Springer

KSME

Journal of Mechanical Science and Technology 24 (10) (2010) 2083~2089
www.springerlink.com/content/1738-494x
DOI 10.1007/s12206-010-0709-5

Effect of raceway geometry parameters on the carrying capability and the service life of a four-point-contact slewing bearing†

Xuehai Gao, Xiaodiao Huang*, Hua Wang, Rongjing Hong and Jie Chen

Jiangsu Key Laboratory of Digital Manufacturing for Industrial Equipment and Control Technology, Nanjing University of Technology, Nanjing, Jiangsu province, China, 210009

(Manuscript Received October 27, 2009; Revised June 8, 2010; Accepted June 25, 2010)

Abstract

The carrying capability and the service life of a four-point-contact slewing bearing are influenced significantly by its raceway geometry parameters. In this paper, the carrying capability is characterized by the maximal contact stress in the paper. The usual industry practice is to specify the service life as the L_{10} life. The geometry parameters, cross section clearance, ratio of curvatures, rolling ball diameter, and initial contact angle, are also considered. The sensitivity of each parameter influencing the carrying capability and the service life is discussed as well. The results show an interesting phenomenon, indicating that the maximal contact load and the maximal stress do not always have the same varying trend when discussed parameters vary. The developed design can be utilized as a reference for an optimal slewing bearing design. If a slewing bearing of high static carrying capability is required, it can be designed with large ratio of curvatures and rolling ball diameter. In terms of service life, the cross section clearance and the initial contact load should also be considered.

Keywords: Slewing bearing; Raceway geometry parameter; Carrying capability; Service life

1. Introduction

Slewing bearing is a major element in a rotational connection. It is basically a large-sized bearing subjected to a compound set of loads, including axial force F_a, radial force F_r, and turnover moment M. A slewing bearing can be found frequently in various machineries, such as cranes, civil engineering machines, wind power turbines, radars, and tanks. Different types of slewing bearings, such as single or double row ball slewing bearings, double or three row roller slewing bearings and cross roller slewing bearings, are applied in different fields. Among the most commonly used are the single row four-point-contact slewing bearings. In this paper, a single row four-point-contact slewing bearing, hereafter referred to as a slewing bearing, is discussed.

The carrying capability is characterized by the maximal contact stress with large maximal contact stress, indicating low carrying capability. The maximal contact stress depends on both the maximal contact load and area of contact surface. Thus, the maximal contact load should be determined first. Several models have been developed previously to calculate the maximal contact load. The previous models can be presented as:

$$Q_{\max} = \frac{HF_a}{Z\sin\alpha_o} + \frac{IM}{DZ\sin\alpha_o} + \frac{JF_r}{Z\cos\alpha_o}, \tag{1}$$

where α_o is the initial contact angle, D is the average diameter, Z is the number of rolling balls and H, I, J are coefficients modified by the users [1-3]. The previous models are based on an over simplified contact load distribution theory, which supposes that the actual contact angles are constant and always equal to the initial contact angles. Raceway geometry parameters are not all involved in the model.

To determine the maximal contact load, load distribution over the raceways of a slewing bearing has to be calculated. Iterative models to calculate the load distribution have been established by previous researchers by taking actual contact angles into consideration [3-5]. Zupan, Daidié, and Smolnicki et al. [5-7] studied the effect of the support structure stiffness on the load distribution over raceways of a slewing bearing with FEM (Finite Element Method).

Their works contributed mainly in the establishment of an exact load distribution model. However, their studies did not indicate any methods in optimizing the proposed slewing bearing designs. Therefore, the main objective of this paper is to create a method to optimize the slewing bearing design by discussing how each parameter influences its carrying capabil-

† This paper was recommended for publication in revised form by Associate Editor Jeonghoon Yoo
*Corresponding author. Tel.: +86 25 83172582, Fax: +86 25 83366783
E-mail address: 84gaohai@163.com

ity and service life. The results of this paper could aid designers in optimizing a slewing bearing design according to its requirements.

The service life of a slewing bearing is specified by L_{10} life, which is life with a 10% probability of failure, and is presented as:

$$L_{10} = \left(\frac{Ca}{Pa}\right)^3, \tag{2}$$

where Ca is the basic dynamic capability, and Pa is the dynamic equivalent thrust load. Eq. (2) is derived from the theory of commonly used bearing, because a slewing bearing usually appears as a thrust type bearing that primarily endures thrust and turnover loads. Although the actual service life of a slewing bearing is usually two or more times longer than the L_{10} life for the crack propagation [8-10], L_{10} life still has great significance in the evaluation of the service life of a slewing bearing.

