International Journal of Advanced Science and Technology
Vol. 29, No. 9s, (2020), pp. 5804-5812

Justification Of Thrust Bearings Parameters For Increasing The Reliability Of Downhole Pumps

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Abstract

This article provides a study of factors affecting the reliability and durability of thrust bearings in electric centrifugal pumps of the ECW type. For these factors, a number of analytical equations are presented that can be used in determining the parameters for friction pairs of thrust bearings of turbine generators of hydroelectric stations. The methodology for testing to determine the operational roughness of friction pairs established after the end of the running-in process is described in detail. During of the experiments, values of roughness parameters differing from the initial values were obtained. The initial production of surfaces of friction pairs with parameters closer to operational gives a faster transition to equilibrium roughness, a decrease in initial wear and an increase in the reliability of friction pairs. It is recommended to produce sliding friction pairs of downhole pumps with roughness parameters Ra=0.31-0.37 μm.

***Key words**: thrust bearing pair, thrust bearing, equilibrium roughness, bearing capacity, downhole pumps.*

1. Introduction

Of particular importance in the world is the development of modern and energy-saving methods for the efficient and rational use of water resources, including the widely using of friction pairs, which increase the efficiency of pumping units. In this regard, improving the reliability and developing the operational parameters of pumping units, in particular for downhole pump devices, is one of the urgent tasks [1].

Researches are being carried out to improve the parameters of thrust bearings of electric centrifugal water (ECW) pumps in the world [2]. In this direction, special attention deserves research to increase the accuracy of the manufacture of pump parts, efficiency and working life of well pump units, to reduce the wear of friction elements [3]. It is considered important to improve the geometric parameters of thrust bearings for pumps of ECW type, in particular [4]. In the pumping park, a special place is occupied by submersible electric pumps, which are the mass production of the pump structure [5].

The demand for pumps of this type is constantly growing. The analysis of the main types of failures of parts and submersible electric pump assemblies was carried out on the basis of an examination of the repair fund of pumps at Suvmash JSC (Tashkent), as well as by studying the repair documents available at these enterprises.

Well pumps operate in very difficult conditions - water, high pressure, vibration, abrasive particles, low temperatures, etc. For this reason, the design of the pumps and the materials used in them should have a large margin of safety [6]. Indirect signs of improper pump operation - pressure, increased noise, vibration, uneven water supply are reduced, power consumption has increased, etc.

Depending on the design of the pump and the type of draw-well, abrasive particles and natural fibers adversely affect the impellers or valves, clog the filter, and accumulate, reducing work efficiency. With a high content of sand in the water, frequent replacement may require a valve operating in the mode of increased wear [7, 8]. As a result of the examination of 100 pumps (20 pumps of each size), the data obtained on the replacement of worn parts and assemblies are shown in table 1.

`Table 1. The number of replaced or restored parts during repair, %

№	Name of parts and assemblies	Типоразмеры насосов				
		ECW 10-120-60	ECW 10-160-35	ECW 12-210-25	ECW 12-255-30	ECW 12-375-30
1.	Body parts	50	45	60	60	60
2.	Impellers	70	90	75	100	90
3.	Sliding bearings	60	80	80	80	80
4.	Seals	60	60	50	50	50

