Estimation of Hydrodynamic Parameters of Connecting Rod Plain Bearings of Pumping Plant of Powered Mine Support

M V Zernin, A V Mishin and N N Rybkin

Braynsk State Technical University, 7, Bulvar 50-letiya Oktyabrya, Bryansk, 241035, Russia

E-mail: zerninmv@mail.ru

Abstract. Expert evaluation of damage the connecting rod bearing shells UNP55-250 pump unit was performed. The finite elements method is used to calculate oil flow in the bearing clearance. Influence of operating defects on the parameters of hydrodynamics is also estimated. Calculations for various designs of connecting rod bearings are carried out and impact of various surface compliances is estimated.

1. Introduction

High-lift pumping plant UNP55-250 is an essential element of a powered mine support [1]. One of the most frequently damaged elements of the pumping plant is a connecting rod plain bearing. Several series of computational studies were carried out to improve the reliability of this unit. But first of all, an expert evaluation of the types of damage during the operation of connecting rod bearings was carried out. A calculation of the influence of the main types of shell defects on the parameters of the bearing hydrodynamics was made. The computational analysis of the bench test results of several variants of the bearing unit structural design, differing in size, as well as the presence of the oil groove system, was done. Possible changes of bearing surface compliance were also estimated.

2. Analysis of operating damages of bearing shells

The shells damaged in operation were examined and the nature and causes of their damage were revealed. The classifiers of damage to plain bearings [2, 3] and some others were used in this case.

A small part of the shells have catastrophic damages, overheating, melting and almost complete overlay extrusion and even plastic deformation of the steel base; the wear of steel base back surface in friction on the steel surface of the connecting rod and its cap (significant wear is observed on all specified steel surfaces). These damages resulted from the violation of bearing lubrication rate, which were eventually followed by seizure and adhesion of shaft and overlay surfaces, fixture key cutting, and rotation of the shells together with the shaft and friction of steel surfaces.

With intense overheating, which occurred with such a rigid friction regime of the shell, melting centres were born and the molten metal transfer appeared. With a sharp temperature rise thermal expansion coefficients of the overlay and the steel base become different. High tension stresses and plastic flow of metal occur in the layer. Sliding surfaces change their colour due to intense overheating.

For most shells (let’s call them normally operated) three types of damages, appeared in different degrees, were discovered: wear in boundary friction conditions (running-in or more significant);
abrasion wear scars - scratches on the surface by the particles, moving in the direction of shaft rotation, and in some cases, indentation, or smearing of the particles on the surface; fatigue damages. In addition, traces of manifestations of such damages as erosion and cavitation were observed.

3. The program of finite element analysis of bearing hydrodynamics

Bryansk State Technical University is developing [6 - 8] finite elements methods of calculating the hydrodynamics of dynamically loaded bearings on the basis of solving Reynolds two-dimensional equations. The estimate of the efficiency of the finite-difference liquid-flow models in cylindrical friction bearings performed in work [6] showed the preferability of the simplest triangular finite elements of the first order using the corresponding dependences, particularly presented by D. Booker and K. Hubner [9]. The algorithm of K. Murti for defining the boundaries of the oil wedge [10] is effective in the case of a fairly fine mesh of finite elements when it is initially assumed that all of the nodes are included in the zone of the oil wedge. The sequential exclusion of the outermost nodes from this zone during the iterative solution allows one to find the real shape of the boundary with an accuracy defined by the sizes of the finite elements.

The main algorithms and their implementation taking into account various factors that influence bearing hydrodynamics are described in the papers [6 - 7, 11 - 12]. Rospatent records [8] the first version of software system BBFEM (Bearing Builder Finite Element Method) that implements finite element algorithms for unstrained surfaces of the bearing. The software system has a modern interface [7] to prepare initial data and present calculation results. The hydrodynamic pressures, flow of the flowing liquid, the shaft trajectory in the bearing and other hydrodynamic parameters are calculated. Bearing capacity criteria such as minimum clearance, maximum pressure, power losses are determined based on these data.

4. Estimation of influence of surface defects on the parameters of hydrodynamics

Manufacturers of bearings give considerably different recommendations and criteria for replacing shells during periodic inspections. So, it is important to justify these criteria shell rejection.

