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THE SELECTION OF PRELOAD IN ANGULAR CONTACT BALL BEARINGS ACCORDING TO THE DURABILITY CRITERION

DOBÓR NAPIĘCIA WSTĘPNEGO W ŁOŻYSKACH KULKOWYCH SKOŚNYCH WEDŁUG KRYTERIUM TRWAŁOŚCI

Key words: angular contact ball bearings, selection of preload, basic rating life, durability of bearing, durability criterion.

Abstract The regulation of the preload of angular contact ball bearings is mentioned in specialist literature, but only as a practical activity carried out based on experience, still without selection methodology. At present, there exists no widely accessible method of this load selection. Neither the literature on the subject nor catalogues of bearings producers give recommendations for the selection of this parameter, which is most often perceived intuitively. Still, it is known that its wrong selection can have a catastrophic influence on the durability of bearings. In consequence, there exists a problem of the accurate choice of preload (i.e. of such a preload that the durability of bearings will not be significantly decreased).

The aim of the study is to determine the maximum preload in angular contact ball bearings by the criterion of durability, taking into account shaft flexibility and the flexibility of bearings.

In order to determine a general influence of preload on fatigue life of bearings, numerous calculations were made for different locations of the plane of load.

Słowa kluczowe: łożysko kulkowe skośne, dobór napięcia wstępnego, trwałość łożyskowania, kryterium trwałości.

Streszczenie Regulacja napięcia wstępnego łożysk kulkowych skośnych jest wzmiankowana w literaturze przedmiotu, ale tylko jako działanie praktycznie stosowane w oparciu o doświadczenie, natomiast bez metodyki doboru. Obecnie nie istnieje powszechnie dostępna metoda doboru tego napięcia. Literatura fachowa ani też katalogi producentów łożysk nie podają dokładnej metody doboru tego parametru. Jest natomiast wiadome, że błędny jego dobór może katastrofalnie zaważyć na trwałości łożysk. W związku z tym istnieje problem trafnego doboru napięcia wstępnego (tzn. takiego, że nie będzie istotnie zmniejszona trwałość łożysk).

Celem tej pracy jest określenie maksymalnego napięcia wstępnego w łożyskach kulkowych skośnych, kierując się jego trwałością, z uwzględnieniem sprężystości wału oraz sprężystości łożysk.

INTRODUCTION

Adjusting the preload of angular contact bearings is mentioned in the literature, but only as practicable action based on experience, but without the methodology of selection. Currently, there is no widely available methodology for the selection of this tension. The scientific literature or directories from bearing manufacturers do not provide guidance on the selection of this parameter, and it is usually selected intuitively.

Preload can be expressed as a force or as a displacement (distance), and though it is the force of preload that performs the part of the basic parameter in specification **[L. 9]**. Depending on the regulation method, the preload is also indirectly associated with the moment of friction in a bearing.

The optimum values of preload can be obtained from proven constructions, and then applied in similar constructions **[L. 1, 9]**. In case of new constructions, it is recommended to calculate the force of preload and to check the accuracy of the calculations by means of experiments. In practice, it may be necessary to introduce corrections, because not all real work parameters can be known in detail. The credibility of calculations primarily depends on the extent to which calculations made in relation to temperature conditions during work and

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elastic deflections of cooperating elements (first of all of a fitting) are in accordance with the real conditions.

As given in **[L. 8],** when determining a preload, one has to calculate the aimed preload force assuring an optimum combination of rigidity, exploitation time, and the reliability of the work of the bearing in the first place. Then, the preload for application during setting and assembling of the bearings has to be calculated. During the assembling, the bearings should have the temperature of the environment and cannot be loaded with a working load.

As given in **[L. 1],** the preload of bearings may be necessary in order to compensate for the clearance resulting from the elastic deflection of the bearing overtaking axial load and in order to achieve better distribution of loads in the second bearing (not axial-loaded). The preload leads to increasing the rigidity of the bearing system. When considering the issue of rigidity, it has to be remembered that it is influenced not only by the resilience of the bearing, but also by the elasticity of shaft and its fitting, the fitting of the rings, and the elastic deformation of all elements in the field of forces, including the resistance elements. All these factors have great influence on the elasticity of the whole system of the shaft.

According to **[L. 1],** the axial (longitudinal) and radial (transversal) elasticity of a bearing depends on its internal structure, i.e. on the conditions of contact (point-based or linear), the number and diameter of roller elements, and the angle of operation. The larger the angle of operation, the higher is the stiffness of the bearing in the axial direction. If the first assumption is the linear relation between the load and flexibility, i.e. a constant coefficient of flexibility, the comparison shows that axial shift in the system of bearings with preload is lower than in a system of bearings without preload for the same external axial force F_x (**Fig. 1**).

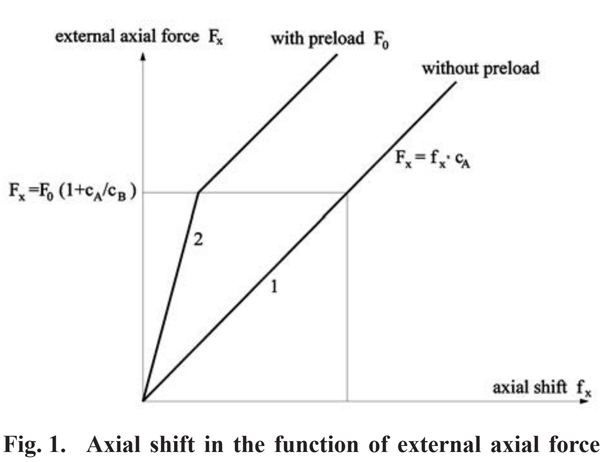


Fig. 1. Axial shift in the function of external axial force [L. 9]

Rys. 1. Przemieszczenie osiowe w funkcji zewnętrznej siły osiowej **[L. 9]**

Past calculations of the durability of angular contact ball bearings do not include the effect of preload and the bearings are treated as perfect rigid support of an articulated character.

