Special Lubrication problems for bearings Running at High Speed

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Abstract: Generally the conditions of EHD lubrication working conditions are accomplished at the level of the contacts between the rolling bodies and the ring races of the bearings. At the level of the contacts between the cage and the rolling frames, the lubrication systems vary among large limits (limited, mixed, EHD or HD conditions), while at the contacts between the cage and the guiding level the mixed or HD lubrication conditions prevail. This paper presents the theoretical and experimental importance of studying the thickness of the lubricant film and proposes methods of determining it. The diminished quantity of lubricant generates the phenomenon which determines the reduction of the film's thickness.

Key Words: radial bearings, high speeds bearings, thickness of lubricant film.

1 Introduction

In the case of the bearings, the accidental contact pressure reaches high values (1000 - 3000) MPa, therefore, it is important to utilize lubrication systems dependent on factors such as: load, springiness of the interlocking gear's materials, variation of the lubricant's viscosity with pressure, peripheral speed and geometry [1].

Experimental tests referring to the running of bearings at high speeds were conducted on a stand which was designed and developed by the author. The stand is equipped with complex equipment for registering temperatures, friction moments, speeds and lubricant film thickness.

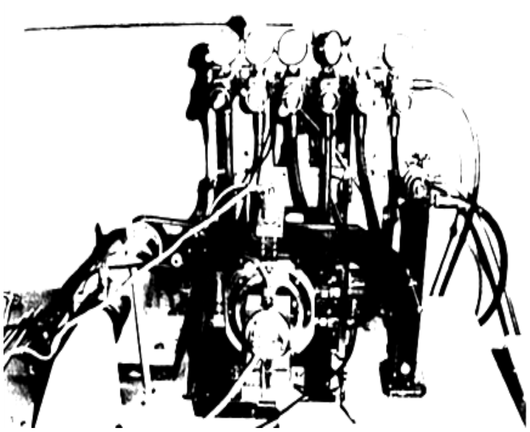

Fig.1 The stand for testing bearings at high speed (I).

In Fig. 1 and Fig. 2 the stand for testing bearings at high speeds is presented. It is made up of a testing head, intermediate broach, driving unit, lubrication installation, loading system, electrical installation and frame.

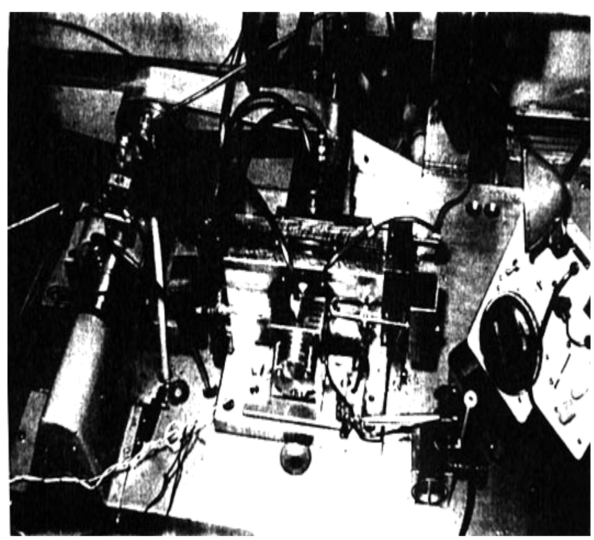

Fig.2. The stand for testing bearings at high speed (II).

The test head includes a front and a rear broach between which is a housing enclosing a hydrostatic

bearing supported elastic, within which the testing bearing is mounted.
The hydrostatic bearing is designed to ensure an evenly distributed lubricant film, which has the role of ensuring minimum friction torque of bearing measurement. The test bearing is mounted on the shaft and in the housing through interchangeable bushings.

2 Theoretical aspects

In the case of the point contact, solving the Reynolds equation becomes difficult because of the lateral leaks that cannot be ignored. So, in this case, the starting point is the Reynolds equation for the bidirectional leak:

$$\frac{\partial}{\partial \mathbf{x}}\left(\rho \frac{\mathbf{h}^3}{\eta}\cdot\frac{\partial \mathbf{p}}{\partial \mathbf{x}}\right)+\frac{\partial}{\partial \mathbf{y}}\left(\rho \frac{\mathbf{h}^3}{\eta}\cdot\frac{\partial \mathbf{p}}{\partial \mathbf{y}}\right)=\mathbf{12}\frac{\partial}{\partial \mathbf{x}}(\rho\cdot\mathbf{h}\cdot\mathbf{v_x}) + \mathbf{12}\frac{\partial}{\partial \mathbf{y}}(\rho\cdot\mathbf{h}\cdot\mathbf{v_y}) \quad (1)$$

