SPECIFIC PROBLEMS OF HIGH SPEED BEARINGS

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Abstract: The requirements of very high performance (stress, rotational speed, stability to vibrations, etc.) for the compounds of mechanical engineering also involve an adequate fiability. The bearings can run at over 70 m/s peripheral speed of high fiability conditions (gas turbines in the aviation industry), with some high precessions and depressed frictions (gyroscopes, grinding broaches), with low/depressed noise and vibrations level. The increase of the bearings speed over the limits considered “ordinary” influences negatively the durability due to some kinematic, dynamic and lubrication significant changes. Major centrifugal forces appear, friction forces modify sensitively, starvation phenomenon occurs, and vibrations are established, too.

Keywords: cage, starvation phenomenon in ball bearings, lubricant's viscosity, rings

1. INTRODUCTION

The main cause of the bearings running at normal speed is the spoiling due to contact fatigue. In the case of the high-speed bearings, due to the increase of the friction processes and of speed, the thermal regime is amplified, having as result the temperature increase, both on local and on the whole bearing level.

A bearing must be stable, this means that it must change its sizes very few under thermal and mechanical solicitations. In this paper are presented some constructive changes in order to improve the performances of the ball radial bearings running at high speeds.

In the case of high speeds, it is experimentally determined the critical role the cage has in the deterioration of the bearing. The tests were done on bearings having metal cages. The repeated collisions between the cage and the rolling elements, regarding the aspect of some energy generation and dissipation phenomena, in many cases lead to the acceleration of the cage vibrations on the rotation running direction and implicitly to the loss of the cage running stability.

2. THEORETICAL AND EXPERIMENTAL CONSIDERATIONS

If the elastic deformations of couplers’ surfaces, as well as the changes of lubricant viscosity with the contact pressure become significant, the lubrication regime is elastohydrodynamic, EHD. In some practical cases, either because of scarcity of lubricant, or because of very high speed, the point at which pressure starts to rise in the input area is closer to the Hertzian area. Small amount of lubricant generates the phenomenon of starvation, causing the reduction of film thickness. Figure 1 shows two distinct situations:
a) – excessive lubrication,
b) - poor lubrication (starvation) – the input area in contact with the length *mb* is finite.

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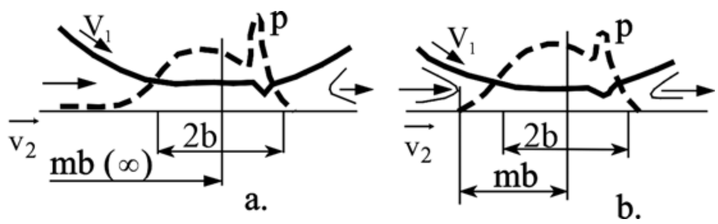

Fig. 1. Lubrication in an EHD contact.

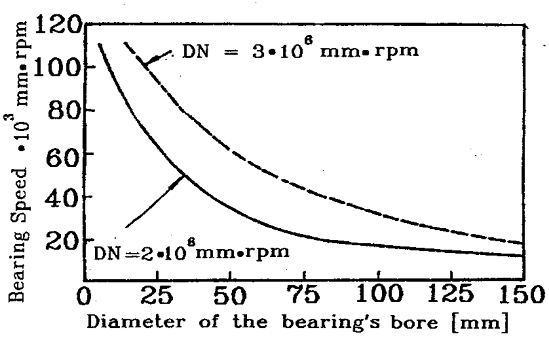

Fig. 2. Variation of the running speeds.

The running speeds which result from Figure 2 correspond to some values of the DN product (the product between the diameter of the bearing bore and the running speed) of $2\cdot10^6$ (mm.rot/min), with the prospects of exceeding the $3\cdot10^6$ (mm.rot/min)- limit, in conditions of some contact tensions at the bearing race level of (1.4-2) GPa and (5000-15000) hours andurance. Generally, at the contact level between the rolling and the bearing races are performed the conditions of some EHD lubrication regimes. At the contact level between the cage and the rolling, the lubrication regimes vary between large limits (bound regime, mixed, EHD or HD), while at the contact between the cage and ring gear prevail the mixed lubrication regime or HD.