Besides shaft flexibility, the flexibility of bearings, radial, axial, and flexural elasticity have to be considered in calculations of the durability of bearings. Radial elasticity is expressed by the interdependence of a radial force influencing the bearing (and at the same time radial reaction of the bearing) and radial displacement of the inner ring in relation to the outer one. Axial elasticity is expressed analogously; whereas, flexural resilience is expressed by the dependence of the bending moment generated in the bearing (i.e. the reaction moment of the bearing) on the angular deflection of the inner ring in relation to the outer one. Radial displacement and angular deflection of the inner ring corresponds to the local deflection and the angle of deflection of the shaft line. At the same time, not only the radial and axial reaction, but also the reaction moments of each bearing are taken into consideration in the equations of the statics of the system. It is worth highlighting the feedback in the system: Radial displacements in bearings correspond to shaft deflections on supports; whereas, the angles of the deflection of the rings of bearings are determined by the line of shaft deflection. This deflection line, in turn, is influenced by the reaction moments of bearings depending on the angles of rings deflection.

A method of determining the maximum preload for angular contact roller bearings is presented below.

CALCULATION METHOD

When solving the problem, the following issues were put together:

1) The line of deflection of the machine shaft with a complex external load,
2) Displacements of inner bearing rings as a result of loads and as a result of preload,
3) Elastic contact deformations at the contact of the rolling parts and the race in both the bearings of the system,
4) Calculations of contact forces in bearings on the basis of contact deformations,
5) The balance between internal (contact) forces in bearings and the external load of the whole bearing system, and
6) Calculating durability of bearings basing on the contact forces.

Adopting a calculation model is a decisive step on the way to theoretical solving of a scientific problem. It influences the degree of approximation to reality, the range of considered phenomena, but also the workload involved in the solution. In the publications on the phenomena taking place in roller bearings, calculation models are usually not revealed in detail. This study uses a modelling method developed by the authors and applied in the studies **[L. 2–5, 7, 8]** and other papers.

Figure 2 illustrates a half section of an angular contact ball bearing. In Part (a) of the image, a state without load is illustrated. In this state, a ball is adjacent to both the rings without deep deformation. The centre of the curvature of the inner ring race is located in Point P; whereas, the centre of curvature of the outer ring race is located in Point Q. The distance between these centres equals A.

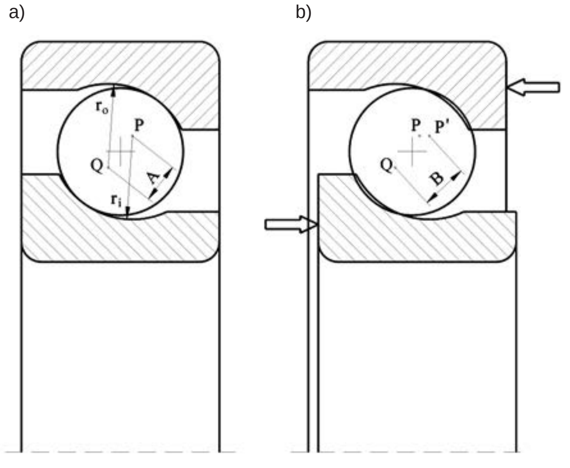


Fig. 2. Displacement of the ring as a result of load: a) a state without load, b) a state after the axial load of the bearing
Rys. 2. Przemieszczenie pierścienia wskutek obciążenia: a) stan bez obciążenia, b) stan po obciążeniu osiowym

Part (b) of the illustration presents the state after the axial load of the bearing (of course exaggerated). The inner ring was dislocated in relation to the outer one, and the centre of curvature of the race moved to point P'. Now a circle symbolising a ball penetrates the profiles of both rings at a certain depth, which illustrates contact deformations. The sum of normal contact deformations equals the difference of lengths of the sections P'Q and PQ.

$$\delta = B - A = P'Q - PQ \qquad (1)$$

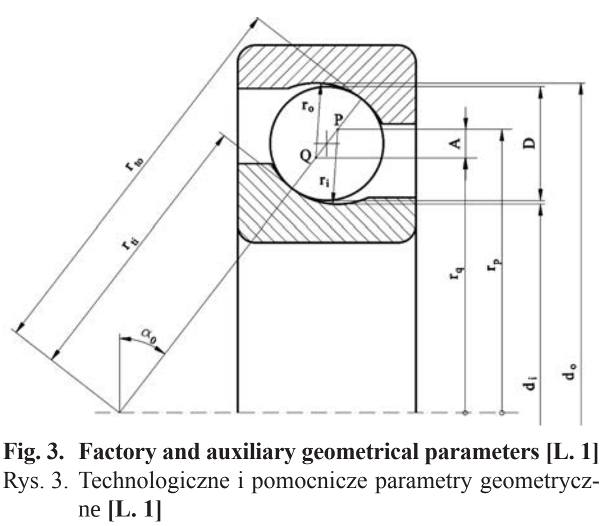


Fig. 3. Factory and auxiliary geometrical parameters [L. 1]
Rys. 3. Technologiczne i pomocnicze parametry geometryczne **[L. 1]**

Upon this rule, calculating the sum of normal contact deformations in a general case was defined, i.e. for a randomly located ball with a complex displacement of one ring in relation to the other. Before describing this special analysis, it is necessary to specify certain geometrical parameters. They result from the main dimensions of working surfaces of an angular contact ball bearing, in accordance with **Fig. 3**.

