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CHECKING CALCULATIONS OF SLIDING BEARINGS WITH FLOATING RING

In the paper, the authors investigate the assumptions concerning checking calculations of a sliding bearing with a floating ring. The adiabatic model of the bearing is used for the calculations. Particular design stages are depicted in the form of a structural chart. The proper work conditions of the bearing are formulated and discussed. For the presented calculation example, the authors have determined the area of feasible solutions. In the conclusion, the authors formulate suggestions that might be useful for the designers who deal with these types of bearings.

Nomenclature

B – bearing bush width [m], $B^{*}= B/D_1$ – bearing bush relative width, c_p – specific heat measured at the constant pressure [J/kg·°C], C_{Ri} – radial clearance [m], C_R^* – quotient of radial clearance, D – diameter [m], e – eccentricity [m], F – bearing load [N], $F^* = \dfrac{4 \cdot \psi_1^2}{B^* \cdot D_1^2 \cdot \eta_o \cdot \omega_1} \cdot F$. – dimensionless bearing load, F_L – oil film load capacity [N], h – lubricant gap height [m], $h_i^* = \dfrac{h_i}{C_{Ri}} = 1 + \varepsilon_i \cos(\varphi_i - \beta_i)$ – dimensionless lubricant gap height, M – moment [Nm], n – rotational speed [rps], p – pressure [N/m2], $p_i^* = \dfrac{p_i \cdot \psi_i^2}{\eta_o \cdot \omega_i}$ – dimensionless pressure in oil film, R – radius [m], T – temperature [°C], $T_i^* = \dfrac{\rho \cdot c_p \cdot \psi_i^2}{\eta_o \cdot \omega_i}(T_i - T_o)$ – dimensionless temperature in oil film, v – linear velocity [m/s], x = φ· R – Cartesian coordinate in circumferential direction,

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y – Cartesian coordinate in radial direction, z – Cartesian coordinate in axial direction, β – angle contained between straight line going through the centre of journal and floating ring – i= 1, or floating ring and fixed bearing bush – i=2 and positive turn of axis y, $\varepsilon_i = e_i / C_{Ri}$ – eccentricity ratio, $\varepsilon_0 = e_0 / C_{R0} = e_0 / (C_{R1} + C_{R2})$ – resultant relative eccentricity, η – oil dynamic viscosity [Pa·s], η_o – oil viscosity in reference temperature, λ – thermal conductivity factor [W/m ·°C], ρ – oil density [kg/m^3], $\psi_i = C_{Ri}/_{Ri}$ – relative clearance, ω_i – angular speed [1/s],

Subscripts; J – relates to journal for inner oil film it is journal rotor, whereas for outer oil film it is floating ring bearing, i=1- relates to inner oil film, i=2- relates to outer oil film, all- relates to allowable dimension, max- maximum dimension, min- minimum dimension.

1. Introduction

The design calculations can refer to either bearing or its particular elements [2], [5], [6], [7]. The selection of parameters can be done by means of:

– catalogues,
– graphs or charts,
– computer programs.

The design calculations can be done in a dimensional or dimensionless form, as well as in a mixed form. This work will discuss assumptions concerning checking calculations of a floating ring bearing (Fig.1) with the use of graphs.

The constructional elements of a bearing are (Fig.1): the fixed bearing bush (2), and the loosely fixed floating ring (1) separating the journal and the fixed bearing bush- known as the floating ring bearing. The oil is supplied into the outer and the inner bearing under the pressure through the holes (3) which are in the fixed bearing bush and the floating ring bearing. The circumferential lubricant grooves (4) in the fixed bearing bush or the floating ring bearing provide regular oil feed to the lubricant gaps. The directions of oil flow in a bearing are marked with arrows (6).