If shaft working surfaces and sleeves are perfectly cylindrical, then nominal oil film thickness at any point (angular coordinate) for the certain eccentricity is determined by simple formulas. General modelling of all deviations of the surface from the nominal is setting additional amount of clearance (thickness of oil film) in the places of deviations. These additional clearances can be with plus or minus depending on the type of surface damage. In the places of deviations the oil film thickness is formed as a sum of nominal and additional thicknesses. Next, the oil layer evolvent discretization to the triangular finite elements is performed in an automatic mode. As a result, the thicknesses of the oil film within each triangular finite element are represented by bilinear interpolation of real thicknesses.

Influence of the main operating surface defects of shells on the parameters of hydrodynamics is estimated. Only those from the whole range of defects which, because of their relatively large size, can influence the parameters of bearing hydrodynamics were studied. Thus, fatigue cracks due to their small size do not a priori affect the parameters of hydrodynamics. Larger fatigue damages such as overlay particle spalling of arbitrary shape and depth can be significant.

Fatigue damages are taken into account in the finite element model by adding values equal to the depth of damage into the pivotal value of oil film thickness. Figure 1 shows the oil film scan with the spall area dimensions 12 mm by 15 mm. Fatigue spalling defect gives a local "dip" in the pressure profile (Figure 2). And significant influence on such hydrodynamic parameters of the bearing as maximum pressure and minimum clearance is not observed. But after appearance of these defects accelerated increase of their dimensions is possible. It results in reduction of the work area and deterioration of lubrication rate. It is also important that spalling overlay particles can get into the bearing clearance or contaminate fine oil filter.
Wear of surface areas. Worn surface areas are characterized by blending from wear-free part of the surface to the worn one. Modelling of surface wearing is as follows. An approximating formula for such a smooth surface can be constructed on the basis of wear measurements at several points. Additional clearances are defined in three stages: on the basis of a priori analysis it is identified what approximating surface should be built; least squares method is used to obtain parameters of approximating dependence; for all scan units within the worn area additional clearances are calculated. Figure 3 shows pressures in the oil film in the presence of a worn area. Pressure profile has become smoother, contact angle (of oil-film wedge) increases, and maximum pressure reduces to some extent. It turns out that the initial wear of surface areas does not affect the oil flow in the clearance. But with more intensive wear it is possible to significant clearance increase in the bearing, reduction of the contact angle, growth of maximum pressure and decrease of minimum clearance.

If there are some solid particles passed by the filters in the bearing clearance, they can have different effects on the surface of the shell. Particles, being smaller than the clearance, can hit the surface causing erosion damage. For particles with a size comparable to the clearance, different options of interaction with the surface are possible, which depend on the size of the particles (compared to the bearing clearance) and the ratio of material properties of the shaft and of the particle.

Scratches from abrasive action on the shell surface. If the particles are slightly larger than the clearance, then their abrasive action results in occurring a system of continuous or noncontinuous longitudinal scratches on the surface of the bearing shell. These scratches have no significant influence on hydrodynamic parameters (Figure 4).
Abrasive particle embedded in the overlay. Abrasive particle embedded in the overlay usually projects from the working surface and can charge the shaft. Pressure increasing is possible at a local area (Figure 5). But the key hydrodynamic parameters (pressure, minimum clearance) do not change significantly.

5. Analysis of benchmark test results of various unit models
Benchmark tests were carried out at joint-stock company Bryansk Machine-building Plant and they aimed at increasing durability of various units of pumping plant, including connecting-rod plain bearing. Several embodiments of this unit were offered: with grooves of various sizes and differently located on the bearing surface. Namely two variants of connecting-rod bearing were tested (Table 1), besides, the second variant had several versions: smooth, with grooves, with grooves and also condenser and grooving. Some bearing parameters were the same for all the embodiments: oil dynamic viscosity 0.0277 (Pa s), crank radius 34 (mm), connecting-rod length 245 (mm), angular velocity of crank rotation 43.04 (rad s\(^{-1}\)).

