З питання про розподіл радіального навантаження між тілами кочення однорядного радіального підшипника запропоновані аналітичні та комп'ютерні моделі. Розглянуто особливості геометрії і деформації деталей, а також зазори в підшипнику без урахування різного положення центрів внутрішнього і зовнішнього кілець. Останнє може суттєво впливати на розподіл радіального навантаження між тілами кочення і бути причиною проковзування тіл кочення по кільцях.

Це дослідження спрямоване на уточнення розрахункового значення розподілу навантаження на тіла кочення підшипника. Чим ближче до істинного значення результат розрахунку розподілу навантаження і, особливо, на центральне тіло кочення, тим точніше визначений ресурс підшипника. Це багато в чому визначає надійність машини, а також витрати на ремонт і заміну підшипників. Не менш важливо при визначенні втомної міцності і зносостійкості сепаратора як можна точніше знати величини і напрямки дії радіальних сил на тіла кочення по краях зони навантаження підшипника. Саме ці сили спричинюють навантаження, що діють на сепаратор.

Отримано геометричні рівняння зв'язку радіальних зближень кілець, фізичні рівняння зв'язку зближень тіл кочення з кільцями і силами, умова рівноваги внутрішнього кільця з урахуванням різного положення центрів зовнішнього і внутрішнього кілець підшипника. Виявлено дотичні сили на бігових доріжках кілець, що викликають проковзування тіл кочення в зоні навантаження підшипника, та отримано формулу для визначення їх значень. Розрахунок розподілу навантаження між тілами кочення підшипника за розробленою аналітичної моделі показує зниження на 5 % радіальних сил на центральне тіло кочення і підвищення на 3 % радіальних сил на ролики по краях зони навантаження підшипника в порівнянні з відомою моделлю. Це забезпечує підвищення розрахункового ресурсу підшипника за контактно-втомними пошкодженнями кілець і тіл кочення на 18,6 %.

Нова аналітична модель розподілу радіального навантаження може знайти застосування в теорії розрахунку ресурсу підшипників підвищеної вантажопідйомності за контактно-втомними пошкодженнями кілець і тіл кочення з урахуванням конструкції сепаратора

Ключові слова: модель розподілу навантаження, тіла кочення підшипника, сили на доріжках кілець

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CONSTRUCTION OF A MODEL FOR THE DISTRIBUTION OF RADIAL LOAD AMONG THE BEARING'S ROLLING BODIES

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1. Introduction

To calculate the resource of bearings based on the criterion of contact endurance of rings and rolling bodies, it is very important to acquire data on the distribution of radial load. The more precisely this distribution is defined, especially a load on the bearing's central rolling body, the closer the result of resource estimation to the actual value. It is equally important in determining the fatigue strength and durability of the separator to know as accurately as possible the magnitudes and directions of the action of radial forces on the rolling bodies at the edges of the bearing loading zone. Therefore, a more detailed study into the distribution of radial load among rolling bodies, which would refine the magnitudes of forces, is a relevant task aimed at developing the theory of calculations of all elements in the bearing.

2. Literature review and problem statement

A problem on the static distribution of radial load among rolling bodies in the perfect single-row bearings is presented in [1]. In this case, the radial convergence of rings with the single center was estimated based on the magnitude of contact deformation of rolling bodies while the radial gap of the bearing and the ring curvature were not taken into consideration.

The analytical model from [2] provided a detailed analysis of the impact of the inner geometry of a roller bearing on distribution of the load among rolling bodies taking into consideration a radial gap. It was established that depending on the combination of magnitudes of the outer load and the radial gap, the load distribution changes. Great load magnitudes and lower values for a radial gap form a more favorable load distribution among rolling bodies. The draw-

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back of paper [2] is that the ring curvature was completely ignored.

Numerical calculation of static load distribution in a single-row angular contact ball bearing when it is exposed to the combined (radial and axial) load is given in [3]. The proposed procedure requires that Z+2 nonlinear equations should be iteratively solved simultaneously, where Z is the number of balls. This makes it difficult to derive an exact solution for the axial and radial deformations of the bearing with taking the contact angles into consideration.

