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# THE INFLUENCE OF LOCATION OF THE SURFACE OF LOAD ON THE VALUE OF PRELOAD IN THE SYSTEM OF ANGULAR CONTACT BALL BEARINGS

## WPŁYW POŁOŻENIA PŁASZCZYZNY OBCIĄŻENIA NA WIELKOŚĆ ZACISKU WSTĘPNEGO W UKŁADZIE ŁOŻYSK KULKOWYCH SKOŚNYCH

**Key words:**

angular contact ball bearings, bearing system, preload, bearing stiffness, bearing rating life

**Słowa kluczowe:**

łożyska kulkowe skośne, układ łożysk, napięcie wstępne, sztywność łożyskowania

**Abstract**

Angular contact ball bearings are commonly used when high stiffness of the bearing is needed. However, a significant stiffness increase can be achieved only through introducing preload (mounting tightness) to the system of angular ball bearings.

The aim of the study is to determine how the location of a plane of a load influences the selection of preload in a system of angular contact ball bearings, and, in consequence, the durability of the bearing.

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## INTRODUCTION

A preload can be expressed as a force or as a displacement (distance), although the force of preload is the basic parameter in specification **[L. 7]**. Depending on the regulation method, a preload also indirectly influences the frictional moment in the bearing.

The optimum values of preload can be obtained from proven constructions, and then used is similar constructions **[L. 1, 7]**. In case of new constructions, it is recommended to calculate the force of preload and to check the correctness of calculation via experiments. In practice, it might be necessary to introduce corrections, because not all real work parameters can be known precisely. The credibility of calculations depends on the extend at which the assumptions made in relation to temperature conditions during work and the elastic strains of the cooperating elements – first of all of the holder – are in accordance with the real conditions.

As given in **[L. 7]** while determining the preload, primarily, one has to calculate the target force of the preload assuring optimum combination of stiffness, exploitation time, and the reliability of the bearing work, then the force of preload to be applied during arranging the bearings for assembly. During the assembly, the bearings should have the temperature of the surroundings and cannot be a subject to a working load.

The correct preload in a normal working temperature, according to literature, depends on the bearing load. Angular contact ball bearings can carry at the same time radial and axial loads. With the axial load in the bearing, a force is produced working in the axial direction, and it has to be overtaken by the second bearing located in the direction opposite to the first one. Strictly axial displacement of one bearing ring towards the second one means that half of the bearing perimeter (i.e. half of rolling elements) is under load and the axial force in the bearing will equal **[L. 7]**:

– For angular contact ball bearings:

$$F_a = e \cdot F_r \tag{1}$$

– For taper roller bearings:

$$F_a = 0.5\ F_r/Y \tag{2}$$

Where: $F_r$ – axial load of the bearing;
e – the value characterizing the inner construction of radial bearing in the area of capacity of carrying loads; for angular contact ball bearings with the operation angle of 40º e = 1,14;
Y – the calculation factor of calculating axial load into equivalent load. Its values can be found in the tables published in the catalogues of bearings.

If a single bearing is a subject to radial load $F_r$, then the external axial load $F_a$ of the above value has to be applied in order to fulfil the above initial requirements assumed while determining dynamic load rating (half of the perimeter of a bearing under load) **[L. 3, 4]**. If the applied force is lower, the number of rolling elements carrying the load will be smaller, and the ability of carrying the load by the bearing will be also adequately decreased.

In the bearing arrangement consisting of two one-raw angular contact ball bearings in a convergent and divergent system, each of the bearings has to overtake the axial forces of the other bearing. If the system of bearings is settled for zero clearance, a distribution of load, in which half of rolling elements is under load, will be automatically achieved.

As further given in **[L. 3]**, in other cases of the load, in particular in the situation of external axial load appearance, it may be necessary to use a preload of the bearings in order to compensate for the clearance appearing in one bearing as a result of the elastic deformation of the bearing overtaking the axial load. At the same time, thanks to preload, a better load distribution on the balls in the bearing not loaded axially is achieved. The preload leads to increasing the rigidity of the bearing system. Considering the issue of rigidity, it should be remembered that, not only the elasticity of the bearing has influence on it, but also the elasticity of the shaft and holder, the fit of the rings and elastic deformation of all elements within the field of the forces, including the retaining elements. All these factors have a great influence on the elasticity of the whole system of the shaft.

## CALCULATION METHOD

The study presents a modelling method developed by the author and applied, among others, in papers **[L. 4, 5]**. It has been assumed, for the purpose of the analysis, that, in the calculations of the distribution of loads, the contact of a ball with the course is point-based, under a load transforms into an elliptic contact, a subject to Hertz theory, and that elastic deformations of the elements of two bearings take place only at a contact point of rolling elements with both the races.