The contact load and stress distribution model in this paper has been developed based on previous studies. The effect of raceway parameters on the carrying capability and service life for the optimization of a slewing bearing design are discussed, and results shown illustrations.

2. Theories on contact load distribution and service life

2.1 The load and stress distribution of a slewing bearing

All analyses are based on the contact load and stress distribution model. The contact load and stress distribution influences not only the carrying capability and the service life of a slewing bearing but also its reliability. This will be discussed in a future publication.

Relations between geometry parameters under initial states (without loads) are shown in Fig. 1. A slewing bearing with positive clearance is commonly applied [Fig. 1(a)]. In some fields, zero or negative clearance is required [Fig. 1(b)]. C_{id}, C_{iu}, C_{ed}, and C_{eu} are raceway curvature centers, and contact only occurs along diagonally opposed curvature centers. The initial distance between diagonally opposed curvature centers is:

$$A_0 = \left|C_{id}C_{eu}\right| = \left|C_{iu}C_{ed}\right| = 2R - d - 2P_r . \tag{3}$$

If contact begins, the distance between diagonally opposed curvature centers is:

$$A = 2R - d . \tag{4}$$

The ratio of curvatures which commonly varies from 1.02 to 1.08 is defined as:

$$t = 2R/d . \tag{5}$$

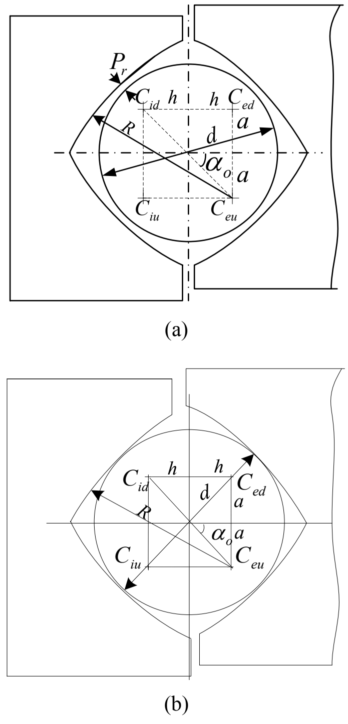


Fig. 1. (a) Cross section of a slewing bearing with positive clearance; (b) Cross section of a slewing bearing with zero or negative clearance.

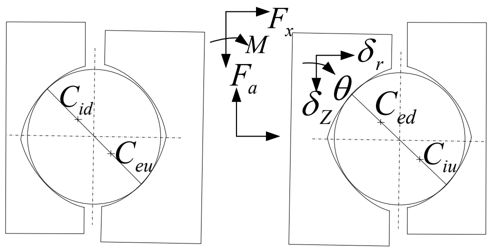


Fig. 2. Relative displacement between raceways.

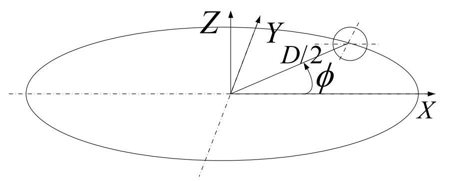


Fig. 3. Position of each ball inside a slewing bearing.

There are three components for relative displacements (Fig. 2) when a set of loads are applied to a slewing bearing: the radial relative displacement δ_r, the axial relative displacement δ_Z, and the relative rotation θ. Actual contact angles vary at each contact point, thus, the elastic deformation at each contact point should be taken into account. In this work, the coordinates system (Fig. 3) was established to define the position of each ball.

Raceway curvature centers at each contact point under a compound set of loads were calculated using the following equation:

$$\begin{bmatrix} XC_{idF} & YC_{idF} & ZC_{idF} \\ XC_{euF} & YC_{euF} & ZC_{euF} \\ XC_{iuF} & YC_{iuF} & ZC_{iuF} \\ XC_{edF} & YC_{edF} & ZC_{edF} \end{bmatrix} = f\left(D,\phi,d,\alpha_0,t,P_r,\delta_r,\delta_Z,\theta\right), \quad (6)$$

where the subscript F represents the final positions of the raceway curvature centers. It is not necessary to calculate the coordinates of each curvature center directly; instead relative displacements (δ_r, δ_Z, θ) can be used to illustrate their coordinates. The equation used in this work was established from geometrical relations. An example is provided as Eq. (7) presented as:

$$XC_{idF} = \left(\frac{D}{2} + h\right)\cos\phi + \delta_r \ . \quad (7)$$

The final distances between diagonally opposed curvatures centers are represented by:

$$A_{1F} = \left|C_{id}C_{eu}\right| = \sqrt{\left(XC_{idF} - XC_{euF}\right)^2 + \left(YC_{idF} - YC_{euF}\right)^2 + \left(ZC_{idF} - ZC_{euF}\right)^2} , \quad (8)$$

and

$$A_{2F} = \left|C_{iu}C_{ed}\right| = \sqrt{\left(XC_{iuF} - XC_{eiF}\right)^2 + \left(YC_{iuF} - YC_{eiF}\right)^2 + \left(ZC_{iuF} - ZC_{eiF}\right)^2} . \quad (9)$$

If $\Delta_1 = A_{1F} - A > 0$, contact works on the diagonal $C_{id}C_{eu}$. Thus, the contact load is:

$$Q_1 = K\left(\frac{\Delta_1}{2}\right)^{\frac{3}{2}}, \quad (10)$$

and the actual contact angle is:

$$\alpha_{1F} = \arcsin\frac{ZC_{idF} - ZC_{euF}}{A_{1F}} . \quad (11)$$

If $\Delta_2 = A_{2F} - A > 0$, contact works on the diagonal $C_{iu}C_{ed}$. Thus, the contact load is:

$$Q_2 = K\left(\frac{\Delta_2}{2}\right)^{\frac{3}{2}}, \quad (12)$$

and the actual contact angle is:

$$\alpha_{2F} = \arcsin\frac{ZC_{iuF} - ZC_{edF}}{A_{2F}} . \quad (13)$$

Finally, a set of non-linear equations, linking the relative displacements and the external loads, can be developed as the following expressions:

$$\sum_1^Z \left(Q_1 \sin\alpha_{1F} + Q_2 \sin\alpha_{2F}\right) + Fa = 0 , \quad (14)$$

$$\sum_1^Z \left(Q_1 \frac{XC_{idF} - XC_{euF}}{A_{1F}} + Q_2 \frac{XC_{iuF} - XC_{edF}}{A_{2F}}\right) + Fr = 0 , \quad (15)$$

and

$$\begin{aligned} &\sum_1^Z \left(Q_1 \sin\alpha_{1F}\left(\frac{D}{2} - \frac{d}{2}\cos\alpha_{1F}\right)\cos\phi + \right. \\ &\left. Q_2 \sin\alpha_{2F}\left(\frac{D}{2} - \frac{d}{2}\cos\alpha_{2F}\right)\cos\phi\right) + M = 0 \end{aligned} . \quad (16)$$

The contact load at each contact point can be calculated based on the above equations. Contact stress can be obtained using Eq. (17) presented as:

$$S_{\max} = \frac{1.5Q}{\pi ab} , \quad (17)$$

where Q is the contact load, and $\pi ab = f(Q,D,d,t)$ is a nonlinear equation presenting the area of contact surface.

2.2 The L_{10} life of a slewing bearing

Eq. (2) shows that the effect of raceway geometry parameters on the basic dynamic capability (*Ca*) and the dynamic equivalent thrust load (*Pa*) should be determined first. *Ca* and *Pa* models are always established on the basis of ISO and AFBMA load rating and fatigue life standards [11,12], However, some modification should be made for the large size and complex loads of a slewing bearing. NREL (National Renewable Energy Laboratory of U.S.) has modified the models as:

$$Ca = fcm(\cos\alpha_o)^{0.7} Z^{2/3} d^{1.8} \tan\alpha_o \quad (d \le 25.4mm), \quad (18)$$

$$Ca = 3.647 fcm(\cos\alpha_o)^{0.7} Z^{2/3} d^{1.4} \tan\alpha_o \quad (d > 25.4mm), \quad (19)$$

and

$$Pa = \left(\frac{1}{Z}\sum_{n=1}^{Z} Q^3\right)^{1/3} Z \sin\alpha_o \quad (20)$$

The models are applicable for a slewing bearing with zero or negative clearance. The service life analysis used in this paper was based on these models.

