-speed electric submersible vane pump installations largely depend on a comprehensive solution to reliability problems.

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