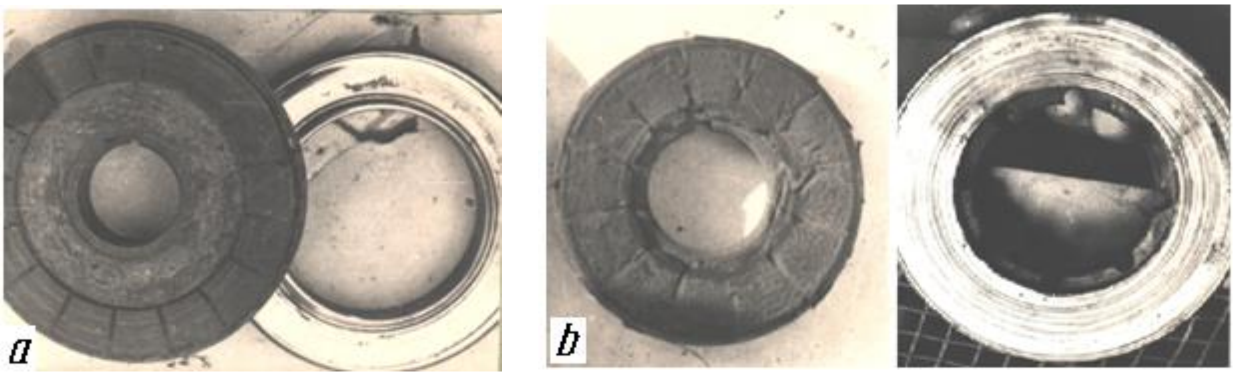

Fig. 1. Wear of the thrust bearing of the electric pump ESW 10-120-60 (a - initial stage, b - ultimate state)

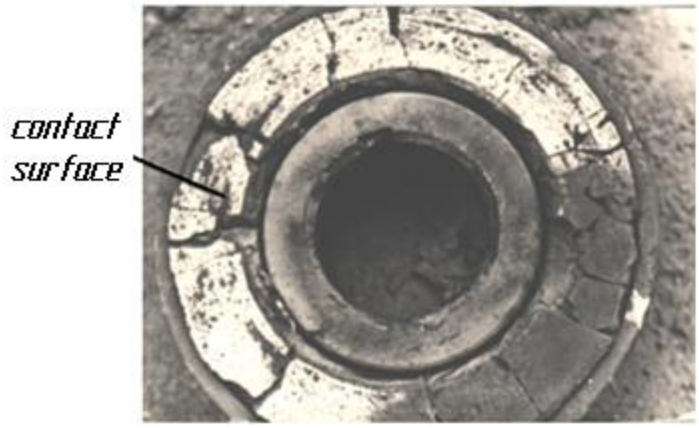

Fig. 2. Cracking of trust bearing.

Unlike radial bearings, the theoretical load of which is zero, thrust bearings work under more hard conditions and are therefore subject to more intensive wear [5].

Through the use of different methods, including various methods of draw-well prevention and the use of additional devices (impurity separators, containers, filters, etc.), an increase in the resource of ECW pumps is achieved. However, the use of highly wear-resistant hard alloys, ceramic materials, bearings significantly increases the cost of pump production. To save costs, design optimization in terms of the use of expensive materials and the rational arrangement of intermediate bearings is important.

Despite the achieved scientific results of the study of processes in sliding friction pairs of machines and mechanisms, many unresolved problems remain. The surface and geometric parameters of thrust plain bearings have not been adequately studied, including the friction processes of this type of bearings and the resulting hydrodynamic sliding modes on them, the patterns of oil film pressure distribution in bearing surfaces, changes in surface roughness parameters and dependencies associated with process factors. To solve the above problems, it is necessary to conduct research to analyze the processes occurring in sliding friction bearings, to improve the parameters of a thrust bearing, to calculate the bearing lifting capacity, to develop a methodology for conducting experimental studies.

2. Methodology

2.1. The main causes of wear and failure of the thrust bearing pair are the following:

1) Insufficient amount of crushing strength of the materials of the bearing pair;

2) Excess axial load on the bearing;

3) The presence of hydroabrasive friction;

4) The gyroscopic effect in the process of achieving revolutions of the pump shaft from a minimum to the operating value of rotation.

5) Incorrectly selected surface roughness parameters of the sliding friction pair;

6) The absence of a boundary separation between the heel and tilted pad unit.

There is a method of increasing the wear resistance of sliding friction pairs using sliding bearings made of ceramic-composite materials [9]. One of the ways to increase the wear resistance of ECW pumps is considered to be the “compression” mounting method, in which the pump stages are assembled in a position where the impellers are joined together without a gap.

