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# Study of Plain Bearing Destruction Processes Using Mathematical Modeling Methods

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***Abstract* — This paper contains the detailed study of processes and technical causes of the wear and destruction of plain bearings in modern rolling cutter bits using mathematical modeling methods. The studies performed and presented in the paper have demonstrated that at the start of friction, redistribution of specific load occurs with decrease from the inner diameter of the ring to the outer diameter, while the wear rate of small journal plain bearings is greater than the wear rate of big journal plain bearings. Redistribution of specific load on the contact surface of friction pairs is related to the wear of these components. As the result of the study, several methods for technological improvement of bearing mechanism in rolling cutter bits have been developed. These methods enable considerable reduction of plain bearing wear, which, in its turn, leads to the increase in the service life and performance efficiency of rolling cutter bits. Due to their simplicity and easy availability, the proposed methods can be used for any rolling cutter bits, which creates a considerable practical advantage**

***Keywords — plain bearings, wear rate, friction couples***

## I. INTRODUCTION

In modern rolling cutter bits journal as well as thrust plain bearings based solutions are finding more and more increasing use [1]. They are the most effective in oil-filled sealed bearing structures. Thrust plain bearings used in rolling cutter bits bearing structures provide a possibility to minimize axial play, enhance stability and service life of sealing devices and improve dimensions and bearing capacity of peripheral radial bearing and the whole bearing support by diminishing the dimensions of a retaining bearing [2].

Rolling cutter bits with sealed bearings are known for their operational instability showing as high durability dispersion [3]. First of all this is the case of rolling cutter bits with AU (based on two and more plain bearings with pressurized oil-filled bit bearings) type bearings. In this regard we studied technical reasons for the bearings durability loss. In order to do this we analyzed laboratory studies data of wear of rolling cutter bits operated field conditions.

The analyze of distribution of bit cutters over the range of wear value of radial bearings revealed that about 85% of larger plain bearings and 97% of smaller plain bearings have aggregate wear exceeding 0.30 mm if the bearing sealing is depressurized. There were no bearings with safe pressurized sealing and wear exceeding 0.50 mm. If we consider 97.5% of bit cutters as a limit wear value (the value equal to 2.5% significance value), thus the limit wear value is equal to 360 µm.

If the sealing is depressurized about 75% of bearings have multiply more significant wear than the maximum wear in conditions when the sealing is safe pressurized [4].

## II. METHODS OF IMPROVE THE AU-TYPE BEARINGS QUALITY

At the first glance this may be possible due to the long-term operation of the bearing after the sealing is depressurized. To check this supposition we calculated bit cutters durability and their cutting structures wear values. According to our calculations the bit cutters with the sealing pressurized till the end of drilling process generally operate longer or for the same time, and have lower wear of the structure than the cutter bits the sealing of which are depressurized during the trip.

Given the fact that in radial plain bearings the distribution of load in friction couple – shaft and hub - contact zone to a great extent depends on difference of bend radiuses of mating surfaces - during experiments with such bearings we revealed huge dispersion of reading variables, what made the decision making process longer and more complicated.

In this regard most of the experiments we conducted with flat samples – in the shape of flat grooves

### *A. Calculation of the working conditions of bearings*

During the detection of admissible bearing capacities and the resulting friction velocity the experiment method included stepwise increase of load every 30 minutes at preset rotation frequency of samples until the indications of sticking (scoring) of samples are revealed. We conducted experiments with samples of different materials in different lubrication environment: in order to provide planeness and parallel alignment of mating surfaces they were grinded and polished, and in certain cases even were bedding to each other. The rings were produced with different thickness [5].

