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Jaroslaw KACZOR*

THE INFLUENCE OF PRELOAD IN RELATION WITH LOAD ON A SYSTEM OF ANGULAR BALL BEARINGS

WPŁYW ZACISKU WSTĘPNEGO W POWIĄZANIU Z OBCIĄŻENIEM NA PRACĘ UKŁADU ŁOŻYSK KULKOWYCH SKOŚNYCH

Key words: preload, angular ball bearing, stiffness of a system of bearings, bearing durability, friction in a system of bearings.

Abstract A proper preload has a decisive influence on the correct work of a system of bearings. According to specialist literature **[L. 1, 2, 8, 12]**, the main benefits of preload application are the following: increasing the stiffness of a system of bearings, reducing noise during work, increasing the precision of the shaft driving, compensation for the wear and settle processes during work, and thus assurance of a longer life of the system.
The aim of the study is to determine how application of preload influences the operation of angular bearings, i.e. their durability, stiffness, and friction. The influence of preload will be expressed in a relative way: through variations of ratios.

Słowa kluczowe: zacisk wstępny, łożysko kulkowe skośne, sztywność układu łożysk, trwałość łożyskowania, tarcie w układzie łożysk.

Streszczenie: Odpowiedni zacisk wstępny ma decydujący wpływ na prawidłową pracę układu łożysk. Jak podaje literatura **[L. 1, 2, 8, 12]** głównymi powodami stosowania zacisku wstępnego są następujące korzystne jego efekty: zwiększenie sztywności układu łożyskowego, zmniejszenie hałasu podczas pracy, zwiększenie dokładności prowadzenia wału, kompensacja procesów zużycia i osiadania podczas pracy, zapewnienie dłuższego czasu eksploatacji.
Celem pracy jest określenie, jak wpływa zacisk wstępny na pracę układu łożysk skośnych, tj. na trwałość, sztywność i tarcie w łożyskach. Wpływ zacisku wstępnego będzie wyrażony przez zmianę odpowiednich wskaźników, czyli w sposób względny.

INTRODUCTION

The stiffness of a bearing (often expressed in kN/μm) is defined as the ratio of the force acting on the bearing to elastic deformation of the bearing. For a given typical load elastic deformation caused by the load in preloaded bearings is lower than in the bearings that are not subjected to preload.

It is known that the higher the working clearance in a bearing, the poorer the running of roller elements in unloaded zone, and the more intense are the vibrations of the bearing system and the louder are the bearings in operation. Reducing the clearance to achieve a preload of the bearings results in noise reduction **[8]**. However, the reduction is achieved only for a certain specific value of preload. Excessive preload causes an increase in contact loads between the roller parts and the rings in the bearing. The above results in the bearing working more "heavily", which is manifested by a higher noise level. Thus, the noise emission by a bearing is an indicator of overpassing the optimum value of preload.

Bearings with preload assure more accurate shaft driving because preload facilitates reducing the tendency of a shaft to deflect under a load. For example higher shaft driving accuracy and higher stiffness assured by preloaded pinion bearings and differential bearings guarantee that tooth adhesion will be more accurate, the contact ratio will be stable and dynamic forces

* ORCID: 0000-0003-2315-980X. Institute of Environmental Engineering and Building Installations, Lodz University of Technology Lodz, Poland, e-mail: jaroslaw.kaczor@p.lodz.pl

will be minimised **[L. 8]**. As a result the system during operation will emit less noise, and the life of gear teeth will be prolonged.

Compensation for wear and settling

Wear and settle processes taking place during the operation of a system of bearings contribute to the formation of clearances, but this phenomenon can be compensated by a gradual increase in preload in time.

For some applications, systems of bearings with preload demonstrate higher reliability of work and longer life. Properly selected preload can have a beneficial influence on the distribution of loads of roller parts in bearings, and thus on their durability **[L. 11]**.

Preload can be expressed as force or as dislocation (distance). However, the force of preload is a basic specification parameter **[L. 12]**. Depending on the regulation method, preload is also directly associated with the tightening torque.

Optimum values of preload can be modelled on successful designs and applied in similar designs **[L. 1**, **12]**. In the case of new designs, it is recommended to calculate the force of preload and to check the correctness of calculations by means of experimenting. In practice, it might be necessary to introduce corrections, because not all real parameters of work can be accurately known. The reliability of calculations depends primarily on how the assumptions are made in relation to temperature conditions during work and the elastic deformations of cooperating elements (first of all of the housing), which are accordant with real conditions.

