International Journal of Mechanical Sciences 100 (2015) 209–215

Contents lists available at ScienceDirect

International Journal of Mechanical Sciences

journal homepage: www.elsevier.com/locate/ijmecsci

Determination of the precise static load-carrying capacity of pitch bearings based on static models considering clearance

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ARTICLE INFO

Article history:
Received 6 December 2013
Received in revised form
12 June 2015
Accepted 28 June 2015
Available online 11 July 2015

Keywords:
Pitch bearing
Negative clearance
Static model
Static load-carrying capacity

ABSTRACT

Static models for a single row and a double row four-point contact pitch bearing, taking into account the clearances therein, were presented. A precise computation method was proposed for the static load-carrying capacity curves of such pitch bearings, which can be used to pre-select pitch bearings and slewing bearings supposing rigid bearing rings. The effects on the static load-carrying capacity induced by changing the clearance, the raceway groove radius of curvature, and initial contact angle were analysed. The clearance has a significant effect on the static load-carrying capacity of the bearing only in load cases with a rather small axial force and a large tilting moment. When the coefficient of raceway groove curvature radius increases, the static load-carrying capacity decreases. The smaller the radial load, the more significant the effect of the coefficient of raceway groove curvature radius on the static load-carrying capacity of the bearing. When radial loads range from 0 to 800 kN, the load-carrying capacity increases with increasing initial contact angle. When the radial loads are above 800 kN, the load-carrying capacity decreases with increasing initial contact angle.

1. Introduction

The pitch bearing of a wind-power generator is basically a single row four-point contact ball bearing or a double row four-point contact ball bearing. The running speed of a pitch bearing is usually very small, therefore, the static model and static load-carrying capacity, which can be described by the static load-carrying curves, are mainly considered in pitch bearing design. The impact load acting on the pitch bearing is very large, so a zero, or negative, clearance is considered to reduce fretting wear. Negative clearance not only affects the start-up torque, rotation precision, and stiffness of the bearing, but also affects the load distribution, load capability, and the life of the bearing. It is worth establishing a static model considering the effect of clearance and using it to study the static load-carrying capacity of pitch bearings.

Göncz [1] analysed the contact force distribution for a three-row roller slewing bearing considering no clearances and no ring deformations. Aguirrebeitia [2–4] presented the derivation of the general static load-capacity of three-row roller slewing bearings and four-contact point slewing bearings, based on Rumbarger's method. Their calculations assume zero clearance in the contact and rigid rings. A procedure for obtaining the load distribution in a four-contact point slewing bearing considering the effect of the structural elasticity was provided elsewhere [5]. Kunc [6,7] studied the actual carrying capacity of the rolling contact in single-row ball bearings using a computation model considering the material properties, which includes isotropic and kinematic hardening, and cumulative damage. Amasorrain [8] provided a static model taking into account the clearance for a single-row slewing ball bearing and analysed the load distribution; however, the effects induced by the supporting surface of the bearing were not considered. Zupan [9] discussed the carrying angle and carrying capacity of a large single-row ball bearing taking into account the osculation, actual carrying angle, and clearance. Kania [10] presented an FE method for computing the catalogue capacity of a slewing bearing taking into account the flexibility of its rings and clamping bolts. Kania also discussed the problem of calculating the real carrying capacity of a single-row roller slewing bearing by FEM, but did not give the analytical expressions for the proposed static model [11]. Göncz [12] provided a new computational model which considered ring deformations and clearances for determination of the internal contact force distribution and the determination of static load curves for a three-row roller slewing bearing. Glodež [13] presented a similar model for a large slewing ball bearing. However, the vector approach was used in the two reports [12,13] to give a mathematical description of the bearing geometry and the static force and moment equilibrium calculation. Potočnik [14] developed a design method for assessing the static capacity of a large double-row slewing ball bearing, but only gave the vector expression of the static model for a single-row slewing ball

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http://dx.doi.org/10.1016/j.ijmecsci.2015.06.023
0020-7403/

bearing because it is thought that the contact deformation and contact force of the upper row are same as those of the lower row for the double-row slewing ball bearing, which does not suit actual loading conditions.

In the aforementioned papers discussing a double-row slewing ball bearing, the static model in the case of a single-row ball bearing has been described and the differences in contact formation and contact force of each row have not been pointed out [14]. In this paper, the proposed static model for a double-row four-point contact ball bearing was felt to be more reasonable than existing models as it could differ from the contact deformation and contact force between the upper row and the lower row. The determination of the contact deformation in existing static models is based on the changes in the initial and final coordinates of the centres of curvature of the raceways, which are complicated, especially for double-row ball bearings. In this research, new static models for single- and double-row ball bearings are presented, in which the determination of the contact deformation was based on the changes in the initial and final distances between the diagonally opposed centres of curvature of the inner and outer raceways. The new static models are as precise as the existing models taking the clearances into account, but they are easier to describe and realise than existing static models.

In the aforementioned literatures and elsewhere [15], the static load-carrying capacity curves were presented in the form of a moment-axial-force diagram and the radial force was not taken into consideration; the method of establishing a static load-carrying curve considering the radial force was not provided. In this research, the method used to establish the precise static load-carrying curve, considering the effect of radial load, was presented on the basis of the static force model considering the effect of clearance. The effects of structure parameters on the load-carrying capacity of pitch bearings were also analysed.