This illustration presents a bearing in a nominal state (with no load). Dimensions D (the diameter of a ball in the bearing), d_i (the diameter of the inner ring), d_o (the diameter of the race in the outer ring), r_o (the radius of curvature of the outer ring race), and r_i (the radius of curvature of the inner ring race) are factory dimensions, given in the executive documentation of bearings. On this basis, the following was determined:

– The radii of the circles OK_p and OK_q, on which points P and Q (**Fig. 4**) are located (the circles are geometrical places of the centres of curvature of the inner r_p and outer ring race r_q):

$$r_p = 0{,}5\, d_i + r_i \qquad (2)$$

$$r_q = 0{,}5\, d_o - r_o \qquad (3)$$

– The difference of the radii:

$$A = r_p - r_q = r_i + r_o - 0{,}5\left(d_o - d_i\right) \qquad (4)$$

– The nominal angle of the bearing operation:

$$a_0 = \arccos\frac{A}{PQ} = \arccos\frac{A}{r_i + r_o - D} \qquad (5)$$

– The radii of curvature of the raceway rings on the inner ring r_{ti} and on the outer ring r_{to}:

$$r_{ti} = \frac{r_p}{\cos\alpha} - r_i \qquad (6)$$

$$r_{to} = \frac{r_q}{\cos\alpha} + r_o \qquad (7)$$

The term "nominal angle of the bearing operation" means the angle that gets constituted after all elements get in contact under a minimum axial load.

Calculating deformations at a contact of a randomly located ball with bearings can be made with the use of a vector equation **[L. 6]**, but it seems more convincing to explain the procedure on a special drawing showing linear and angular displacements. Such a method is used in this proceeding and illustrated in **Fig. 4**. A starting point is the approximation of the inner ring to the outer one at the place where the ball is located. A local distancing from each other of the circles OK_p and OK_q is a measure of this approximation. It can be determined by analysing the displacement of Point P to the location P'. This displacement consists of the shifts of the inner ring towards x, y, z and its tilts in relation to the perpendicular axis towards the shaft axis, i.e. in relation to the axis y and z: displacement towards x: f_x, displacement towards y: f_y, displacement towards z: f_z, displacement resulting from the tilt in relation to the axis y: $r_p \cdot \Theta_y \cdot \sin\psi$, displacement resulting from the tilt in relation to the axis z: $r_p \cdot \Theta_z \cdot \cos\psi$.

Point P' is located in a different axial plane than Point P, because it is a subject to a certain circumferential displacement. This new plane is marked on the **Fig. 5** by the vertices of the triangle B'P'Q'. In this situation, a local distance between the circles OK_p and OK_q must be measured from Point P' to Point Q'. In order to illustrate the elements of displacement of Point P to P' in a more accessible way, an additional **Fig. 4** presents

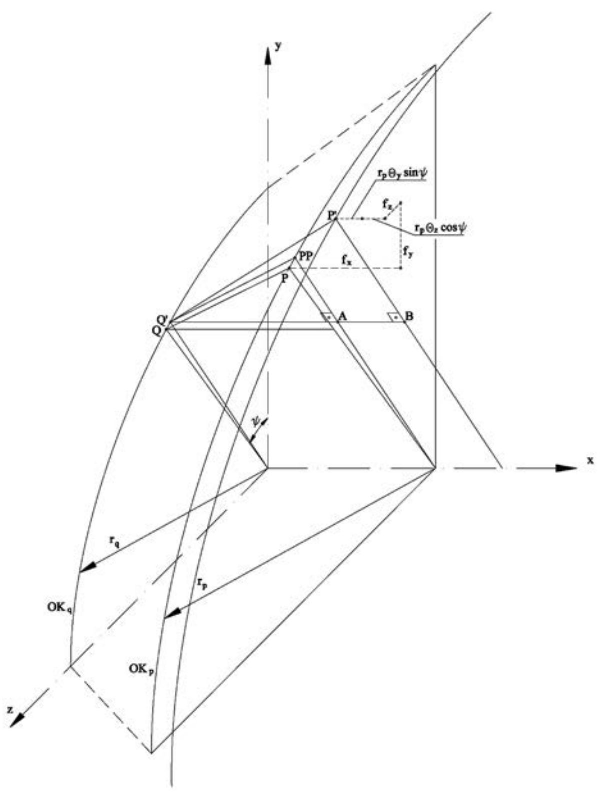


Fig. 4. Spatial illustration of the rule of calculating the local approchement of rings

Rys. 4. Przestrzenna ilustracja zasady obliczania lokalnego zbliżenia pierścieni

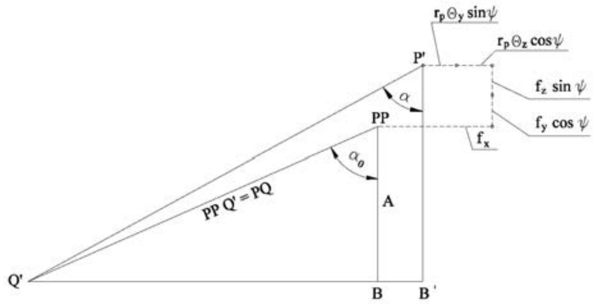


Fig. 5. Illustration in the plane B'P'Q' of the rule of local approchement of rings

Rys. 5. Ilustracja w płaszczyźnie BP'Q' zasady obliczania lokalnego zbliżenia pierścieni

the plane B'P'Q'. In this figure, Point P is replaced by point PP, located on the same circle OK_p. Displacements f_y and f_z are casted on the plane B'P'Q'. The difference between the distance P'Q' and the initial distance PP Q' determines the value of approximation of the rings δ. The distance P'Q' is specified as the hypotenuse of the triangle B'P'Q'.

