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THE INFLUENCE OF CLEARANCES IN BALL BEARINGS ON WORK FAILURE OF THREE-BEARING SHAFTS

WPŁYW LUZU W ŁOŻYSKACH KULKOWYCH NA NIEPRAWIDŁOWĄ PRACĘ WAŁU TRZYPODPOROWEGO

Key words:

bearings system, ball bearing, bearing load, bearing durability

Słowa kluczowe:

układ łożysk, łożyska kulkowe, nośność łożysk, trwałość łożysk

Summary

Durability of deep groove ball bearings depends on factors (attributes) of design, technology, and operation. Among the design features, one of the most important is the bearings. Polish standards list five groups of looseness in bearings in the range from 0 to 105 microns.

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Manufacturers of roller bearings do not normally give the exact value of the bearing looseness. The aim of this study is to determine how looseness affects the play three-bearing shafts, including elasticity and resilience.

INTRODUCTION

The period (measured in time units or in the number of work cycles realised at a specific speed) in which bearings perform their functions in working conditions assumed by the engineer is referred to as their durability **[L. 6]**. Durability is also the resistance of an object to changes in properties characterising the quality of bearings **[L. 12]**.

Due to improvement of methods of calculating the durability of bearings, it is indispensable to precisely define conditions in which the bearings will function. Regardless of the conditions, fatigue is the basic form of wear considered in all cases.

The following two stages can be differentiated on examining the fatigue of roller bearings **[L. 11]**:

– The appearance of fatigue phenomena (weakening and strengthening of material, plastic deformation) at the local scale leading to the occurrence and development of micro-cracks; and,
– The occurrence and development of micro cracks leading to damages.

The fatigue crack appearance process depends on the bearing load and the quality of lubricant **[L. 2]**. Depending on the type of load, the stages occur throughout the whole element or only in its surface layer. The share of the enumerated stages in the total fatigue varies and depends on the following **[L. 10]**:

– The type of bearing,
– The structural features of the elements in the bearing, and
– The level of load.

In all basic methods applied for calculating fatigue life, stress is the value indicating the level of the material effort, which determines durability.

The criteria most commonly used for calculations of roller bearing durability are based on the analysis of the state of tensions and deformations as the result of load.

Within the complete engineering activity, i.e. in the phases of designing, constructing, producing, and usage, it is possible to take decisions concerning structural properties determining the application features of particular elements as well as of whole products **[L. 9]**.

The most important structural properties include the radial clearance of bearings.

According to the specialist literature, clearance influences the following:

- The distribution of radial loads on particular balls – the bigger the radial clearance, the smaller the number of balls carrying the load;
- The value of the reaction moment appearing as a result of the angular tilt of the inner ring – the smaller the clearance, the bigger the flexural stiffness of the bearing (i.e. the bigger the torque); and,
- The reactions of the supports of the roller bearings, e.g., if the radial displacement of the shaft in the middle bearing area is smaller than half of the bearing clearance, it does not cause a reaction force in the bearing.

The values of radial clearance depending on the class (group) of clearance and the inner diameter of a bearing are included in the standard ISO 5753 **[L. 5]**.

Table 1 presents the statement of standards for roller bearings.

Table.1. The value of radial clearance for ball bearings with slotted races having radial contact with a cylindrical bore [L. 5]

Tabela.1. Wartość luzu promieniowego dla łożysk kulkowych z bieżniami rowkowymi o styku promieniowym z otworem walcowym **[L. 5]**

diameter d [mm]		group 2		group N		group 3		group 4		group 5	
above	including	min. [μm]	max. [μm]	min. [μm]	max. [μm]	min. [μm]	max. [μm]	min. [μm]	max. [μm]	min. [μm]	max. [μm]
2,5	6	0	7	2	13	8	23	-	-	-	-
6	10	0	7	2	13	8	23	14	29	20	37
10	18	0	9	3	18	11	25	18	33	25	45
18	24	0	10	5	20	13	28	20	36	28	48
24	30	1	11	5	20	13	28	23	41	30	53
30	40	1	11	6	20	15	33	28	46	40	64
40	50	1	11	6	23	18	36	30	51	45	73
50	65	1	15	8	28	23	43	38	61	55	90
65	80	1	15	10	30	25	51	46	71	65	105