A durability of a system of angular contact ball bearings can be approximately calculated with the use of a catalogue method **[L. 6]**, basing on the radial and axial load of each bearing and on the catalogue ratios of load. Yet this method is not appropriate for calculations used for accounting for the influence of preload, because it does not take this preload into consideration.

When solving the problem the influence of this, the following was taken into account:

- Radial and axial load influencing the ball raced shaft;
- Elastic shaft deflection, caused by the internal tilt of the rings of bearings;
- Preload.

All these element influence the loading of balls in a bearing and accordingly on the loading of a bearing.

**Fig. 1** presents the illustration of a contact deformation in a bearing.

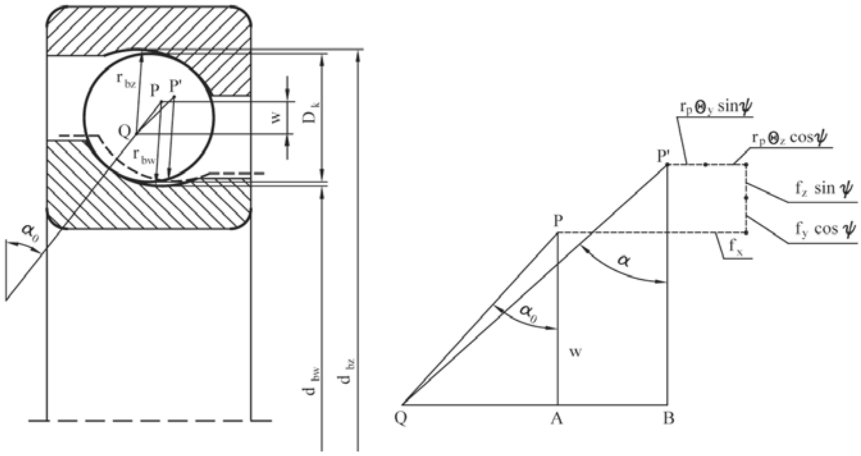



**Fig. 1. Presentation of calculating contact deformation: P – centre of curvature of the inner ring raceway in nominal position, Q – centre of curvature of the inner ring raceway in nominal position, P' i Q' – centres of the same curvatures, but after displacement, $D_k$ – diameter of a ball, $r_{bz}$ – the radius of curvature of outer raceway, $r_{bw}$ – the radius of curvature of inner raceway, $d_{bz}$ – diameter of outer raceway, $d_{bw}$ – diameter of inner raceway**

Rys. 1. Ilustracja odkształcenia stykowego w łożysku: P – środek krzywizny bieżni pierścienia wewnętrznego w położeniu nominalnym, Q – środek krzywizny bieżni pierścienia zewnętrznego w położeniu nominalnym, P' i Q' – środki tych samych krzywizn, ale po przemieszczeniu, $D_k$ – średnica kulki, $r_{bz}$ – promień krzywizny bieżni zewnętrznej, $r_{bw}$ – promień krzywizny bieżni wewnętrznej, $d_{bz}$ – średnica bieżni zewnętrznej, $d_{bw}$ – średnica bieżni wewnętrznej

## CALCULATIONS

Angular-contact ball bearings operate in systems and have to be considered in systems, especially when a preload is to be taken into account. The forces operating in bearings, i.e. the durability of the bearings, depend on numerous factors, among others, on the load per each bearing in the system, and shaft deflection. The loads per bearings depend among others on the type, size, and number of gear wheels assembled on the shaft and on the location of cooperating wheels. The number of possible cases is infinite. Therefore, the authors assumed a specific construction.

The adopted construction was a model shaft (**Fig. 2**) ball raced with two angular contact ball bearings 7206B, with parameters: $D_k$ = 9.525 mm, $d_{bw}$ = 36.387 mm, $d_{bz}$ = 55.636 mm, $r_{bw}$ = 4.9 mm, $r_{bz}$ = 5.0 mm, and the number of balls Z = 13. Dynamic load of the bearings according to **[L. 8]** is C = 20400 N.

The bearing situated on the left end of the shaft was marked “A” and the one on the right – “B”.