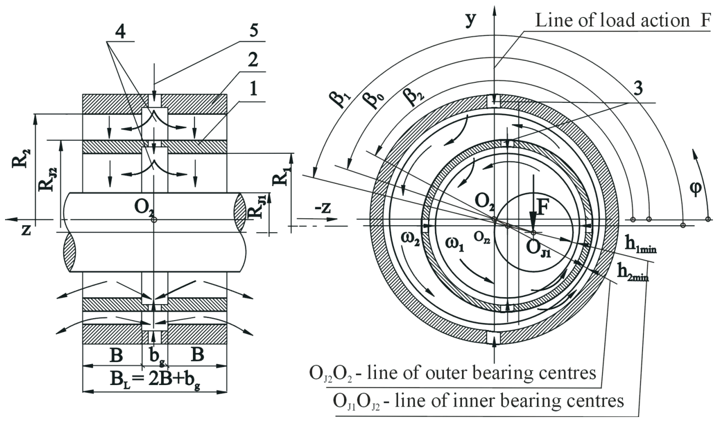

Fig. 1. Sliding bearing with a floating ring

Pressures' resultants in oil films, the hydrodynamic bearing load capacity (F_L) balances the outer load on the bearing (F), which is applied to the bearing journal. The adiabatic model of a bearing was accepted for calculations. A planar model of the oil flow was accepted, which means the flow consistent with circumferential direction – x and an axial direction – z. The produced heat is carried away by the oil flowing from the bearing. Other assumptions concerning physical and mathematical model are presented in work [8]. The system of forces and moments is described in Fig. 2.

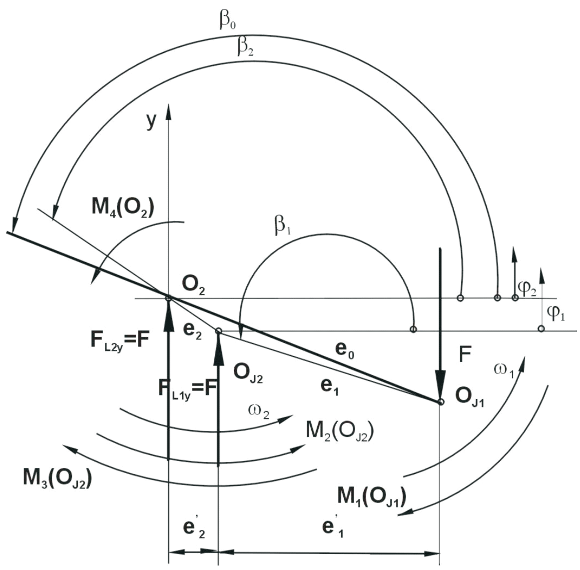

Fig. 2. System of forces and moments in a bearing

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2. General principles of the design method

According to the general construction theory [1], [11], the construction process can be depicted in the form of a structural diagram [3], Fig.3. In construction process, we can distinguish the following stages:

1. Assumptions, concepts, initial solutions.
2. Analysis of the assumptions and initial solutions, division of structural parameters into self-imposed, assumed, resultant.
3. Initial calculations.
4. Preliminary project.
5. Checking calculations.
6. Optimization analysis, the choice of the best solution.
7. Structural specification.

Preliminary calculations determine characteristics of a construction being designed. These characteristics are: geometric, material, and kinematic ones. Nominal values are used to determine construction characteristics. In the initial calculations of a sliding bearing with a floating ring [9], and [10], we use the characteristics of the bearing, where we look for a resultant parameter – the position of the journal centre relative to the fixed bearing bush, taking into account the specific self-imposed and assumed parameters of the bearing:

$$\varepsilon_0 = \varepsilon_0(\varepsilon_1, \varepsilon_2), \tag{1}$$

where, ε_1 – relative eccentricity of inner oil film, ε_2 – relative eccentricity of outer oil film.

The static characteristics, which are used to determine the resultant relative eccentricity, are described by the dependence:

$$F^* = F^*\left(n_1, \varepsilon_0, B^*, \psi_1, C_R^*, \eta_o, T_z, p_z\right) \tag{2}$$

Exemplary characteristics are presented in the paper [9] and in Fig. 4.

The checking calculations require changing bearing characteristics in such a way that the checking of work conditions is done for specific criteria based on dynamic, kinematic, and operating qualities. These dependences are expressed by the function:

$$\varepsilon_0 = \varepsilon_0\left(p^*_{i\max}, T^*_{i\max}, p^*_{iall}, T^*_{iall}, h^*_{i\min}, h^*_{ilall}\right) \tag{3}$$

The characteristics for the static balance position are depicted in the form of a chart in Fig. 5-7. The result of this calculation stage is a final design of the construction including the tolerances: dynamic, kinematic, and operating ones.