2.2. Calculation of the force on the supporting surface of the ECW

To conduct tests to determine the equilibrium roughness for the supporting surfaces of pumps of the ECW type, which works as part of a hydraulic unit, it is necessary to set the required speed, rated power, and the required pressure of the pump system. It is also necessary to evaluate the axial hydraulic and weight load from the pump, which should be perceived by the thrust unit [10].

The axial load perceived by the upper support element is transmitted through the ring element to the levers, which in turn transfer it to the heel of the lower support element moving axially, pressing it to the thrust bearing.

Axial load to the tilted pad unit consist of three main components: pump weight G_p, weight of electrical motor G_{em} and axial hydraulic force to tilted pad P_{ax}^{hyd}.

$$P_{ax} = G_p + G_{em} + P_{ax}^{hyd}, \quad (1)$$

where G_p and G_{em} is found from reference data for pumps of type ECW;

Hydraulic axial force on the heel:

$$P_{ax}^{hyd} = S_{icp}\ H_{max}\ \rho_в(\kappa H), \quad (2)$$

where S_{icp} – area of internal cross section of pipe, which connected to pump,

$S_{icp}=\frac{\pi \cdot d_{in}^2}{4}$, in this case d_{in} – internal diameter of pipe, which connected to pump;

H_{max} – the maximum pressure value is determined from the technical data of the ECW pump;

$\rho_в$ – fluid density [11].

2.3. Calculation of bearing capacity of thrust bearing pair

To eliminate the harmful effects of the gyroscopic effect and nutation of the bearing heel, we divide it into 12 sectors, which should ensure uniform distribution of the load P and the lifting force F_f (Fig. 3).

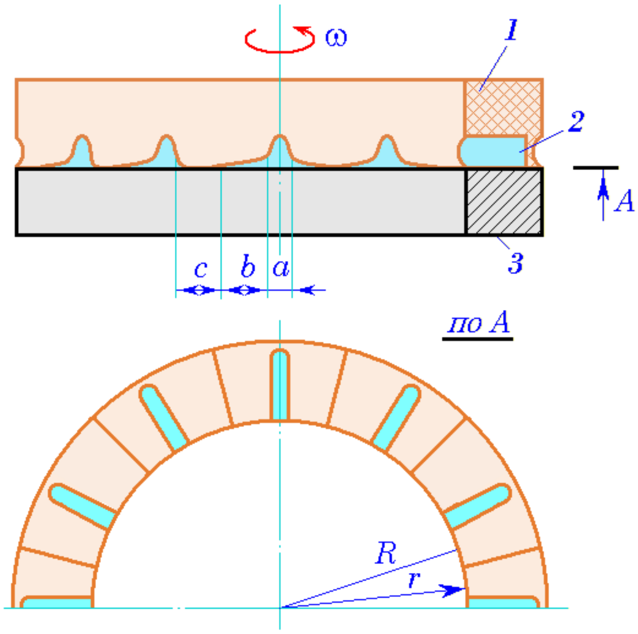

Fig. 3. The calculation model of the end pair of sliding friction of the pump unit: 1 - heel; 2 - water; 3 – tilted pad unit.

As the initial conditions, it's accepted that:
- the liquid (water) is Newtonian, i.e., incompressible, therefore we neglect the vector $\mathbf{v}_\rho$;
- the fluid motion regime between the heel and the tilted pad unit is laminar, and we neglect the influence of capillary forces;
- the influence of gravity and inertia is negligible compared to the forces due to the viscosity of the liquid;
- surface phenomena at the boundary of the fluid and heel layers do not violate the adhesion to the wall and do not affect the velocity distribution in the clearance;
- the thickness of the liquid layer h_{min} is equal to the value of the abrasive particles, a multiple of the total roughness of the friction pair.

A feature of the end pair of sliding friction, as shown in Fig. 3, is the linear velocity decrease as the radius of the heel changes from R to r.