In the theory of lubrication EHD the minimum thickness of the lubricant film at the level of contacts between the rolling and the bearing races is:

$$h_{\min} = 3.63 R_Y U^{0.68} G^{0.49} W^{-0.073} (1 - e^{0.68K}) \text{ [mm]} \tag{1}$$

where R_y is the radius of curvature equivalent on the moving direction; U, G and W are the speed, the corporal and the stress parameters and K is the factor of ellipticity (the ratio of the semi-axis of contact ellipses ratio). Relation (1) uses Barus' model for the variations of viscosity with the pressure ($\eta=\eta_0 e^{\alpha p}$), feeding plenty the "full flooded" contact with the lubricant and keeping a constant temperature in the contact area.

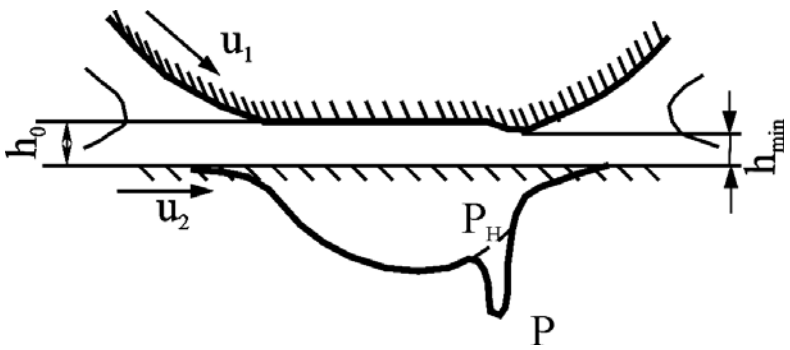

Fig. 3. Thickness of the central faceplate.

According to Figure 3, the lubrication regime EHD is characterized in the contact time by the presence of a central faceplate with a h_0 thickness, which pinches at the minimum value h_{min} in the outlet area, while the pressure distribution is changed in comparison with the hertzian contact, a peak of pressure occurring in the outlet area.

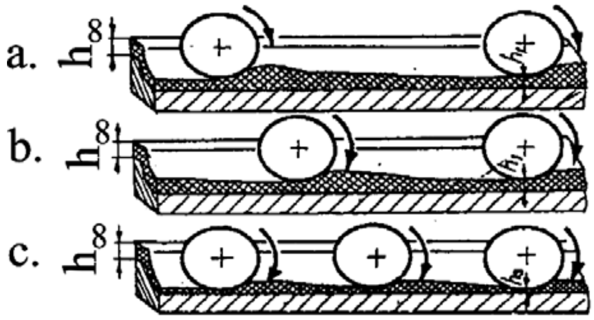

Fig. 4. Variation of the lubricant film thickness.

It presents the practical importance of λ, the parameter of the film defined by the relation:

$$\lambda = \frac{h_{min}}{\sqrt{\sigma_1^2 + \sigma_2^2}} \tag{2}$$

in which σ_1 and σ_2 are the heights' square deviation of the roughness from the both surfaces in contact. In Figure 4 is presented the variation of the lubricant film thickness according to the speed of rotation and the distance between the rollings (h_∞ is the thickness of the film at contact distance, h_l and h_s in contact) [2]. The estimation of the lubricant film thickness in the case of starvation phenomenon, cumulated with the thermal effects is done with the help of the relation:

$$h_{min,ef} = h_{min} \cdot C_S \cdot C_T \tag{3}$$

where C_S is a dimensionless coefficient of the film thickness with values of (0,15- 0,8) in the high-speed bearings, while C_T,is a thermal coefficient which depends on speed and lubricant's viscosity, with the values indicated in Figure 5.