ASSUMPTIONS FOR THE ANALYSIS

- The following simplifying assumptions have been adopted:
- The problem is considered statically (the forces and their changes caused by rotation of the shaft and inner parts of bearings have been omitted);
- The shift load takes place in one axial plane;
- No errors in the shape of the balls or rings appear in the bearing;
- The elastic deformations of bearings occur only in the contact points of the roller parts with rings;

– The location of bearings is geometrically error free, i.e. the axes of outer rings of the bearings are placed on one line;
– The setting of bearings in supports is perfectly rigid (outer rings of the bearings do not get dislocated); and,
– The clearances associated with the fitting of the bearings are omitted.

METHODS OF CALCULATIONS

The system consisting of a shaft and three bearings is statically undeterminable. In textbooks, the problem is solved by completing the equations of the statistics with the equation of shaft elasticity. The approach applied by the author of this study has been described in the article **[L. 3]**. Besides the elasticity of the shaft, the approach takes into consideration the elasticity of bearings – the radial, axial, and flexural ones. Radial elasticity is expressed by the correlation of the radial force applied on the bearing (with subsequent radial reaction of the bearing) and radial displacement of the inner ring in relation to the outer one. Axial elasticity is expressed in a similar way. However, flexural elasticity is expressed by the correlation of the bending moment affecting the bearing (at the same time torque of the bearing) and the angular deflection of the inner ring in relation to the outer one. The radial displacement and angular deflection of the inner ring corresponds to the local deflection and bending angle of the shaft line. At the same time, not only radial and axial reaction, but the reaction time of each bearing is included in the equations of the system statics. It is worth highlighting the feedback occurring in this system: The angles of the tilt of the bearing rings are determined by the line of deflection of the shaft, but, in turn, the deflection line is influenced by the reaction moments of the bearings, depending on the angles of rings deflection.

There exists a non-linear dependence between the angle of the tilt of the bearing ring and the reaction torque of the bearing. Together with the increase of the angle, contact stresses in the bearing increase non-linearly, despite the stability of the forces loading the bearing. This results in a rapid drop of its durability. It is practically impossible to determine elastic displacements (shifts and tilts of the rings) in a bearing based on its load. The reason is lack of knowledge about the number of loaded balls, the distribution of balls' loads, and the angles of operation of particular balls in the bearing. It is possible to apply a reverse procedure, as presented in **[L. 1]** and proposed in publication **[L. 3]**. The displacement of the inner ring in relation to the outer ring in three directions (f_x, f_y, and f_z) is presented in **Fig. 1** (where axis x is the axis of the bearing, and axes y and z are perpendicular to it), and the tilt of the inner ring in relation to the outer one in two planes (γ_y – in x-y plane and γ_z in y-z plane) are assumed. On this basis, the operation and deformation of particular balls in relation to the contact angles of particular balls with the rings are calculated.

Depending on these deformations, normal forces Q_i for all balls are specified. These forces are the basis for calculating the resultant forces F_x, F_y, and F_z and the torques of the bearing M_y, and M_z defined by Formulas 1 to 5, i.e. subsequently the outer forces and the moments affecting the bearing:

$$F_x = \sum_{i=1}^{Z} (-Q_i \sin\alpha) \tag{1}$$

$$F_y = \sum_{i=1}^{Z} (-Q_i \cos\alpha \cos\psi) \tag{2}$$

$$F_z = \sum_{i=1}^{Z} (-Q_i \cos\alpha \sin\psi) \tag{3}$$

$$M_y = \sum_{i=1}^{Z} (Q_i r_p \sin\alpha \sin\psi) \tag{4}$$

$$M_z = \sum_{i=1}^{Z} (Q_i r_q \sin\alpha \cos\psi) \tag{5}$$

$$r_p = 0.5d_m - 0.5d_k - 0.25\Delta_r + r_i \tag{6}$$

$$r_q = 0.5d_m + 0.5d_k + 0.25\Delta_r - r_o \tag{7}$$

where

- α – operating angle (load) of the ball in the axial plane of the ball,
- ψ – nominal angle of the ball position in the frontal plane of the bearing,
- r_p – the radius of the circle on which the centres of curvature of the inner ring race are situated,
- r_q – the radius of the circle on which the centres of curvature of the outer ring race are situated,
- d_m – the average diameter of the bearing ; d_m = 0,5(d+D),
- d – inner diameter of the bearing,
- D – outer diameter of the bearing,
- d_k – diameter of the bearing ball,
- Δr – radial clearance in the bearing,
- r_i – transversal radius of the race in the inner ring of the bearing,
- r_o – transversal radius of the race in the outer ring of the bearing.

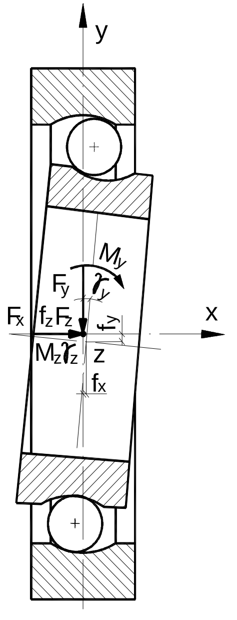

Fig. 1. Dislocations in a bearing [L. 3]
Rys. 1. Przemieszczenia w łożysku **[L. 3]**

The average load of any Q_{sr} ball is determined (averaged over its revolution around the bearing) based on the normal forces influencing all balls in the bearing at a specific moment:

$$Q_{sr} = \left(\frac{1}{Z} \sum_{i=1}^{Z} Q_i^{\;3} \right) \tag{8}$$

where z – the number of balls in the bearing,

and thus a substitute load of the bearing P:

$$P = Z \frac{J_r(\varepsilon)}{J_1} Q_{sr} \tag{9}$$

where $J_r(\varepsilon)$ i J_1 – Sjövalla integrals, defining the relationship between the outer loads of the bearing and the load of the most loaded ball. In accordance with [8], in order to make durability calculations based on dynamic load capacity of the bearing, the value of integrals corresponding to the angle of load is assumed to be $2\psi = \pi$ **(Fig. 2)**, $J_r(\varepsilon) = 0.2288$, $J_1 = 0.5625$.

The load capacity of the bearing L_{10} (in millions of rotations) is determined using the following well-known relation:

$$L_{10} = \left(\frac{C}{P}\right)^3 \tag{10}$$

where C – dynamic load capacity of the bearing.

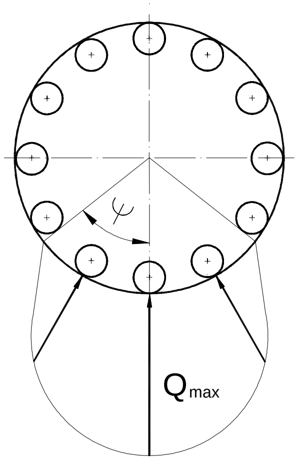

Fig. 2. Distribution of load on the rolling elements in the radial bearing [L. 4]
Rys. 2. Rozkład obciążenia na elementy toczne w łożysku promieniowym **[L. 4]**

In this way, durability has been expressed based on the actual load of the balls in the bearing. For comparison, the calculation of nominal durability with the catalogue method recommended by ISO (based on substitute load determined from the radial force F_r and axial force F_a) has been added according to **[L. 7]**:

$$P = XF_r + YF_a \tag{11}$$

where X – lateral load factor,
Y – longitudinal load factor.

