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EFFECT OF GEOMETRICAL PARAMETERS ON LOAD DISTRIBUTION IN DOUBLE ROW BALL SLEWING RING

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1. Introduction

The calculation method of double row ball slewing rings was presented in the paper (Gibczyńska et al. (to be published)). In this paper the mathematical model was formulated and on its base the algorithm and the computer programme were elaborated. The programme permits the determination of forces acting on each rolling element of the bearing at a given spatial state of load constituted by an axial force P_x, two radial forces P_y and P_z and two moments M_y and M_z acting in planes perpendicular to each other (fig. 1).

The mentioned paper is an extention of the paper (Gibczyńska and Życzkowski (1975)) in which an approximate analitycal solution for a bearing loaded with a flat system of loads were presented. In the model a clearance, a change of the contact angle and a deflection of supporting structures with rings were considered. The conditions of the equilibrum of the bearing inner ring were formulated. Forces acting on rollig elements were determined from Hertz's formula. Deformation of the raceway and of the i-th rolling element were presented as a function of geometrical parameters of the bearing, five generalized displacements of the inner ring X, Y, Z, η_y, η_z, clearance g, and moreover three components x, y and z of deflection of the carrying system with rings were ascribed to each rolling element. Five equations of equilibrum were solved numerically.

The elaborated computer programme permits the determination of the sought generalized displacements, the forces acting on each ball and the respective contact stress for the given outer load on the bearing. The effect of a clearance and deflection of the supporting structures on the load distribution in a double row ball slewing ring was presented in the paper (Gibczyńska and Marciniec (1989)). Here we should confine ourselves to examinations of the change of the contact angle in consequence of deformations, clearance and the effect of this change on the maximum stress as well as the influence of the coefficient of contact determined by the quotient

$$k_{1,} = \frac{r_{bi} - r_k}{r_k} \quad ,$$

(where: r_k - radius of the ball, r_{bz} - radius of the outer raceway, r_{bw} - radius of the inner raceway) on maximum values of contact stress.

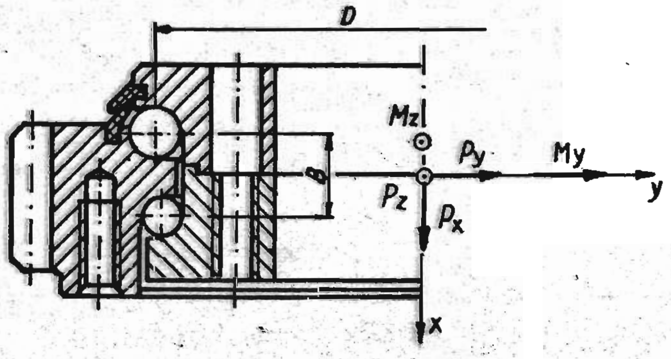

Fig.1. Model of the bearing.

Considerations were carried out for a double row ball slewing ring (fig.1) of the followig parameters: rolling diameter D=2572mm, distance between rows B=71.8mm. An identical number of balls z=157 and identical dimentions of balls and raceways in both rows was assumed: diameter of the ball $2r_k$=41.275mm ($1^5/_8$"), raceway curvature radius r_b=21.5mm what corresponds to the coefficient k=0.04. The axial force P_x=2400kN and the momentum M_z=61700kNm was the adopted load.

2. Effect of contact angle and coefficient of contact on maximum stress.

The diagram of changes of contact stress occuring between the raceway and the ball was presented in fig.2 in function of the nominal contact angle for the examined bearing. Investigations were carried out in the range of contact angles from 45° to 90°, used in such kind of bearings.

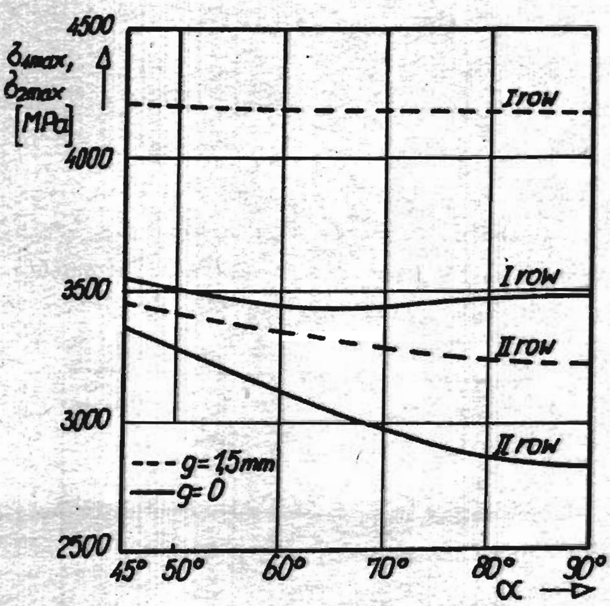

Fig.2. Maximal stress in function of the nominal contact angle for clearance g=0 and g=1.5mm .