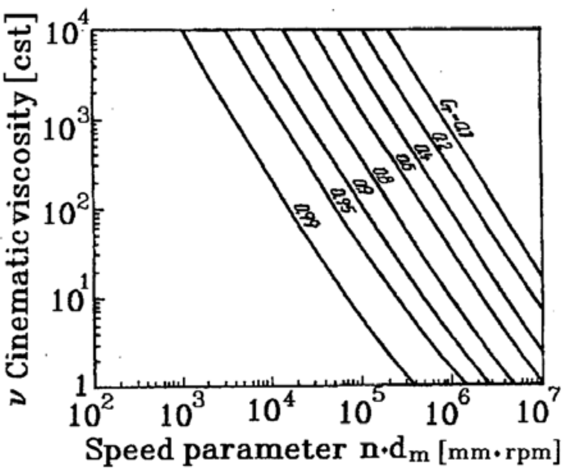

Fig. 5. Kinematic viscosity versus speed parameter.

By reducing the thickness of the film because of the thermal effects and of the lubricant in the bearing, more kinds of lubrication regimes can be developed, in conformity with the λ parameter. From Figure 6, the result is that the complete regime EHD is obtained only for λ≥3.

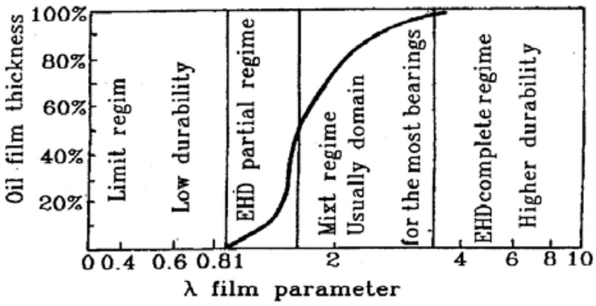

Fig. 6. Thickness versus parameter film.

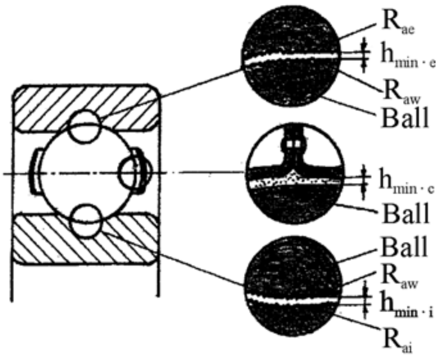

Fig. 7. Representing points of interest regarding the lubrication in a radial ball bearing.

Figure 7 shows points of interest, in terms of lubrication, in a radial ball bearing. In the contacts between the balls and rolling elements, EHD lubrication regimes are achieved due to the loads and their reduced conformities, with film thicknesses of h_{min}, (≈ 1μm) [3].

It has to be verified the cages' rate of wear in seats, controlling the outside surface of the cage which must not be in contact with the collar of the bearing ring. It also must not appear deposits from the cage material on the contact surfaces of the rolling units and rings [4].

In Figure 8 are presented the typical wear forms which develop on the cage. There are pronounced wear prints observed on the cage guiding shoulders (A surfaces) as well as in the balls seats (B surfaces).

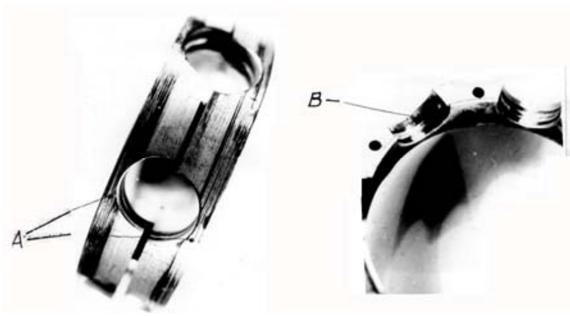

Fig. 8. The typical wear forms which develop on the cage.

By testing done on TALYROND, it was pointed out the non-uniformity of the wear on the circumference, the wear being manifested only on certain sections.

As result of the oil pollution with wear products, it was obtained an exaggerated increase of temperatures in the bearing, parallel to the increase of the friction moment.
If in Figure 8 are observed strong wear prints, in Figures 9 and 10 these wear prints are much more attenuated, which denotes a positive running speed, and also retrieved by the depressed moment of friction. With the help of an attachment designed by the author, a series of diagrams was traced to determine the correlation between the total film thickness at the level of balls-ball races contacts and the electric capacity.