The cast of the distance P'Q' on the direction x, i.e. the distance B'Q', is as follows:

$$B'Q' = BQ' + f_x - r_p \times \left(\Theta_y \sin\psi + \Theta_z \mathrm{cosn}\psi\right) \tag{8}$$

The distance BQ' in the equation (8) is as follows:

$$BQ' = \sqrt{(PQ)^2 - A^2} \tag{9}$$

Projection of the distance P'Q' on the plane y-z, i.e. the distance B'P', is (**Fig. 5**)

$$B'P' = A + f_y \cos\psi + f_z \sin\psi \tag{10}$$

Thus, the distance P'Q' is

$$P'Q' = \sqrt{\left(B'P'\right)^2 + \left(B'Q'\right)^2} \tag{11}$$

Whereas, in accordance with the **Fig. 2**, the distance between Points P and Q (tantamount to the distance PP Q') is

$$PQ = r_i + r_o - D \tag{12}$$

When knowing the dimensions PQ and P'Q', it is possible to calculate local approximation of rings, and thus the sum of deformations at the contact of a selected ball with the two rings, with the use of the relation (1):

$$\delta = P'Q' - PQ \tag{13}$$

At the same time, a local real angle of operation of bearing α, more precisely, the real angle of operation of the considered ball is calculated as follows:

$$\sin\alpha = \frac{B'Q'}{P'Q'} \tag{14}$$

$$\cos\alpha = \frac{B'P'}{P'Q'} \tag{15}$$

$$\tan\alpha = \frac{B'Q'}{BP'} \tag{16}$$

Knowledge of the real angle of operation of the considered ball allows one to define the radii of curvature of the raceway rings on the inner ring r_{ti} and on the outer ring r_{to}:

$$r_{ti} = \frac{r_p}{\cos\alpha} - r_i \tag{17}$$

$$r_{to} = \frac{r_q}{\cos\alpha} + r_o \tag{18}$$

For calculating the force acting on the ball, it is necessary to know Hertz parameters of contact, which are the sum of curvatures in the contact point and non-dimensional parameter of deep deformation δ^*. Because the radii of the area of the inner and outer rings are usually different, Hertz parameters can be treated as different for the contact point ball 0 inner ring and ball – outer ring.

ANALYTICAL CALCULATIONS

Angular contact ball bearings of the basic type, i.e. in the series 72.B, were the subject for calculations. Four sizes of bearings were selected, that is 7206B, 7209B, 7212B, 7224B. They are characterised by the diameter of the aperture, respectively, being 30, 45, 60, and 120 mm. For calculation purposes, the dimensions of working areas of the bearings are needed. They were adopted following the archive documentation CBKŁT. No contemporary data were available (producers of bearings do not reveal the dimensions, treating the knowledge as a production secret), but current deviations from these archive data are not major (measurements of bearing are unchangeable) Thus, the possible differences cannot affect the quality of the calculations results. Dimensions of working surfaces of selected bearings are presented in Table 1. In the last line, the dynamic load of the bearings is presented following **[L. 10]**.

For the selected bearings, simplified (model) shafts were designed, the dimensions of which are determined in accordance with **Fig. 6** and given in **Table 2**. It was assumed that, in each calculation case, a shaft is supported by two identical angular contact ball bearings set in the system X.

Table 1. Dimensions of working surfaces of bearings adopted for calculations
Tabela 1. Wymiary powierzchni roboczych łożysk przyjętych do obliczeń

Bearing →	7206B	7209B	7212B	7224B
D [mm]	9.525	12.700	15.875	30.162
d_i [mm]	36.387	52.181	68.976	137.058
d_o [mm]	55.636	77.850	101.059	198.017
r_i [mm]	4.900	6.540	8.180	15.530
r_o [mm]	5.000	6.670	8.330	15.840
The number of rolling elements Z	13	14	15	15
Dynamic load C [N]	20400	37700	57200	165000

The shape of the shaft was selected following a typical shape of shafts in which bearings are located on the ends of the shaft, and diameters at particular sections are the same as in theoretical outline, built upon the rule of equal resistance to bending. In technical reality, there exist an infinite number of types of shaft shapes, but this one was considered the most representative.

