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EFFECT OF TAPERED ROLLER BEARINGS PRELOAD ON BEARING SYSTEM DURABILITY

WPŁYW ZACISKU WSTĘPNEGO ŁOŻYSK STOŻKOWYCH NA TRWAŁOŚĆ ŁOŻYSKOWANIA

Keywords: tapered roller bearing, preload, shaft loading, durability of bearing, elasticity of bearing.

Abstract: Tapered roller bearings are usually used in situations in which high bearing stiffness is required. However, a significant increase in stiffness can only be achieved by introducing initial tension (mounting clamp) to the angular contact bearings arrangement. Just like in case of angular contact ball bearings and tapered roller bearings, the preload is an axial displacement of one bearing relative to the other, measured from the neutral state and causing an increase in the internal forces in the bearings. Positive preload causes contact pressures of the rolling parts to appear even when no external force acts.
The purpose of this work is to present the influence of the initial tension of a tapered roller bearings system on life of these bearings.

Słowa kluczowe: łożysko stożkowe, zacisk występny, obciążenie wału, trwałość łożyskowania, sprężystość łożysk.

Streszczenie: Łożyska stożkowe są zwykle stosowane w takich sytuacjach, kiedy potrzebne jest uzyskanie dużej sztywności łożyskowania. Jednakże znaczące zwiększenie sztywności można uzyskać dopiero dzięki wprowadzeniu napięcia wstępnego (zacisku montażowego) do układu łożysk skośnych. Identycznie jak w wypadku łożysk kulkowych skośnych, i w wypadku łożysk stożkowych, zacisk wstępny jest przesunięciem poosiowym jednego łożyska względem drugiego, mierzonym od stanu neutralnego a powodującym wzrost sił wewnętrznych w łożyskach. Dodatni zacisk wstępny wywołuje pojawienie się nacisków kontaktowych części tocznych nawet wtedy, kiedy nie działa żadna siła zewnętrzna.
Celem tej pracy jest przedstawienie wpływu napięcia wstępnego układu łożysk stożkowych na ich trwałość.

INTRODUCTION

In the work presented, two related terms are used: initial tension, which is defined as the force of the assembly interaction between the rolling parts and the bearing raceways, and preload, presented as a displacement of one bearing relative to the other, causing initial tension.

It should be noted that the adjustment of tapered roller bearings preload is mentioned in the literature of the subject but without a precise selection methodology. There is currently no widely known and available method for selecting this tension. However, it is known that a wrong selection may catastrophically influence bearings' life. Therefore, there is the problem of accurate initial tension selection.

The solution of the problem requires relating the following issues:

– The deflection line of a machine shaft under a complex external load,
– The displacements of the inner bearing rings against the outer ones due to loads and the preload,

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– Deformations in the point of contact of the rolling parts and the raceways of both of the bearings in the system,
– The calculation of contact forces in bearings based on contact deformations,
– A balance between internal (contact) forces in bearings and the external load on the entire bearing arrangement, and
– The calculation of bearings' life based on contact forces.

GENERAL INFORMATION

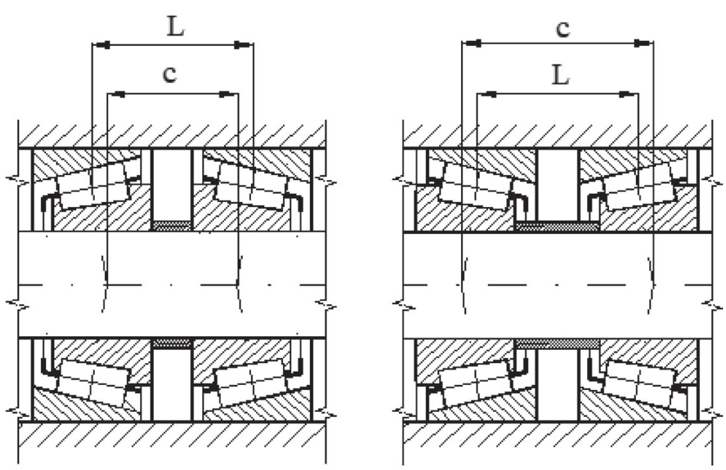
In order to solve the problem, the following issues have been connected:

Depending on the bearing type, the initial tension can be radial or axial. For instance, cylindrical roller bearings, due to their design, can only be subject to radial initial tension (on the principle of a tight fit of raceways and shafts). On the other hand, thrust ball bearings and thrust roller bearings can only be subject to axial initial tension. Tapered roller bearings (**Fig. 1**) are generally mounted in an arrangement of two bearings of the same type in the divergent system "O" or in the convergent system "X." This enables initial tension by axial displacement. Initial tension applies equally to the convergent and divergent systems; although, in the "O" arrangement, it is achieved by slightly different technical means than in the "X" arrangement.

The initial tension of bearings results from the introduction of a preload, which means a negative working clearance. However, both negative and positive working clearance can be found in bearings. For example, in vehicle wheel bearings, the working clearance should be positive, i.e. the bearing should have a residual clearance when running. In many other applications (e.g., machine tool spindle bearings, pinion bearings in vehicle drives, bearings of small electric motors and oscillatory mechanisms), a negative clearance is used, i.e. a preload and in result an initial tension **[L. 9]**. It is required to increase the rigidity of the bearing system, which is followed by higher motion accuracy and reducing vibration **[L. 8]**.