That's why

$$v_\varphi = \frac{2\omega\rho}{\pi} = \frac{n\rho}{30} = \frac{n}{60}\int_r^R \rho\, d\rho \qquad (3)$$
,

To derive the lifting force equation of a hydrodynamic wedge in cylindrical coordinates, imagine a bearing (Fig. 3) in section AD, in order to highlight significant geometric parameters. For this purpose, we turn to Fig. 4, which shows a section of the heel sector.

The condition for the normal functioning of the bearing is the excess of the lifting force of the hydrodynamic wedge over the pressure force acting on the support, i.e., $\boldsymbol{F_f} > \boldsymbol{P}$.

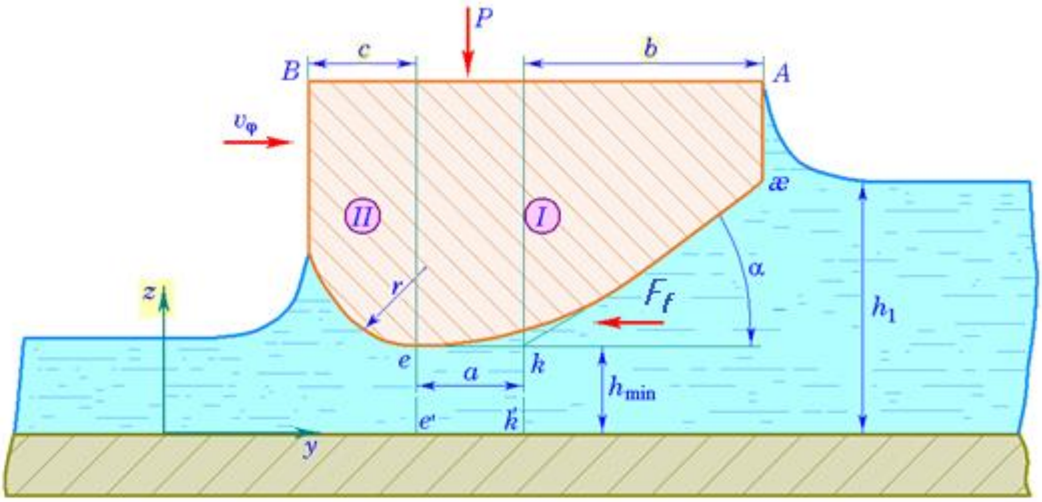

Fig. 4. Sectorial model of the heel lift using a hydraulic wedge.

Pressure at any point in the loaded fluid layer:

$$p = \frac{24 v_\varphi \mu}{h^2} \int_{\varphi}^{\varphi_{\min}} \int_{r}^{R} tg\,\alpha (R - r)\, d\varphi_c d\rho \,, \quad (4)$$

where φ_c - is the sectorial angle, varying from φ_0 (corresponding to the onset of pressure in the liquid layer, i.e., at point *æ*) to φ_{min} (corresponding to the minimum thickness of the liquid layer).
Summing up the projection of all the forces of the loaded liquid layer on the direction of the load, we obtain the bearing capacity of the bearing:

$$k = \frac{i 24 v_\varphi \mu \rho^2}{\pi (R^2 - r^2) h^2} \int_{\varphi}^{\varphi_{\min}} \int_{r}^{R} tg\,\alpha (R - r)\, d\varphi_c d\rho \,, \quad (5)$$

Here i - is the number of sectors of the bearing heel;

α - is the angle of elevation of the heel profile;

ρ - is the current radius of the heel, varying from R to r;

μ - dynamic coefficient of fluid viscosity.

To use the obtained mathematical models when designing bearings for ECW pumps, it is sufficient to have the following initial data:

n - is the rotational speed of the rotor of the electric motor, *rpm*;

p or Δp - pressure or pressure created by the pump, *MPa*;
R and r - are the heel radiuses, external and internal, *m*.

3. Results and Discussion

One of the important reasons of wear and failure of friction pairs is the incorrect running-in process. It is known that the quality of running after assembly affects the reliability of the machines. During running-in, technological defects are detected, friction surfaces are run-in at the micro and macro levels. It was experimentally revealed that in the process of friction, an equilibrium roughness is established that differs from the initial one. The value of the equilibrium roughness depends on the load applied to the friction unit, the sliding speed, and the properties of the materials of the friction pair.