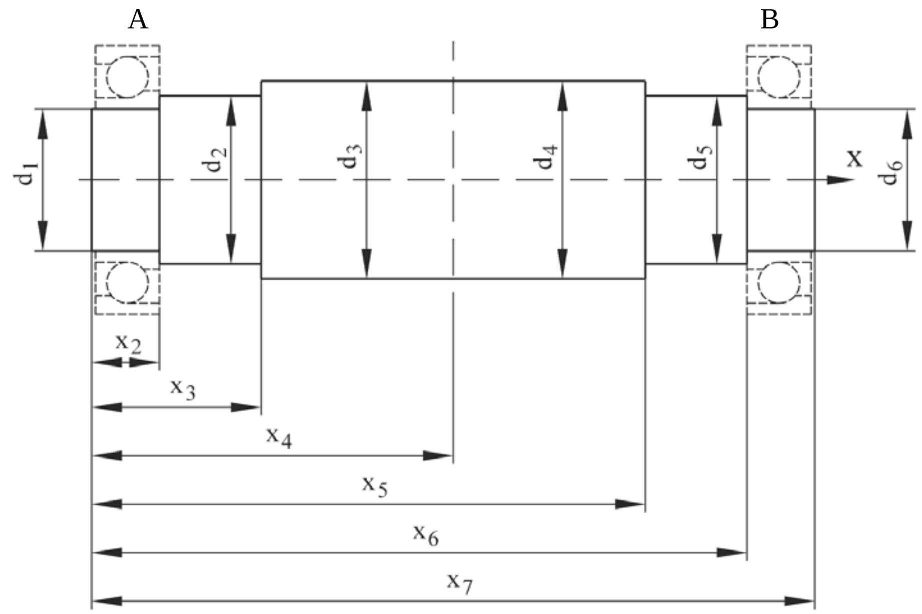



**Fig. 2. A draft of a model shaft [L. 2]**
Rys. 2. Szkic modelowego wału

The dimensions of the model shaft are as follows: $x_2 = 16$ mm, $x_3 = 50$ mm, $x_4 = 100$ mm, $x_5 = 150$ mm, $x_6 = 184$ mm, $x_7 = 200$ mm, $d_1 = 30$ mm, $d_2 = 35$ mm, $d_3 = 40$ mm, $d_4 = 40$ mm, $d_5 = 35$ mm, and $d_6 = 30$ mm. Nodal points of the bearings (points of reaction concentration) are determined by the coordinates: $x_A = 27$ mm, $x_B = 173$ mm.

The adopted shaft model was subjected to calculations for load option, which is shown in **Figure 3**. Is assumed that the load is applied on both sides of one gear wheel, located at a distance $x_L$ from the beginning of the shaft. The locations of the planes of loads are respectively: $x_L = 0.4 \cdot L_{w,}$ $x_L = 0.5 \cdot L_w$ $x_L = 0.6 \cdot L_w$, for the following assumptions:

1) $F_{c1} = F_{c2}$, $F_{p1} = F_{p2}$, $F_{x1} = F_{x2}$,
2) $F_c = 0.1C$,
3) $F_p = 0.36\ F_c$,
4) The axial force $F_x$ was adopted in five values in subsequent pre-established relations to the circumferential force: 0,0.049 · $F_c$, 0.098 · $F_c$, 0.196 · $F_c$, 0.392·$F_c$.

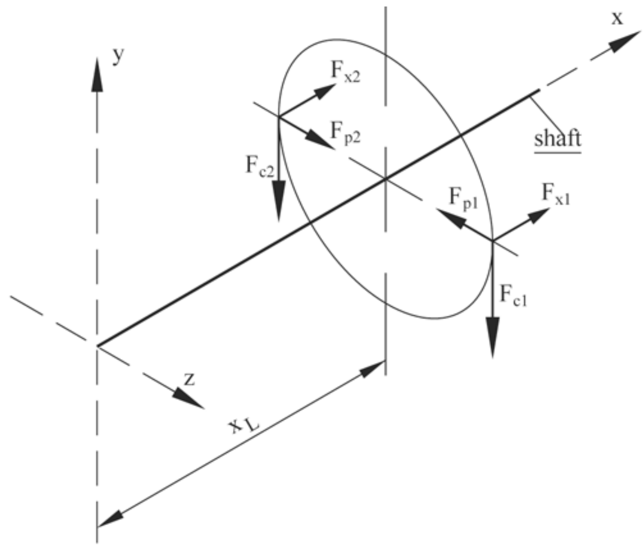
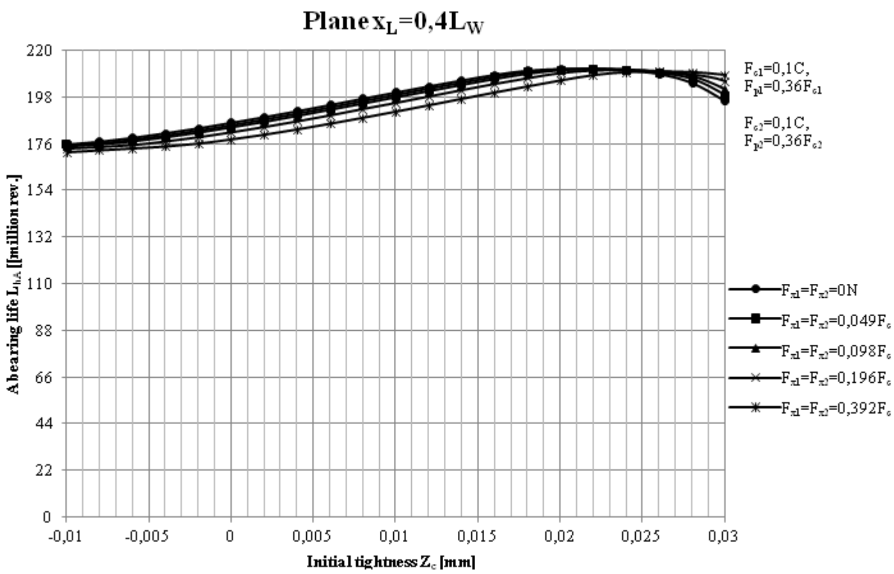