A new approach to mathematical modeling of load distribution among the bearing rolling bodies was developed in [4]. Geometrical parameters of the bearing are selected directly from the respective catalogs, which makes it possible to compare any type of them. However, the computer model [4], as well as [3], does not elucidate the effect of rolling bodies slipping on rings of the bearing.

Experimental study into load distribution among rolling bodies using sensors based on optical fibers was proposed in [5]. However, there is no answer to the question about the patterns of interaction between rolling bodies and rings, related to slippage, in paper [5].

An analytical study on the influence of a radial gap on load distribution among rolling bodies, reported in work [6], provides a reasonably accurate solution. Error in the calculations amounted to no more than 0.5 %. However, the model from work [6] does not take into consideration differences in the positions of geometrical centers of the inner and outer rings.

Paper [7] describes a computer model and reports results of research into interaction between a rolling body and the inner ring of an angular contact ball bearing at different values for radial load. The derived dependences of change in the stresses in a contact area on the magnitude of load made it possible to estimate the performance of the bearing; they, however, did not explain the nature of rolling bodies slipping on rings.

An analysis of the published scientific papers [1–7] reveals that up to now there has been no solution to the problem on accounting for various positions of the centers of the inner and outer rings. An attempt to solve a specified task could involve a study to investigate the distribution of radial load among the bearing's rolling bodies depending on the various positions of centers of the inner and outer rings. A new analytical model for the distribution of radial load could be applied in the development of the theory to calculate the resource of bearings that have increased lift capacity [8] with the improved separator design [9].

3. The aim and objectives of the study

The aim of this study is to construct a refined analytical model of the distribution of radial load among rolling bodies of the bearing, considering the various positions of centers of the outer and inner rings, their curvature, and their radial gap.

To accomplish the aim, the following tasks have been set:

– to construct geometrical equations that would link the radial encounters of rings, physical equations that would relate encounters between the rolling bodies and the rings and forces, a condition for the equilibrium of the inner ring;

– to explore the distribution of radial load among the bearing's rolling bodies, derived using the developed, as well as known, analytical models.

4. Mathematical model of the radial load distribution among the bearing's rolling bodies

Schematic of static interaction between rolling bodies and bearing's rings is shown in Fig. 1.

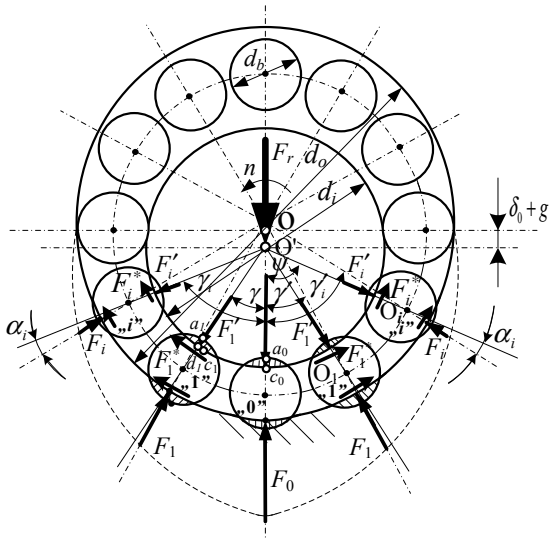

Fig. 1. Schematic of encounter between the outer and inner rings under load

When constructing a model of the radial load distribution among the bearing's rolling bodies considering the various positions of centers of the outer and inner rings, their curvature and radial gap, the following assumptions were accepted:

– the bearing's parts are of an ideal geometric shape;

– distortions of rings and twists of rollers are not taken into consideration;

– radial encounters between the outer and inner rings and rolling bodies at the expense of a contact deformation are the same along the assigned radial direction;

– dynamic effects predetermined by rotation of the inner ring do not affect the operation of parts. In their original state, centers of the outer and inner rings coincide. At the inner ring displacement vertically downwards by magnitude (δ_0+g), distance OO′; formed between the centers of the rings; g is the initial radial gap in the radial bearing (Fig. 2). The magnitude of g is determined along the radial direction and is computed from known expression [4] $2g=d_o-d_i-2d_b$.