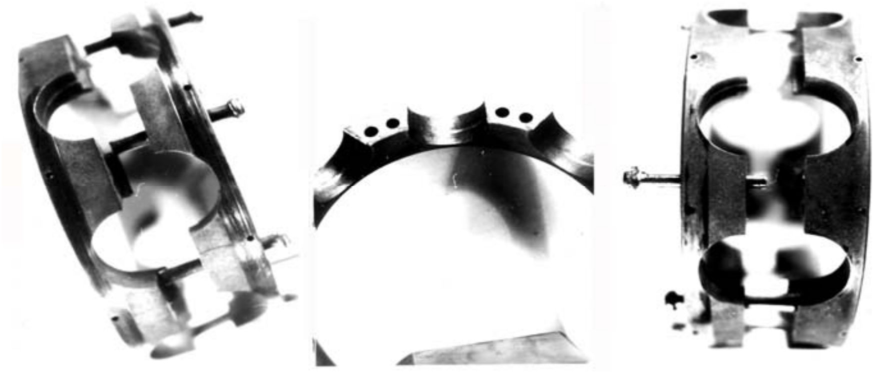

Fig. 9. The typical wear forms which develop on the cage.

Fig . 10. The typical wear forms which develop on the cage.

In Figure 11 is presented such a correlation, observing that in the domain of high thickness the method can be used with good results, and with the risk of electric penetration of the film in the case of small thickness [1].

In Figure 12 are presented the evolutions of the electric resistance of the lubricant film, as well as of temperatures, for the 6306 MAUP bearing in two versions of the radial play [1].

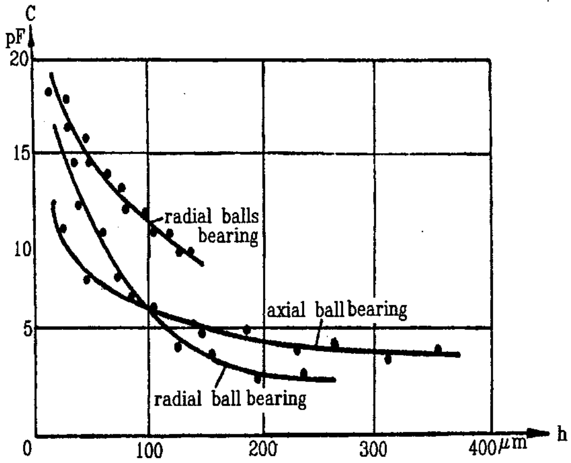

Fig.11. Variation of film thickness capacity.

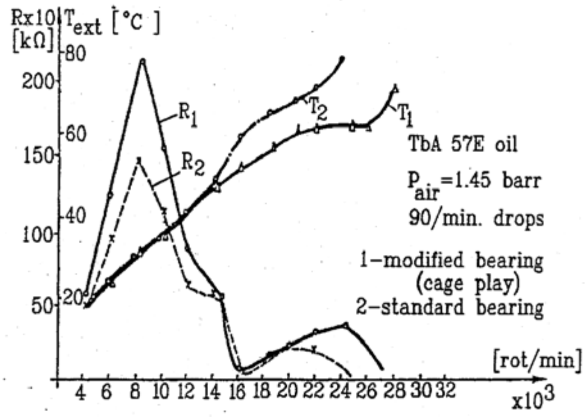

Fig. 12. Evolutions of electric resistance and temperatures.

Starting from the presented faults, there were executed six modified bearings classes; in the case of the first two (class 2 and 3), the radial play was increased from 0.3 mm to 0.4 mm, and respectively 0.5 mm; for the 4th and 5th class, the ball seat diameter was changed from 12.1 mm to 12.03 mm, respectively 12.25 mm; for the 6th class, the cage alignment on the bath rings was realized, and for the 7th class the number of the balls in the bearing was changed from 7 to 8 (Table 1).

