OPERATING CONDITIONS OF BEARINGS RUNNING AT HIGH SPEEDS

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ABSTRACT:
IN ORDER TO PREVENT AGGRESSIVE DAMAGE, VIBRATION CONTROL IS REQUIRED ALONG WITH MONITORING DURING OPERATION. THE EASIEST METHOD TO ACHIEVE THIS IS WITH AN ACOUSTIC PROBE PLACED AS CLOSE AS POSSIBLE TO THE BEARING WHICH IS TO BE CONTROLLED. THE ANALYSIS REGARDING THE FIELD OF TIME REQUIRES SIMPLE EQUIPMENT, PROVIDING THE POSSIBILITY OF FAULT DETECTION THROUGH THE APEXES POSITIONED AT REGULAR INTERVALS IN TIME, FOR THE FAULTS FROM THE INTERNAL OR EXTERNAL RING, AND QUASI-REGULAR FOR THE FAULTS ON THE ROLLING BODIES.

KEYWORDS*:* BEARINGS, FRICTION MOMENT, HIGH SPEED BEARINGS, BEARING LOAD, ROLLING RESISTANCE

1. INTRODUCTION

The Bearing cage plays an important role in the proper functioning of a bearing, the cage which has the purpose of maintaining a fixed distance between the revolving bodies. The cage is characterized by its form and by the materials of its construction. The form of the cage can vary in function with the type of bearing it houses as well by the conditions or functions imposed on the bearing. Metallic cages pressed from sheet metal or massive cages that are machined mechanically can achieve moderate or small work revolutions, whereas light cages composed of phenolic resins or polyamides allow for high and very high revolutions that are currently used commercially. Therefore, an important factor in the proper function of a cage is the composition of the materials used in its construction. There are two types of synthetic resins used in the fabrication of bearing cages: thermorigid and thermoplastic.

For the fabrication of bearing cages with characteristics corresponding to aero-bearings, respective to the large domain of working temperatures and elevated revolutions, hence, only plastics that show elevated stability under these conditions can be utilized in these bearings. The mechanical and physicochemical properties of the plastics can be modified through the addition of materials of a different nature (ruggedizing materials). This imposes the use of a synthetic resin that can resist high temperatures for prolonged periods, maintaining its physicochemical properties unchanged, a synthetic resin that can resist a series of chemical

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agents and combustibles, resist humidity, exhibit low thermal expansion and be easy to fabricate.

In working at high speeds, it is experimentally ascertained that the cage has a decisive role in the faulting weight (in the case of bearings having aerose cages). The repeated collisions between the cage and the rolling elements, as regards the aspect of energy generation and dissipation phenomenon, lead in many cases to the acceleration of the cages' vibrations on the rotation running direction and are implicit to the loss of the cages' running stability. It must be verified that the outer surface of the cage is not in contact with the collar of the bearing ring, the rate of wear of the cage seats must be determined and there should be no debris or settlements from the cage material on the contact surfaces of the rolling units and rings.

The friction due to the contact deformations are manifested in two ways:

- internal friction occurring within the material of the bodies in contact;
- friction obtained by increased resistance at the rolling moment due to the contact deformations.

The first category refers to the energy loss due to internal friction from the material of the bodies in contact, caused by the fact that the energy needed for a bodies' deformation is higher than the one obtained after the action stop of the external load and, consequently, the material's relaxation. This provokes a resistance against the rolling motion. Mechanical deformational work is directly proportional to the bearing's rotational speed.

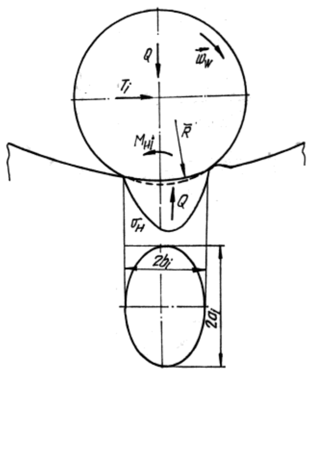
The moment in the bearing generated by the elastic hysteresis is:

$$M_H = \sum M_{Hi} = 5{,}8 \cdot 10^{-7} \cdot d_m \cdot D_w^{-\frac{2}{3}} \sum Q_i^{\frac{4}{3}} \ [\mathrm{Nm}] \qquad (1)$$

where D_w is the ball's diameter in [mm]; d_m is the medium diameter of the bearing in [mm]; Q_i is the load requiring a certain body "i" in [N].

While functioning, the rolling bodies are required by the forces perpendicular on the contact surface, coming into contact with the bearing race. Due to these forces, the bodies in contact are deformed, the dimensions of the deformed surface is based on Hertz's relations.