In analyses of a shaft supported by tapered roller bearings, a usual assumption is that the support reaction is concentrated at a nodal point of the tapered roller bearing. In the case of tapered roller bearings, the distance “c” between the nodal points is larger in the divergent bearing arrangement (**Fig. 1**) and smaller in the convergent arrangement than the distance “L” between the bearing centres. This means that bearings in the "O" divergent arrangement can take over relatively large overturning moments, even if the distance between the bearing centres is relatively small. The transversal forces resulting from the moment load and deformations in the bearings caused by them in the divergent arrangement are smaller than in the case of the convergent arrangement **[L. 2, 12]**.

According to the literature **[L. 1, 2, 8, 10]**, the main reasons for using initial tension are its following profitable effects:
– Increasing the rigidity of the bearing system,



a) “O” arrangement b) “X” arrangement

Fig. 1. A pair of tapered roller bearings in two different arrangements [L. 10]
Rys. 1. Para łożysk stożkowych w dwóch różnych układach **[L. 10]**

– Noise reduction during operation,
– An increase in the shaft guiding accuracy,
– The compensation of wear and settling processes during operation, and
– Ensuring longer service life.

ASSUMPTIONS AND METHOD

Assumptions for analysis

In the presented work, a modelling method devised by the author is described, which is similar to the method for angular contact ball bearings and used in works **[L. 3–7]**. Due to the complexity of the problem, the following most important simplifying assumptions were made:

1) The bearing material is isotropic and subject to Hooke's law.
2) The work surfaces of bearings are perfectly smooth.
3) There are no shape errors of rolling elements or bearing rings or shaft.
4) The axes of bearing outer rings are always in one straight line (geometrically flawless seating of bearings, no clearances connected with bearing fit).
5) Mass forces, cage effects, and lubricant resistance are not taken into account in rolling load analysis.
6) Tangential forces have no significant effect on elastic deformations and are ignored.
7) The pressure distribution in the point of contact of rolling elements and raceways is the same in motion as in static load.
8) Elastic deformations of bearing elements occur only in points of contact of rolling element and rings.

Calculation method

The life of the angular contact bearing system can be determined to a good approximation using the catalogue method **[L. 10]**. However, this method is not suitable for calculations which take into account the influence of preload. Therefore, in the presented work, a method based on the value of the average shaft load (rolling element) **[L. 3]** is used.

It is not possible to exactly determine forces acting in bearings if the elasticity of bearings and shaft are not taken into account at the same time. The bearings together with the shaft are a coupled system in which a shaft deflection forces an angular deflection of the bearing rings, but the angular deflection of the bearing rings causes a reaction torque in the bearings (except for self-aligning bearings). This reaction torque impacts the reduction of the shaft deflection. On all of this, a preload, i.e. a close-up of the outer bearing rings towards each other, must be applied. This clamp will manifest itself as the sum of axial deflections in both bearings, but it is not known in advance how this sum is distributed between the two bearings of the arrangement.

The influence of the following factors on the rolling element load in the bearing was therefore considered:
– Radial and axial load acting on the bearing shaft,
– Elastic deflection of the shaft causing deflection of the inner bearing rings, and
– Preload.

Figure 2 shows the set of loads taken into account. The external loads from hypothetical gear wheels (F_x, F_y) are the basis for calculating the Rx and Ry forces on the bearings. When calculating these forces, it was assumed that the bearing reactions are concentrated at the points of intersection of the lines of action of the shafts, i.e. according to the "c" dimension marked in **Fig. 1**.

The determination of forces in bearings is statically impossible. To solve it, a procedure used earlier by the author **[L. 4, 5]** was adopted. This procedure relies on iterative search for such displacements in bearings (displacements and deflections) that all equilibrium conditions are fulfilled:
– Compliance of the radial bearing reactions with the radial forces influencing the bearings according to the scheme in **Figs. 2** and **3**,
– Compliance of the sum of bearing axial reactions with the sum of external axial loads,
– The compatibility of bearings' deflection angles and shaft's deflection angles under bearings,
– The compatibility of the reaction torques arising in the bearings and the bending moments taken into account while calculating the shaft deflection line, and
– The compatibility of the sum of axial displacements in the bearings with the value of the preload.

From the knowledge of the Q forces acting between the bearing shafts and rings, it is possible to determine all bearing reactions understood in general, i.e. reaction forces in three directions of the coordinate system and reaction torques regarding the axes which are perpendicular to the bearing