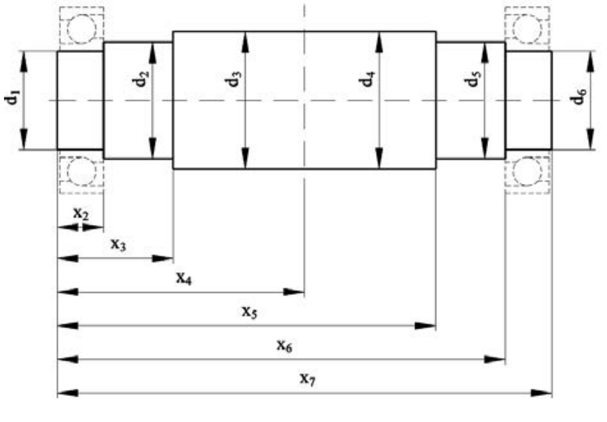


Fig. 6. Draft of a model shaft
Rys. 6. Szkic modelowego wału

For all the model shafts, the coordinate of the shaft starting place x_1 was adopted to be zero.

Table 2. Measurement parameters of model shafts [mm]
Tabela 2. Wymiary modelowych wałów [mm]

Bearing →	7206B	7209B	7212B	7224B
x_2	16	19	22	40
x_3	50	75	100	200
x_4	100	150	200	400
x_5	150	225	300	600
x_6	184	281	378	760
x_7	200	300	400	800
d_1	30	45	60	120
d_2	35	52	67	135
d_3	40	60	75	150
d_4	40	60	75	150
d_5	35	52	67	135
d_6	30	45	60	120

The adopted model shafts were loaded with a balance of forces as presented in **Fig. 7**. The locations of the planes of loads x_L were adopted in an established relation to the length of shaft L_w: $x_L = 0.5 \cdot L_w$.

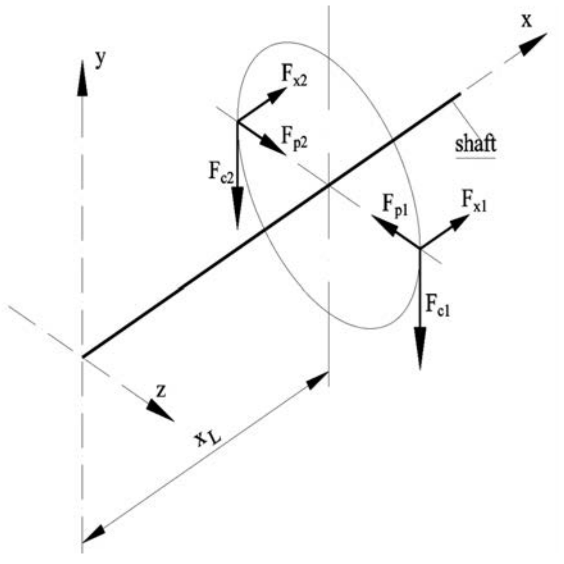


Fig. 7. Adopted variations of loading of the bearing system

Rys. 7. Przyjęty wariant obciążeń łożyskowania

The values of loads were assumed in specific relations concerning the dynamic load of bearings C. Circumferential force on the assumed gear wheel F_{c1} was established on the level equalling respectively: 0.075 C; 0.1 C; 0.125 C, while it was assumed that $F_{c1} = F_{c2}$, $F_{p1} = F_{p2}$, and $F_{x1} = F_{x2}$. Radial force F_p was established as approximately 0.364 of the circumferential force for the assumption that the angle of the contact of the meshing in a gear wheel is approximately 20º. The axial force F_x was adopted in five values: 0, $0.049 \cdot F_c$, $0.098 \cdot F_c$, $0.196 \cdot F_c$, and $0.392 \cdot F_c$. These forces are established in relations to circumferential force.

In the first place, the influence of preload on the durability of the Bearings “A” and “B” was determined where it was stated that the durability of bearing decreases when the one for the other bearing increases. It could not be concluded without doubt from the observation what value of preload is optimum for a system of bearings. Thus, the characteristics of W_T indicator, connecting together the durability of both bearings were introduced **[L. 3]**:

$$W_T = \frac{L_{hA}}{L_{hA0}} \times \frac{L_{hB}}{L_{hB0}},$$

where

L_{hA} – fatigue life of the bearing A determined in specific conditions with applied preload,

L_{hB} – fatigue life of the bearing B determined in the same conditions with applied preload,

L_{hA0} – fatigue life of the bearing A determined in specific conditions without preload,

L_{hB0} – fatigue life of the bearing B determined in the same conditions with without preload.