The catalogue method is characterized by not taking into consideration the dislocation and tilt of the inner ring in relation to the outer one, as well as the reaction torques of bearings. In addition, the forces in bearings are dependant only on the load and the shape of shaft. This determined durability is appropriate only for a "rigid" bearing.

For the needs of the analysis, the examples of bearings as in **Figs. 3** and **4** have been adopted.

The following bearing variations have been assumed for Example I:

- Variation I: bearing A, B and C are the bearings in the same group of clearances (group 2) with radial clearance $\Delta r = 0.001$ mm;
- Variation II: bearings A and B are in the same clearance Group 2 with the radial clearance $\Delta r = 0.001$ mm, and bearing C is in clearance Group 5, having the radial clearance $\Delta r=0.073$ mm;
- Variation III: bearings A and C are in clearance Group 2 with the radial clearance $\Delta r = 0.001$ mm, and bearing B is in clearance Group 5 with the radial clearance $\Delta r = 0.073$ mm;
- Variation IV: bearings B and C are in clearance Group 2, with the radial clearance $\Delta r = 0.001$ mm, and bearing A is in clearance Group 5 with radial clearance $\Delta r = 0.073$ mm.

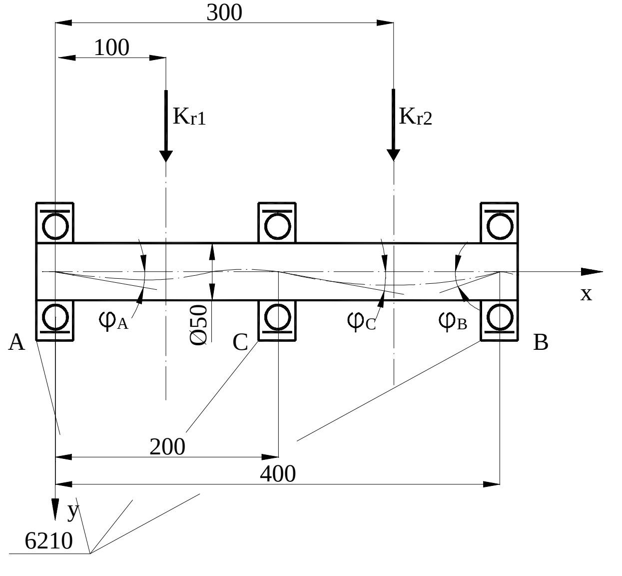

Fig. 3. Example I of a bearing adopted for the analysis
Rys. 3. I. przykład łożyskowania przyjętego do obliczeń

The load of the shaft with a torque of 200000 Nmm are the radial forces K_{r1} and K_{r2}, applied in two points, with the values $K_{r1} = 1000$ N and $K_{r2} = 1500$, 2000, 2500, 3000 N.

In the second analysed example, the shaft loads are the radial forces K_{r1} and K_{r2}, and their values are respectively $K_{r1} = 200$ N, $K_{r2} = 0$, 200, 400, 600, 800, 1000 N. Torque of the shaft is 60000 Nmm. The bearings are in clearance Group N with the following values: bearing A and B: $\Delta r=0.006$ mm, and bearing C: $\Delta r = 0.023$ mm.