Disregard of different methods of surfacing, in all experiments sticking and scouring started from the inboard edges and adjacent areas of rings as shown on the ring diagram (Figure 1). This fact is a non-repudiation evidence that in spite of exact mating of friction surfaces in an initial position, after the beginning of friction the specific load is transferred diminishing from the inner to the outer diameter of the ring. The contact friction surface specific load transfer is

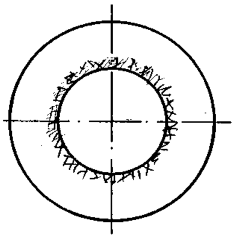

Fig.1: *Ring diagram connected to the wear out of element-members of friction couples.*

Because the wear value (Δ) of friction surfaces is proportional to the work applied through these surfaces, let us write the following equations for the first and second elements of the friction couple, eqn.(1) [6]:

$$\Delta_1 = K_1 \cdot q \cdot L ; \; \Delta_2 = K_2 \cdot q \cdot L , \qquad (1)$$

where: q – effective value of specific load at friction couple contact surface, L – sliding distance, $K_1$ and $K_2$ – proportionality factors.

For rotating friction couple, eqn.(2) (Figure 2)

$$L = V \cdot t = \varpi \cdot \rho \cdot t , \qquad (2)$$

where: ω - normalized angular velocity of friction couple rotation;
ρ – reference radius at the friction couple surface ($r<\rho<R$);
t – friction couple operation time.

By the virtue of the fact that the relative approach value of the friction couple elements is equal in any part of it, let's write, eqn.(3):

$$\Delta_1 + \Delta_2 = \Delta_{1,2} = (K_1 + K_2) q \cdot \omega \cdot \rho \cdot t = const \qquad (3)$$

If we divide both sides of this equation by it, we shall obtain, eqn.(4)

$$\gamma_{1,2} = \frac{\Delta_{1,2}}{t} = (K_1 + K_2) q \cdot \omega \cdot \rho = const \qquad (4)$$

where $\gamma_{1,2}$ – linear wear value for this couple at given values of q, ω and ρ.

Whereas for this friction couple at the given angular velocity the variables are q and ρ, from the expression (4) it follows that, at stabilized regime (e.g. for t >> 1 min.), eqn.(5)

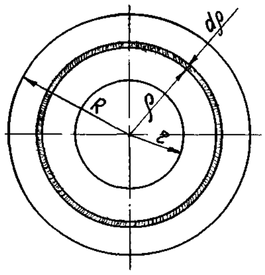

Fig. 2: Rotating friction couple

$$q \cdot \rho = N = const \qquad (5)$$

To determine aggregate thrust load influencing the couple by P, let's write, eqn.(6):

$$P = \int_r^R 2\pi \cdot \rho \cdot d\rho \cdot q = 2\pi \cdot N \int_r^R d\rho = 2\pi \cdot N (R-r) =$$
$$= 2\pi \cdot \rho \cdot q (R-r) \qquad (6)$$

From which, eqn.(7): $q = \frac{P}{r \cdot \pi (R-r)} \cdot \frac{1}{\rho}$ (7)

From (7) it follows that for rotating friction couple (when one element of the couple is in rotational motion only in relation to the other element) specific load q – is a variable value, changing from the center to the periphery areas of rotation in inverse proportion to the radius of friction circumference - $\rho$, which confirm the experimentally obtained data of anticipatory destruction of ring inner edges.

From (6) it follows that, eqn.(8)

$$N = \frac{P}{2\pi (R-r)} \qquad (8)$$

From (7) we can determine minimum and maximum values of specific load in a ring-generating friction couple, eqn.(9) and eqn.(10):

$$q_{\min} = \frac{P}{2\pi \cdot R(R-r)} , \qquad (9)$$

$$q_{\max} = \frac{P}{2\pi \cdot r(R-r)} \qquad (10)$$

Specific load values expressed in formulas (9) and (10) are the theoretic explanation of the beginning of process of destruction of ring shaped plain bearings at stepwise growing axial load applied.

Taking into consideration that the ring-shaped axial plain bearings are the critical elements of rolling cutter bits bearings and in other fields of machine-building industry, and the limiting factor of their bearing capacity and service life is for the great extent the specific load, it is rather interesting to analyze the influence of inner and outer diameter ratio of a ring-shaped bearing at preset values of axial load and outer diameter of a bearing to the $q_{max}$ value [7].