Here, the new static models are based on the supposition that the bearing rings were ideally stiff and elastic contact deformations only existed on the contact area between balls and their raceways. The study of the static models and static load-carrying capacity curves has significance with regards the pre-selection of slewing bearings.

2. Static model considering clearance

Published research results [15] show that clearance greatly affects the magnitude of the contact force and its distribution. Accurate calculation of the contact force is essential when establishing the precise static load-carrying curve. So the clearance must to be taken into account when the static model used to compute the contact force is established. In this article, the static models considering clearance for a single-row four-point contact bearing, or a double-row four-point contact bearing, are provided, and a procedure for establishing the static model is not given as it may be found elsewhere [15].

The zero or negative clearance is always used in pitch bearings. Negative clearance not only affects the load distribution and load capability, but also affects the friction torque of the bearing.

Taking a double-row four-point ball bearing with the geometry shown in Table 1 as an example, the start-up friction torques measured by the spring balance scale were 2.8 kN·m and 3.5 kN·m when the clearances were 0 and −0.01 mm respectively, with no external loads. So it was necessary to take the clearance into consideration in the static model of such a pitch bearing.

The flexibility of rings and the flexibility of their fastening also affect the distribution of the contact force on the balls in a four-point contact ball bearing. Kania [10] has already discussed this problem. In this research, the deformed states of the rings and support structures of the bearing are not taken into account. The procedure described here supposes that the rings are ideally stiff and only assumes elastic Hertzian contact deformations on the contact area between the balls and their raceways. The outer ring was supposed to be fixed and the external loads, which are an axial load F_a, a radial load F_r, and a tilting moment load M, were applied on the inner ring.

Before loading, the distance between the diagonally opposed centres of curvature of the inner and outer raceways, with clearance, can be written as:

$$A=(f_i+f_e-1)D_W-\frac{1}{2}u_a\cos\alpha_0 \tag{1}$$

The initial distance between diagonally opposed centres of curvature with zero clearance is:

$$A_0=(f_i+f_e-1)D_w \tag{2}$$

where $f_i=r_i/D_w$ is the coefficient of the inner raceway groove curvature radius, $f_e=r_e/D_w$ is the coefficient of the outer raceway groove curvature radius, D_w is the ball nominal diameter, r_i and r_e are the inner and outer raceway groove curvature radii, respectively, u_a is the bearing axial clearance, and α_0 is the initial contact angle.

After loading, the distances between diagonally opposed centres of curvature along the direction of contact pairs k ($k=1, 2$) for a single row four-point contact bearing were defined as $S_{k\varphi_j}$, and can be computed thus:

$$\begin{cases} S_{1\varphi_j}=[(A\sin\alpha_0+\delta_a+R_i\theta\cos\varphi_j)^2+(A\cos\alpha_0+\delta_r\cos\varphi_j)^2]^{\frac{1}{2}} \\ S_{2\varphi_j}=[(A\sin\alpha_0-\delta_a-R_i\theta\cos\varphi_j)^2+(A\cos\alpha_0+\delta_r\cos\varphi_j)^2]^{\frac{1}{2}} \end{cases} \tag{3}$$

where the angular position of each ball inside the bearing is $\varphi_j=2\pi(j-1)/Z$ ($j=1, 2, 3...Z$), Z is the number of balls in a single-row of the bearing, δ_a, δ_r and θ are the axial, radial, and angular displacements, respectively generated in the inner ring relative to the outer ring, and R_i is the radius of the track of the raceway groove curvature centre of the inner ring. R_i is given by:

$$R_i=\frac{1}{2}d_m+(f_i-0.5)D_w\cos\alpha_0-\frac{1}{4}u_a(\cos\alpha_0)^2 \tag{4}$$

where d_m is the pitch diameter of the bearing.

The distances between diagonally opposed centres of curvature along the direction of contact pairs k ($k=1, 2, 3, 4$) for a double-row four-point contact bearing are defined as $A_{k\varphi_j}$, they are:

$$A_{1\varphi_j}=\left[\left(A\sin\alpha_0+\delta_a+R_i\theta\cos\varphi_j\right)^2+\left(A\cos\alpha_0+\delta_r\cos\varphi_j+0.5d_c\theta\cos\varphi_j\right)^2\right]^{\frac{1}{2}} \tag{5}$$

$$A_{2\varphi_j}=\left[\left(A\sin\alpha_0-\delta_a-R_i\theta\cos\varphi_j\right)^2+\left(A\cos\alpha_0+\delta_r\cos\varphi_j+0.5d_c\theta\cos\varphi_j\right)^2\right]^{\frac{1}{2}} \tag{6}$$

$$A_{3\varphi_j}=\left[\left(A\sin\alpha_0+\delta_a+R_i\theta\cos\varphi_j\right)^2+\left(A\cos\alpha_0+\delta_r\cos\varphi_j-0.5d_c\theta\cos\varphi_j\right)^2\right]^{\frac{1}{2}} \tag{7}$$

$$A_{4\varphi_j}=\left[\left(A\sin\alpha_0-\delta_a-R_i\theta\cos\varphi_j\right)^2+\left(A\cos\alpha_0+\delta_r\cos\varphi_j-0.5d_c\theta\cos\varphi_j\right)^2\right]^{\frac{1}{2}} \tag{8}$$

where d_c is the distance between the centres of the two balls in the upper and lower rows.

Under load, the contact angles at angular position φ_j along the directions of contact pairs k ($k=1, 2$) for a single-row four-point