As given in **[L. 12]**, in order to determining the right preload, the target value of preload assuring optimum compromise between stiffness, life time, and the reliability of the bearing operation has to be calculated in the first place. In the next step, calculations should be made for the force of the preload proper during settling the bearings in the assembly. The bearings should have the temperature of the environment, and they cannot be subjected to a working load during assembly.

Correct preload in standard working temperature depends on the loading of the bearing. Angular ball bearings and tapered roller bearings have to carry radial and axial loads simultaneously. Together with radial load, a force working in the axial direction operates in the bearing. In general the force has to be taken over by the other bearing, which is turned in the opposite direction to the first one. Strictly radial dislocation of one ring of the bearing in relation to the other means that half of the bearing circuit (i.e. half of all roller elements) is under the load and the axial force appearing in the bearing will equal **[12]**:

$$F_a = e \cdot F_r,$$

where:
F_r – radial load of the bearing,
e – value characterising inner construction of a radial bearing within the range of relation F_a/F_r; for angular ball bearings with the operation angle of 40º e=1.14.

If a single bearing is a subject to radial load F_r of the above value, external axial force F_a has to be applied, so as to fulfil prerequisites in accordance with the assumptions made while determining dynamic load capacity (half of the bearing circuit before applying load). If the applied external force is lower, the number of roller elements carrying the load will be lower, and the capacity to carry the load by the bearing will be also respectively reduced.

In a bearing system consisting of two angular single-raw ball bearings, each of the bearings has to overtake the axial forces of the other one. If a system of bearings is set on zero clearance, the distribution of load in which half of the roller elements are loaded will be achieved automatically.

In the case of an external axial load in the bearing system, elastic deformations of elements increase in the bearing overtaking the load. This phenomenon may cause a clearance and affect the distribution of load in the opposite bearing. In order to avoid this situation, it is necessary to introduce preload in bearings. The preload also contributes to the increase in the stiffness of a system of bearings. While considering the question of stiffness, it ought to be remembered that stiffness is influenced not only by the resilience of the bearing but also by the elasticity of shaft and of the housing**,** the fit of the rings, and the elastic deformation of all other elements in the field of forces, including the thrust elements. All of these factors have great influence on the elasticity of the whole system of the shaft.

CALCULATING METHOD

In order to solve the problem, the following issues require associating:

1. The deflection lines of the machine shaft with a complex external load,
2. The dislocations of bearing inner rings in relation to the outer ones as a result of load plus preload,
3. Contact elastic deformations at the contact point of roller parts and the tracks in both the bearings of the system,
4. The calculations of contact forces in bearings on the basis of contact deformations,
5. The balance between inner (contact) forces in bearings and the loading of the whole bearing, and
6. The calculations of the durability of bearings based on the contact forces.

Assuming the appropriate calculation model is a decisive step on the way to finding a theoretical solution to the problem. The approximation to reality, the scope of the considered phenomena, and the effort behind finding a solution depend on the model accuracy.

Calculation models usually remain unrevealed in detail in publications on the phenomena observed in roller bearings. In this study, the author used a method of modelling developed by himself and applied in their studies **[L. 3**, **4, 5, 6, 7, 8, 9]**, and other.

The following simplifying assumptions have been made:

1. The material of the bearing is isotropic and a subject to Hooke's law.
2. The working surfaces of the bearings are perfectly smooth.
3. There exist no errors in the shape of balls, bearing rings, or shaft.
4. The placing of bearings in housing**s** is geometrically faultless, and clearances associated with the fitting of the bearings are omitted, which means that axes of outer bearings are always located along one straight line.
5. No mass forces, basket interactions and lubricant resistance are taken into consideration in the analysis of the balls loads.
6. Static forces in the contact of roller parts with rings have no significant influence on elastic deformations, and they are omitted.
7. In the calculations of the distribution of pressures, the contact of the ball with the toroidal treadmill is modelled as an elliptical contact, and subjected to Hertz's theory.
8. The distribution of pressures in the contact of roller elements with tracks is the same in motion and under a static load.
9. Elastic deformations of the elements of bearings occur only in the contact points between roller elements and rings, thanks to which, the non-working surfaces of the rings retain a cylindrical or flat shape.

Dislocations in bearing and durability index W_T are determined and presented in detail in **[5]**, while **[6]** describes the way of determining the moment of friction in a system of bearings and the index of the moment of friction W_M.