Let's rewrite the (10) formula as following, eqn.(11):

$$q_{max} = \frac{P}{2\pi \cdot R^2} \cdot \frac{1}{\left(1 - \frac{r}{R}\right) \cdot \frac{r}{R}} \qquad (11)$$

And introduce the notations: $\frac{r}{R} = x$ (0 < x < 1), thereupon, eqn.(12):

$$\varphi(x) = q_{max} \frac{2\pi \cdot R^2}{P} = \frac{1}{(1-x) \cdot x} \qquad (12)$$

Minimum peak value of $q_{\max}^{\Pi}$ we shall obtain when solve an equation $\varphi'(x) = 0$:

$$\frac{2x-1}{x^2(1-x)^2} = 0; \; x = \frac{1}{2}; \; r = \frac{R}{r};$$

$$q_{\max}^{\Pi} = \frac{2}{\pi} \cdot \frac{P}{R^2} = 0.6366 \frac{P}{R^2}$$

For example, at values of P = 4000 kgf and R = 3 cm, minimal specific peak load in bearing inner circumference zone at $r = \frac{R}{r} = 1.5cm$ shall be equal to: $q_{\max}^{\Pi} = 0.6366 \frac{4000}{32} = 282.93 \frac{kgf}{cm^2}$

Formula (12) describes concave curve (Figure 3) [8].

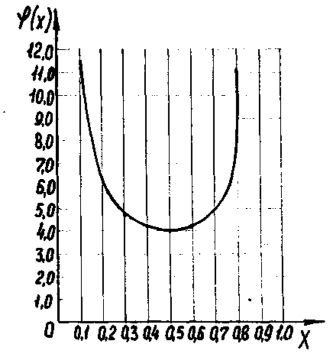


Fig.3: *Curve chart* $\varphi(x) = q_{\max} \frac{2\pi \cdot R^2}{P} = \frac{1}{(1-x) \cdot x}$

On the Figure 4 there is a curve of $q_{max}$ changing at the abovementioned preset values P and R and different values of r.

During design calculations of axial plain bearings with ring-shaped contact surface it is not acceptable to consider specific load value equal to $q = \frac{P}{F}$, where $F = \pi(D^2 - r^2)$ - total friction area surface. In our case we would obtain q = 188,64 kgf/cm$^2$, while in a real worn-in bearing, even with an optimal ratio of contact surface radiuses $r = \frac{R}{2}$, due to the transfer of specific load theoretically calculated minimum of its value at inner circumference of friction surface is

$$q_{\max}^{\Pi} = 282.93 \frac{kgf}{cm^2},$$

which is close to its critical value after what the scouring effect is inevitable what is shown at Figure 1.

In order to ensure against error during the calculations of frictional moment in axial plain bearings and the value of heat emission it is recommended to consider the effect of specific load transfer.

Elementary circumference friction moment (Figure 2), eqn.(13).

$$dM_{TP} = 2\pi \cdot \rho \cdot d\rho \cdot g \cdot f \cdot \rho, \qquad (13)$$

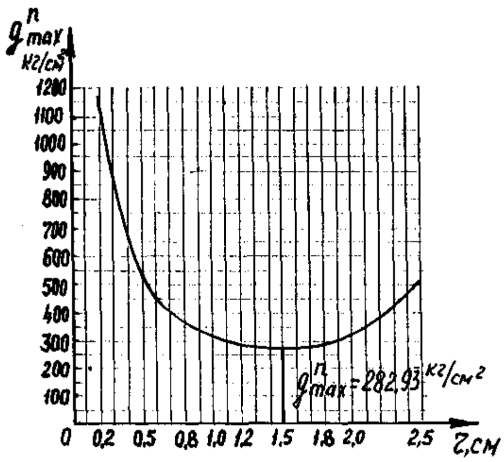


Fig. 4*: Dependence of $q_{max}$ at preset P and R values and different values of r*

where f – frictional factor.

