Working in of roller ends and bars of cylindrical roller bearings at combined radial and axial loads

I M Klebanov¹, V V Murashkin², M I Kondratev², I E Adeyanov¹,³ and K A Polyakov¹

¹ Samara State Technical University, Samara, Russia
² OAO «EPK Samara», Samara, Russia
³ E-mail: adigorev@gmail.com

Abstract. The paper investigates the influence of the process of running-in the ends of rollers and sides of cylindrical roller bearings on the conditions of hydrodynamic contact of these parts: the formation of an oil film and friction. For this purpose, the results of finite element modeling of the stress-strain state of the bearing parts in contact: the inner ring, the side flange and a fragment of the axle of the wheelset, and the multi-mass simulation of the dynamics of the bearing operation are used. The influence of the yaw angle of the roller on the size and shape of the contact area on the side is investigated. The dependences for calculating the thickness of the oil film in the contact between the ends of the rollers and the flanges of the bearing rings are being refined, which now make it possible to take into account the misaligned position of these parts during their force interaction. The results are illustrated using the example of a roller bearing of standard size 232926.

1. Introduction

Cylindrical roller bearings with collars on the inner and outer rings take not only radial but also moderate axial loads in one or both directions. The axial load is transmitted through the contacts of the roller ends and the guide flanges of the rings.

The features of the contact interaction of the ends of the rollers and sides are considered in a number of theoretical and experimental works. The formation of a hydrodynamic oil film was first observed in [1]. Measurements of the thickness of this film, performed in [2], showed that it is noticeably smaller than it could be under similar conditions with a full-fledged hydrodynamic contact. Works [3-4] showed that due to poor conditions of film formation, liquid contact of the ends of the rollers and rings of cylindrical roller bearings does not occur in all cases. The thickness of the film is noticeably reduced with a decrease in the height of the sides of the rings and an increase in the yaw angles of the rollers - the rotation of the axis of the rollers relative to the axis of the ring in the tangential plane. At the same time, in most works, when calculating the thickness of the oil film, they proceed from the coaxial position of the rollers and rings [3-5]. The consequence of this is an overestimated value of the thickness of the oil layer. For example, in work [5], the calculation of the minimum thickness of the oil layer in the contact between the ends of the rollers and sides is proposed to be performed according to the formula of work [6]

\[ h = 3.17\left(\eta \nu \alpha \right)^{0.75} \alpha^{0.6} \text{eq}^{0.4} q^{0.15}. \]  

(1)
where \( V_\Sigma \) is the total speed of the bodies at the point of contact with the separator stopped, \( \eta_0 \) is the dynamic viscosity of the oil at the considered temperature and atmospheric pressure, \( q_H \) is the axial load on the roller divided by the length of the contact line, \( R_{eq} \) is the equivalent radius of curvature in contact, direction of the rolling speed, \( \alpha \) is the piezo coefficient of viscosity.

The equivalent radius of curvature is calculated from the radii of curvature of the roller and bevel of the side \( R_{req} \) and \( R_{feq} \)

\[
\frac{1}{R_{eq}} = \frac{1}{R_{req}} + \frac{1}{R_{feq}}.
\]

For these radii, the following formulas were used in [5]

\[
R_{req} = \frac{R_m}{\sin \gamma}, \quad R_{feq} = \frac{R_m - R_r}{\sin \gamma},
\]

where \( \gamma \) is the bevel angle of the side, \( R_m \) is the radius of the location of the centers of the rollers, \( R_r \) is the radius of the roller.

The running-in of the ends of the rollers and sides is an important factor affecting the conditions of their interaction during the operation of the bearing. For the first time, this was indicated in [7-8], where, based on the test data, it was shown that after running-in, the friction forces at the ends significantly decrease.

2. Theoretical background

The purpose of this work is to study the influence of the process of running-in the ends of rollers and sides of cylindrical roller bearings on the main characteristics of their contact interaction. The running-in process is illustrated by the example of a cylindrical roller bearing 232926.

![Figure 1](image1.png) Figure 1. Inner ring attachment flange with running-in marks.  

![Figure 2](image2.png) Figure 2. Roller end with running-in marks.

When inspecting the surfaces of the bearing flanges of bearings 232926 installed in the axle box unit of a freight car, two running-in strips are found: one with an exit to the chamfer of the outer diameter and the second approximately in the middle of the bevel (figure 1).

When examining the rollers, it was noticed that all their ends, facing the side flanges, have a running-in of the surface in the form of annular strips with the formation of a slight surface wear on them with an exit to the chamfer (figure 2).