ANALYTICAL CALCULATIONS

The calculations were made for an angular ball bearing of the basic type, i.e. produced in the series 72.B. The selected bearing had a specific number 7209B. The diameter of its opening equalled 45 mm. For calculation purposes, the dimensions of the working surfaces of the bearings were needed. They were assumed in accordance with archival documentation CBKŁT. No current data on the subject was at the disposal of the author (producers of bearings keep the data in secret), but actual derogations from archival datadata would not have been significant. Therefore the differences could not influence the quality of the calculation results. Dimensions of working areas of selected bearings are presented in **Table 1**. In the last row dynamic load capacity of the bearing is presented according to **[L. 13]**.

Table 1. Dimensions of working areas of the bearing chosen for calculations

Tabela 1. Wymiary powierzchni roboczych łożyska przyjętego do obliczeń

Bearing →	7209B
Diameter of the rolling element D_k [mm]	12,700
Diameter of inner track d_{bw} [mm]	52,181
Diameter of outer track d_{bz} [mm]	77,850
The radius of curvature of the inner track r_{bw} [mm]	6,540
The radius of curvature of the outer track r_{bz} [mm]	6,670
The number of rolling elements Z	14
Dynamic load capacity C [N]	37700

Model shafts were determined for the selected bearing. Their size was specified in accordance with **Fig. 1** and given in **Table 2**. It was assumed that, in each calculation case, the shaft is supported by two identical angular ball bearings set in the X arrangement. The shape of the shaft was determined following a typical shape of transmission shafts, with bearings located at their ends; whereas the diameters of particular segments were in accordance with the theoretical outline, built by the rule of equal resistance to bending. In technical reality there exist an infinite number of types of shafts, but this one was regarded as the model one for most of them.

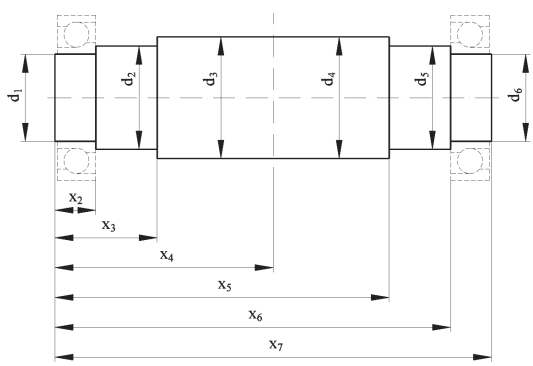


Fig. 1. An outline of a model shaft

Rys. 1. Szkic modelowego wału

The bearings were subjected to calculations for the application of loads with different values and different locations. The variations of the loads locations are presented in **Fig. 2**. In the first variation, it is assumed that the load is applied on two sides of one gear, located at a distance x_L from the beginning of the shaft. Load forces (peripheral, radial, and axial) are applied in points determined by a straight line running through the circle parallel to the z-axis; therefore, peripheral forces F_c are oriented parallel to the y-axis. Radial forces F_p are oriented towards the centre of the circle, and axial forces F_x operate following the x-axis. In the second variation the loads are placed on two gears located at a distances x_{F1} and x_{F2} from the beginning of the shaft. The location of load application points is determined by the angles β1 and β_2. The directions of radial and axial forces are defined in the same way as in the previous case.

Table 2. Dimensional parameters of a model shaft [mm]
Tabela 2. Parametry wymiarowe modelowego wału [mm]

Bearing →	7209B
x_2	19
x_3	75
x_4	150
x_5	225
x_6	281
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, a coordinate of the beginning of the shaft x_1 equal to zero was assumed.

Locations of load planes were assumed to be at specific ratios to the length of shaft L_w, equalling the dimension x_7 from the **Table 2**.

For Variation I of the location (**Fig. 2**): $x_L = 0.4\ L_w$, $x_L = 0.5\ L_w$ or $x_L = 0.6\ L_w$.

For Variation II of the location (**Fig. 2**): $x_{L1} = 0.4\ L_w$, $x_{L2} = 0.6\ L_w$, $\beta_1 = 90º$, $\beta_2 = 180º$.

Derived from the above are the distances from the beginning of the shaft to the load planes given in **Table 3**.

It was decided that loads in both the points presented in **Fig. 2** are identical ($F_{c1} = F_{c2}$, $F_{p1} = F_{p2}$, $F_{x1} = F_{x2}$). As the diameters of the gearwheel were also assumed identical, the torques acting on the shaft were balanced.