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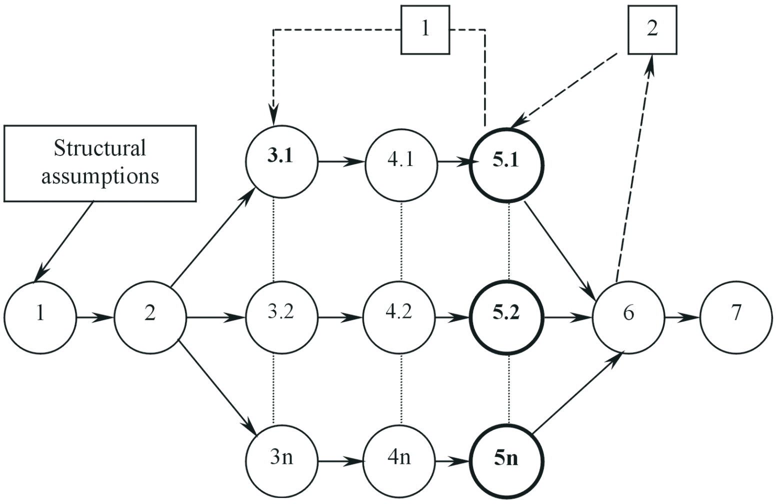

Fig. 3. Structural chart of construction process

3. Determination of the area of feasible solutions

The following inequalities describe the conditions of a regular work of the bearing :

$$\begin{aligned} \varepsilon_0^{Ti\max} &\le \varepsilon_0^{Tiall} \\ \varepsilon_0^{pi\max} &\le \varepsilon_0^{piall} \\ \varepsilon_0^{hi\min} &\le \varepsilon_0^{hiall} \end{aligned} \tag{4}$$

The above conditions describe:

– The position of the bearing journal relative to the fixed bearing bush taking into account maximum and acceptable temperature of the oil film. This condition specifies acceptable temperature which can exist in the bearing, which ensures normal operating qualities of the used lubricant agent.
– The position of the bearing journal relative to the fixed bearing bush taking into account maximum and acceptable pressure in the oil film. This condition specifies the permissible thrust at which there are no plastic strains in the bearing's material.
– The position of the bearing journal relative to the fixed bearing bush taking into account the minimum and the acceptable height of the oil film. This condition guarantees low friction of fluid in the bearing.

The area of feasible solutions is determined by the resultant relative eccentricities, which are calculated from equation (3), together with the limitations imposed by the limiting work conditions, done by inequalities (4).

This investigation has revealed that the maximum values of pressure and temperature, and the minimal values of oil film height can appear in the outer or the inner oil film.

Calculation example.

To determine the area of feasible solutions, we have choosen a bearing specified by geometry and feeding conditions in the form of: D_1, C_R^*, ψ_{1eff}, T_z, p_z, B^*, $\eta(T)$, $\rho(T)$, $c_p(T)$, λ. Also, three different values of the quotient of radial clearance have been assumed: $C_R^* = 0{,}5;\ 1{,}1;\ 2{,}0$.

The bearing journal works in the range of rotational speed $< n_1^{min} \div n_1^{max} >$ and is loaded by a static force F^* (Table1).

Table 1.