Figures 1 and 2, it can be seen that during running-in, the ends of the rollers interact with the bevel of the attached sides in the middle of the sides and with an exit to their chamfer. This arrangement of
the contact area is due to the relatively large yaw angle of the rollers. The main reason for the yaw is a significant value of the working radial clearance.

To determine the conditions of contact interaction between the roller and the bead in the CAE ANSYS finite element analysis package, calculations of the stress-strain state of the assembly, consisting of an inner bearing ring, an added bead and a fragment of the wheelset axle, were performed. In figure 3 shows one of the results of these calculations, the contact pressure field in the case of a yaw angle of 15 arc minutes.

![Figure 3. Contact pressure field (Pa).](image)

Calculations have shown that with an increase in the yaw angle, the contact area on the bead shifts towards its outer diameter, and its length decreases. This is shown schematically in figure 4. The position of the center of the contact line is determined by the central angles of the roller and bead $\alpha$ and $\beta$. The vector $\mathbf{u}$ in figure 4 is the vector of the rolling speed at the center of the contact area.

![Figure 4. The location of the contact line of the end of the roller and the rim of the inner ring during the yaw of the roller.](image)

In the case of the coaxial position of the roller and the ring, the center of the contact area is located on the $OO_1$ line (figure 4), and the contact area itself does not come out on the surface of the roller in the area of conjugation of the end face and the roller chamfer [4, 5, 9]. When yawing, the center of the contact area is displaced from the $OO_1$ line, and the contact of the roller with the rim occurs along the above mating zone. Both of these circumstances affect the thickness of the oil layer.
To take into account the influence of the above factors, the dependences for calculating the thickness of the oil layer given in [5] were refined. The zone of conjugation of the end face and the chamfer of the roller has the shape of a torus, the radius of the generating circle of which is the radius of conjugation $R_{rr}$. At the point of contact with the end face of the side, this radius is the radius of curvature and the first dependence in equation (3) is replaced by the equality

$$R_{req} = R_{rr}.$$ (4)

As the total speed in equation (1), it is necessary to substitute the double rolling speed $u$, which is determined taking into account the non-parallelism of the speed vectors of the roller and the bead in the center of contact when the separator is “stopped”. All dimensions required for calculations are easily determined depending on the central angles $\alpha$ and $\beta$ (figure 4).

3. Results
When calculating the thickness of the oil layer according to equation (1), the actual length of the contact line must also be taken into account. The length of the contact line and its location were determined from the results of finite element modeling. And based on the results of modeling the dynamics of the bearing, the yaw angles of the rollers and the forces acting on the rollers were determined. For this, a numerical dynamic model of the bearing was constructed, similar to that considered in [10].

Figures 5 and 6 illustrate the dynamic behavior of the roller: the change in the yaw angle of the roller, as well as the forces on the ends of the roller from the side of the side flange of the inner ring and the opposite flange of the outer ring. These calculations were carried out at a wheelset axle speed of 761 rpm, a radial force on the bearing of 63098 N, an axial force of 21312 N, and a working radial clearance of 0.12 mm.

![Figure 5. Change of the yaw angle of the roller for one revolution of the separator.](image1)

![Figure 6. Graphs of changes in the forces at the ends of the roller for one revolution of the cage: 1 - from the side of the side flange, 2 - from the side of the opposite flange of the outer ring.](image2)

From the obtained results of modeling the dynamics of the bearing, it follows that the coaxial position of the roller and the ring is observed only in the absence of an axial force acting on the roller. The greatest values of the yaw angle are achieved, as a rule, when the maximum axial forces are applied to the roller. The main effect on the load on the rollers is the misalignment of the bearing rings. The maximum axial forces on the rollers in the absence of misalignment of the rings are within the range from 4500 N to 5400 N; at the same time, their values depend on the difference in length of the set of rollers. When the rings are skewed by two angular minutes, the range of axial forces acting on the rollers increases to 5900 N - 6400 N.
In the axial direction, besides the forces from the sides, the rollers are subjected to friction forces between the rollers and the raceways, caused by the cyclic axial displacement of the rollers. These forces are especially significant in the load zone, which leads to an additional force on the side flange (figure 6).