Values of loads were assumed to be in specific ratio in relation to the dynamic load capacity of the bearing C. It was decided that peripheral force on the presumed toothed wheel F_{c1} would be considered on level 0.075 C.

Assuming that the angle of tooth contact of gearwheels was 20º, the radial force F_p was determined to be approx. 0.36 of peripheral force. Five values of axial force F_x were adopted, their ratios to the peripheral force being the following: 0, 0.049 F_c, 0.098 F_c, 0.196 F_c and 0.392 F_c. Because, in both the assumed variations of load, two identical axial forces and two identical peripheral forces act on the bearing, the relation ratio between the total of axial forces to the total of peripheral forces is described by the same series of numbers. The peripheral force F_c is not the only transverse load. (The other one is the radial force); however, due to the assumed constant ratio of radial force to the peripheral one, the ratio of peripheral force and of the carrying capacity of the bearing system F_c/C is widely treated as a parameter characterising the level of transversal load in a bearing system.

It is obvious that the presented models of bearing system and loads of the bearings do not cover the whole range of real working conditions related to angular ball bearings. Such coverage is impossible. The adopted plan for calculations should be used to illustrate the influence of preload on longitudinal stiffness, fatigue durability, and the frictional moment of the bearing system in selected cases considered frequent.

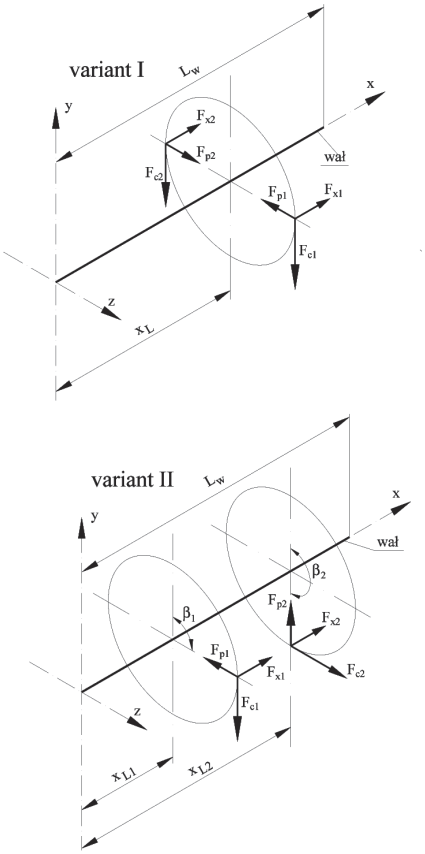


Fig. 2. Models of location of the loads in a bearing system
Rys. 2. Modele umiejscowienia obciążeń łożyskowania

Table 3. Locations of load planes and diameters D_t [mm] used in calculations
Tabela 3. Przyjęte do obliczeń położenia płaszczyzn obciążeń i średnice D_t [mm]

Bearing →		7209B
Variation I of the load location	x_L =	120
		150
		180
Variation II of the load location	x_{L1} =	120
	x_{L2} =	180
Diameter of rolling wheel	D_t	150

The main objective of the research was to obtain characteristics of the parameters enumerated above, depending on the preload of bearings, which are represented numerically by preload Z_c [mm]. While choosing the testing values for preload, the rule of flexible adjustment of its value was applied for further calculations – so as to achieve characteristics that are extensive enough. Through the phrase "extensive enough", one should understand the characteristics

that, together with the increase in preload, reveal the range of the acceptable deterioration of the enumerated parameters with the added margin. Flexible determining means carrying out further calculations when using increasing values Z_c , as long as the decrease in fatigue durability or the increase in the moment of friction are not critical in their value.

RESULTS

The graphs below present the results of calculations of the influence of preload on fatigue durability, and the moment of friction of the bearing expressed by the durability indexes W_T and the indexes of the moment of friction W_M.