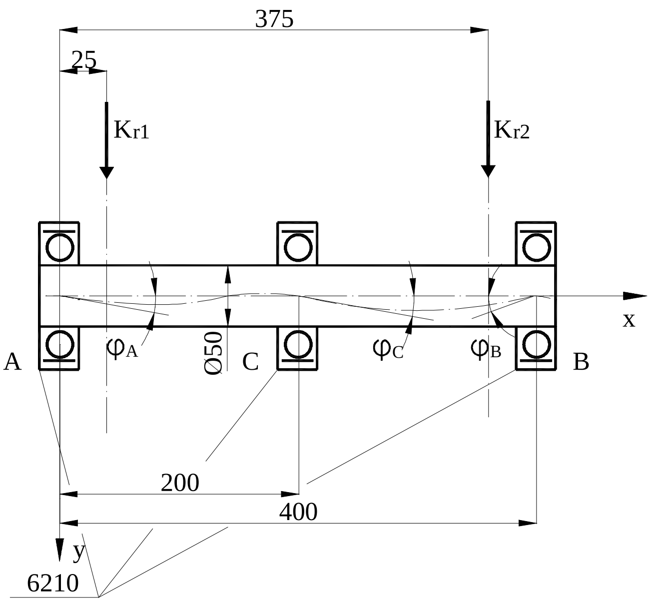

Fig. 4. Example II of a bearing adopted for the analysis
Rys. 4. II. przykład łożyskowania przyjętego do obliczeń

The durability of bearings for Example I for particular variations of bearings for each case of loading is presented in **Fig. 5**. **Fig. 6** refers to Example II for the clearance values accordingly assumed for the calculations.

Based on the above examples of bearings, the influence of poorly chosen clearances that impairs the operation of the whole system of bearings can be determined. By analysing the graphs **(Figs. 5** and **6)**, a complex relation in the cooperation of all bearings can be noticed. When applying a bearing with excessive backlash in comparison with the backlash in the other bearings, a situation can be caused when this particular bearing is not loaded, whereas the load it should carry is carried by the remaining bearings. The fourth variation of the bearing of Example II assumed for calculations is an illustration of that situation. It can also be observed in Variation II for this example of a bearing that, until a particular moment, the bearing B is not loaded. The loading of the bearing occurs only at the force $K_{r2} \approx 2400$ N.

The second example of a bearing accepted for calculations is more probable, because commonly used bearings are used in it (the bearings in the backlash Group N).