2. THEORETICAL AND EXPERIMENTAL ASPECTS



To overcome the resistance in the rolling moment, caused by these deformations, a tangential T_i force (Figure 1) is needed. The size of the T_i force will be determined by the necessity of overcoming the resistance of prominent material occurring before the rolling body. At the same time a phenomenon caused by rolling motion generates a negative pressure behind this body[2].

The moment of friction in a bearing may be considered as being made up of two components: the friction moment at the idle running M_o independent of the load, and the friction moment M_1 dependent of the load. M_o depends on the size of the bearing, the kinematic viscosity of the lubricant ν at the temperature during operation, speed and the coefficient f_o dependent on the constructive material type of the bearing and lubrication system.

[2] M. Gafiteanu, *Bearings, Vol. I* (Bucharest: Editura Tehnica, 1985), 93-105.

$$M_0 = f_0 \cdot (\nu n)^{2/3} \cdot d_m^3 \qquad (2)$$

where M_o is expressed in [Nmm]; the medium diameter d_m in [mm]; the kinematic viscosity ν in [mm^2/s] and the speed n in [min^{-1}]. The values of the coefficient f_o are given in the catalogues of various companies.

Chart 1

The type of the bearing construction:	Type	F_0		
		FAG	SKF	INA
Ball bearings	Grease	$(0{,}7\text{-}1)\cdot 10^{-7}$	$(1{,}5\text{-}2\cdot 10^{-7}$	-
	Oil pan	$(1{,}5\text{-}2)\cdot 10^{-7}$	$(1{,}5\text{-}2)\cdot 10^{-7}$	-
Cylindrical roller bearing with cage	Grease	$(1{,}5\text{-}2)\cdot 10^{-7}$	$(2\text{-}3)\cdot 10^{-7}$	-
	Oil pan	$(2\text{-}3)\cdot 10^{-7}$	$(2\text{-}3)\cdot 10^{-7}$	-
Cylindrical roller bearing without cage	Grease	$(2\text{-}2{,}5)\cdot 10^{-7}$	$(2{,}5\text{-}4)\cdot 10^{-7}$	$(0{,}2\text{-}1)\cdot 10^{-6}$
	Oil pan	$(2{,}5\text{-}3{,}5)\cdot 10^{-7}$	$(2{,}5\text{-}4)\cdot 10^{-7}$	$(0{,}4\text{-}1{,}2)\cdot 10^{-6}$
Needle bearings	Grease	$(3\text{-}6)\cdot 10^{-7}$	$(1\text{-}10)\cdot 10^{-7}$	$(0{,}1\text{-}1)\cdot 10^{-6}$
	Oil pan	$(6\text{-}12)\cdot 10^{-7}$	$(3\text{-}9)\cdot 10^{-7}$	$(0{,}3\text{-}0{,}9)\cdot 10^{-6}$

Chart 1 presents the values of the coefficient f_o which are used to calculate the friction moment at the idle running of M_o bearing. Analyzing the values in the chart, it is found that, beside the large dispersions from one catalogue to another, there are also dispersions of the coefficient f_o even within the same catalogue (Chart 1). At the same time, the indicated values for f_o do not take into account a series of particularities related to the lubricant quantity from the bearing and the heat exchange etc. In case of oil lubrication, M_o depends on the oil quantity given to the bearing. In case of greasing, if the quantity is too excessive, M_o increases, because the excess must be removed from the bearing's components. And in case of oil mist lubrication (minimum quantity), the value of f_o may decrease until the half value of the oil pan.

The friction moment M_1 dependent on the load can be calculated, according to the Palmgren relation:

$$M_1 = f_1 \cdot g_1 \cdot P_0 \cdot d_m \qquad (3)$$

where: - f_1 is a coefficient depending on the construction and relative load of the bearing
- g_1 is a coefficient dependent on the direction of the load
- P_o represents a static load equivalent of the bearing
- d_m is the medium diameter.

The coefficient f_1 is a constant dependent on the equivalent static load, radial or axial load provoking in the bearing the same deformation as the real load.