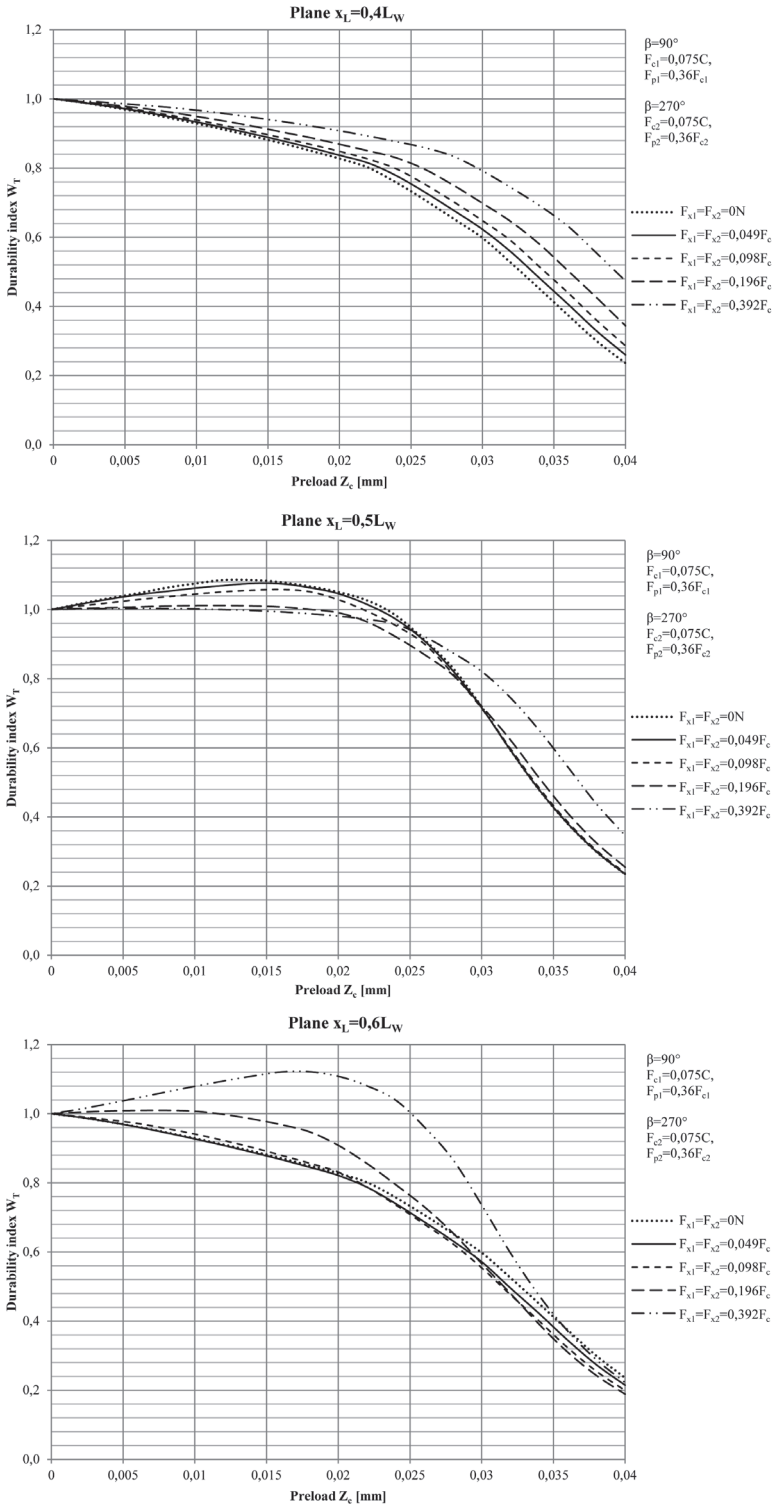


Fig. 3. Influence of preload Z_c on the durability index W_T for Variation I the load
Rys. 3. Wpływ zacisku wstępnego Z_c na wskaźnik trwałości W_T dla I wariantu obciążenia