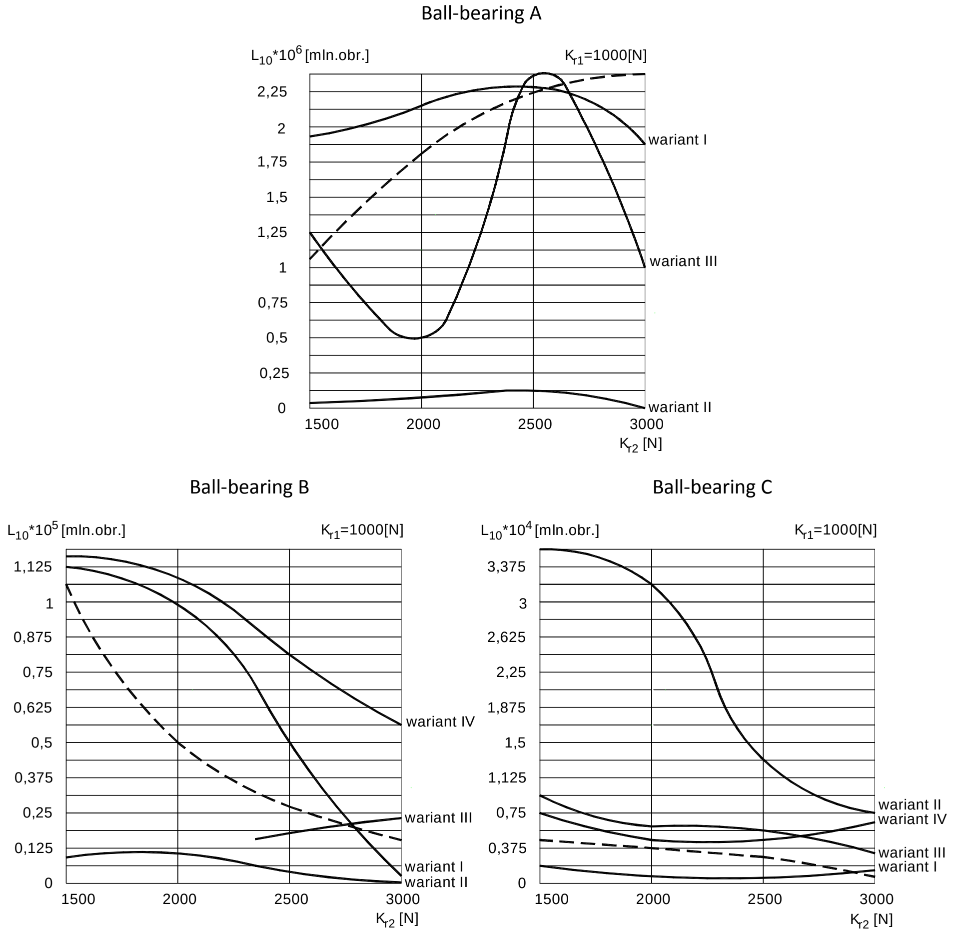

Fig. 5. Durability L_{10} of ball-bearings for the Example I. Continuous line – according to the author's calculation, dashed line – according to catalogue calculations

Rys. 5. Trwałość L_{10} łożysk kulkowych dla I. przykładu. Linia ciągła – wg obliczeń własnych, linia kreskowa – według obliczeń katalogowych

In this case, it is also possible mismatched bearing the same group clearances that in certain situations it is not possible to load one of the bearings. In the adopted bearings system, bearing C is that bearing, for the Kr2≈760 N is not loaded.

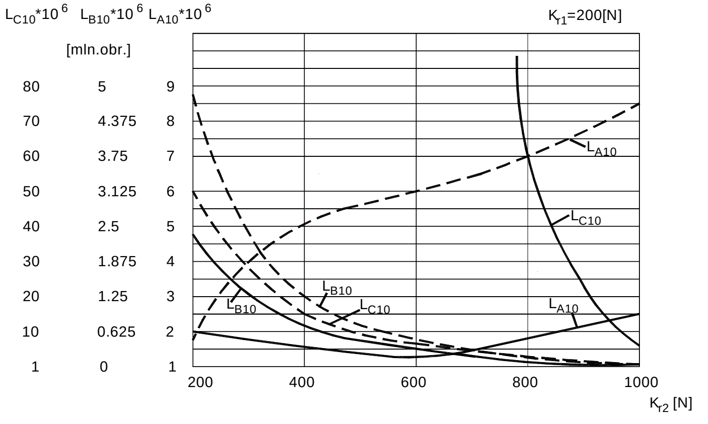

Fig. 6. Durability L_{10} of ball bearings for Example II. Continuous line – according to the author's calculation, dashed line – according to catalogue calculations

Rys. 6. Trwałość L_{10} łożysk kulkowych dla II. przykładu. Linia ciągła – wg obliczeń własnych, linia kreskowa – według obliczeń katalogowych

CONCLUSIONS

Based on these results it can be concluded that the method list, based on the standard ISO 281, is a method of simplified and does not account for the impact of many factors, such as looseness in the bearings, bearing displacements and internal forces in the bearings on bearing life. In fact, these factors have a significant impact on bearing life, resulting in an increase or decrease bearing life and, in extreme cases, cause a malfunction of the bearings.

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Streszczenie

Trwałość łożysk kulkowych zwykłych zależy od czynników (tzw. cech) konstrukcyjnych, technologicznych i eksploatacyjnych. Wśród cech konstrukcyjnych jedną z najistotniejszych jest luz w łożyskach. Polska Norma wskazuje 5 grup luzu w łożyskach, w których wartości luzu wahają się od 0 do 105 µm.

Producenci łożysk tocznych podają tylko grupę luzu, jaką posiada dane łożysko, bez podawania dokładnej wartości luzu. Celem pracy jest określenie, jak luz w łożyskach wpływa na pracę łożysk w układzie wału trzypodporowego, z uwzględnieniem sprężystości wału trzypodporowego i sprężystości łożysk.