Sides $\Delta OO'O_1$ (O_1 is the first rolling body's center) are determined from:

$$O_1O=\left[0{,}5d_o-\left(0{,}5d_b-0{,}5\delta_\gamma\right)\right],$$

$$O_1O'=\left[0{,}5d_i+\left(0{,}5d_b-0{,}5\delta_\gamma\right)\right],$$

$$\mathrm{OO'}=\delta_0+g, \qquad (1)$$

where d_o is the diameter of the rolling track of the outer ring; d_i is the inner ring's rolling track diameter; d_b is the diameter of a rolling body; δ_γ is the radial deformation in a contact between the rolling body and the outer and inner rings.

In a general case, the following ratio holds for any rolling body in the zone of radial load on the bearing between sections O_iO, O_iO' and OO' (Fig. 1):

$$[0,5d_o-(0,5d_b-0,5d_{\gamma i})+u_o)]^2=$$
$$=[(0,5d_i+(0,5d_b-\delta_{\gamma i})+u_i)]^2+(g+\delta_0)^2+$$
$$+2(g+\delta_0)[0,5d_i+(0,5d_b-\delta_{\gamma i})+u_i)\cos\gamma_i'. \quad (2)$$

Upon transform of expression (2), excluding low order higher than the second, we obtain geometrical equations that link radial encounters $\delta_{i\gamma}$ and δ_0 of rings in the neighborhood of any (i-th) and the central rolling bodies:

$$\delta_{i\gamma}=\delta_0\cos\gamma_i'+g(\cos\gamma_i'-1)-(u_o-u_i), \quad (3)$$

where the magnitudes u_o and u_i are the deformations, respectively, of the outer and inner rings (they are not shown in Fig. 1 because of their smallness compared to the initial radial gap in a bearing); u_o and u_i are computed for thick rings by integrating differential equation [10] of the curved axis in a crooked beam

$$\frac{\partial^2 u}{\partial\theta^2}+u=\frac{MR^2}{EI}, \quad (4)$$

where u is the deformation; θ is the angular coordinate; M is the bending moment; R is the ring's radius; I is the moment of inertia of the ring's cross section; E is the modulus of elasticity of a material. Expressions for the deformations of the outer and inner rings are then:

$$u_o=\frac{F_i'\cdot R_o^3}{4E\cdot I_o}\cos\gamma_i-\frac{F_i'\cdot R_o^3}{E\cdot I_o\cdot\pi}+\frac{F_i'\cdot R_o^3}{4E\cdot I_o}\gamma_i'\cdot\sin\gamma_i';$$

$$u_i=\frac{F_i'\cdot R_i^3}{4E\cdot I_i}\cos\gamma_i-\frac{F_i'\cdot R_i^3}{E\cdot I_i\cdot\pi}+\frac{F_i'\cdot R_i^3}{4E\cdot I_i}\gamma_i'\cdot\sin\gamma_i',$$

where F', F_i' are the forces acting on the outer and inner rings; γ', γ_i' are the angles of contact between the outer and inner rings; I_o, I_i are the moments of inertia of cross sections of the outer and inner rings, respectively; R_o, R_i are the radii of the outer and inner rings.

The angle of the bearing's loading zone ψ is determined from expression (3) under condition $\delta_{i\gamma}=0$, when $\gamma_i'=\psi$,

$$\psi=\arccos\frac{g+u_o-u_i}{g+\delta_0}.$$

Physical equations for the relation between balls' encounters $\delta_{i\gamma}$ with rings and forces F_i acting on the balls, with respect to expression (3), take the form:

$$\delta_0=C_b\cdot F_0^{2/3},$$
$$\delta_1=\delta_0\cos\gamma_1'+g(\cos\gamma_1'-1)-(u_o-u_i)=C_b\cdot F_1^{2/3},$$
$$\delta_n=\delta_0\cos\gamma_n'+g(\cos\gamma_n'-1)-(u_o-u_i)=C_b\cdot F_n^{2/3}, \quad (5)$$

where C_b is a constant magnitude [10], defined by the mechanical properties of materials and the geometry of parts

$$C_b=1,31\cdot\sqrt[3]{\frac{1-\mu^2}{E}\cdot\frac{R_1+R_2}{R_1\cdot R_2}}.$$

Ratio between the left and right parts of expressions (5) yields relationship between F_0 and F_i