and

$$\mathbf{v_x}=\frac{\mathbf{v_{1x}}+\mathbf{v_{2x}}}{\mathbf{2}},\ \mathbf{v_y}=\frac{\mathbf{v_{1y}}+\mathbf{v_{2y}}}{\mathbf{2}}, \quad (2)$$

where v_x and v_y represent the medium peripheral speeds on the rolling Ox direction and on the lateral Oy direction, according to Fig. 3.
The contact between the two bodies (1) and (4) may be equated with a contact between a plane and an equivalent revolving body which has the equivalent curve shaped Rx and Ry beams in the contact point, where:

$$\frac{\mathbf{1}}{\mathbf{R_x}}=\frac{\mathbf{1}}{\mathbf{R_{1x}}}+\frac{\mathbf{1}}{\mathbf{R_{2x}}},\ \frac{\mathbf{1}}{\mathbf{R_y}}=\frac{\mathbf{1}}{\mathbf{R_{1y}}}+\frac{\mathbf{1}}{\mathbf{R_{2y}}}. \quad (3)$$

In the case when two bodies are elastically distorted under the action of the normal load Q, the vertical distance between 2 points situated on the distorted surfaces is identical with the similar distance between the equivalent body and the plane. This is shown in Fig.4, where:

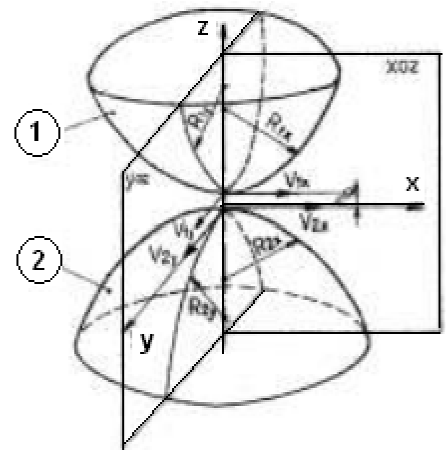

Fig.3. The geometry of the Hertzian rolling point contact.

This is shown in Fig.4, where:

$$\mathbf{h(x,y)=h_0+s(x,y)+w(x,y)}. \quad (4)$$

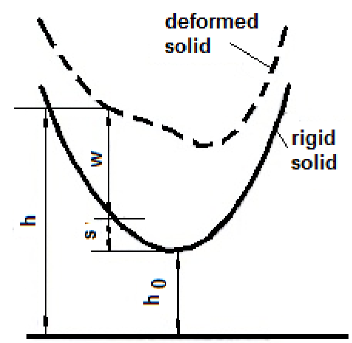

Fig.4. The thickness of the separation film between an equivalent ellipsoid and a flat surface.

In (4), h_0 is the thickness of the lubricant film in the central area, s(x,y) represents the space between the two bodies or clearance, and w(x,y) is the elastic distortion [2].
The clearance s(x,y) is given by the formula:

$$\mathbf{S(x,y)=\frac{x^2}{2R_x}+\frac{y^2}{2R_y}}. \quad (5)$$

The elastic distortion, in the case of perfectly elastic, homogenous and isotropic bodies, is given by the relation:

$$w(x,y)=\frac{2}{\pi E'}\int_{-\infty}^{+\infty}\int_{-\infty}^{+\infty}\frac{p(\xi,\delta)d\xi d\delta}{\sqrt{(x-\xi)^2+(y-\delta)^2}} \quad (6)$$

In (6), $p(\xi,\delta)$ represents the pressure under a point (ξ,δ) from the contact area.
Changes in the viscosity and the lubricant's density have to be taken into consideration as a consequence of the high pressures from the contact area. Thus, Barus's relation for viscosity can be applied, where:

$$\eta=\eta_0 e^{\alpha p} \quad (7)$$

Dowson's formula for the variation of the lubricant's density with the pressure can be used:

$$\rho=\rho_0\left(1+\frac{0.6p}{1+1.7p}\right). \quad (8)$$

Solving (1) is very difficult, but as a consequence of experimental determinations, the presence of a central plate was emphasized, where h_0 is the thickness of the film of a horseshoe-shaped notch, situated in the exit area of the contact. [3]
The practical importance presents h_{min} (the minimum thickness of the lubricant) and h_0 (the central thickness). Essentially, these depend on U, the speed parameter, and w the loading parameter (Fig. 5).