Operating parameters of the bearing

Given parameters	Structural task					
Rotational speed of journal [rps]	,n_1^{min}	n_1^{max}	n_1^{min}	n_1^{max}	n_1^{min}	n_1^{max}
	100	400	100	400	100	400
Diameter of journal D_{J1} [m]	$120 \cdot 10^{-3}$					
Relative width B^*	0,52					
Relative real clearance of inner oil film ψ_{eff}	$1{,}4\,^0/_{00}$					
Quotient of radial clearance C_R^*	0,5		1,1		2,0	
Dimensionless bearing load F^*	0,1		0,1		0,1	
Reference temperature T_0 [°C]	20					
Feeding temperature T_z [°C]	30					
Pressure of oil feeder p_z [MPa]	0,1					
Type of oil: Turbo Shell 29, $\eta(T)$ [Pa·s], $\rho(T)$ [kg/m^3], $c_p(T)$ [J/kg·^{0}C], λ [W/m·^{0}C],	$\eta(T) = \eta_O \cdot e^{a_\eta \cdot (T-T_0) + b_\eta \cdot (T-T_0)^2}, \rho(T) = a_\rho + b_\rho \cdot T + d_\rho \cdot T^2$ $c_p(T) = a_c + b_c \cdot T + d_c \cdot T^2, \lambda = 0,145$, where: η_o=0,1084, $a_\eta = -0,55291 \cdot 10^{-1}, b_\eta = -0,239 \cdot 10^{-3}, a_\rho = 896,25, \cdot b_\rho = -1,437$, d_ρ=0,625 $\cdot 10^{-2}$, $a_c = 1802,1$, $b_c = 2,878$, $d_c = 87 \cdot 10^{-2}$					
Allowable temperature of oil film T_{all} [°C]	110					
Allowable pressure of oil film p_{all} [MPa]	7					
minimal height of oil film h_{all}[mm]	0,02					
Results of initial calculations						
Quotient of radial clearance C_R^*	0,5		1,1		2,0	
Eccentricity $\varepsilon_0(n_1{=}100)$	0,15		0,3		0,38	
Eccentricity $\varepsilon_0(n_1{=}400)$	0,3		0,4		0,5	

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The values of resultant relative eccentricity ε_0 (Table1), which have been determined from static characteristics (Fig. 4) are consistent with the specified operating parameters (F^*, n_1).

The limitation of normal work of the bearing is defined in a dimensional form through: p_{all}, T_{all}, h_{all}. The investigation has revealed that the influence of journal rotational speed on the minimal height of oil film is insignificant in the examined case (n_1= 100- 400 rps). Due to this fact, it has been accepted that for n_1 = 100 and n_1 = 400 rps the value of minimal height of the oil film is the same.

The values of allowable resultant relative eccentricity, presented in table 2, were calculated for the values of quotient of radial clearance equal to 0,5; 1,1 and 2,0, and rotational speeds of the journal n_1 =100 and 400 rps.

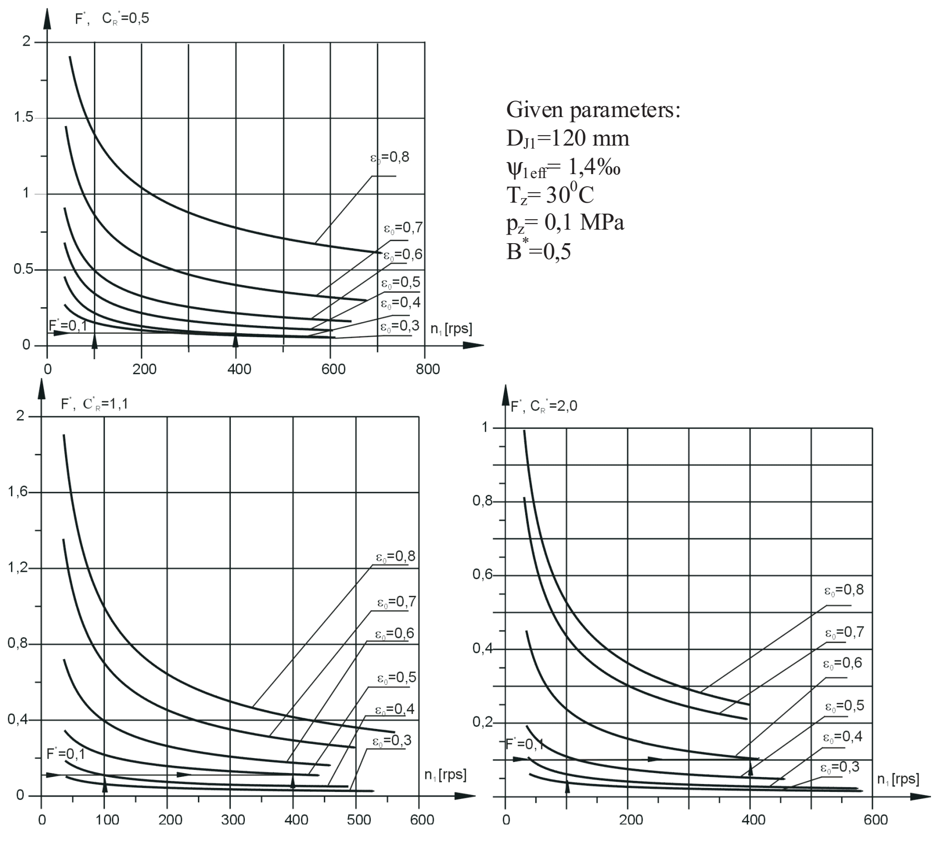

Fig. 4. Static characteristics of the bearing

Having examined the results shown in Table 2 and in Fig. 5 or 6, we can state:

Table 2.