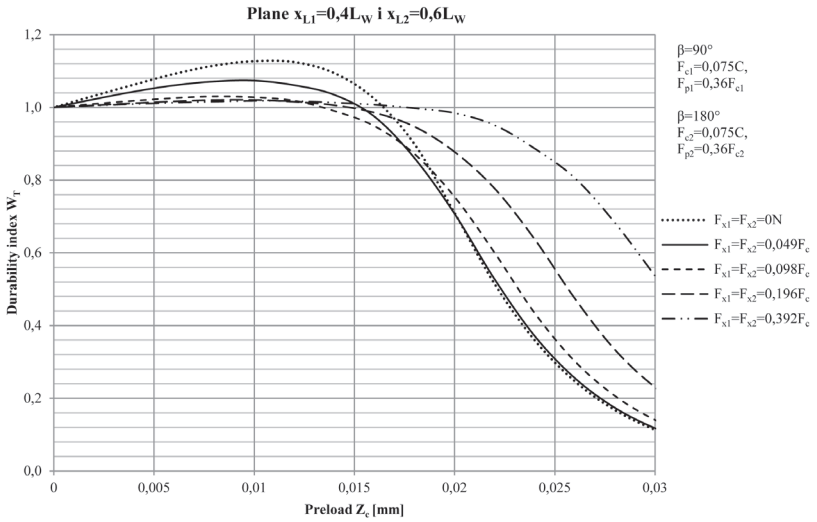
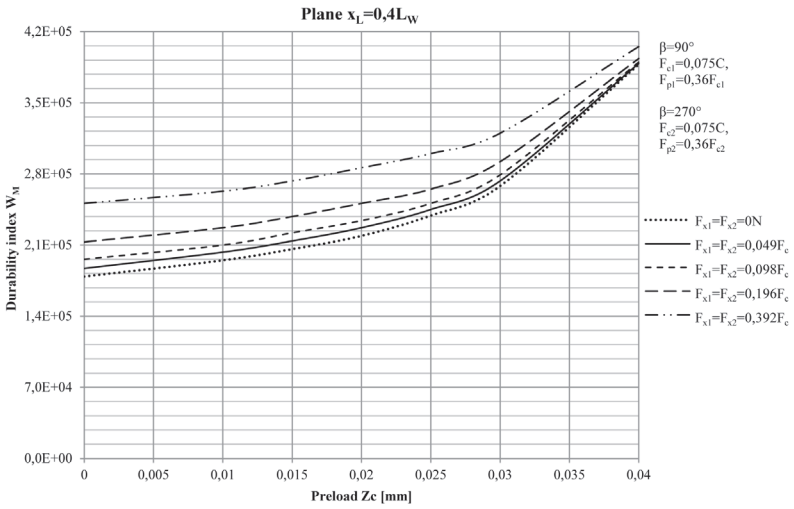
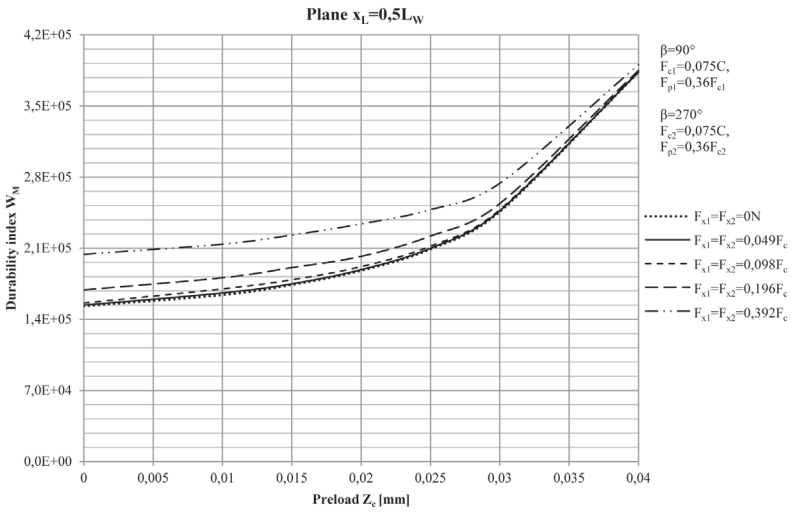
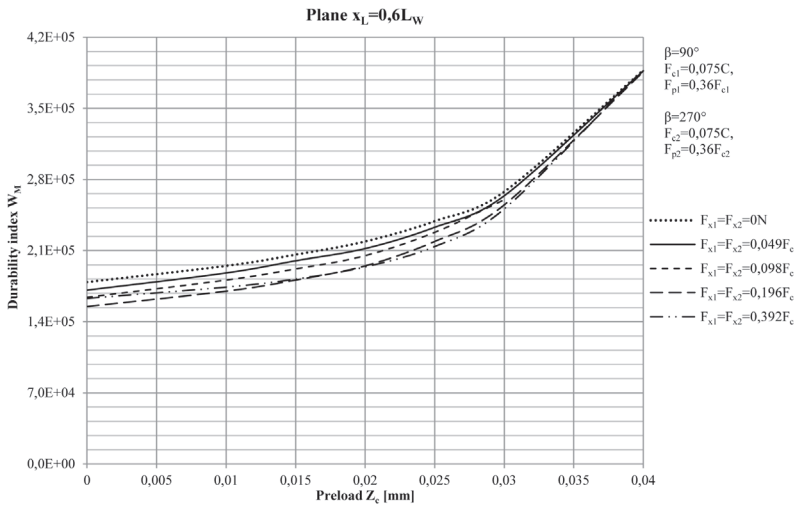