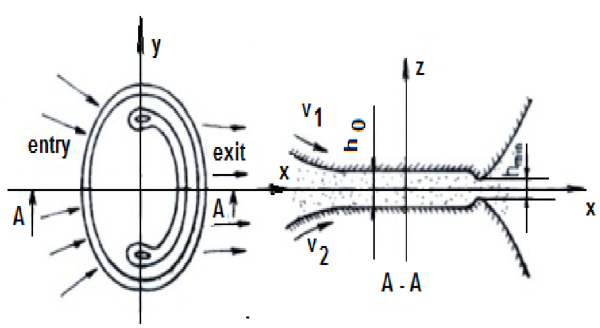

Fig. 5 Entry-Exit The shape of the profile in an EHD point contact.

Using the principle of the resistive method and considering the contacts between balls and bearing races, as a consequence of their separation through the lubricant film realizing electrical resistances when the flow passes from one level to another, the electrical resistance of a contact varies exponentially with the thickness of the film:

$$R=Ke^{h}, \quad (9)$$

where proportionality factor K, depends on the pressure, temperature, lubricant type, contact's dimension, rigor, etc. The resistive method allows qualitative, comparative measurement, of the lubrication system with diverse lubrication mediums. If it is considered a cylindrical roller bearing and the equivalent electrical scheme, it can be written that the global resistance of the bearing is:

$$R=\sum_{n=1}^{2}\frac{1}{\left(R_{c,i}+R_{c,e}\right)_n}, \quad (10)$$

where $R_{c,e}$ and $R_{c,i}$ represent the electrical resistances at the level of interior and exterior contacts and "2" is the number of balls. The use of the capacitive method means using a low voltage radio frequency flow applied to the frames between which the lubricant film is located. The lubricant film functions as a dielectric and the films thickness has to be determined. The method requires that while testing, the dielectric constant of the oil should not vary. The method is also practicable because, when the thickness of the film is small, the measured capacity is higher and therefore, the measurement's precision is correct. The difficulty was caused by the frequency of the piercing of the film by the contacts of low resistance, among the surfaces' asperities or conductive particles [4].

3 Experimental aspects

By reducing the thickness of the film due to the thermal effects and the lubricant in the bearing, more kinds of lubrication regimes can be developed, in conformity with the λ parameter. From Fig. 6, the result is that the complete regime EHD is obtained only for $\lambda \geq 3$. In Fig. 6, λ represents the film parameter that is selected in the calculation according to the lubrication regime.
Fig. 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}, ($\approx 1\mu m$) [5].

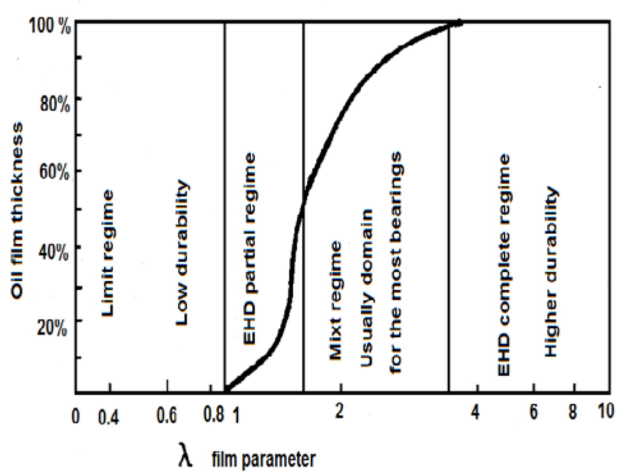

Fig. 6. Thickness versus parameter film.

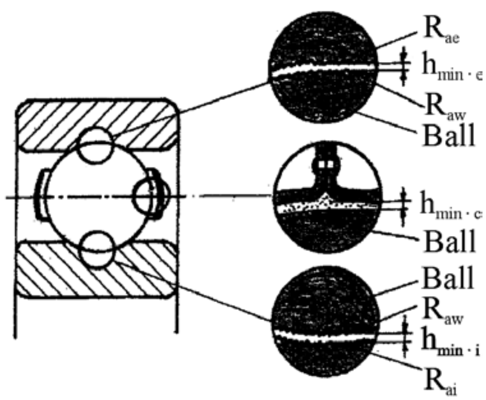

Fig. 7. Points of interest regarding the lubrication in a radial ball bearing.