$$\delta_0/\delta_1=\delta_0\Big/\left(\delta_0\cdot\cos\gamma_1'+g(\cos\gamma_1'-1)-(u_o-u_i)\right)=(F_0/F_1)^{2/3}, \quad (6)$$
$$\delta_0/\delta_n=\delta_0\Big/\left(\delta_0\cdot\cos\gamma_n'+g(\cos\gamma_n'-1)-(u_o-u_i)\right)=(F_0/F_n)^{2/3},$$

hence, the distribution of forces between rollers

$$F_1=F_0\left(\cos\gamma_1'+\frac{g}{\delta_0}(\cos\gamma_1'-1)-\frac{1}{\delta_0}(u_o-u_i)\right)^{3/2},$$

$$F_n=F_0\left(\cos\gamma_n'+\frac{g}{\delta_0}(\cos\gamma_n'-1)-\frac{1}{\delta_0}(u_o-u_i)\right)^{3/2}. \quad (7)$$

By applying an equilibrium condition for the inner ring, which is under the action of radial load F_r and forces F_i,

$$F_r=F_0+2F_1\cdot\cos\gamma_1'+...+2F_n\cdot\cos\gamma_n',$$

one can determine the force acting on the most loaded ball

$$F_0=F_r\Bigg/\left(1+2\sum_{i=1}^{n}\cos\gamma_i'\left(\cos\gamma_i'+\frac{g}{\delta_0}(\cos\gamma_i'-1)-\frac{1}{\delta_0}(u_o-u_i)\right)^{3/2}\right).$$

At g=0, u_o=u_i=0, we obtain an expression for the force acting on the most loaded rolling body in a perfect ball bearing.

Physical equations of relationship for encounters $\delta_{i\gamma}$ between rollers and rings and forces F_i, which act on rollers, taking into consideration expression (3), take the following form:

$$\delta_0=C_{r0}\cdot F_0,$$
$$\delta_1=\delta_0\cos\gamma_1'+g(\cos\gamma_1'-1)-(u_o-u_i)=C_{r1}\cdot F_1,$$
$$\delta_n=\delta_0\cos\gamma_n'+g(\cos\gamma_n'-1)-(u_o-u_i)=C_m\cdot F_n, \quad (8)$$

where C_{ri} is the variable from forces F_i, defined by the mechanical properties of materials and the geometry of parts [10],

$$C_{ri}=\frac{0,579}{l\cdot E}\left[\ln\frac{1,727l\cdot E\cdot(R_1+R_2)}{F_i}+0,814\right].$$

Ratio of the left and right parts of expressions (8) yields a relationship between F_0 and F_i

$$\delta_0/\delta_1=\delta_0\Big/\left(\delta_0\cos\gamma_1'+g(\cos\gamma_1'-1)-(u_o-u_i)\right)=$$
$$=c_{r01}(F_0/F_1),$$

$$\delta_0/\delta_n=\delta_0\Big/\left(\delta_0\cos\gamma_n'+g(\cos\gamma_n'-1)-(u_o-u_i)\right)=$$
$$=c_{r0n}(F_0/F_n), \quad (9)$$

where

$$C_{r01}=C_{r0}/C_{r1},...,C_{r0n}=C_{r0}/C_m;$$

hence, the distribution of forces between:

$$F_1=C_{r01}\cdot F_0\left(\cos\gamma_1'+\frac{g}{\delta_0}(\cos\gamma_1'-1)-\frac{1}{\delta_0}(u_o-u_i)\right);$$

$$F_n = C_{r0n} \cdot F_0 \left(\cos\gamma_n' + \frac{g}{\delta_0}\left(\cos\gamma_n' - 1\right) - \frac{1}{\delta_0}\left(u_o - u_i\right) \right). \quad (10)$$