A bearing clearance free and with clearance equal 1.5mm was assumed. A clearance free bearing is treated as a model bearing used to compare values of the calculated quantities. The taken clearance value g=1.5mm is the mean value of clearance limits given in (Rothe Erde's catalogue (1989)) for the assumed bearig diameter. It follows from the curve shapes that in the case of the clearance free bearing contact stress between the ball and the raceway of the first (upper) row attain the lowest value for the angle α=67.5°, whereas, in a bearing with clearance the lowest value of contact stress occurs at the contact angle α=90°. In this case the difference between the highest and lowest value is very small equalling

0.8% .Contact stress in the second (lower) row demonstrate the lowest values in both cases at the contact angle $\alpha=90^\circ$. Considering the effect of contact angle on force distribution in the bearing and on maximum stress values the change of the angle under the action of load i.e. due to local deformations and clearance cannot be neglected.

The contact angle of each loaded ball is different, and the contact angle of the ball loaded most undergoes greatest changes. The difference between the real contact angle α_r and the nominal one α is determined by

$$\Delta\alpha = \alpha_r - \alpha .$$

In diagrams presented in fig.3 the traces of the curve of maximum difference $\Delta\alpha_{max}$ were given in function of the nominal contact angle α. The change of the contact angle increases markedly in the angle range $\alpha<<90^\circ$ and increases evidently with clearance. Only for the contact angle 90° it can be considered neglegibly small. Contact angles smaller than 90° to-

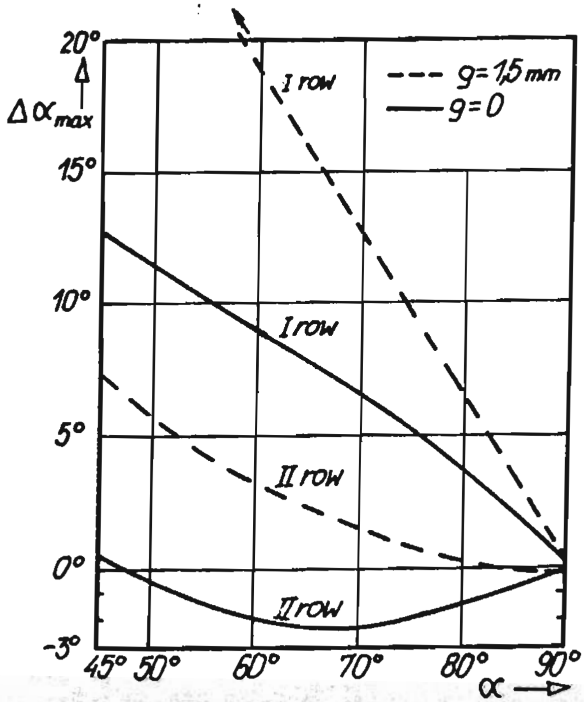

Fig.3. Maximum changes of the contact angle in function of the nominal contact angle for clearance g=0 and g=1.5mm.

gether with their changes exert a positive effect on the distribution of load on the circumference of the raceway. If in a clearance free brearing at the contact angle $\alpha=90^{\circ}$ only 56% of balls in the first row carries the load, so at the contact angle 70°, 60°, 45° the load is carried by 65%, 73%, 88% respectively. Hence, this distributions are more regular.

The range of changes of the contact angle $\alpha=70^{\circ}$ was presented in function of clearance in fig.4. The continuous lines refer to the contact angle corresponding to the most loaded ball, the broken lines - to the ball lying on the border of the zone of loading (least loaded).