**Fig. 3. Assumed variations of bearing loads [L. 2]**
Rys. 3. Przyjęty wariant obciążeń łożyskowania

**Figures 4** and **5** present separate characteristics of the basic rating life of the left (A) and right bearing (B) in the function of the initial tightness $Z_c$, specified for the presented variations of load. Minus values of initial tightness mean that the system was assembled with the operating clearance.



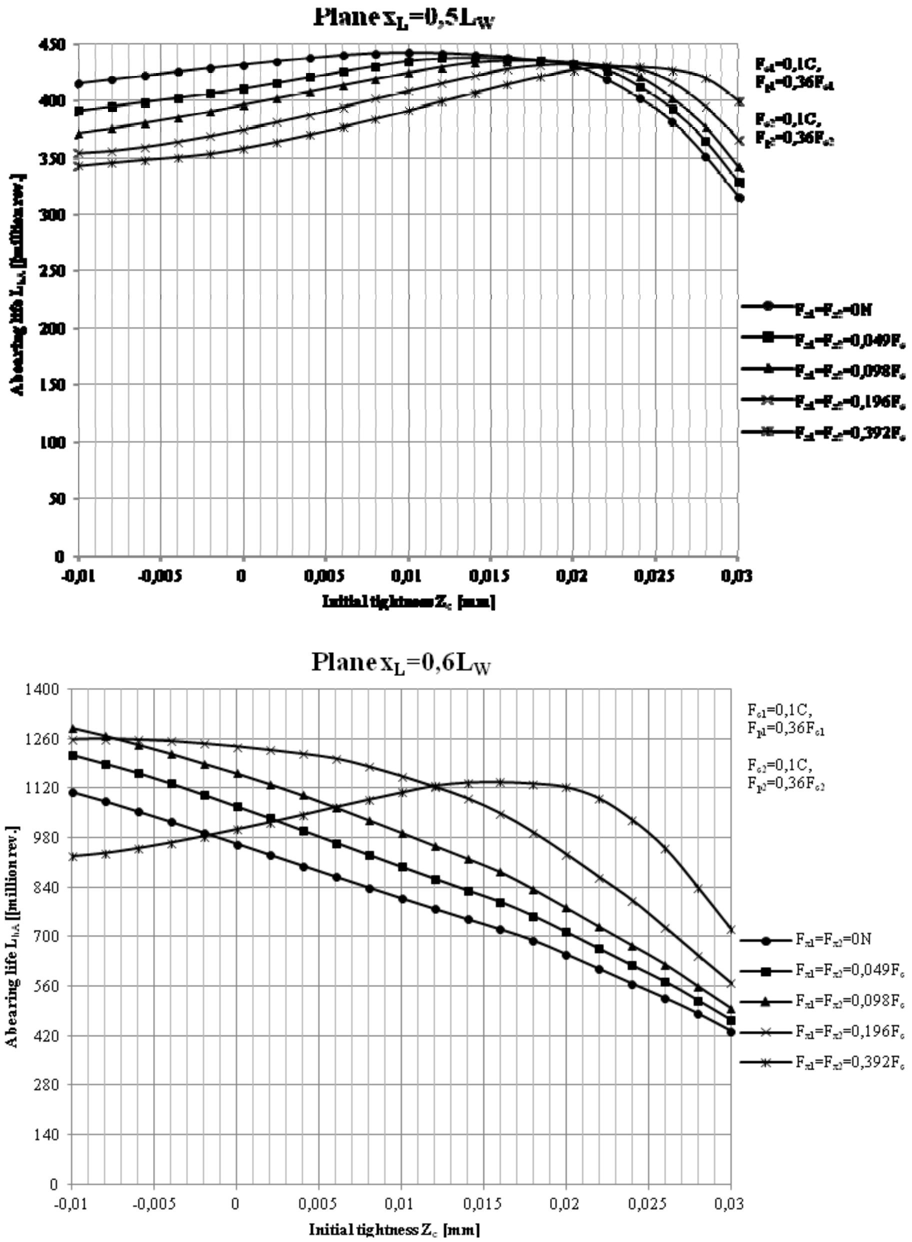
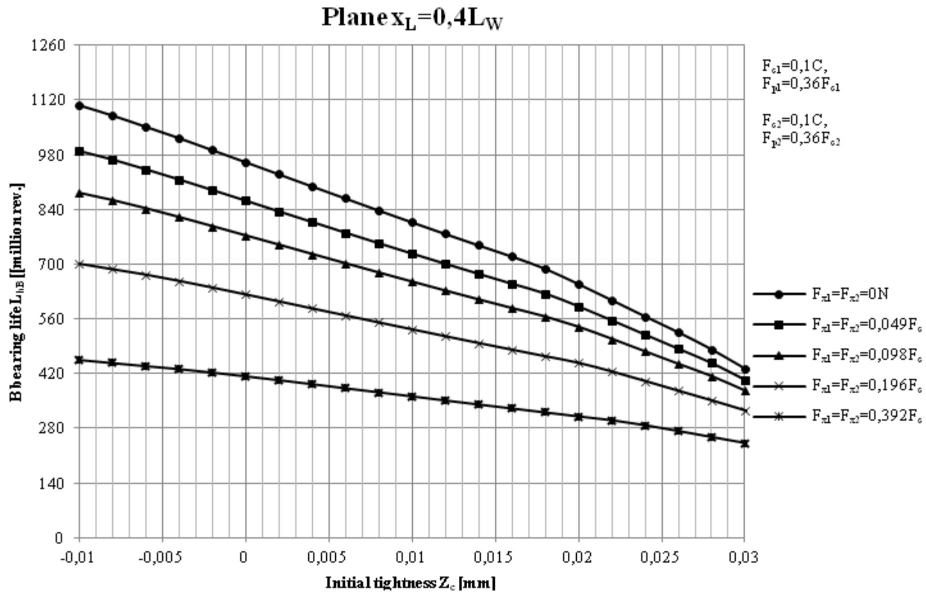
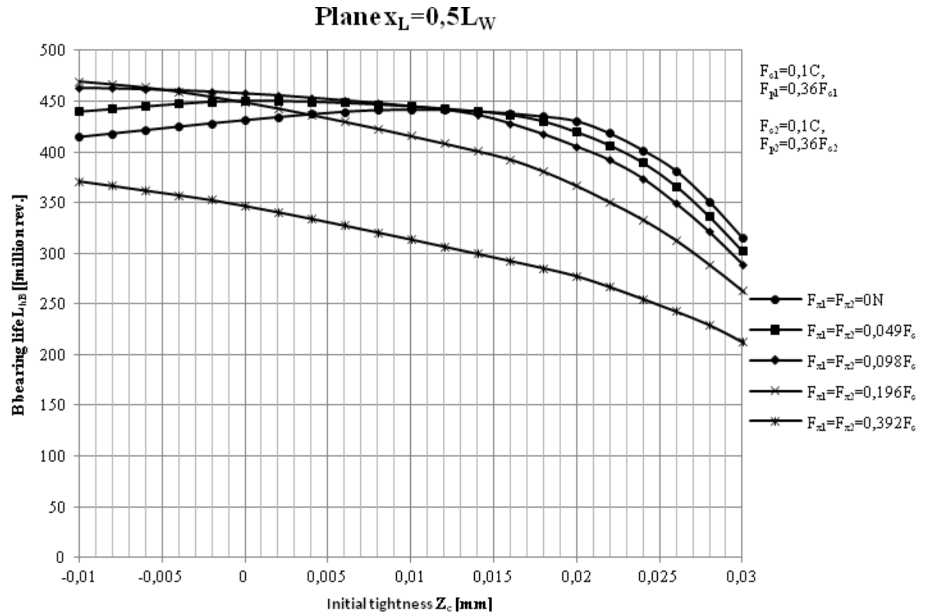



**Fig. 4. Durability of bearing A for assumed load**
Rys. 4. Trwałość łożyska A przy przyjętym wariancie obciążenia


Plane $x_L$=0,4$L_W$
1260
1120
980
840
700
560
420
280
140
0
-0,01
-0,005
0
0,005
0,01
0,015
0,02
0,025
0,03
Bbearing life $L_{hB}$ [million rev.]
Initial tightness $Z_c$ [mm]
$F_{o1}$=0,1C,
$F_{p1}$=0,36$F_{o1}$
$F_{o2}$=0,1C,
$F_{p2}$=0,36$F_{o2}$
$F_{x1}$=$F_{x2}$=0N
$F_{x1}$=$F_{x2}$=0,049$F_o$
$F_{x1}$=$F_{x2}$=0,098$F_o$
$F_{x1}$=$F_{x2}$=0,196$F_o$
$F_{x1}$=$F_{x2}$=0,392$F_o$