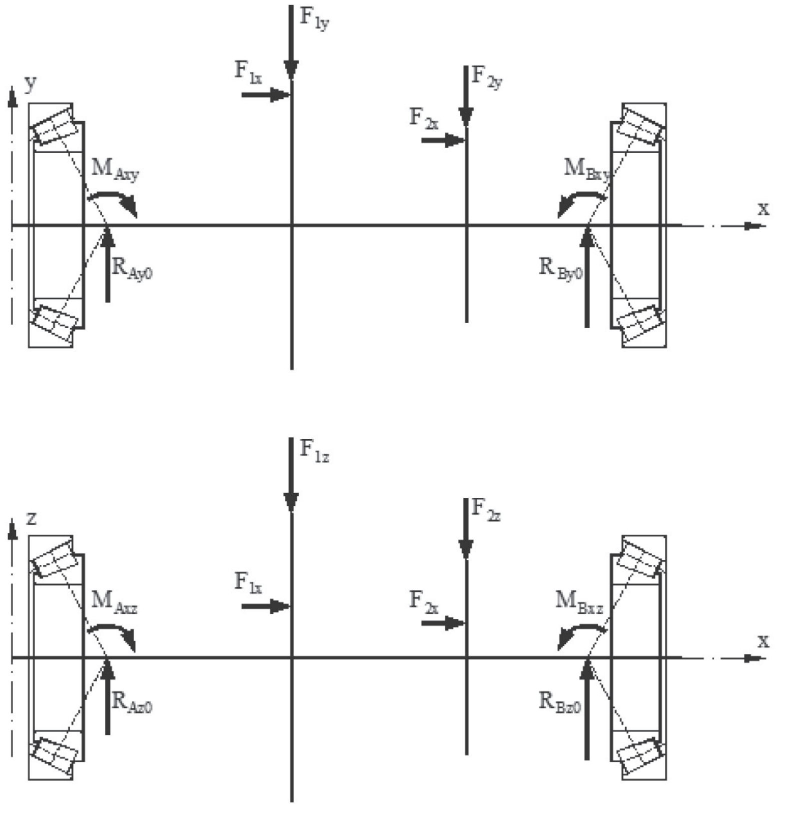


Fig. 2. Calculation scheme of transverse support reactions
Rys. 2. Schemat obliczeniowy poprzecznych reakcji podpór

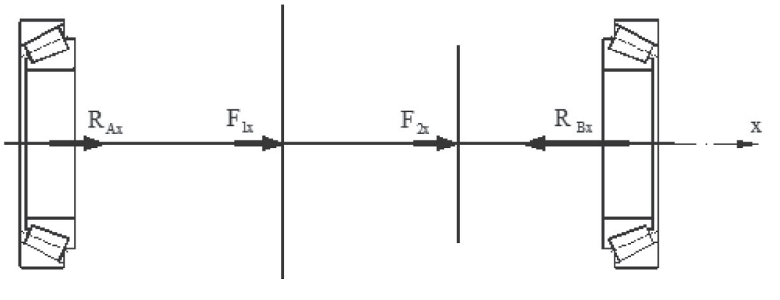


Fig. 3. Calculation scheme of axial support reactions
Rys. 3. Schemat obliczeniowy osiowych reakcji podpór

axis of rotation. Creation of reactions and marking them down is illustrated in **Fig. 4**.

In the upper part of the drawing there is shown, in perspective, an arc of the centres of shafts' points of contact with the bearing outer ring raceway. Reaction Q forces acting between the shaft and the raceway of the ring are applied to the points lying on this arc. The forces are deflected from the y-z plane by α bearing action angle. From the projection of the Q force on x-y-z directions the following formulae result:

$$Q_x = Q \cdot \sin\alpha \tag{1}$$

$$Q_r = Q \cdot \cos\alpha \tag{2}$$

$$Q_y = Q_r \cdot \cos\psi = Q \cdot \cos\alpha \cdot \cos\psi \tag{3}$$

$$Q_z = Q_r \cdot \sin\psi = Q \cdot \cos\alpha \cdot \sin\psi \tag{4}$$

The bending moment resulting from Q force is calculated in regard to the nodal point of the nominal bearing reaction, because, according to

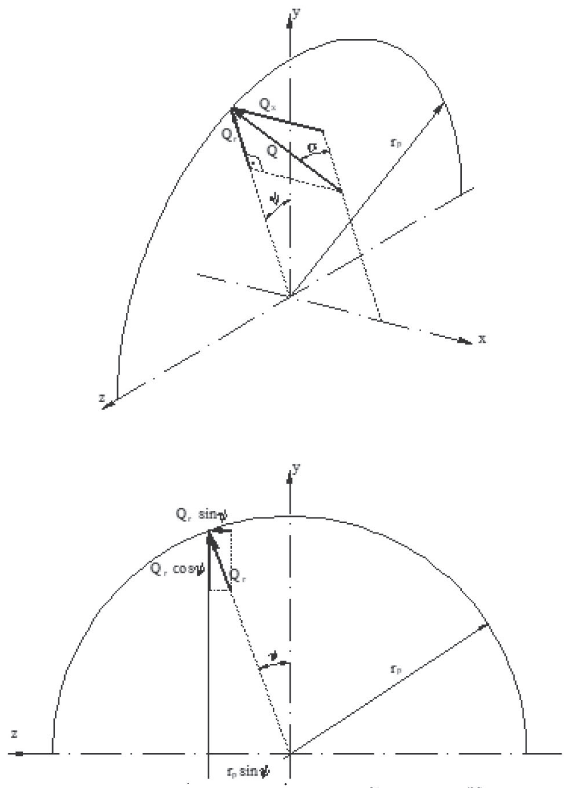


Fig. 4. Illustration of determination of bearing reactions
Rys. 4. Ilustracja wyznaczania reakcji łożyska

assumptions, this point is assumed to be the place where the shaft is supported by the bearing. This calculation is illustrated in **Fig. 5**.

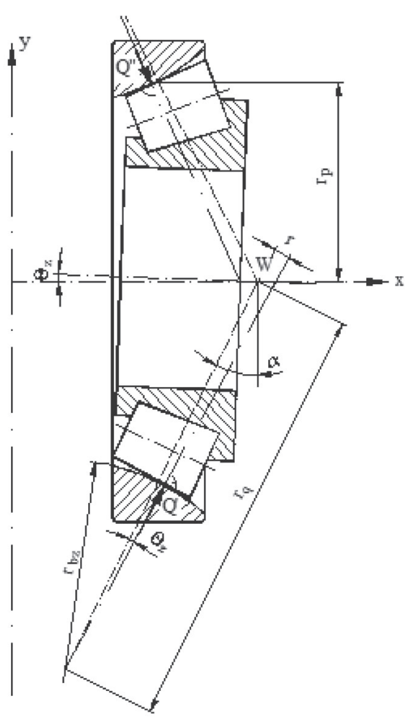


Fig. 5. Illustration of determination of bearing reaction torque
Rys. 5. Ilustracja wyznaczania momentu reakcyjnego łożyska