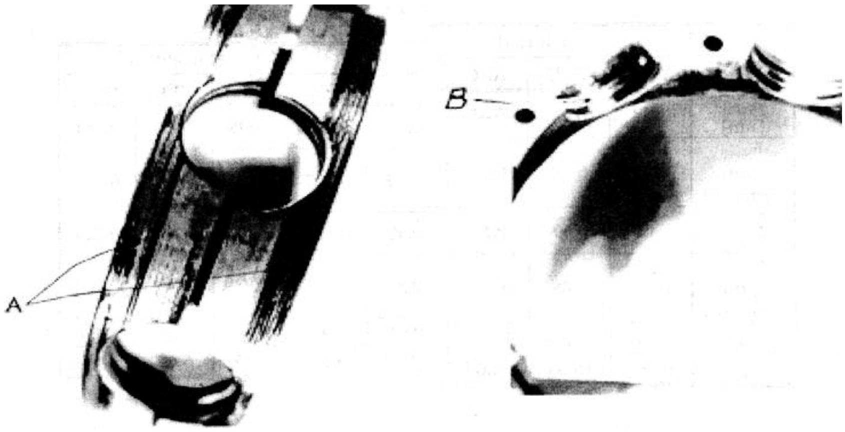


Fig. 2 The typical wear forms which develop on the cage

In Figure 2 are presented the typical of wear forms which develop on the cage in case of very high speeds. There are pronounced wear prints observed on the cage guiding shoulders (the A surfaces), as well as in the balls seats (the B surfaces).

Because of the presented failures it has been ascertained a change of the roughness on the ball races. For a bearing which presented very high rubbings on the outside surface of the cage, the roughness has increased on the inner ball races from 0.02μm to 0.04μm, and on the outer ball race, placed in the area where the cage-ring wear prevails, the roughness has changed from 0.02μm to 0.1μm. This roughness increase has led to the accentuated decrease of the λ parameter of the film, the calculated values for the modified roughness being presented in Chart 2.

Chart 2

	Oil Type							
	Oil L 4/1				Oil LA 32 modified			
	30°C	50°C	70°C	90°C	30°C	50°C	70°C	90°C
λ_{int}	3,58	3,58	2,98	2,44	1,18	1,52	1,40	1,20
λ_{ext}	4,42	4,45	3,69	3,02	1,45	1,87	1,73	1,49

Analyzing the experimental and theoretical results, it is noticed a good correlation between the thickness of the lubricant film values. In the same time diverse factors which influence film thickness have different weights, being noticed the decisive influence which roughness has on the X film parameter.

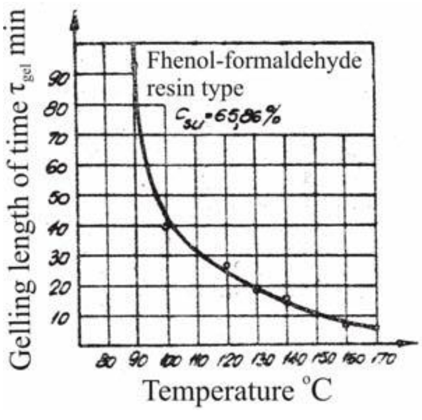


Fig. 3 The influence of temperature on gelling length of time

Experimental test tubes, similar to those determining traction strength of phenoplasts, have been submitted to the following treatments:

a - Condensation appropriate to τ_c^o and heat treatment for 60 minutes. At the same temperature, τ_c=(τ_c^o+60) min., (Figure 3);

b - Condensation appropriate to τ_c^o followed by heat treatment for 120 min. at 130˚C;

c - Immersion in acetone, for 2 h at 20oC;

a/d; b/d- immersion in acetone for the test tubes of type a and b (Figure 2 and Figure 3).

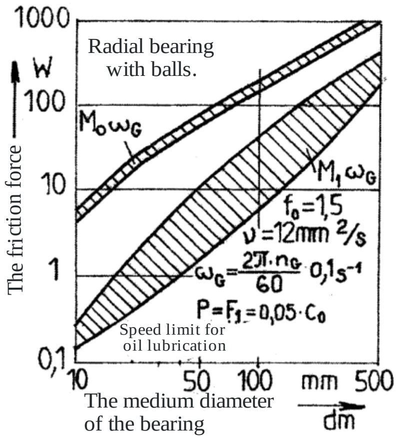


Fig. 4 Variation of the rolling power for the ball bearing

The assessment of the resin's condensation degree has been done through acetone strength (losing the weight ΔG) and that of mechanical characteristics through traction strength, σt [MPa].

Figure 4 presents the force of idle rolling, independent of the load ($M_o\omega_G$) and the friction force dependent on the load ($M_1\omega_G$), for a ball bearing at the limit value of the speed and the corresponding load at $0.05C_o$. For a given viscosity of the lubricant and a constant speed, $M_o\omega_G$ depends on the value of the coefficient f_o[3]. The values of the friction moment M_o, obtained when the bearing functioned without radial load, together with the product (ν n) have been represented in the diagram in Figure 5.