| Parameter                      | Type 1 | Type 2.1 | Type 2.2 | Type 2.3 |
|--------------------------------|--------|----------|----------|----------|
| Width (mm)                     | 60     | 70       | 70       | 70       |
| Shaft diameter (mm)            | 110    | 92.21    | 92.21    | 92.21    |
| Bush diameter (mm)             | 110.04 | 92.25    | 92.25    | 92.25    |
| Geometric deviations from circular cylindrical shape | No | No | Cross grooves with the size 6x0.7mm per 30\(^0\), 150\(^0\), 210\(^0\) and 330\(^0\), two condensers 24x0.7mm per 90\(^0\) and 270\(^0\), and grooving 10x1.4mm from -90\(^0\) to 90\(^0\) | Grooves 6x0.7mm per 30\(^0\), 150\(^0\), 210\(^0\) and 330\(^0\), two condensers 24x0.7mm per 90\(^0\) and 270\(^0\), and grooving 10x1.4mm from -90\(^0\) to 90\(^0\) |
| Minimum clearance, (mm)        | 0.007987 | 0.00981 | 0.007744 | 0.00527 |
| Maximum pressure (MPa)         | 16.023 | 16.2     | 18.561   | 27.2     |

To analyse the test results of the given embodiments some finite element calculations of bearing hydrodynamic parameters according to our program developed [7, 8] were carried out. Oil film of bearings is discretized by a mesh of triangular finite elements taking into account grooves, groovings and condensers. As an example Figure 6 shows a scan of oil film for type 2.3 discretized into finite elements. Figure 7 demonstrates pressure profile for this type.
The main results of hydrodynamic calculations are given in Table 1. It is evident that smooth variant (type 2.1) of modifying the standard bearing variant (type 1) reduces maximum pressure and increases minimum clearance to some extent. Thus, this modification is possible but it is hardly reasonable to change the design for achieving such improvement of parameters. Further modifications (with grooves and groovings) decrease bearing surface area. In consequence, maximum pressure increase and minimum clearance reduces.

Variant 2.3 is prohibitive, as minimum clearance 0.00525mm becomes close to the size of fine filter cells 0.005mm. In this case, particles passing through the filter will have an abrasive effect on the working surfaces. Because of decreasing the working surface area, pressure profile is characterised by very high maximum value and high pressure gradients on the groove edges in comparison with other embodiments (Figure 7). Stretching components of tension can appear in the overlay in these zones and as a consequence – fatigue damages.

On the whole, the results of designed bearing embodiments correspond to the results of benchmark test of these embodiments.

6. Influence of surface compliance on the parameters of hydrodynamics

The second version of program BBFEM realises the method of allowance of shaft and shell radial yield using hypothesis of “Winkler foundation”. A series of calculations of connecting-rod bearing types with overlay which has various yielding coefficients was carried out. Figure 8 shows that the more yielding is surface, the more are eccentricities: the path of shaft centre motion relative to the shell centre is more remote from the shaft centre. At the same time, minimum clearances increase subject to yielding. Figure 9 demonstrates pressure profiles at the section with maximum pressure. It is clear that the more yielding are surfaces, the less is maximum pressure.

7. Conclusion

The analysis of shell damages during operation shows that normal damages are fatigue ones, wear in marginal friction conditions and damages because of abrasive particle impact. A tool set [5, 6] was developed which helped to estimate the influence of damages on hydrodynamic parameters. This influence has been calculated and it shows that initial stages of these damages have inessential influence on bearing hydrodynamics. Hydrodynamic calculations of several embodiments of the bearing proved inexpediency of these changes. Also the impact of surface radial yield on hydrodynamic parameters has been calculated.

8. Acknowledgments

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**Figure 8.** Path of shaft centre motion relative to the shell centre with rigid surfaces (1) and yielding coefficients; 2 – 40.6·10^{-14} \text{ m}^3/\text{N}; 3 – 27.4·10^{-14} \text{ m}^3/\text{N}; 4 – 22.1·10^{-14} \text{ m}^3/\text{N}.

**Figure 9.** Two dimensional pressure profile in the section with maximum values and with rigid surfaces (1) and yielding coefficients; 2 – 40.6·10^{-14} \text{ m}^3/\text{N}; 3 – 27.4·10^{-14} \text{ m}^3/\text{N}; 4 – 22.1·10^{-14} \text{ m}^3/\text{N}.

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