By applying an equilibrium condition for the inner ring, which is under the action of forces F_r and F'_i,

$$F_r = F_0 + 2F_1' \cdot \cos\gamma_1' + \ldots + 2F_n' \cdot \cos\gamma_n',$$

one can determine the force that acts on the most loaded roller:

$$F_0 = F_r \Bigg/ \left(1 + 2\sum_{i=1}^{n} C_{r0i} \cdot \cos\gamma_i' \left(\begin{array}{l} \cos\gamma_i' + \frac{g}{\delta_0}\left(\cos\gamma_i' - 1\right) - \\ -\frac{1}{\delta_0}\left(u_o - u_i\right) \end{array} \right) \right). \quad (11)$$

A deviation in radial forces F_i of the interaction between the outer ring and the rolling bodies from the directions of radial forces F_i' $(F_i = F_i')$ of interaction between the rolling bodies and the inner ring on the contact angles α_i predetermines the occurrence of tangential forces F_i^* along the rolling tracks of the rings (Fig. 1). The magnitudes of forces F_i^* are determined from expression:

$$F_i^* = F_i \sin\left\{ \arccos \frac{\left[0{,}5d_o - 0{,}5\left(d_b - \delta_\gamma\right) + u_i\right]^2 + \left[0{,}5d_i - \left(d_b - \delta_\gamma\right) + u_i\right]^2 - (g + \delta_0)^2}{2\left[0{,}5d_o - 0{,}5\left(d_b - \delta_\gamma\right) + u_o\right]\left[0{,}5d_i - 0{,}5\left(d_b - \delta_\gamma\right) + u_i\right]} \right\}. \quad (12)$$

The direction of forces F_i^* is such that the motion of a rolling body that enters the loading zone is decelerated, but it is accelerated at the output from this zone. For example, in bearings of type 2726, installed at anchor nodes of wheel sets of railroad carriages, forces F_i^* can contribute to the slip of rollers, wear and destruction of their separators [11].

5. Results of studying the distribution of radial load among rolling bodies of the bearing

Results of calculation of the radial load distribution F_r= =50 kN among the rollers of bearing, type 2726, are shown in Fig. 2.

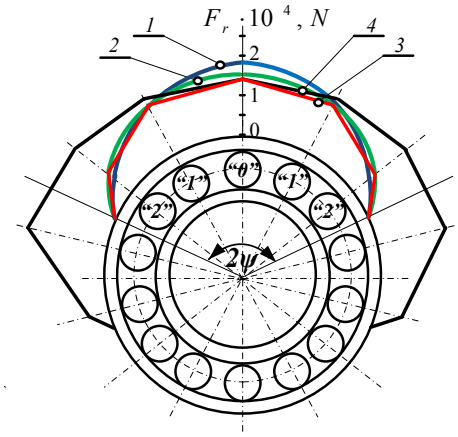

Fig. 2. Distribution of radial load: 1 – estimated based on known model [6]; 2 – estimated based on the constructed model (formulae 10, 11); 3 – under conditions of bench testing [12]; 4 – in a train [12]

Radial load distributions among rolling bodies of bearing, derived based on the constructed model taking into consideration the different positions of centers of the outer and inner rings (the angles of contact between rolling bodies and rings), radial gap, rigidity of rings, are not much different from those obtained from known model (maximum deviation does not exceed 5 %). In this case, the constructed model produces 5 % lower values for the radial force on the central roller, and 3 % larger values for the radial forces on rollers at the edges of the bearing's loading zone.

Results of calculation of tangential forces F_i^* in the bearing's load zone using the constructed model depending on the rigidity of rings and the number of rollers z are shown in Fig. 3.

Tangential forces F_1^* in the neighborhood of a first roller are significantly (by 1.5...2 times) higher than tangential forces F_2^* in the neighborhood of a second roller, though an increase in rigidity of the outer ring leads to a decrease in the difference between them. An increase in rigidity of the outer ring leads to a decrease in tangential forces in the vicinity of a first roller, while they increase in the neighborhood of a second roller. In this case, tangential forces in the neighborhood of a second roller grow in proportion to an increase in the number of rollers in the bearing.

Results of calculation of force F_i^* in the bearing's loading zone depending on the number of rollers, their location, and the magnitude of a radial gap, are shown in Fig. 4.