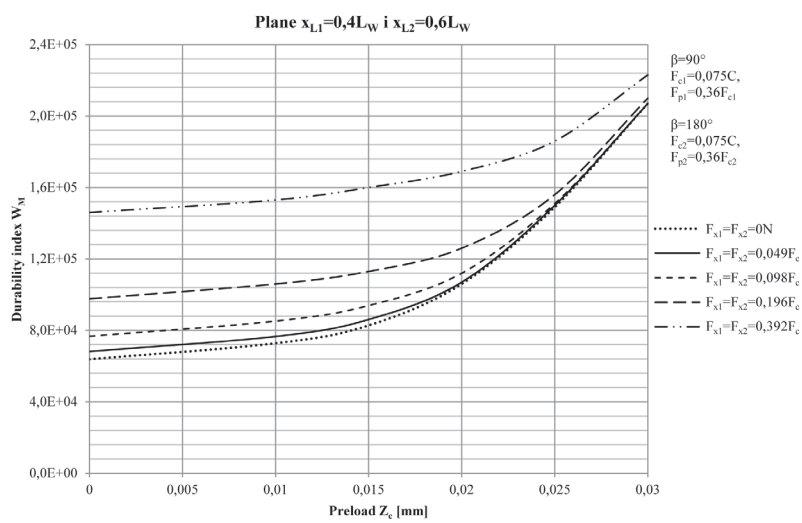
Fig. 4. Influence of preload Z_c on the durability index W_T for Variation II the load
Rys. 4. Wpływ zacisku wstępnego Z_c na wskaźnik trwałości W_T dla II wariantu obciążenia







Rys. 5. Influence of preload Z_c on the durability index W_M for Variation I the load
Rys. 5. Wpływ zacisku wstępnego Z_c na wskaźnik trwałości W_M dla I wariantu obciążenia



Rys. 6. Influence of preload Z_c on the durability index W_M for Variation II the load
Rys. 6. Wpływ zacisku wstępnego Z_c na wskaźnik trwałości W_M dla II wariantu obciążenia

FINDINGS

After the analysis of characteristics of the index W_X it can be stated that longitudinal stiffness increases together with the increase in preload. That increase is observed in every case, without any exception, which was naturally expected both by intuition and on the basis of literature information. The higher the preload, the higher is the first derivative of the increase.

The increase in longitudinal stiffness of a bearing system is a demanded phenomenon (and the main purpose of using a preload); therefore, from this point of view, there is no limitation on the value of preload. Limits appear due to another phenomenon, which is the increase in internal forces in the bearing, which causes the decrease in their durability, and as a result of the moment of friction.

The following observations result from the characteristics of the index W_T:

1. If loads are applied to the shaft closer to the left bearing (x_L = 0.4 L_w), regardless of other factors (value of transversal and axial loads), as the preload Z_c increases, the index of fatigue durability W_T decreases. The relative gradient of the decrease in the value of index W_T is quite similar for all the studied values of transversal loads. Moreover it can be noticed that the lower the values of axial force acting on the bearing system, the higher is the gradient of

the decrease. General observation: A decrease in the durability index in all cases in the series proves that the use of preload in this case is at the price of a significant decrease in the total durability of the bearing system.

2. If the loads are applied in the middle of the distance between the bearings ($x_F = 0.5\ L_w$), together with the increase in preload from zero value, characteristics of durability index W_T run differently: In some cases, they increase, in other cases – decrease. Still, as most of the W_T characteristics increase together with the increase in Z_c, this case was treated as demanding special interest. The following regularities can be observed on the charts:
   - The lower the axial force operating on the bearing system, the higher is the tendency to increase W_T characteristics in the initial phase of preload increase. Characteristics $W_{T,}$ in these cases, reach a certain maximum, and, after that, they begin to decrease aiming at the values lower than the ones for the zero load.
   - The higher the axial load operating on the bearing system, the lower is the tendency to increase W_T characteristics, and, in some cases, this tendency does not appear at all (W_T characteristics run initially horizontally or almost horizontally, and then they drop).
   - A stronger tendency to the increase in W_T characteristics above the level corresponding to $Z_c = 0$ manifests itself with lower transversal force loading on the bearing system.
   - It can be concluded, that, even the cases of a minimum decrease in W_T characteristics appear optimistic, because, at a certain range of preload, the drop is insignificant.
3. If the loads are applied closer to the right bearing ($x_F = 0.6\ L_w$), the characteristics of durability index have varied runs. These lines of durability index, which correspond with the highest assumed axial force ($F_x = 0392\ F_c$), are elevated above the level W_T with the value 1.0. Lines corresponding with $F_x = 0.196\ F_c$ at a very short distance exceed the level $W_T = 1$, and, for a minor preload they usually already start to decline. The remaining lines, corresponding to a lower axial load, have a clearly declining character. That tendency indicates the application of a low, acceptable preload.
4. In case when the second variation of the load location is applied (as in the **Fig. 3**), the obtained characteristics of the durability index rise to their highest and reach their maximum when they correspond to the lowest axial forces. This situation is predictable, because in this case the resultant of transversal loads is located exactly in the middle of the shaft's length. The difference in the courses can be explained by the fact that, in this case, the load of the shaft coming from axial forces is asymmetrical, which causes certain moments, in the plane x-z as well as in the plane x-y.