The main calculations of the thickness of the oil film between the ends of the rollers and the sides of the bearing 232926 were carried out at a wheelset axle speed of 761 rpm, an axial force on the roller of 6000 N, and a roller yaw angle of 15 arc minutes. In this case, the rolling speed at the center of the contact line is 3.25 m/s, and the sliding speed is 1.27 m/s. The following properties of the lubricant were taken: \( \eta_0 = 0.08 \text{ Pa}\cdot\text{s}; \ \eta = 2 \times 10^{-8} \text{ Pa}\cdot\text{s}^{-1}. \) According to the measurement data for unworked rollers, the radius of conjugation of the bevel of the roller and its end was taken equal to 0.01 mm, and for running-in rollers - 27 mm. The calculation according to the accepted dependences showed that with unworked surfaces the thickness of the oil layer is 0.07 \( \mu \text{m} \), and with running-in surfaces - 3.8 \( \mu \text{m} \).

The effect of the thickness of the oil layer on the nature of contact interaction and wear of parts is estimated using the oil film parameter

\[
\Lambda = \frac{h}{1.25\sqrt{R_{\alpha1}^2 + R_{\alpha2}^2}},
\]

where \( R_{\alpha1}, R_{\alpha2} \) are the arithmetic mean deviations of the absolute values of the deviations of the profiles of the contacting surfaces.

When calculating the parameter of the oil layer for unworked surfaces, the roughness \( R_{\alpha1} = 0.32 \mu \text{m} \) and \( R_{\alpha2} = 0.63 \mu \text{m} \) were taken. For the worn-in surfaces, two times lower values were taken. It turns out that in the first case, the parameter of the oil layer is 0.08, and in the second - 3.8.

The values of the parameter of the oil layer with unworked surfaces indicate that in this case the boundary friction mode is realized [9]. When the surfaces are run-in at nominal rotation speeds, the full fluid friction mode is established.

At boundary friction, the coefficient of friction can be determined by the Kragelsky formula [11]

\[
\mu = (-0.1 + 22.28s)e^{-181.46s} + 0.1,
\]

where \( s \) is the slip coefficient, which is the ratio of the slip speed to the rolling speed.

To calculate the coefficient of hydrodynamic friction depending on the properties of the lubricant, the thickness of the oil layer and other characteristics of the contact interaction, the Muraki-Kimura model was used [12].

The friction coefficient calculated by these models for the case of unworked end surfaces was 0.095, and for running in 0.027. Thus, as a result of running-in, the friction forces at the ends of the rollers decrease by about 3.5 times.

4. Conclusions

The obtained theoretical results confirm that the process of running-in the end surfaces of the rollers and sides of cylindrical roller bearings under combined radial and axial loading is largely determined by the misalignment of the rollers and rings during their force interaction. The yaw angle of the rollers when the rollers pass the zone of greatest load in the bearing determines both the localization of the contact areas on the surfaces of the rollers and ends, and the forces arising from their interaction, thereby affecting the intensity of the running-in process.

For a full assessment of the operating conditions of a cylindrical roller bearing under the action of a combined radial and axial load, it is necessary to use the data of modeling the dynamics of the bearing and the stress-strain state of the contacting regions of the rollers and beads, as well as the results of calculating the thickness of the hydrodynamic oil film that appears between them under the conditions of the misaligned position of the hydrodynamic oil film.
References

[1] Korrenn H 1970 ASME Journal of Lubrication Technology 92 129-37
[2] Gadallah N and Dalmaz G 1984 ASME Journal of Tribology 106 265-74
[3] Aramaki H, Cheng H S and Zhu D 1992 ASME Journal of Tribology 114 311-6
[4] Prisacaru Gh, Cretu Sp and Olaru D N 2001 Proc. of the 2nd World Tribology Congress 3-7 Vienna Austria
[5] Gajdamaka A V 2011 Roller Bearings of Axleboxes of Cars and Locomotives: Modeling and Improvement (Harkov: NTU HPI) 312
[6] Kodnir D S, Zhilnikov E P and Bajborodov Yu Z 1988 Elastohydrodynamic Calculation of Machine Parts (Moscow: Mechanical engineering) 160
[7] Brown S R and Poon S Y 1982 Proc. 8th Leeds-Lyon Symposium on Tribology 91-111
[8] Brown S R and Poon S Y 1983 ASLE Transactions 26(3) 317-24
[9] Harris T A and Kotzalas M N 2007 Advanced Concepts of Bearing Technology. Rolling Bearing Analysis (Fifth ed. Boca Raton FL: CRC Taylor & Francis) 360
[10] Klebanov I M, Murashkin V V, Polyakov K A and Danilchenko A I 2017 Mechanical Engineering Bulletin 11 3-9
[11] Kragelskii I V 1965 Friction and Wear (London: Butterworths) 346
[12] Muraki M and Kimura Y 1983 Journal of Japan Society of Lubrication Engineers 28(10) 753-60