Allowable values of resultant relative eccentricity

<table>
<tr><td rowspan="9">Fig.5 and 6</td><td colspan="5">C_R^*=0,5</td></tr>
<tr><td>H-
$\varepsilon_0^{\mathrm{hlall}} = 0,78$
n_1= 100- 400 rps</td><td>A-
$\varepsilon_0^{\mathrm{plall}}$=0,49
n_1= 100 rps</td><td>L-
$\varepsilon_0^{\mathrm{plall}}$=0,37
n_1= 400 rps</td><td>$T^*_{1\max} < T^*_{1\mathrm{all}}$ in the examined case
n_1= 100 rps</td><td>T-$\varepsilon_0^{\mathrm{Tlall}} = 0,54$
n_1= 400 rps</td></tr>
<tr><td>I-
$\varepsilon_0^{\mathrm{h2all}}$ =0,45
n_1= 100- 400 rps</td><td>D-
$\varepsilon_0^{\mathrm{p2all}}$ = 0, 5
n_1= 100 rps</td><td>O-
$\varepsilon_0^{\mathrm{p2all}}$= 0,38
n_1= 400 rps</td><td>$T^*_{2\max} < T^*_{2\mathrm{all}}$
in the examined case
n_1= 100 rps</td><td>$T^*_{2\max} < T^*_{2\mathrm{all}}$
in the examined case
n_1= 100 rps</td></tr>
<tr><td colspan="5">C_R^* =1,1</td></tr>
<tr><td>$h^*_{1\min} > h^*_{1\mathrm{all}}$
in the examined case
n_1= 100- 400 rps</td><td>B-
$\varepsilon_0^{\mathrm{plall}}$ = 0, 54
n_1= 100 rps</td><td>M-
$\varepsilon_0^{\mathrm{plall}}$= 0,48
n_1= 400 rps</td><td>$T^*_{1\max} < T^*_{1\mathrm{all}}$
in the examined case
n_1= 100 rps</td><td>U-
$\varepsilon_0^{\mathrm{Tlall}}$= 0,66
n_1= 400 rps</td></tr>
<tr><td>J-
$\varepsilon_0^{\mathrm{h2all}}$= 0,66
n_1= 100- 400 rps</td><td>E-
$\varepsilon_0^{\mathrm{p2all}}$= 0,58
n_1= 100 rps</td><td>P-
$\varepsilon_0^{\mathrm{p2all}}$= 0,48
n_1= 400 rps</td><td>$T^*_{2\max} < T^*_{1\mathrm{all}}$
in the examined case
n_1= 100 rps</td><td>$T^*_{2\max} < T^*_{2\mathrm{all}}$
in the examined case
n_1= 100 rps</td></tr>
<tr><td colspan="5">C_R^* =2,0</td></tr>
<tr><td>$h^*_{1\min} > h^*_{lall}$
in the examined case
n_1= 100- 400 rps</td><td>C-
$\varepsilon_0^{\mathrm{plall}}$= 0,66
n_1= 100 rps</td><td>N-
$\varepsilon_0^{\mathrm{plall}}$= 0,61
n_1= 400 rps</td><td>$T^*_{1\max} < T^*_{1\mathrm{all}}$
in the examined case
n_1= 100 rps</td><td>W-
$\varepsilon_0^{\mathrm{Tlall}}$= 0,78
n_1= 400 rps</td></tr>
<tr><td>K-
$\varepsilon_0^{\mathrm{h2all}}$= 0,57
n_1= 100- 400 rps</td><td>F-
$\varepsilon_0^{\mathrm{p2all}}$= 0,72
n_1= 100 rps</td><td>R-
$\varepsilon_0^{\mathrm{p2all}}$= 0,56
n_1= 400 rps</td><td>E-
$\varepsilon_0^{\mathrm{T2all}}$= 0,73
n_1= 100 rps</td><td>$T^*_{2\max} < T^*_{2\mathrm{all}}$
in the examined case
n_1= 400 rps</td></tr>
</table>