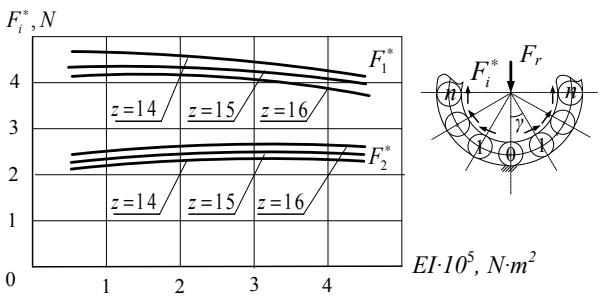

Fig. 3. Changes in the estimated values for tangential forces F^*_i depending on the number of rollers and rigidity of rings

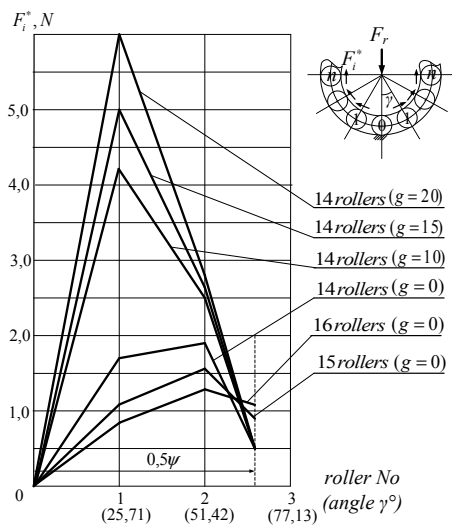

Fig. 4. Changes in the estimated values for tangential forces F^*_i depending on the number of rollers, their location in the bearing, and a radial gap

For bearings (Fig. 4) with radial gaps (g>0), the tangential forces in the vicinity of the central roller are absent (F_0^*=0); in the neighborhood of a first roller, they reach maximum values; when they approach the boundaries of a loading zone, they decrease. In the absence of radial gaps in bearings (g=0), tangential forces reach the maximum value in the neighborhood of a second roller close to the boundaries of a loading zone. An increase in the number of rollers in the bearing contributes to an increase in tangential forces at the edges of a loading zone.

6. Discussion of results of studying the distribution of radial load among rolling bodies of the bearing

The need to refine the calculation of static distribution of radial load among rolling bodies in a single-row radial bearing arose from the experimentally observed phenomenon of rollers slipping at the input and output from the loading zone of bearings in the railroad carriages' axle nodes [11]. The slipping of rollers along the rings' rolling tracks must inevitably lead to the occurrence of friction forces, that is the tangential forces in the neighborhood of boundaries of the bearing's loading zone. However, known theoretical solution to the problem on static distribution of radial load among rolling bodies in a single-row radial bearing contains no information about any tangential forces at all. Such tangential forces should contribute to an increase in load on the separator and, hence, affect its strength and durability. In this context, refinement of the calculation of static distribution of radial load among rolling bodies in a single-row radial bearing is of particular relevance.

The emergence of tangential forces along the rings' rolling tracks is the consequence of divergent lines along which the interaction forces between rolling bodies and the outer and inner rings act, due to the different position of their centers, which has not been taken into consideration up to now. The direction of tangential forces is that they slow down the motion of rolling bodies towards the middle of the bearing's loading zone, but accelerate the motion at the output from the specified zone. Tangential forces are absent in a contact between the central rolling body and the rings' rolling tracks.

All the facts and conclusions established above are valid for any type of single-row radial bearings; the model for the refined calculation of static distribution of radial load among rolling bodies was tested experimentally [12]. The difference between the refined estimated distribution of radial load among the rolling bodies of bearing in the journal box of railroad cars and that experimentally derived is explained by special features in the structure of parts in the running gears of railroad cars. Under operating conditions for railroad cars, the bearings are exposed to the torque loads both in the vertical and in the horizontal planes with subsequent distortions of rings and load concentration at the edges of rollers. The latter is the reason for the edge contact between the rollers and the bearing's rings, which is accounted for only experimentally. A theoretical solution to the problem on assessing the impact of distortion of bearing's rings on its operational capability has not been investigated in detail up to now. The spread of permissible dimensions for bearing parts, which is not considered in this study, significantly affects the distribution of radial load among the rolling bodies of bearing. Thus, the factors related to the skewness of rings and spread in the dimensions of parts impose a limit on solving the problem on the refinement of an analytical model of the distribution of radial load among the bearing's rolling bodies. However, consideration of various positions of centers of the outer and inner rings shows the way to reasonably prolong the service life of bearing operation, which is an advantage of this study compared with known analogs. Therefore, undertaking a theoretical research into the influence of ring distortions and a random character of change in the shape and size of parts in a radial bearing on the distribution of radial load among rolling bodies could further advance our study