Taking into consideration the medium values for M_o and the product (ν n), the coefficient f_o was determined from the relation (2), whose value resulted to be $1.4 \cdot 10^7$.

Compared to the recommendations of different catalogues, the value determined experimentally is closer to the inferior limits, due to the decrease of the friction with the lubricant, in the case of using the drop lubrication. In conclusion, also for high speed, the calculus relation of the moment M_o is usable.

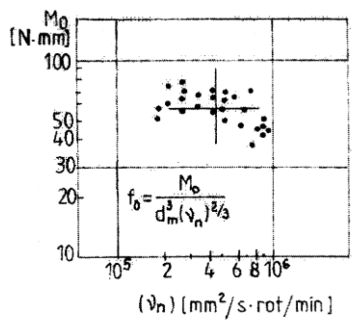


Fig. 5 Value representation of the

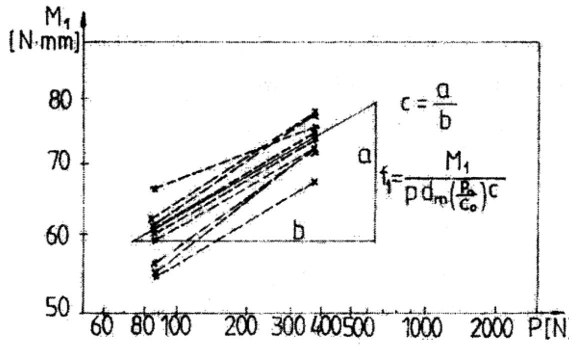


Fig. 6 Value representation of the

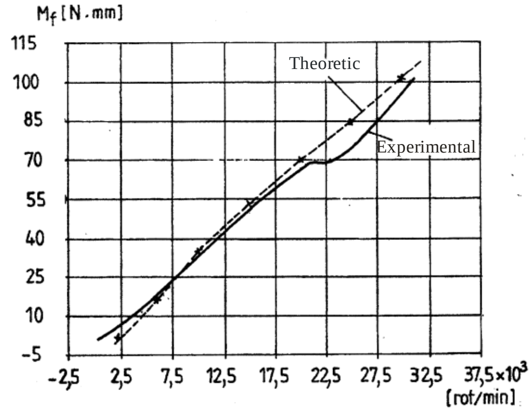
Loading the bearing with radial forces, under the same speed and lubrication conditions, determine the values of the total moment (M_o+M_1). Knowing the values of M_o, (previously determined), one can find the values of M_1, represented graphically as loading function (Figure 6), in a logarithmic coordinate system.

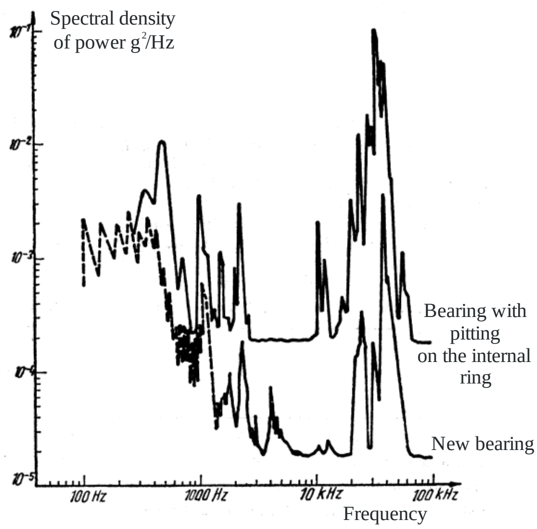
A regression line has been calculated for the drawn lines, whose slope *c* represents the exponent of the relation (P_o/ C_o), which has the value of 0.55, in the given situation. This value is consistent with the indications given in the catalogues.

[3] T. Bolfa, *PhD Thesis: Contributions to improving the quality performance of high speed bearings* (1991).

It can be concluded that for standard bearings which are drop lubricated the catalogue relations can be used at high speed using the minimum values from the catalogue for the factor f_o. Making constructive changes related to the decrease of the friction from the bearing and using mist lubrication may reduce the factor f considerably (1/10 of the catalogue values)[4].

In Figure 7 is presented the graphic representation of the variation for the friction moment dependent on the speed. Some perturbations occur when the speed is greater than (18.000- 23.000) rpm, because of the loss of the cage's stability.