– for rotational speed of journal 100 rps, the bearing can work with a quotient of radial clearance C_R^* = 0,5; 1,1; 2,0 and resultant relative eccentricity $\varepsilon_0 < \varepsilon_0^{\mathrm{h2all}} = 0,45$,
– for rotational speed the journal 400 rps, the bearing can work with a quotient of radial clearance C_R^* = 0,5; 1,1; 2,0 and the resultant relative eccentricity $\varepsilon_0 < \varepsilon_0^{\mathrm{p1all}} = 0,37$.

Thus, a bearing with given operating parameters such as described in chart 1 fulfils the conditions of proper work. It should be noticed that the bearing with the quotient of radial clearance C_R^* = 0,5 can transmit much higher loads than the bearings whose quotient of radial clearance take values of C_R^* = 1,1; 2,0. Bearings which are less loaded, and work at higher rotational speeds,

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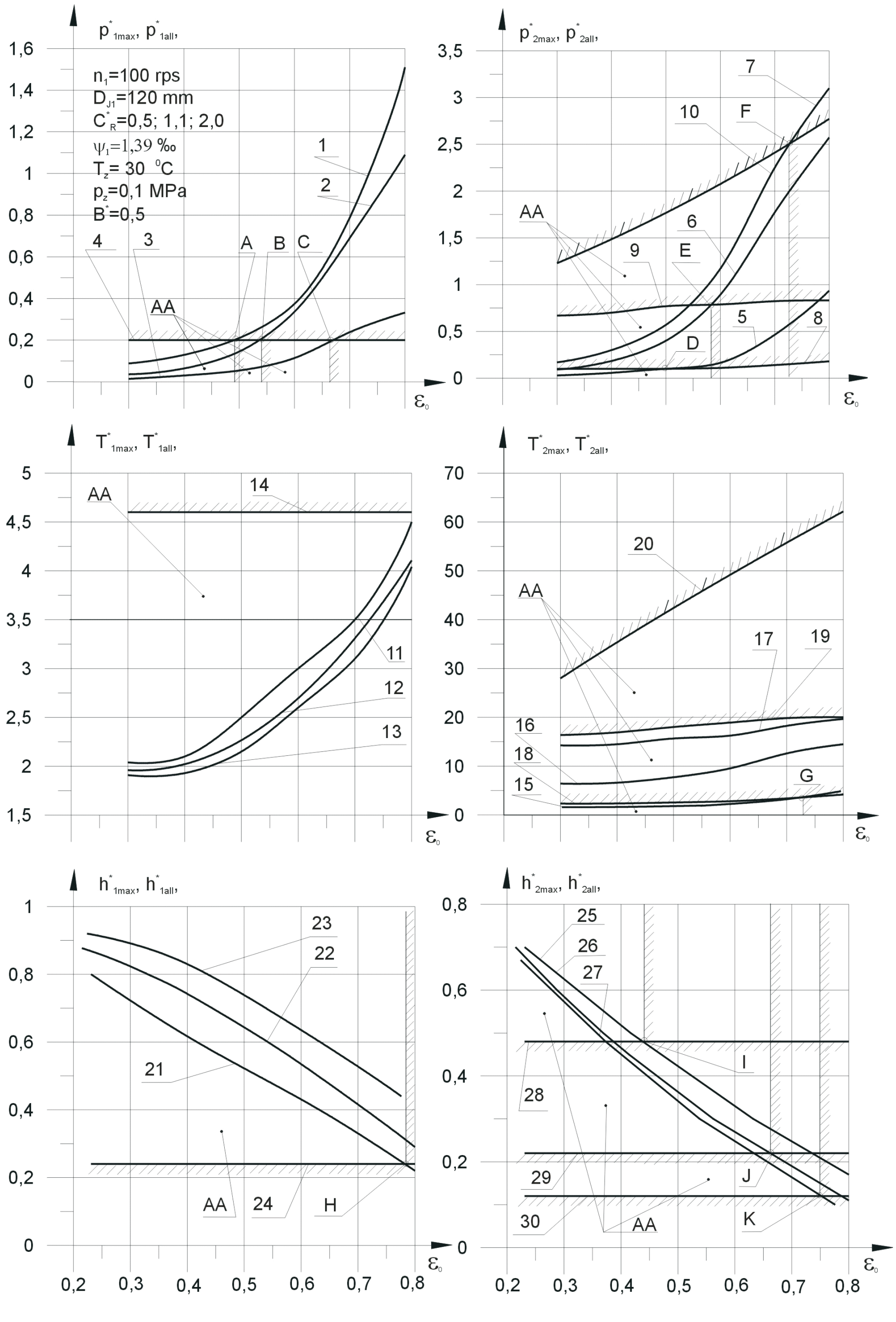