The nodal point is denoted by W. Q force is the reaction of the outer ring on the shaft, so it is applied at the centre of the shaft's point of contact with the outer ring raceway. The moment of Q force coming from an arbitrarily chosen shaft (here the shaft lying in the plane of the figure is chosen) is equal:

$$M = Q \cdot r$$

r arm of Q force in regard to O point is r segment. The length of this segment is

$$r = r_q \cdot \Theta_z,$$

where

$$r_q = r_{bz} + \frac{r_p}{\cos\alpha} \tag{5}$$

The r_p dimension is determined by the following formula:

$$r_p = 0,5 \cdot d_{wm} + D_{km} \cdot \cos(\alpha - \beta)$$

and r_{bz} dimension (outer ring correction radius) is the roller bearings execution parameter.

The reaction torque acting in regard to the z-axis, resulting from the tilt by an angle Θz and coming from one shaft lying in the x-y plane, is equal:

$$M_{\Theta z} = Q \cdot r_q \cdot \Theta_z$$

Shafts lying outside x-y plane due to the same Θ_z tilt generate smaller reaction torques, and, if the shaft lies in x-z plane, $M_{\Theta z}$ reaction torque is zero. This dependence is described as follows:

$$M_{\Theta zi} = Q_i \cdot r_q \cdot \Theta_z \cdot \cos\psi_i$$

The vectors of moments coming from all the shafts are directed circumferentially. Therefore, the resultant M_z moment must be determined by projecting all the individual moments onto z-axis. At the same time, from the same individual $M_{\Theta z}$ moments, the resultant M_y moment is created (with regard to y-axis), deriving from the same Θ_z tilt. This is illustrated in the first quadrant of the y-z system in **Fig. 6**.

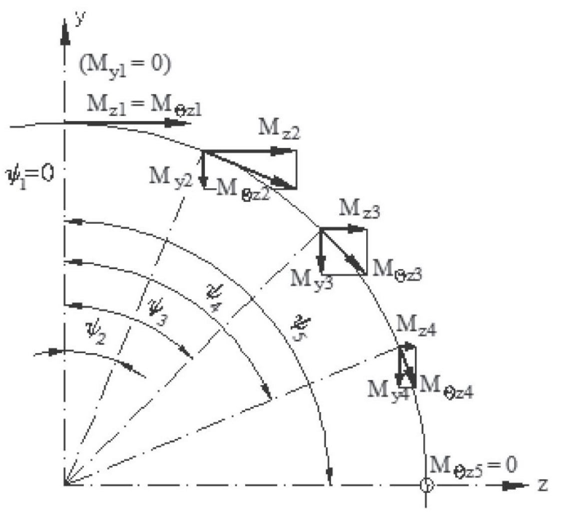


Fig. 6. Vectors of reaction torques resultant from tilt Θz
Rys. 6. Wektory momentów reakcyjnych wynikających z przechylenia Θ_z

Therefore, as a result of Θ_z tilt at the location of each shaft, identified by number "i", the following moments arise:

$$M_{zi} = Q_i \cdot r_q \cdot \Theta_z \cdot (\cos\psi_i)^2$$

$$M_{yi} = Q_i \cdot r_q \cdot \Theta_z \cdot \cos\psi_i \cdot \sin\psi_i$$

The reaction torque proceeds analogously due to Θ_y tilt. According to the illustration presented in **Fig. 7**, each shaft can be assigned to generate a moment particle of value as shown below:

$$M_{\Theta yi} = Q_i \cdot r_q \cdot \Theta_y \cdot \sin\psi_i$$

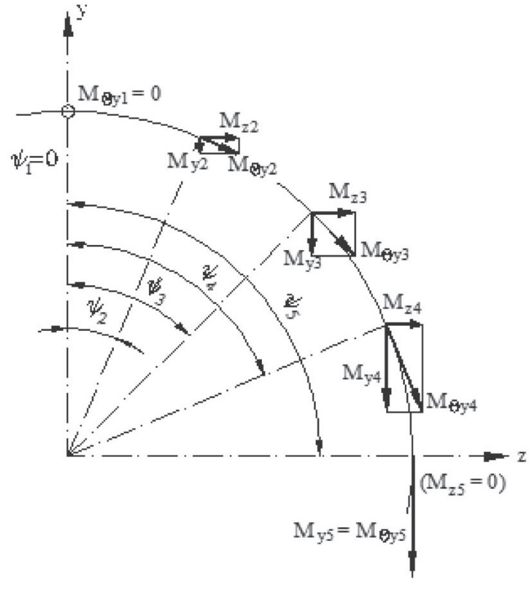


Fig. 7. Vectors of reaction torques resultant from Θy tilt
Rys. 7. Wektory momentów reakcyjnych wynikających z przechylenia Θ_y

The vectors of moments coming from all the shafts are directed circumferentially. As a result of the projection on y and z axes, two resultant moments arise: with regard to the y axis, the M_y moment and, with regard to the z axis, the M_z moment. This is illustrated in the first quadrant of y-z system in **Fig. 7**.