The term "right bearing" stands for the bearing receiving longitudinal loads. The term "left bearing" means the one located opposite to the former one.

When analysing the characteristics of W_M index, it is easy to notice that all these characteristics are illustrated by straight lines rising together with the increase in preload. They represent great similarity in shape. The increase in the resistance of movement results in the increased energy use, heat emission, the deteriorating of the smoothness of movement in delicate appliances (e.g., measuring devices). The rising character of the curves W_M indicates that the value of preload has to be limited due to the increase in movement resistances in the bearing system. At this point, a question arises: What is the maximum acceptable value for the increase in the resistance to motion? There is no finite answer to this question, because a lot depends on specific applications.

CONCLUSIONS

The influence of preload on the work of a system of bearings is not rectilinear. The result of applying preload in the aspect of durability is difficult to generalise. It is recommended to make calculations for a model appropriate for a specific bearing system.

In cases when durability of bearings tends to increase thanks to application of preload, the author suggest introducing preload, even in spite of the simultaneous increase in the resistance to motion. Other benefits from using preload are enumerated in the first paragraphs of the study.

REFERENCES

1. Harris T.A.: Rolling Bering Analysis: John Wiley & Sons, London 2006.
2. Hongqui Li, Yung C. Shin: Analysis of bearing configuration effects on high speed spindles using an integrated dynamic thermo-mechanical spindle model, Int. Journal of Machine Tools & Manufacture 44, 2004, pp. 347––364.

3. Jang G.H., Kim D.K., Han J.H., Kim C.S.: Analysis of dynamic characteristics of a HDD spindle system supported by ball bearing due to temperature variation. Microsystem Technologies 9, 2003, pp. 243–249.
4. Kaczor J., Raczyński 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, 2015, Vol. 229(6) pp. 723–732.
5. Kaczor J., Raczyński A.: The selection of preload in angular contact ball bearings according to the durability criterion, Tribologia 1/2018 (277), pp. 25–34.
6. Kaczor J., Raczyński A.: The selection of preload in angular contact ball bearings according to the criterion of moment of friction, Tribologia 1/2018 (277), pp. 35–44.
7. Kaczor J., Raczyński A.: The influence of preload on the work of angular contact ball bearings, Archive of Mechanical Engineering, Vol. 63, Issue 3/2016, pp. 319–336.
8. Mohammed A. Alfares, Abdallah A. Elsharkawy: Effects of axial preloading of angular contact ball bearings on the dynamics of a grinding machine spindle system, Journal of Materials Processing Technology 136, 2003, pp. 48–59.
9. Raczyński A.: The calculation method of deep groove ball bearings durability with regard to clearance and shaft, Scientific Problems of Machines Operation and Maintenance 34 (1)/1999, pp. 111–120.
10. Raczyński A.: The ball load in the angular contact ball bearing in dependence on the preload, Tribologia 1/2001, pp. 77–88.
11. Young-Kug Hwang, Choin-Man Lee: A Revue on the Preload Technology of the Rolling Bering for the Spindle of Machine Tools Int. Journal of Precision Engineering and Manufacturing, Vol. 11, No. 3/2010, pp. 491–496.
12. Catalog SKF 1991.
13. Katalog internetowy SKF, łożyska kulkowe skośne: http://www.skf.com/group/products/bearings-unitshousings/ball-bearings/angular-contact-ball-bearings/single-row/index.html?prodid=1210010206&imperial=false.