7. Conclusions

1. We have constructed geometrical equations that link the radial encounters of rings, physical equations that relate the encounters of rings and rolling bodies to forces, a condition for equilibrium of the inner ring taking into consideration the different positions of centers of the outer and inner rings of the bearing. The latter predetermines a deviation in the radial forces of interaction between the outer ring and the rolling bodies from the directions of radial forces of interaction between the rolling bodies and the inner ring on contact angles and the occurrence of tangential forces along the rings' rolling tracks, which cause the slippage of the rolling bodies in the bearing's load zone. We have derived a formula for determining the values of tangential forces based on the rings' rolling tracks.

2. The calculation of distribution of radial load among the bearing's rolling bodies based on the constructed analytical model demonstrates a 5 % decrease in radial forces that act on the central rolling body, and a 3 % increase in radial forces that act on rollers at the edges of the bearing's loading zone compared to known model. That improves the estimated resource of the bearing in terms of contact-fatigue damage to rings and rolling bodies by 18.6 %.

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COMPACTION OF POROUS POWDER BODY CONSISTING OF THE ELASTIC-PLASTIC MEDIUM

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При розробці технологічних процесів отримання холоднопресованих спечених деталей малої пористості особлива увага приділяється механізму зміни щільності. На практиці порошкової металургії дуже часто використовують багатокомпонентні шихти, що складаються як з пластичних металів, так з погано стискаємих включень і з'єднань. До такої шихти в рівній мірі можна віднести і шихту, що складається з порошку заліза, чавуну і скла. У цій шихті перший компонент (основа) – це пластичне залізо, а два інших – чавун і скло – пружні компоненти. Викликає певний інтерес, яке ущільнення може бути отримано в такому випадку і які результуючі рівняння можна застосувати для оцінки механіки ущільнення такої порошкової шихти.

Пропонуються результуючі рівняння ущільнення пористих порошкових тіл із залізо-чавун-скло. Наведено аналіз ізотропного, жорсткопластичного зміцнюючого матеріалу типу залізо-чавун-скло. При ущільненні такого матеріалу швидкість дисипації енергії (тиск пресування) визначається швидкостями зміни обсягу тіла і його формозміни. Показано, що відмінність стискаємих (чавун і скло) і пластичних (залізо) матеріалів, що ущільнюються, формує особливі механічні властивості матриці. Отже, гідростатичний тиск може впливати на формозміну тіла, а дотичні напруження – на зміну обсягу. Отримані результати математичного підходу для отримання результуючих рівнянь ущільнення пружнопластичного середовища показали шлях побудови теорії пластичності тіла, що ущільнюється, при якому виключається необхідність врахування виду поверхні навантаження. При обліку поверхні навантаження неможливо отримання універсальних рівнянь з ущільнення пористого пружнопластичного середовища. Показано, що для застосування класичного формулювання моделі пружнопластичного тіла, що ущільнюється, необхідно вважати, що поверхня навантаження опукла-замкнута

Ключові слова: поверхня навантаження, пористе тіло, результуючі рівняння, залізо-чавун-скло, швидкість деформації, інваріант тензора, ізотропний матеріал

1. Introduction

A consistent phenomenological description of the processes of formation of powders and porous bodies of the elastic-plastic medium as the most important element eliminates the choice of governing or rheological equations. For sintering and hot pressing, thanks to the works [1, 2], some clarity in understanding of this issue has been achieved, while for cold molding processes characterized by plastic flow, there is no consensus about the type of governing equations. In this regard, the formation of general restrictions imposed on such equations, based on the current concepts of irreversible thermodynamics and continuum mechanics, is relevant. In this case, an approach to constructing a theory of plasticity should be used, based on setting the properties of the dissipative function [3–6].

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