Plane $x_L$=0,5$L_W$
500
450
400
350
300
250
200
150
100
50
0
-0,01
-0,005
0
0,005
0,01
0,015
0,02
0,025
0,03
Bbearing life $L_{hB}$ [million rev.]
Initial tightness $Z_c$ [mm]
$F_{o1}$=0,1C,
$F_{p1}$=0,36$F_{o1}$
$F_{o2}$=0,1C,
$F_{p2}$=0,36$F_{o2}$
$F_{x1}$=$F_{x2}$=0N
$F_{x1}$=$F_{x2}$=0,049$F_o$
$F_{x1}$=$F_{x2}$=0,098$F_o$
$F_{x1}$=$F_{x2}$=0,196$F_o$
$F_{x1}$=$F_{x2}$=0,392$F_o$

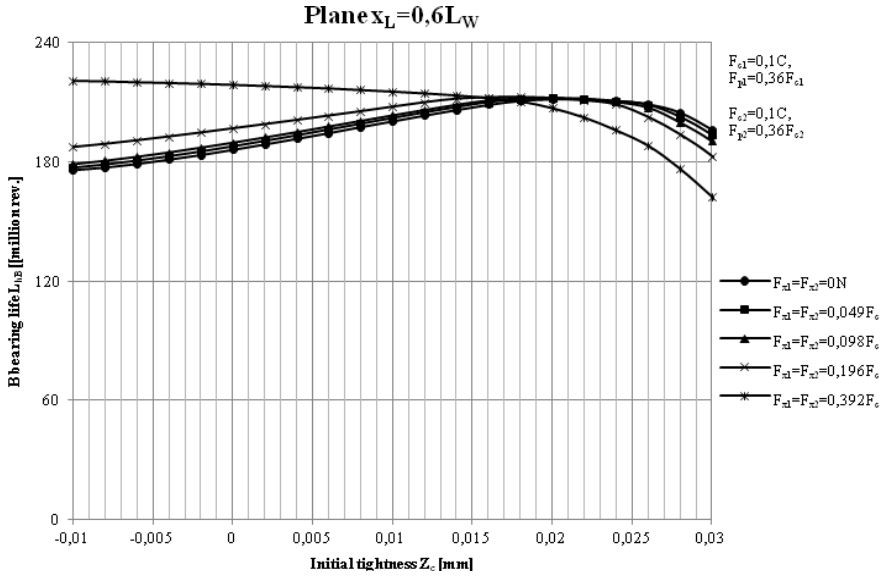



**Fig. 5. Durability of bearing B for assumed load**
Rys. 5. Trwałość łożyska B przy I wariancie obciążenia

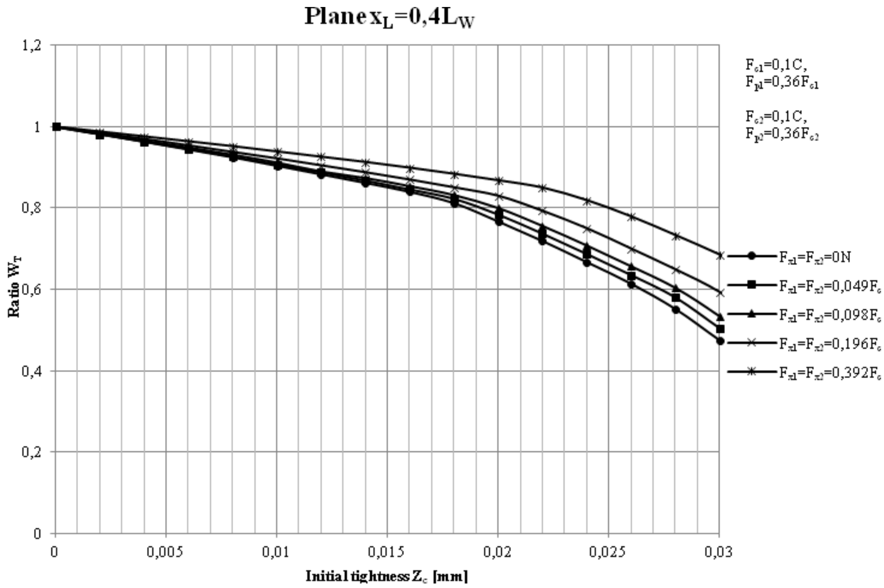
It is observable that, in almost all cases, the characteristics of durability of A bearing and durability of B bearing have different courses. It is usually so that when the durability of one bearing increases, the durability of the other one drops. It means that single-observed characteristics of durability will not provide an answer to the question of which value of preload is the optimum one for a given plane of load. Therefore, it has been decided to create an indicator connecting the durability of both the bearings specified by the following expression:

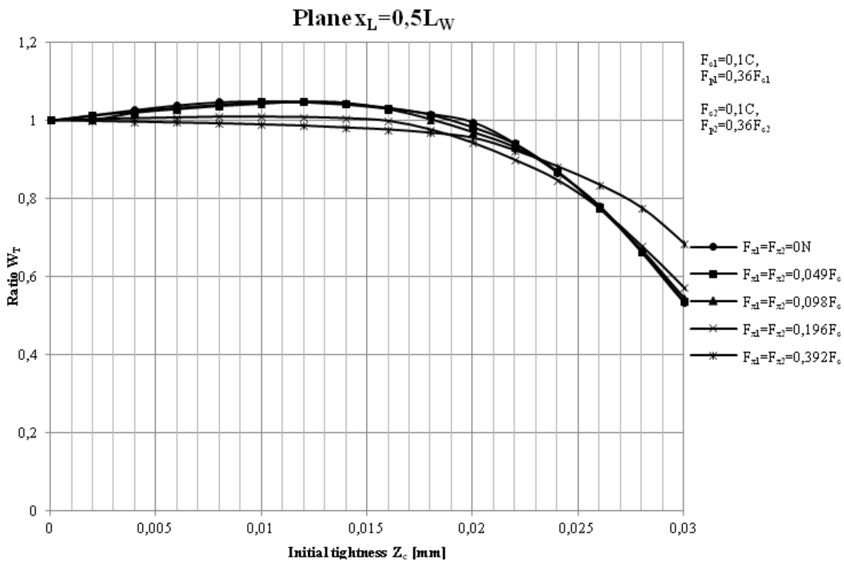
$$W_T = \frac{L_{hA}}{L_{hA0}} \cdot \frac{L_{hB}}{L_{hB0}} \tag{3}$$

where: L – basic rating life of A and B bearings determined in specific conditions with the assumed initial tightness,
$L_0$ – basic rating life determined in the same conditions without the initial tightness.

Creating the indicator $W_T$ embracing both the bearings enabled comparing the durability of bearing systems. It has a great advantage, i.e. a drastic drop of durability of one of the bearings results in the decrease of this indicator. When the durability of one of the bearings drops near zero, the $W_T$ indicator also falls to zero in approximation. Characteristics $W_T$ for the adopted examples are presented in **Fig. 6**.

In order to determine a favourable range of the initial tightness, collective graphs based on agreed border points of the graphs in Fig. 6 have been drawn. The agreed boarder points are the plus values $Z_c$, for which $W_T$ characteristics on the falling side take the value 0.98 (it was an arbitrary assumption that in aiming to increase the axial stiffness of the bearing, the drop of the indicator reaching 2% is acceptable).



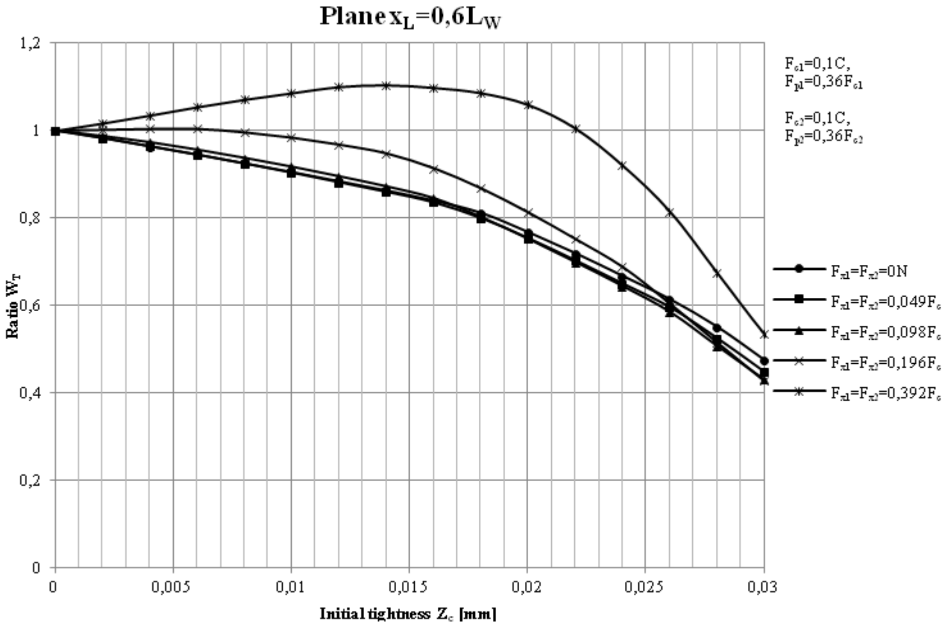



**Fig. 6. Characteristics of $W_T$ indicator for assumed load**
Rys. 6. Charakterystyka wskaźnika $W_T$ przy I wariancie obciążenia

Graphs of tightness limit for the considered variations are presented in **Fig. 7**.