The cages' rate of wear in seats needs to be verified, insuring that the outside surface of the cage can not be in contact with the collar of the bearing ring. Also, deposits from the cage material on the contact surfaces of the rolling units and rings must not be present [6].

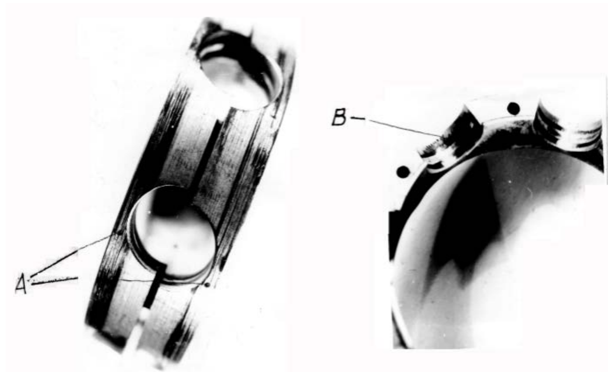

Fig. 8 The typical wear forms that develop on the cage.

Typical wear forms which develop on the cage are presented in Fig. 8. There are pronounced wear prints observed on the cages' guiding shoulders (A surfaces) as well as in the ball seats (B surfaces). By testing carried out on a TALYROND device, circumferential non-uniform wear was exhibited; wear was manifested only on certain sections. As a result of the impurities in the oil through wear products, an exaggerated increase of temperatures in the bearing was obtained, parallel to the increase in the friction moment.

If considerable wear prints are observed in Fig. 8, in Fig. 9 and Fig. 10 these wear prints are greatly attenuated, denoting a positive running speed, and also retrieved by the depressed moment of friction.

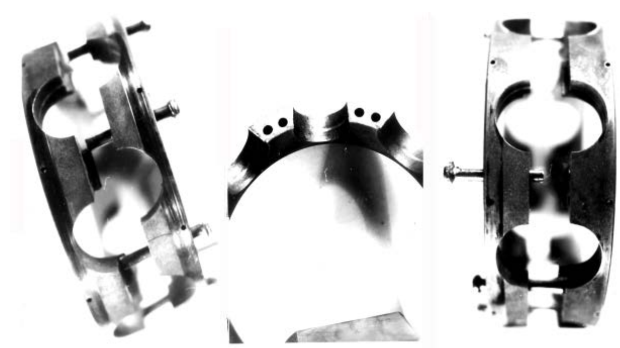

Fig. 9 The typical wear forms which develop on the cage.

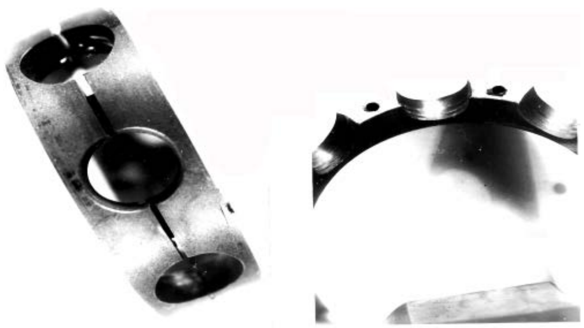

Fig.10 The typical wear forms which develop on the cage.

With the help of an attachment designed by the author, a series of diagrams were traced to determine the correlation between the total film thickness at the level of balls - ball races contacts and the electric capacity. Fig. 11 presents such a correlation, observing that in the domain of high thickness, the method can be used with good results while in the case of a small film thickness there is the risk of electic penetration of the film [7].
Due to the radial loadings, in (10) only the resistances from the loaded balls (balls 1 and 2) occur, in the case of other contacts the electrical resistance is ∞.
Due to the small thicknesses of the lubricant film (<1(μm)), the flow that passes through the contacts is low, varying among the limits (1÷50) (μA). The unique value of the flow is dictated by avoiding the appearance of the electrical loadings at the level of the contacts.
Fig. 11 presents the electrical schematic diagram, the supply voltage is 1 (V) and the additional resistance Ra has the same value as the quantity order of the electric bearing resistance.

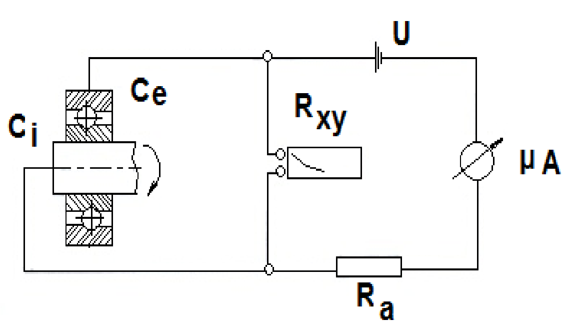

Fig. 11 The electrical schematic diagram.