The purpose of bench testing is to study the process of transition to operational roughness of pumps of type ECW products of the plant of JSC "Suvmash". For testing international standards are widely used [12].

Checking a new pump or a pump that has been repaired should be carried out at the bench for at least three hours, before the running-in process is over.

Information on the parameters measured during the test and the permissible measurement errors are given in the table below (Table 2).

Physical indicators of water such as density, viscosity, saturated vapor pressure in the environment are determined in the physicochemical laboratory.

Since the practical use of energy is determined by the difference in the readings of the counters for the set time interval (at least 2 hours), the instantaneous values of pressure, flow and temperature must be determined at least 5 times with an interval of 20-25 minutes (to find the arithmetic mean).

Before installing the pump on the bench and after the test, the thrust bearing assembly is dismantled and changes in roughness in the heel are checked using the Surface Tester Hommel Etamic profilometer-profilometer. The obtained data are entered in the corresponding test table (Table 3).

To study the change in the roughness parameters of friction pairs, the method of checking downhole pumps of the post-warranty period is used.

Table 2. Parameters measured during centrifugal pump tests

Measured parameters	Measuring devices
Pump inlet and outlet pressure	Primary pressure meters or pressure gauges of accuracy class no more than 1.0
Pump feed	Water metering unit flow meters or portable ultrasonic flow meters
Power consumption of the pump unit	Standard automated system unit of primary meters or mobile sets of K-506 type, accuracy class 0.5
Shaft speed	Tachometers or portable strobotachometers accuracy class 0.5
Pumped liquid temperature	Temperature meters or thermometers with a division value of at least 0.5 ° C

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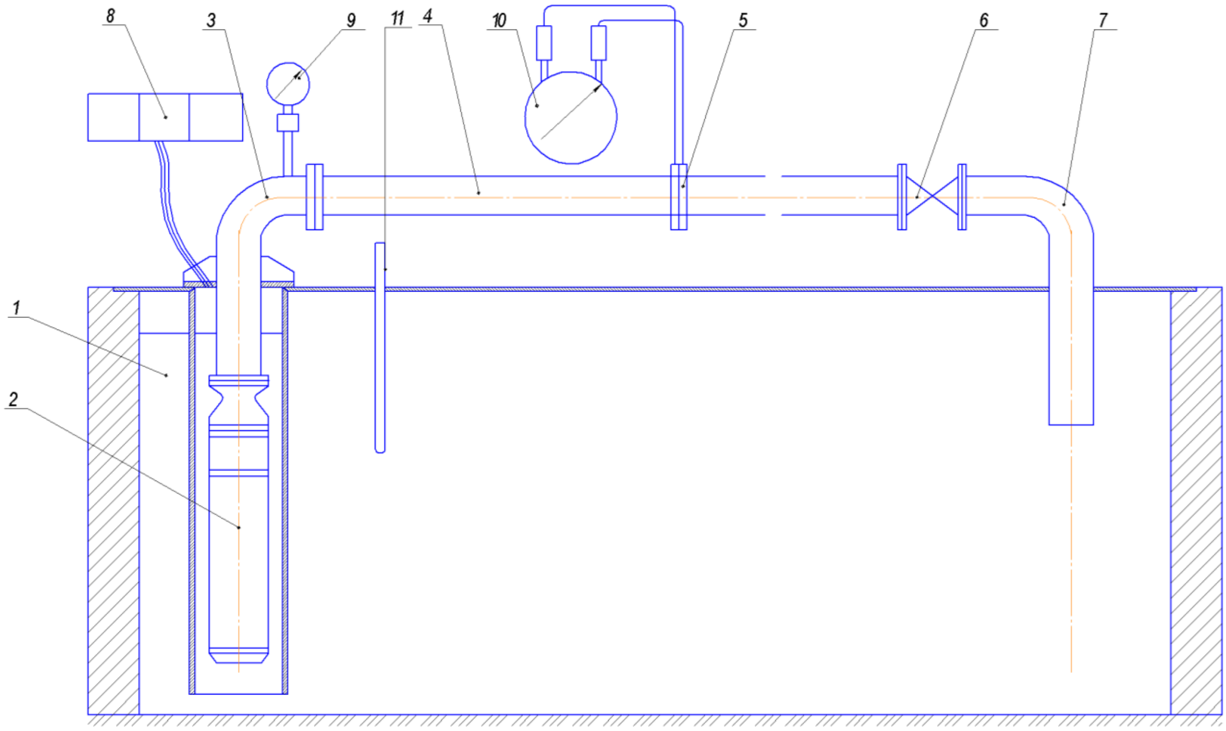