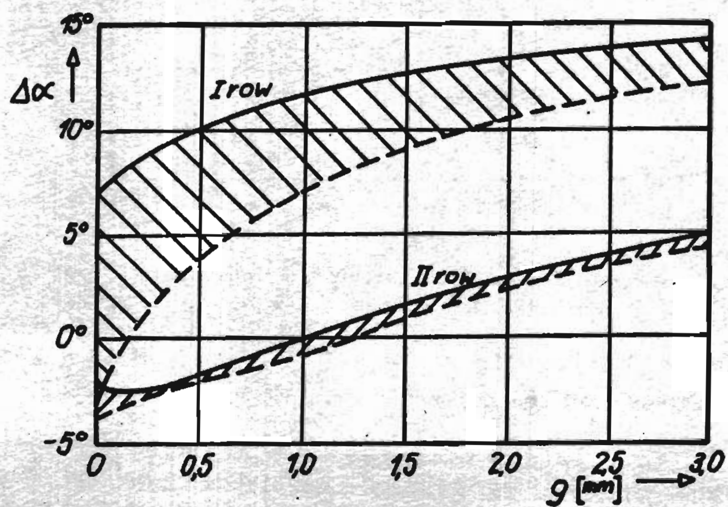

Fig.4. Change of the contact angle value in function of clearance.

The contact angle increases with increase in clearance at concomitant dearease in the range of its changebility. This follows from the fact that with increase in clearance the zone of loaded balls decreases; it means the balls most and least loaded lies closer to each other. Increase in the contact angle causes increase in component forces acting on each ball, parallel to the axis of the bearing and balancing the external load.

The effect of the coefficient of contact on the maximum value of stress was presented in fig.5. In calculations an identical coefficient k was adopted for the inner and outer raceway. The increase in stress with

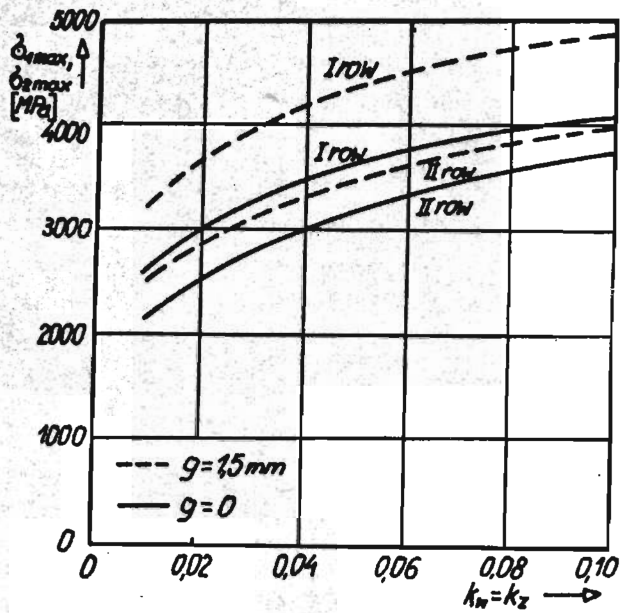

Fig.5. Maximum stress in function of the coefficient of contact for clearance g=0 and g=1.5mm.

increase in that coefficient results not only from Hertz's formula but also from the change of the contact angle. At small values of k the contact angle undergoes great changes in the first row, whereas, at greater ones differences $\Delta\alpha_{max}$ are considerably smaller. And so for a clearance free bearing at k=0.01 - $\Delta\alpha_{max}=10^{\circ}$ and at clearance equal 1.5mm - $\Delta\alpha_{max}=16^{\circ}$. Whereas, if k=0.10 then $\Delta\alpha_{max}=4.5^{\circ}$ for g=0 and $\Delta\alpha_{max}=11^{\circ}$ for g=1.5mm. It should be noticed that with increase in the coefficient k, not only maximum values of contact stress increase but the zone of loading increases as well. On the other hand however, in bearings of higher coefficient k the effect of clearance on increase in stress is smaller than at its low value.

3. Conclusions

With regard to a limited volume of the report only some chosen factors influencing maximum stress values and load distribution in the bearings were here presented. Basing upon carried out investigations great changes in the value of the contact angle may be found. This statemant is very essential with regard to the fact that in numerous papers concerning calculations of this kind of bearings invariability of this angle (Matthias (1962), Mazanek (1982)) was adopted. This assumptions leads, however, to serious errors in calculation of bearing capacity.

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Summary

WPŁYW PARAMETRÓW GEOMETRYCZNYCH NA OBCIĄŻENIE W DWURZĘDOWYM KULKOWYM ŁOŻYSKU WIEŃCOWYM

W pracy przedstawiono przykłady symulacji obciążenia w dwurzędowym kulkowym łożysku wieńcowym. Przeprowadzono analizę wpływu kąta działania, luzu i promienia krzywizny bieżni na wartości maksymalnych naprężeń stykowych.