Table 1. Changes of the radial play, of slots, of balls, of centering type and of balls number.

	Lot I Φ 12.1				
	11	12	13	14	15
D_2	59.7	59.7	59.7	59.7	59.7
d_2 D_c	59.630	59.640	59.625	59.643	59.645
J_{RC}	0.3				
Radial play	22	22	23	20	23
VG	15	10	10	11	11

	Lot II Φ 12.1						
	21	22	23	24	25	26	27
D_2	59.8	59.8	59.8	59.8	59.8	59.8	59.8
d_2 D_c	59.780	59.780	59.795	59.780	59.785	59.780	59.780
J_{RC}	0.4						
Radial play	17	20	20	20	19	25	22
VG	11	11	13	12	11	11	15

	Lot III Φ 12.1				
	31	32	33	34	35
D_2	59.9	59.9	59.9	59.9	59.9
d_2 D_c	59.875	59.870	59.910	59.915	59.885
J_{RC}	0.5				
Radial play	17	23	19	24	23
VG	13	14	11	10	12

	Lot IV Φ 12.03				
	41	42	43	44	45
D_2	59.8	59.8	59.8	59.8	59.8
D_2 D_c	59.820	59.790	59.785	59.790	59.815
J_{RC}	0.4				
Radial play	22	27	25	25	20
VG	14	12	11	11	12

	Lot V Φ 12.25				
	51	52	53	54	55
D_2	59.8	59.8	59.8	59.8	59.8
D_2 D_c	59.760	59.790	59.800	59.780	59.800
J_{RC}	0.4				
Radial play	21	22	20	23	25
VG	12	15	12	12	11

	Lot VI Φ 12.1				
	61	62	63	64	65
D_2	59.8	59.8	59.8	59.8	59.8
D_2 D_c	59. 160 44.7	59.790 44.4	59.776 44.4	59.803 44.4	59.800 44.4
J_{RC}	0.4				
Radial play	18	18	22	23	20
VG	15	15	14	12	14

	Lot VIII Φ 12.1 8 balls						
	71	72	73	74	75	76	77
D_2	59.8	59.8	59.8	59.8	59.8	59.8	59.8
d_2 D_c	59.785	59.820	59.810	59.755	59.810	59.775	59.785
J_{RC}							
Radial play	20		19	19	18	18	18
VG	14		12	15	15	15	14

3. CONCLUSIONS

For high-speed bearings, an important factor is the discrepancy of the rolling motion from the epicycloidal theoretical movement, discrepancy that is translated through a “cage slip”. The stability of the cage must be viewed also from a motional point of view. In the presence of the lubricant and under the temperature influence, a change of the cage's dimensions is produced. Analyzing the obtained results concerning the friction for bearings in the standard and modified versions, some conclusions can be drawn:

1. By increasing the speed for the standard version, generally comes out an increasing of the moment of friction, in conformity with the theoretical model. There are areas of speeds, around the values of 20,000 and 30,000 rot/min, in which heavy reductions of the moment of friction occur. These reductions caused by some changes in the bearing kinematics were pointed also by testing the cage gliding.

2. By increasing the play between the cage and the outer race on which this is guiding from 0.3 mm to 0.5 mm, it is obtained a reduction of the moment of friction in comparison with the theoretical one up to 27,000 rot/min) speeds. Comparative with the standard version results, in this case, a reduction of the power variation with 25% occurs.
3. The increase of the play between the ball and the cage leads to the reduction of the moment of friction in the bearing, but only on the high speeds domain, up to 25,000 rot/min, in the other cases the moments at high values.
4. The centering of the cage on the both rings has as effect a reduction of the moment of friction, more significant at high speeds. At the same time, it is observed the absence of some moment main fluctuations as result of the cage alignment improvement.
5. By increasing the balls number from 7 to 8 it is reduced the moment of friction, especially in the domain of high speeds, parallel to the uniformization of the moment on the whole range of speeds.
6. Producing modified bearings with oulets in seats as well as with oulets at the external ring level, it has been obtained reductions of the moment of friction on the whole range of revolutions.

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