The variations of potential dropping on the bearing and the variation of electrical resistance of the bearing have been registered with an acquisition plank (the collector from the external ring that is immobile, while the collector from the internal ring is mobile) [8].
Fig. 12 presents electrical resistance determinations for 2 tested oils. Simultaneously, on the diagram in the picture, according the λ parameter's values, defined as the proportion between the minimum thickness of the lubricant and the compounded rigor of the 2 surfaces in contact, the lubrication working conditions, according to global electrical resistance of the bearing, have been separated.
The diminishing of the electrical resistance under 3 - 4 (kΩ) leads to the existence of mixed or limited lubrication systems. This reduction is due to the lubricant's deterioration and always precedes it's out of use.

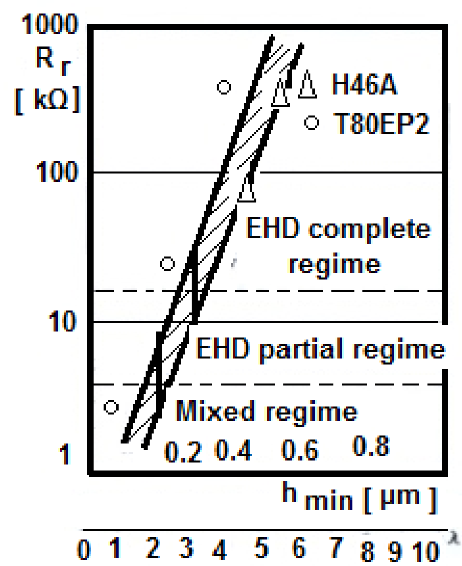

Fig. 12. Determinations of electrical resistance for diverse types of oil.

Fig. 13 depicts the evolutions of the electrical resistance of the lubricant film, and also of the temperatures of 6306 MAUP bearing, in two variants of the cage's play.

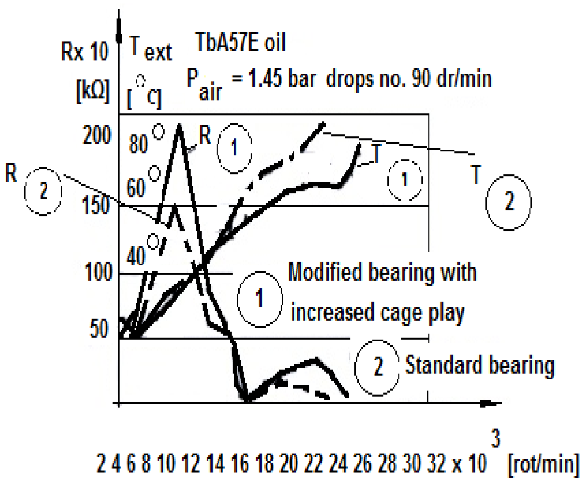

Fig. 13. Electrical resistance of the lubricant film and the temperatures in the case of the particular bearing 6306 MAUP.

According to [8], starting from the presented faults, six modified bearing classes were constructed; in the case of the first two classes (class 2 and 3), the radial play was increased from 0.3 (mm) to 0.4 (mm), and 0.5 (mm) respectively; for the 4th and 5th classes, the ball seat diameter was changed from 12.1 (mm) to 12.03 (mm), 12.25 (mm) respectively; for the 6th class, the cage alignment on the bath rings was achieved, and for the 7th class the number of the balls in the bearing was changed from 7 to 8 (Table I -VII).

TABLE I

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

TABLE II

CHANGES OF THE RADIAL PLAY, OF SLOTS, OF BALLS, OF CENTERING TYPE AND OF BALLS NUMBER

	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	59.7	59.7	59.7	59.78	59.78	59.7	59.7
J_{RC}	0.4						
Radial play	17	20	20	20	19	25	22
VG	11	11	13	12	11	11	15

TABLE III

CHANGES OF THE RADIAL PLAY, OF SLOTS, OF BALLS, OF CENTERING TYPE AND OF BALLS NUMBER

	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

TABLE IV

CHANGES OF THE RADIAL PLAY, OF SLOTS, OF BALLS, OF CENTERING TYPE AND OF BALLS NUMBER

	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

TABLE IV
CHANGES OF THE RADIAL PLAY, OF SLOTS, OF BALLS, OF CENTERING TYPE AND OF BALLS NUMBER

	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	59.7	59.7	59.8	59.7	59.8
	60	90	00	80	00
J_{RC}	0.4				
Radial play	21	22	20	23	25
VG	12	15	12	12	11