Fig. 5. The area of acceptable solutions, n_1 = 100 rps

fulfill the conditions of proper work for higher values of radial quotients: C_R^* = 1,1; 2,0.

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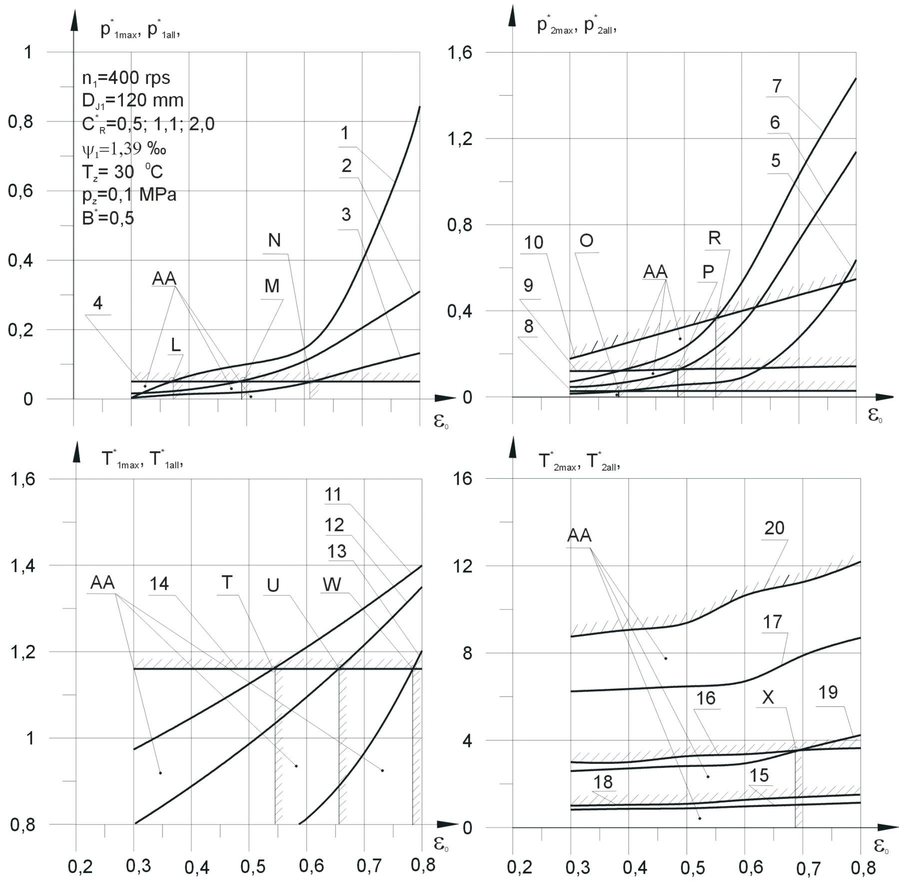