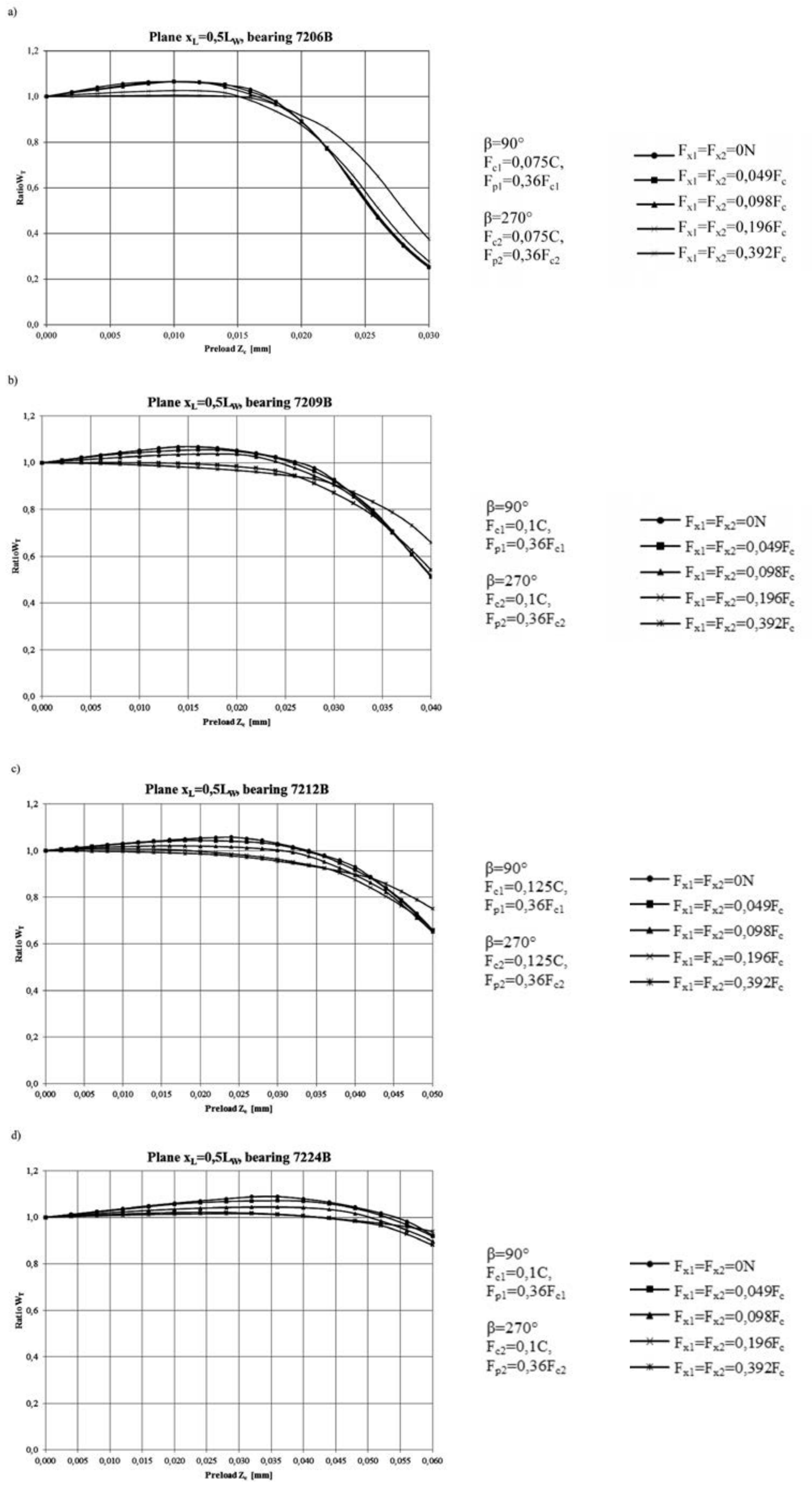
Exemplary characteristics W_T for bearings adopted for calculations and for the adopted values of load are presented in **Fig. 8**.

In order to determine a beneficial range of preload, summary graphs, based on agreed boundary points, were developed. The agreed boundary points were the values Z_c, for which characteristics of W_T indicator have the value 0.98. It was assumed that, when aiming at increasing a longitudinal stiffness of a bearing, the drop of durability indicator of 2% is acceptable.

Summary graphs of the limit load are presented in **Fig. 9**.

The following observations resulted from the characteristics proper for the central location of the plane of load ($x_L = 0.5\ L_w$, **Fig. 7**):

- As far as the bearing 7206B is concerned, it was stated that, in case of a low transverse load in a bearing (F_c = 0.075 C), the boundary value of preload is almost independent from the axial force and equals approximately 16–18 µm. With an average transverse load (0.1 C), the threshold for preload is in the range from 15 µm for the axial force equal 0.4 F_c to 21 µm with no axial force. With the highest applied transverse load (0.125 C), the limit value of preload is from 10 µm with the axial force equal 0.4 F_c to 22 µm with no axial force.
- In the 7209B bearing, a very similar courses of lines were found, with the difference of being on a proportionally higher level (around 1.5 times) than in the 7206B bearing.
- In the 7212B bearing, it was found that, in the case of a low transverse load in the bearing (0.075 C), the limit values of preload are hardly dependant on the axial force, and they equal approx. 26–28 µm (the less, the higher is the axial force),with an average 27 µm for the axial force of 0.4 F_c up to 33 µm for no axial force. For the highest applied transverse load (0.125 C), the limit value of preload is from 24 µm for the axial force 0.4 F_c to 36 µm with no axial force.
- As far as the 7224B bearing is concerned, it was found that, in each case of transverse load in the bearing, the limit values of preload barely depend on the axial force. In case of very small axial forces, the limit values of preload are within the range of 48 µm (for low relative transverse load F_c/C) to 62 µm (for high relative transverse load). For medium and high axial loads ($F_x = 0.2 \div 0.4\ F_c$), the limit value of preload is similar to all cases of relative transverse load F_c/C and is included in the range of 45–50 µm.