TABLE VI
CHANGES OF THE RADIAL PLAY, OF SLOTS, OF BALLS, OF CENTERING TYPE AND OF BALLS NUMBER

	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	59.16	59.7	59.7	59.8	59.80
	0	90	76	03	0
	44.7	44.4	44.4	44.4	44.4
J_{RC}	0.4				
Radial play	18	18	22	23	20
VG	15	15	14	12	14

TABLE VII
CHANGES OF THE RADIAL PLAY, OF SLOTS, OF BALLS, OF CENTERING TYPE AND OF BALLS NUMBER

	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	59.7	59.82	59.8	59.75	59.8	59.7	59.7
J_{RC}							
Radial play	20		19	19	18	18	18
VG	14		12	15	15	15	14

4 Conclusions

This paper presents a set of theoretical and experimental results. Bearings exist in machines and tool components that are used in a wide range in all industry areas.

Using the initial principle of the capacitive method for experiments (Fig. 14), a sectioned angular contact ball bearing was used with the pressure being taken by a single ball while in the third phase a cylindrical roller bearing with many balls was used. The lubrication is essential to assure the bearings' friability and the durability is directly influenced by the parameter λ of the lubrication film.

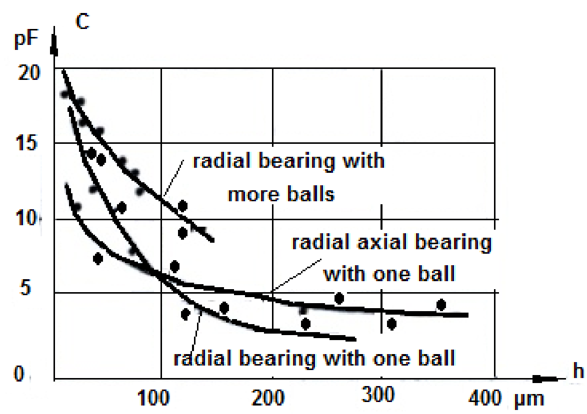

Fig.14.The variation of electrical capacity according to the thickness of the lubricant film

During the experiments, it is observed that there are substantial differences between the bearings working at normal speeds and those working at high speeds. The supplementary effects caused by the speed increase (centrifugal forces, gyroscopic movements, additional frictions) unfavorably influence the speed limits. The role of the lubricant in maintaining reliability at high speeds is decisive both by accomplishing the EHD film and by controlling thermic effects.

Following the analysis of the research results, the conclusion is that the model which provides a minimum moment of friction at the whole range of speeds and at high speeds corresponds to the type of bearing characterized by the presence of longitudinal lubrication channels in seats and of holes in the outer ring.

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 also be viewed from a motional point of view. In the presence of the lubricant and under the thermal influence, a change of the cage's dimensions is produced.

Analyzing the obtained results concerning the friction of the bearings in the standard and modified classes, some conclusions can be identified:

1. By increasing the speed for the standard class, an increase of the friction moment generally occurs, in conformity with the theoretical model. There are areas of speeds, around the values of 20.000 and 30.000 (rpm), in which large reductions of the friction moment occur. These reductions are caused by changes in the bearing kinematics and were also evidenced by testing the cage gliding.

2.

3. By increasing the play between the cage and the outer race on which this is guiding from 0,3mm to 0,5mm, a reduction of the friction moment is obtained in comparison with the theoretical one up to 27.000 (rpm) speeds. In this case, a reduction of the power variation by 25% occurs compared with the standard version (control group).

4. The increase of the play between the ball and the cage leads to the reduction of the friction moment in the bearing only on the high speed domain, up to 25.000 (rpm). In the other classes the moments remain at high values.

5. The centering of the cage on both rings has an effect on the reduction of the friction moment that is more significant at high speeds. At the same time, it is observed that there is an absence of some of the main fluctuations of the moment as a result of the cages' improved alignment.

6. By increasing the number of balls from 7 to 8 the friction moment is reduced, especially in the domain of high speeds, as well as causing a leveling of the moment on the whole range of speeds.

7. Producing modified bearings with outlets in the seats and with outlets at the external ring level, results in reductions of the friction moment on the whole range of revolutions.

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