Thus, at the location of each shaft, identified by number "i", due to Θy tilt the following moments arise:

$$M_{yi} = Q_i \cdot r_q \cdot \Theta_y \cdot (\sin\psi_i)^2$$

$$M_{zi} = Q_i \cdot r_q \cdot \Theta_y \cdot \sin\psi_i \cdot \cos\psi_i$$

Considering the general case when bearing loads cause deflections of the shaft in both planes (i.e. Θ_y and Θ_z tilts occur simultaneously), the following expressions are obtained, describing the particles of the reaction torque, generated at the point determined by ψ_i angle:

$$M_{yi} = Q_i \cdot r_q \, [\Theta_y \cdot (\sin\psi_i)^2 + \Theta_y \cdot \sin\psi_i \cdot \cos\psi_i] \quad (6)$$

$$M_{zi} = Q_i \cdot r_q \, [\Theta_z \cdot (\cos\psi_i)^2 + \Theta_y \cdot \sin\psi_i \cdot \cos\psi_i] \quad (7)$$

Expressions (1)–(7) define the forces and moments coming from one (any) shaft. The total bearing reactions and the total bearing reaction torques are the result of the interaction of all the shafts. These quantities are calculated by summing the forces and moments caused by all those shafts which are subjected to normal deformations, i.e. are loaded with non-zero forces.

$$R_x \, \Sigma(Q \cdot \sin\alpha) \quad (8)$$

$$R_r = \Sigma(Q \cdot \cos\alpha) \quad (9)$$

$$R_y = \Sigma(Q \cdot \cos\alpha \cdot \cos\psi) \quad (10)$$

$$R_z = \Sigma(Q \cdot \cos\alpha \cdot \sin\psi) \quad (11)$$

$$M_y = \Sigma\{Q_i \cdot r_q \, [\Theta_y \cdot (\sin\psi_i)^2 + \Theta_z \cdot \sin\psi_i \cdot \cos\psi_i]\} \quad (12)$$

$$M_z = \Sigma\{Q_i \cdot r_q \, [\Theta_z \cdot (\cos\psi_i)^2 + \Theta_y \cdot \sin\psi_i \cdot \cos\psi_i]\} \quad (13)$$

The formulae presented were used to determine the reactions of both shaft bearings separately, on the basis of their separate internal deformations.

Similarly to the case of angular contact ball bearings and tapered roller bearings, the substitutive bearing load is determined based on the average load of the rolling element in its motion around the bearing axis. The method of proceeding is based on the handbook **[L. 4]** where the values of coefficients are as follows: rolling elements load distribution coefficient Jr (Sjövall integrals) Jr(0,5) = 0.2453 and load function J1(0,5) = 0.6495.

Thus, the substitutive load of a journal bearing with linear contact is calculated from the following formula:

$$P = \frac{J_{r(0.5)}}{J_{1(0.5)}} \cdot Z \cdot Q_{śr} = \frac{0.2288}{0.5625} \cdot Z \cdot Q_{śr} = 0.4068 \cdot Z \cdot Q_{śr} \tag{14}$$

The tapered roller bearing belongs to the group of journal bearings with linear contact. The fatigue strength of the raceways and shafts of a tapered roller bearing depends on their average load in the same way as in a cylindrical roller bearing, provided that the total load, not only its radial component, is taken into account. In the procedure devised, it is the total loads that are calculated. Therefore, it was assumed to use the formula (14) for the tapered roller bearing.

The same method for determining the fatigue life was used for tapered roller bearings as for angular contact ball bearings. The number of millions of revolutions the bearing is likely to complete before significant damage to the raceway or rolling parts occurs is determined by the following formula:

$$L = \left(\frac{C}{P}\right)^p \tag{15}$$

This formula defines nominal bearing life, i.e. life corresponding to the reliability of 90% of the tested number of bearings.

The same life is shown in the graphs in **Figs. 10–12**.

The exponent p is a number that depends on the bearing type. For tapered roller bearings, the exponent p = 10/3 = 3.333 is commonly accepted.

CALCULATION EXAMPLE

Taper roller bearings of the basic type, i.e. series 302, were used as the subject of calculation. Bearing No. 30209 was selected for the calculation example. The dimensions of the bearings' work surfaces are required for the calculation. They were assumed according to the archival documentation of CBKŁT. There were no contemporary data available as regards this topic (bearing manufacturers keep these dimensions as production secrets), but contemporary deviations from these archival data cannot be very big (overall dimensions of bearings are unchangeable). Therefore, these possible deviations cannot qualitatively influence calculation results. Dimensions of work surfaces of selected bearings are shown in **Tab. 1**.

Table 1. Dimensions of work surfaces of bearings taken into account for calculation

Tabela 1. Wymiary powierzchni roboczych łożysk przyjętych do obliczeń

Bearing	30209
D_k [mm]	10.495
L_n [mm]	13.8
r_e [mm]	150
s_f [mm]	0.5
Z	19
D_l [mm]	57.945
r_{bz} [mm]	7500
α [rad]	0.2637096
δ [rad]	1.562070
β [rad]	0.0349066
E	0.40
Y	1.5
C [N]	66000

For the selected bearings, a model shaft was determined, the dimensions of which are defined according to **Fig. 8** and given in **Tab. 2**. It was assumed that, in each calculation case, the shaft is supported by two identical tapered roller bearings being in X arrangement. The shape of the shaft was selected according to the typical shape of gear shafts in which the bearings are placed at the ends of the shaft and the diameters at the individual segments correspond to the theoretical outline being built on the principle of equal bending strength. In technical reality, there are infinitely many types of shaft shapes.