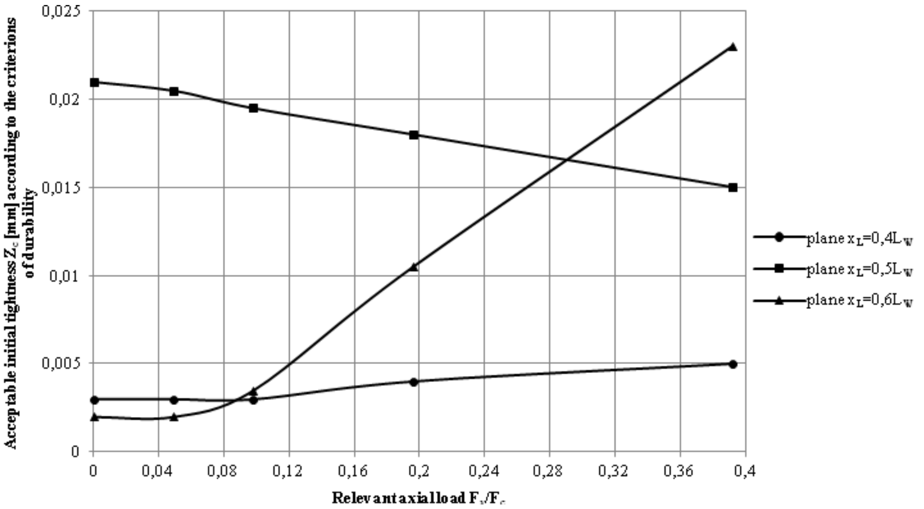



**Fig. 7. Acceptable tightness with the variant I of load**
Rys. 7. Dopuszczalny zacisk przy przyjętym wariancie obciążenia

## CONCLUSIONS

It can be observed that, for the variation of the load, while locating the plane of the load on the axis $x_L = 0.4\ L_w$, the acceptable values of initial tightness are very low, on the level from approx. 3–4 µm. When locating the plane of load on the axis $x_L = 0.5\ L_w$, the border value of initial tightness is from 15 µm with axial force of 0.4 $F_c$ to 21 µm with no axial force. For $x_L = 0.6\ L_w$, the border value of initial tightness is between 2.5 µm with a zero sum of axial forces and approx. 24 µm with the relation $F_x/F_c \approx 0.4$. It should be noticed that a huge dependence of the limit tightness on the relative axial force has appeared, and the quotient of the highest and the lowest values $Z_c$ is around 10.

It can be concluded from the above calculations that the location of the plane of load on the shaft in relation to the return action of axial forces has a huge influence on the effect of applying initial tightness. The most favourable situation is the one in which loads on the shaft are concentrated in the centre of its length or arranged symmetrically towards the centre. In such cases, large initial tightness, considerably increasing stiffness of the shaft, turns up advantageous due to bearing rating life of the bearings.

## REFERENCES

1. Harris T.A., Rolling Bering Analysis. John Wiley & Sons, London 2006.
2. Kaczor J., Raczynski A., The effect of preload of angular contact ball bearings on durability of bearing system. Proceedings of the Institution of Mechanical Engineers, Part J: Journal of Engineering Tribology, Part J: Journal of Engineering Tribology, 2015, Vol. 229(6) 723–732.
3. Krzemiński-Freda H., Łożyska toczne. PWN, Warszawa 1989.
4. Krzemiński-Freda H., Sztywność łożysk skośnych i ich układów. Zagadnienia Eksploatacji Maszyn, z. 1(65)/1986, s. 223–231.
5. Raczyński A., Obliczanie trwałości zmęczeniowej łożysk kulkowych zwykłych z uwzględnieniem luzu i ugięcia wału. Zagadnienia Eksploatacji Maszyn (1) 117, 1999.
6. Raczyński A., Obciążenie kulek w łożysku kulkowym skośnym w zależności od napięcia wstępnego. Tribologia 1/2001.
7. Katalog łożysk tocznych SKF 1991.
8. http://www.skf.com/group/products/bearings-units-housings/ball-bearings/angular-contact-ball-bearings/single-row/index.html?prodid=1210010206&imperial=false.

**Streszczenie**

**Łożyska kulkowe skośne są zwykle stosowane w takich sytuacjach, kiedy potrzebne jest uzyskanie dużej sztywności łożyskowania. Jednakże znaczące zwiększenie sztywności można uzyskać dopiero dzięki wprowadzeniu napięcia wstępnego (tzw. zacisku montażowego) do układu łożysk skośnych.**

**Celem tej pracy jest określenie, jak położenie płaszczyzny obciążenia wpływa na dobór zacisku.**