Fig. 5. Test bench for pumps of ECW type: 1 - capacity, 2 - tested unit, 3 - connecting pipe, 4 - dimensional section, 5 - stone diaphragm, 6 - valve, 7 - discharge pipe, 8 - set K-505 , 9 - pressure gauge, 10 - differential pressure gauge-flow meter, 11 - thermometer.

Fig. 6. The tilted pad unit and the heel of the ECW 10-160-30 pump after the test.

Measurements were taken before the experiment and after the experiment.

Table 3. The results of bench tests

№	Pump type	Heel roughness before testing, Ra, microns	The heel roughness after the test, Ra, microns	Feeding, Q, m^3/h	Power, N, kW	Curr ent, A	Pump speed, n, min^{-1}
1	ECW10-63-65	0,22	0,35	62	22	45	2950
2	ECW10-63-65	0,19	0,34	62	22	45	2950

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3	ECW10-63-65	0,22	0,35	61	22	45	2950
4	ECW10-63-65	0,24	0,34	62	22	45	2950
5	ECW10-63-65	0,23	0,31	62	22	45	2950
6	ECW10-63-65	0,22	0,33	62	22	45	2950
7	ECW10-120-55	0,18	0,31	118	27	55	2950
8	ECW10-120-55	0,22	0,35	117	27	55	2950
9	ECW10-120-55	0,23	0,34	118	27	55	2950
10	ECW10-120-55	0,24	0,37	118	27	55	2950
11	ECW10-120-55	0,23	0,33	118	27	55	2950
12	ECW10-120-55	0,20	0,34	118	27	55	2950
13	ECW12-160-30	0,22	0,31	158	22	47	2950
14	ECW12-160-30	0,23	0,32	159	22	47	2950
15	ECW12-160-30	0,19	0,37	158	22	47	2950
16	ECW12-160-30	0,20	0,31	158	22	47	2950
17	ECW12-160-30	0,21	0,33	159	22	47	2950
18	ECW12-160-30	0,22	0,34	158	22	47	2950

Conclusions

The obtained mathematical models of the bearing capacity of the heel of the end of plain bearing and hydrodynamic pressure at any point of the heel allows optimization of its geometric parameters and operating modes.

The change of the surface roughness of stationary parts does not depend on certain regularity, i.e., the classical approach to changing the surface roughness of soft materials is not appropriate.

Analyzing the data obtained during bench tests, it’s shown that during the transition to operational roughness, a roughness with a different value is established, sometimes differing from the initial (initial) value.

Bench tests to determine the operational roughness of the friction bearings of the ECW type draw-well pumps showed that during running-in the friction surface switches to operational roughness, the values of which are comparable to the roughness parameters indices of operational roughness.

During bench tests, experiments to determine the operational roughness of the ECW10-120-60 pumps showed that at the end of the running-in of the pumps at the IS-4 stand, the operational roughness reaches *0.33-0.36 μm*, which is comparable with the proposed indicators the roughness of the bearings of the friction of the pumps. Based on this, in the factory, it is possible to finish the surfaces of friction pairs not necessarily to $Ra = 0.19 - 0.22\ \mu m$, but closer to the operational one, $Ra = 0.31\text{-}0.37\ \mu m$, which reduces costs by technological processes and running-in time of pumps.

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