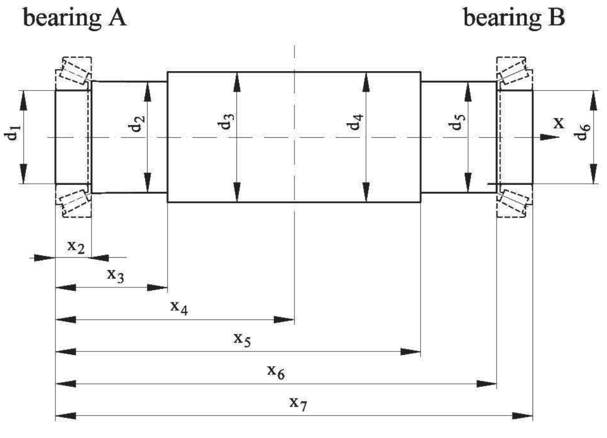


Fig. 8. Model shaft design
Rys. 8. Szkic modelowego wału

Table 2. Dimension parameters of models shafts [mm]
Tabela 2. Parametry wymiarowe modelowych wałów [mm]

Bearing	30209
x_2	21
x_3	75
x_4	150
x_5	225
x_6	279
x_7	300
d_1	45
d_2	52
d_3	60
d_4	60
d_5	52
d_6	45

For all model shafts, the shaft beginning coordinate x_1 was assumed as equal to zero.

The bearing was calculated with loads varying in value. A variant of the load location is shown in **Fig. 9**.

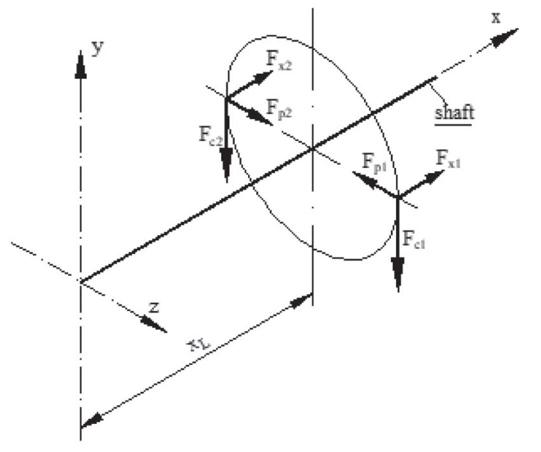


Fig. 9. Assumed variant of bearing load
Rys. 9. Przyjęty wariant obciążenia łożyskowania

The loading forces (circumferential, radial and axial) are applied at points determined by a straight line running through the wheel parallel to the 'z' axis; thus, circumferential forces F_c are directed parallel to 'y' axis, radial forces F_p are directed towards the centre of the wheel, and axial forces F_x are in line with 'x' axis. The location of the load application points is defined by angles β_1 and β_2. The directions of the radial and axial forces are defined in the same way as in the figure. The positions of the load planes are assumed in fixed relations to the shaft L_w length, equal to the x_7 dimension, which is $x_L = 0.6\ L_w$. Rolling wheel diameter $D_t = 150$ mm.

The loads presented in **Fig. 9** are assumed to be identical ($F_{c1} = F_{c2}$, $F_{p1} = F_{p2}$, $F_{x1} = F_{x2}$).

Firstly, it was established that the circumferential force on the alleged F_{c1} gear wheel would be taken at three levels: as 0.075 C, 0.1 C, or as 0.125 C. Assuming that the interlocking buttress angle of the gears is 20º, the radial F_p force was set as approximately 0.36 of the circumferential force. The axial F_x force was assumed in five values in the following fixed relations to the circumferential force, similarly to previous ball bearing calculations. However, due to the greater weight of the axial force in tapered roller bearings than in angular contact ball bearings, the axial force contributions were reduced accordingly. This was done on the principle of inverse proportionality to the thrust load Y factor. In the tapered roller bearings of the 302 series, it is slightly more than 1.4. The contribution of the axial force was, therefore, assumed smaller for the tapered roller bearings in the ratio 0.57:1.4 =

0.41. Hence, the following relative values of the axial forces for the tapered roller bearings of 302 series resulted: 0, 0.03 F_c, 0.05 F_c, 0.0874 F_c, and 0,16 F_c. Since, in the assumed load variant, two equal axial forces and two equal circumferential forces act on the bearing, the ratio of the sum of axial forces to the sum of circumferential forces is described by the same series of numbers. The circumferential Fc force is not the only transverse load (the other one is the radial force), but, due to the assumed constant ratio of the radial force to the circumferential one, it is assumed to treat the ratio of the circumferential force to the bearing Fc/C carrying capacity as a parameter characterizing the level of the transverse load in the bearing arrangement.

The load values summarized in **Tab. 3** result from the above findings.

Table 3. Load values assumed for calculation
Tabela 3. Przyjęte do obliczeń wartości obciążeń

Bearing		30209		
F_c/C		0.075	0.100	0.125
F_c [N]		4950	6600	8250
F_p [N]		1802	2402	3003
F_x [N]	1	0	0	0
	2	149	198	248
	3	248	330	413
	4	433	577	721
	5	792	1056	1320

RESULTS AND CONCLUSIONS

Figures 10–12 show the fatigue life characteristics of the left (A) and right (B) bearing as a function of Z_c preload for assumed load values.