Rig. 8. Exemplary characteristics of W_T indicator for the adopted variation of load

Rys. 8. Przykładowe charakterystyki wskaźnika W_T dla przyjętego wariantu obciążenia

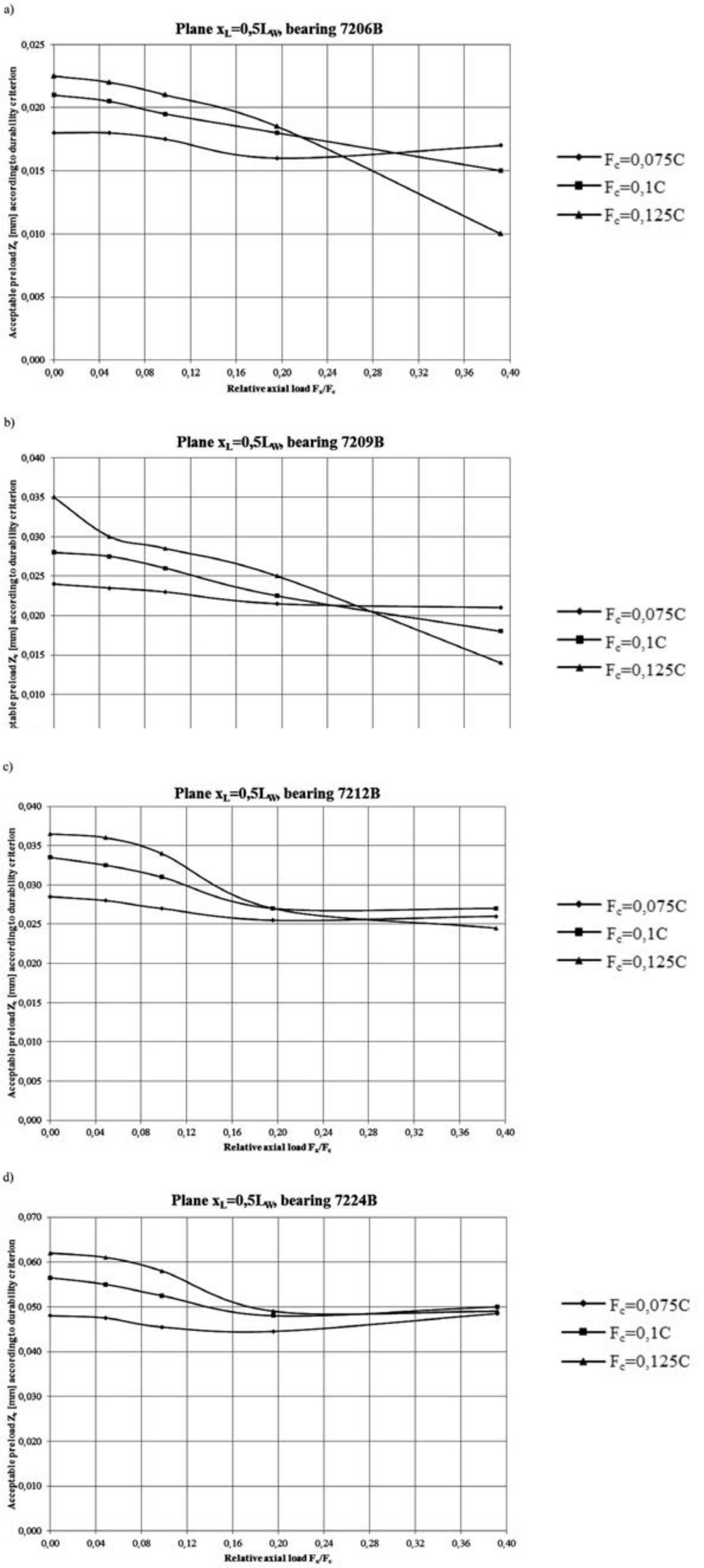


Fig. 9. Acceptable preload Z_c according to durability criterion in the function of relative axial load F_x/F_c for the adopted example

Rys. 9. Dopuszczalne napięcie wstępne Z_c wg kryterium trwałości w funkcji względnego obciążenia osiowego F_x/F_c dla przyjętego do analizy przykładu

SUMMARY

In order to determine a general influence of preload on fatigue life of bearings, numerous calculations were made for different locations of the plane of load. After the analysis, according to the earlier described method, the following conclusions can be made:

1. Applying preload is beneficial in the situation when transverse loads of both identical bearings in a system are similar or identical, which is illustrated in Fig. 10. The limit value of preload is then barely dependant on the transverse and axial load in a bearing, and it does not depend on the direction of axial force.

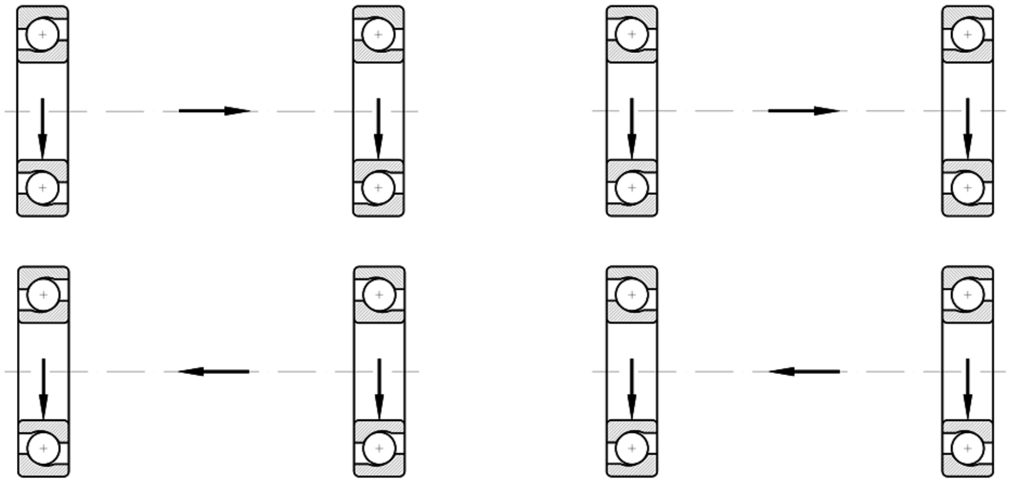


Fig. 10. The cases of load most beneficial for the use of preload
Rys. 10. Przypadki najkorzystniejszego obciążenia dla stosowania napięcia wstępnego