In Figure 8 are presented the outcomes of the analysis for a faulty bearing, compared to a faultless bearing. It can be correlated that the size of the fault with the occurrence of a distinct "shock" frequency, as a counter-value of the time needed for a rolling body to go through the length of the fault. The faults' sizes and their number lead to changes in the frequency spectrum, as the deformations expand, the damaged areas from the spectrum amplify in the field of frequencies (1- 100) kHz. (Figure 8). The contamination particles lead to spectrum changes both in the fields of excitation frequencies, and also in the entire frequency spectrum, this way creating the micro shock conditions.

The analysis in the amplitude field through probability density can bring important specifications, compared to the simple counting of some apexes. If it is marked as p(x) the probability density for a given distribution of the signal, the following values can be defined:

- medium value $M_1 = \int_{-\infty}^{+\infty} p(x)x\,dx = \bar{x};$
- the dispersion $M_2 = \int_{-\infty}^{+\infty} p(x)x^2 dx = D^2 = \sigma^2;$
- the asymmetry coefficient $\sqrt{\beta_1} = \frac{M_3}{\sigma^3} = \left(\int_{-\infty}^{+\infty} p(x)(x-\bar{x})^3 dx \right) / \sigma^3$

[4] T. Bolfa, *Contact Mechanics and Tribology* (Brasov: Editura Lux Libris, 2006), 52-64.

- the influence coefficient of the apex $\beta_2 = \frac{M_4}{\sigma^4} = \left(\int_{-\infty}^{+\infty} p(x)(x - \bar{x})^4 \, dx \right) / \sigma^4$

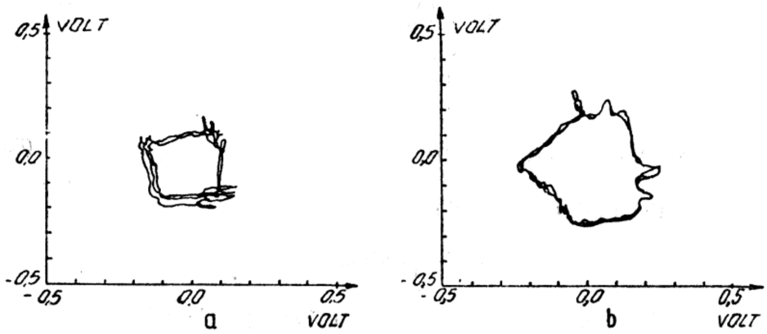


Fig. 9

In Figure 9 are presented the results of the measurements for the position of the shaft center if settled on a sealed bearing in case *a*, and on a bearing with removable cage in case *b*, for which the displacements of the shaft center are higher.

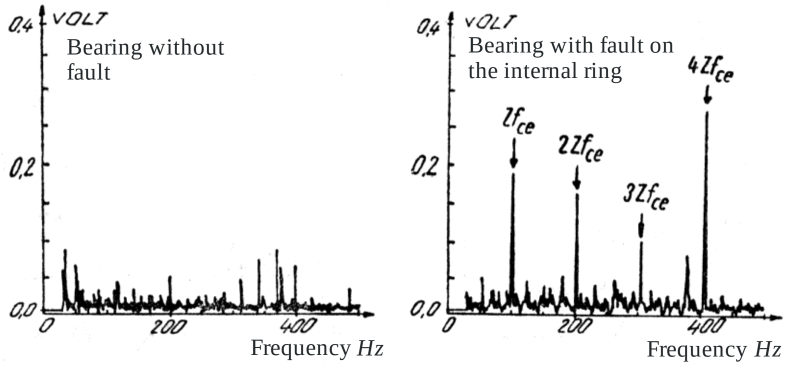


Fig. 10

The faults on one of the rings occur as much as obvious in the frequency spectrum (Figure 10), either for the first harmonic of the excitation corresponding to balls' passing over the faulty area, or for the second or third harmonic.

3. CONCLUSIONS

The studies have pointed out the prevalence of same stressed wear phenomena at the level of the contact between the cage and the guide ring, as well as in the cage seats for balls. The wear phenomena presence, having as a natural consequence the pollution of the lubricant, also affects the contacts between the balls and the bearing races, with the change of the roughness and the reduction of the X film parameter. From these findings it is assessed the execution of constructive modifications especially at the cage level (a smaller outer diameter of the cage, owlets through cage, cage guided on balls, oblong seats of the balls).

An important issue for a structure is the early determination of faults during operation that could lead to serious damage during operation. For the bearings, the level of vibrations and noise is a global quality indicator, showing off the product's competitiveness, correlating it with the failure, indicating the occurrence and size of the faults, acting either through direct effects as acoustic radiation, or through indirect effects at the level of other elements (shafts, shells).

The vibration and noise provide a significant sensitivity to the analysis, developing the processing techniques of the signal.

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