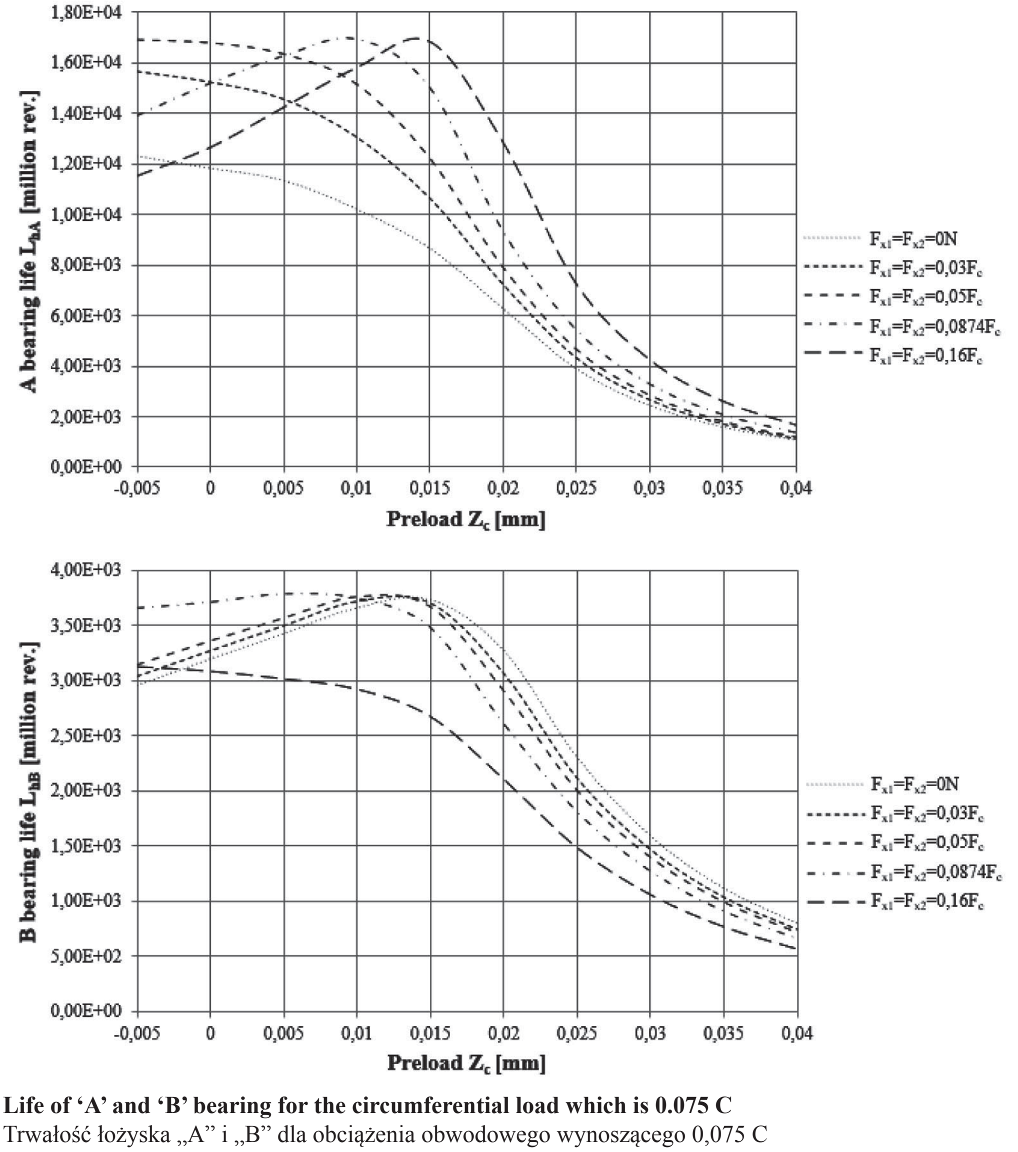


Fig. 10. Life of 'A' and 'B' bearing for the circumferential load which is 0.075 C
Rys. 10. Trwałość łożyska „A" i „B" dla obciążenia obwodowego wynoszącego 0,075 C