Taking into consideration the criterion $W_T \geq 0.98$, the following limit values of preload for the four studied bearings were read from the characteristics made by the authors (**Fig. 5**):

Bearing	7206B	7209B	7212B	7224B
Inner diameter d [mm]	30	45	60	120
Border preload [μm]	11	14	18	30

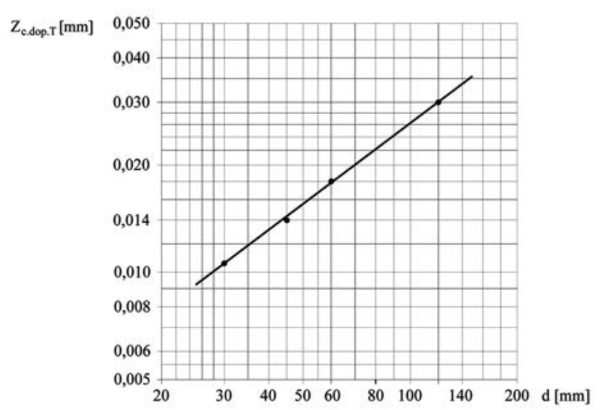


Fig. 11. Dependence of the acceptable preload on the diameter of the bearing in case of similar transverse loads in both the bearings according to durability criterion
Rys. 11. Zależność dopuszczalnego napięcia wstępnego od średnicy łożyska w wypadku zbliżonych obciążeń poprzecznych w obydwu łożyskach wg kryterium wskaźnika trwałości

Out of the obtained values, a graph Z_c was built in the inner function of the radius of the bearing, presented in **Fig. 11**. It is easy to calculate that the straight line presented in the graph represents the proportionality of Z_c to the diameter "d" in the power of 0.75.

2. If transverse loads of bearings vary, and the resultant of axial forces loads the bearing that is less loaded transversally, either in X system or in the "O" system, as presented in **Fig. 12**, the application of preload leads to significant decrease of durability of B in comparison with the state before introducing the preload.

An indicator of durability of bearing drops drastically in this case, yet this does not have to exclude the application of preload. Thus, if identical bearings are used in the described situation, the durability of the bearing B decreases from a much higher level than durability of the bearing A, because the transverse load of bearing B is much lower.

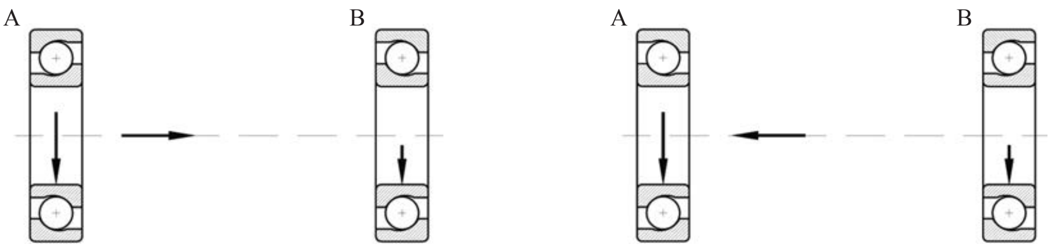


Fig. 12. Cases of loading by axial force of the bearing less loaded transversally
Rys. 12. Przypadki dociążania przez siłę osiową łożyska mniej obciążonego poprzecznie

3. If transverse loads of the bearings are varied and the resultant of axial forces loads the bearing more transversally, either in the system X or in the system "O", as in **Fig. 13**, the application of preload leads to significant lowering of A bearing durability in comparison with the state before introducing the load.

Similarly to the previously discussed case, the indicator of the durability of a bearing drops drastically, yet this does not have to exclude the use of preload. In the described situation, if identical bearings are used, the durability of Bearing A decreases from a much higher level than durability of Bearing B, because the transverse load of Bearing A is much lower. In such a case, the acceptable preload could reach the value of a few dozen micrometers. Yet, such a high value of it cannot be used in real life. The above is associated with the increase of the moment of friction in the bearing system.

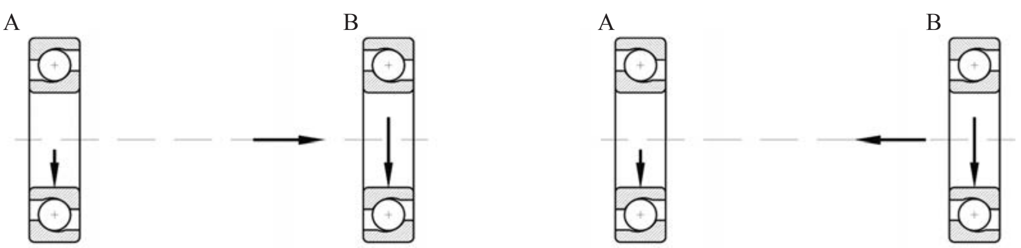


Fig. 13. Cases of loading by axial force of the bearing more loaded transversally
Rys. 13. Przypadki dociążania przez siłę osiową łożyska bardziej obciążonego poprzecznie

Keep in mind that the preload bearings is effected, e.g., increasing the temperature of the bearing, vibration, and the rigidity of the system.

The analysis of preload acceptable due to the moment of the friction of a bearing and to the rigidity of the system will be a subject of future articles.

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