Whereas for this couple $q \cdot \rho = N = const$, let's write, eqn.(14)

$$M_{TP} = 2\pi \cdot f \cdot N \int_r^R \rho d\rho = 2\pi \cdot f \cdot N \frac{R^2 - r^2}{2}, \qquad (14)$$

Whereas as it was shown above, eqn.(15)

$$P = 2\pi \cdot N(R - r), \qquad (15)$$

from which, eqn.(16)

$$N = \frac{P}{2\pi(R - r)}, \qquad (16)$$

then, eqn.(17)

$$M_{TP} = P \cdot f \frac{R + r}{2} \qquad (17)$$

For the same bearing, but not rolled off – with uniform specific load $q = \frac{P}{\pi(R^2 - r^2)} = const$ we would obtain, eqn.(18)

$$M_{TP} = 2\pi \cdot f \cdot q \int_r^R \rho^2 d\rho = \frac{2}{3} \rho \frac{R^3 - r^3}{R^2 - r^2} \qquad (18)$$

Plugging the preset values P = 4000 kgf; R = 3 cm; $r = \frac{R}{r} = 1.5cm$ and f = 0.1 in formulas (17) and (18) we shall obtain:

- for the bearing, operated in stabilized regime (formula 17)

$$M_{TP} = 4000 \cdot 0.1 \frac{3+1.5}{2} = 900.0 kg \cdot cm;$$

- for unrolled bearing (formula 18)

$$M_{TP} = \frac{2}{3} 4000 \cdot 0.1 \frac{3^3 - 1.5^3}{3^2 - 1.5^2} = 933.3 kg \cdot cm$$

With specific value transfer $q^{II}_{\max}$ of unrolled bearings is grown from 188.6 kgf/cm$^2$ to 282.93 kgf/cm$^2$, e.g. per 50%, while the friction moment diminished from 933.3 kg·cm to 900 kg·cm, e.g. per 3.6 % only.

From the calculations above it results that the end thrust bearing operates in the severest conditions because the trunnion axis is passed through its centerline. It results in premature failure of the bearing. That is why it is preferable to place the thrust bearing at the middle bearing face of a trunnion.

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## References

[1] Voronkov, B.D., *Dry friction bearings*, Mashinostroenie: Moscow, 1998.

[2] Atcherkhan, N.S., *Machine component parts*, Mashgiz: Moscow, 1993.

[3] Lelikov, O.P., *Fundamentals of calculation and design of parts and assemblies of machines*, Mashinostroenie: Moscow, 2002.

[4] Anuriev, V.I., (eds), *Reference of Design and Machine Building*, Mashinostroenie: Moscow, 2001.

[5] Ponomarev, S.D., et al., *Stress calculations in engineering manufacture*, Mashgiz: Moscow, 1956.

[6] Nichiporchik, S.N., Korzhentsevskiy, M.I., Kalachev, V.F., and other, *Machine parts in the examples and problems*, Visshaya shkola: Minsk, 1981.

[7] Open Joint stock company Scientific Production Association «Drilling equipment», 9/1 Letnikovskaya str, Moscow, Russia, http://www.vniibt.ru/.

[8] Federal State Autonomous Educational Institution of Higher Professional Education «Ural Federal University named after the first President of Russia B.N.Yeltsin» (UrFU). Institute of Mechanics and Machine Building, 19, Mira str., Ekaterinburg, Russia, http://urfu.ru/ru/.

[9] Gubkin Russian State Oil and Gas University, 65/1, Lininskiy avenue, Moscow, Russia, http://www.gubkin.ru/.

[10] Kazuhisa Miyoshi. Solid Lubricants and Coatings for Extreme Environments: State-of-the-Art Survey. NASA, 2007.

[11] Torbjorn A. Lembke. Induction Bearings. A Homopolar Concept for High Speed Machines. Electrical Machines and Power Electronics. Department of Electrical Engineering. Royal Institute of Technology.Stockholm, Sweden, 2003.

[12] Temis J.M., Temis M.J. Contribution of Bearing Structure in Gas Turbine Power Unit Rotor Dynamics // Proc. 3rd Int. Symp. on Stability Control of Rotating Machinery. - Cleveland, Ohio, USA, 2005.