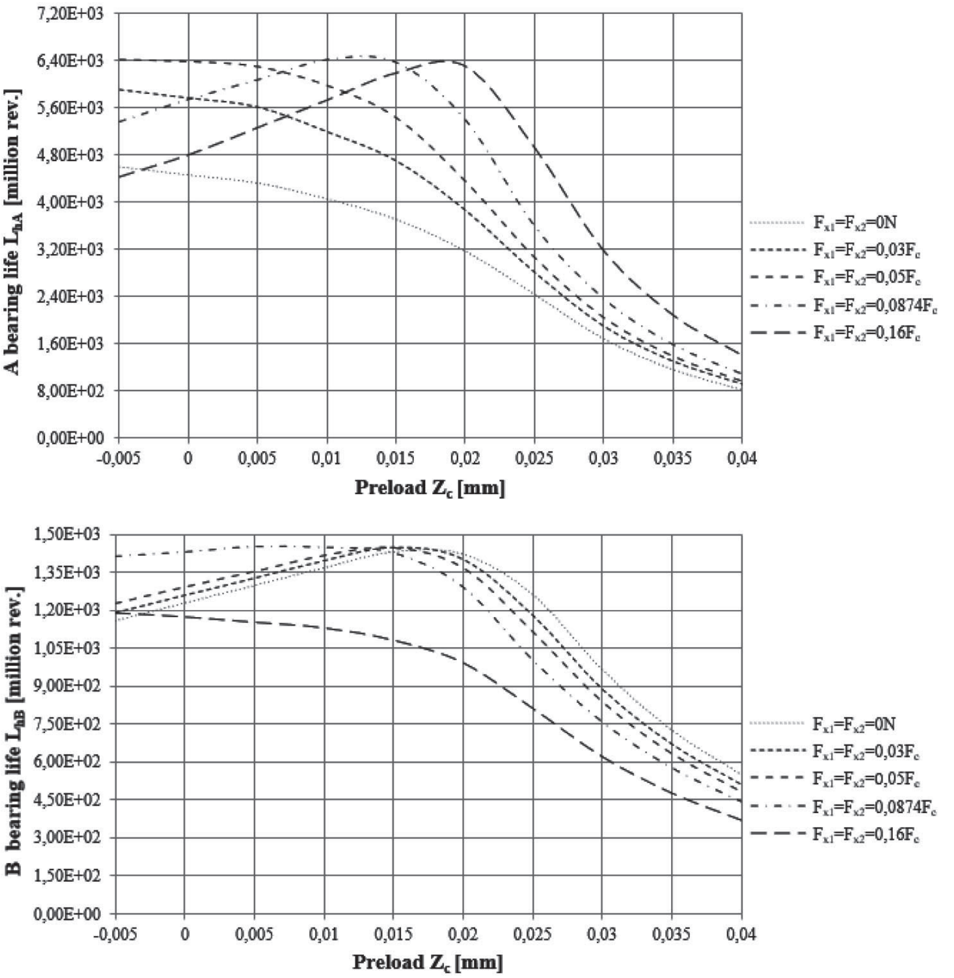


Fig. 11. Life of 'A' and 'B' bearing for the circumferential load which is 0.1 C
Rys. 11. Trwałość łożyska „A" i „B" dla obciążenia obwodowego wynoszącego 0,1 C

If the loads are applied closer to the right bearing, both life characteristics of bearing A and bearing B have diverse shapes, depending on the axial force. At high axial force (above about 0.05 F_c), the life curves of bearing A show an increase to a certain maximum, while at lower axial force, the curves are monotonically falling with increasing from zero clamp values. On the other hand, the life curves of bearing B behave in the opposite way: The occurrence of maximum can be observed when the axial force is lower than 0.03 F_c, while at a higher axial force, the life curves of bearing B are falling, although not very steeply.

In all cases, the life characteristics of bearing A and bearing B are different, i.e. when one is increasing, the other is simultaneously decreasing.

Taking into account the influence of the preload on the fatigue life, it can be claimed that the use of the preload is advantageous in every case of the load of a tapered roller bearing system. In further considerations, recommendations will be made as to the permissible value of preload, taking into account the life of the bearing system, the friction torque in the bearing system, and the bearing rigidity.

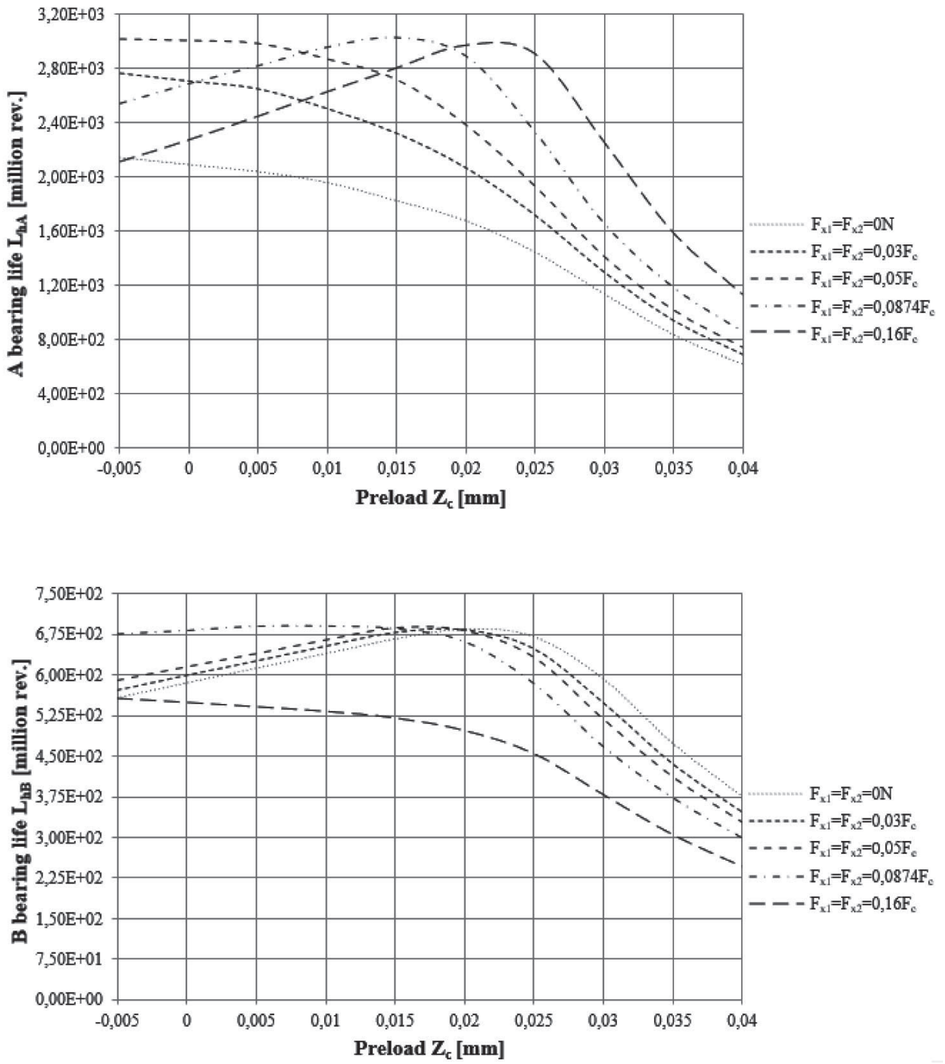


Fig. 12. Life of ‘A’ and ‘B’ bearing for the circumferential load which is 0.125 C
Rys. 12. Trwałość łożyska „A” i „B” dla obciążenia obwodowego wynoszącego 0,125 C

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