Fig. 6. The area of acceptable solutions, n_1 = 400 rps

Where (Fig.5 and 6): AA- area acceptable solutions, 1- $p^*_{1max}= p^*_{1max}(C^*_R =0,5, \varepsilon_0)$, 2- $p^*_{1max}= p^*_{1max}(C^*_R =1,1, \varepsilon_0)$, 3- $p^*_{1max}= p^*_{1max}(C^*_R =2,0, \varepsilon_0)$, 4- $p^*_{1all}= p^*_{1all}(C^*_R =0,5;1;1;2;0; \varepsilon_0)$, 5- $p^*_{2max}= p^*_{2max}(C^*_R =0,5, \varepsilon_0)$, 6- $p^*_{2max}= p^*_{2max}(C^*_R =1,1, \varepsilon_0)$, 7- $p^*_{2max}= p^*_{2max}(C^*_R =2,0, \varepsilon_0)$, 8- $p^*_{2all}= p^*_{2all}(C^*_R =0,5, \varepsilon_0)$, 9- $p^*_{2all}= p^*_{2all}(C^*_R =1,1, \varepsilon_0)$, 10- $p^*_{2all}= p^*_{2all}(C^*_R =2,0, \varepsilon_0)$, 11- $T^*_{1max}= T^*_{1max}(C^*_R =0,5, \varepsilon_0)$, 12- $T^*_{1max}= T^*_{1max}(C^*_R =1,1, \varepsilon_0)$, 13- $T^*_{1max}= T^*_{1max}(C^*_R =2,0, \varepsilon_0)$, 14- $T^*_{1all}= T^*_{1all}(C^*_R =0,5;1;1;2;0; \varepsilon_0)$, 15- $T^*_{2max}= T^*_{2max}(C^*_R =0,5, \varepsilon_0)$, 16- $T^*_{2max}= T^*_{2max}(C^*_R =1,1, \varepsilon_0)$, 17- $T^*_{2max}= T^*_{2max}(C^*_R =2,0, \varepsilon_0)$, 18- $T^*_{2all}= T^*_{2all}(C^*_R =0,5, \varepsilon_0)$, 19- $T^*_{2all}= T^*_{2all}(C^*_R =1,1, \varepsilon_0)$, 20- $T^*_{2all}= T^*_{2all}(C^*_R =2,0, \varepsilon_0)$,21- $h^*_{1max}= h^*_{1max}(C^*_R =0,5, \varepsilon_0)$, 22- $h^*_{1max}= h^*_{1max}(C^*_R =1,1, \varepsilon_0)$, 23- $h^*_{1max}= h^*_{1max}(C^*_R =2,0, \varepsilon_0)$, 24- $h^*_{1all}= h^*_{1all}(C^*_R =0,5;1;1;2;0; \varepsilon_0)$, 25- $h^*_{2max}= h^*_{2max}(C^*_R =0,5, \varepsilon_0)$, 26- $h^*_{2max}= h^*_{2max}(C^*_R =1,1, \varepsilon_0)$, 27- $h^*_{2max}= h^*_{2max}(C^*_R =2,0, \varepsilon_0)$, 28- $h^*_{2all}= h^*_{2all}(C^*_R =0,5, \varepsilon_0)$, 29- $h^*_{2all}= h^*_{2all}(C^*_R =1,1, \varepsilon_0)$, 30- $h^*_{2all}= h^*_{2all}(C^*_R =2,0, \varepsilon_0)$,

4. Conclusion

Taking into account the results of this study, we can state that bearings which work with higher rotational speeds and low loads should be designed with higher quotients of radial clearance. However, the bearings which are

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less loaded and work with higher rotational speeds, fulfill the conditions of proper work for higher values of the radial quotient. In comparison with other methods, the calculation method described in this paper offers the possibility of determining the course of variability of the examined parameters. In effect, it gives the possibility of a better selection of parameters for the bearing. The application of dimensionless characteristics of the bearing gives the opportunity to generalize the design process for a certain category of bearings.

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Obliczanie sprawdzające łożysk ślizgowych z panewkami pływającymi

S t r e s z c z e n i e

W pracy omówiono zagadnienia związane z obliczeniami sprawdzającymi łożysk ślizgowych z panewką pływającą. Do obliczeń przyjęto adiabatyczny model łożyska. Szczegółowe etapy konstruowania przedstawiono w postaci schematu strukturalnego. Sformułowano i omówiono warunki prawidłowej pracy łożyska. Dla przedstawionego przykładu obliczeniowego dokonano wyznaczenia obszaru rozwiązań dopuszczalnych. W podsumowaniu przedstawiono zalecenia projektowe dla projektantów tego typu łożysk.