3. Discussion on the effect of the parameters

It should be noted that there are two assumptions in the model developed in the paper. First, the bearing rings are ide-

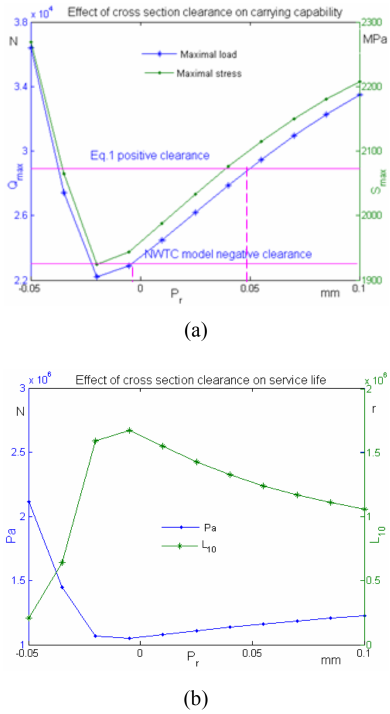


Fig. 4. (a) Effect of cross section clearance on the carrying capability; (b) Effect of cross section clearance on the service life.

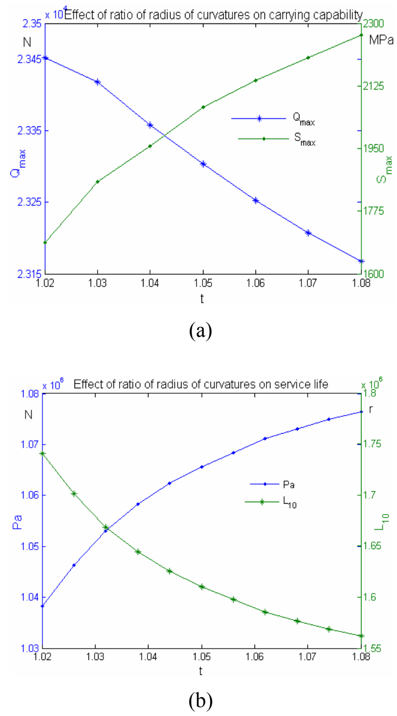


Fig. 5. (a) Effect of ratio of curvatures on the carrying capability; (b) Effect of ratio of curvatures on the service life.

ally stiff, and second, the turnover moment and radial force act in the same plane.

Fortunately, bearing rings are assembled in supporting (fixed) structures that are designed carefully with numerous preloaded blots. The structures and bearing rings can be assumed practically to be stiff enough. On the other hand, in most machines, such as cranes, radars and tanks, turnover moment and radial force act in the same plane, which is in accordance with the assumption. On the other hand, in some special fields where the assumptions are not satisfied, the model may not be applicable.

The calculation was based on a 013.50.1600 slewing bearing with the following parameters: average diameter D=1600mm, ball diameter d=50mm, initial contact angle α_o =45°, number of balls Z=91, axial force Fa=123.3kN, radial force Fr=3kN, and the turnover moment M=555.3kNm. When the slewing bearing is applied in a construction or mine machine where the slewing bearing always has a positive clearance, calculation is carried out using Eq. (1) (H=1, I=5, J=2.5) [1] and the result is Q_{max}=29kN. The result Q_{max}=23kN is calculated using the equation established by NREL [2], where the clearance is zero or negative.

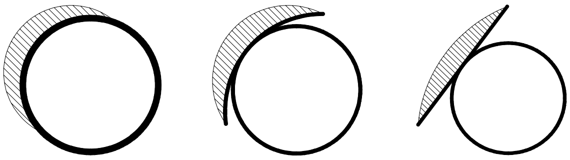
3.1 Effect of the cross section clearance (P_r)

The results of various cross section clearances (P_r), that have been calculated using the model developed in Section 2, are presented in Fig. 4, which shows the maximal contact load and stress with various clearances. The result of Eq. (1) and NREL models are also shown, verifying that the model discussed in Section 2 of this paper is correct to some extent. The least contact stress (largest carrying capability) and the longest service life appear when the slewing bearing has a very small interference [Figs. 4(a) and 4(b)]. In addition, both the carrying capability and the service life decreased sharply as the interference magnitude grew larger. The carrying capability and the service life had a slower decrease as the cross section clearance increased in a positive clearance slewing bearing.

In the positive clearance region [Fig. 4(a)], the maximal contact load is nearly proportional to the maximal contact stress, indicating that the area of contact surface (πab) is independent of the cross section clearance. Thus, the large maximal contact load indicates low carrying capability, which is in accordance with the commonly known. Initial contact accrued when there was interference in the slewing bearing. The area of the initial contact region depends greatly on the magnitude of interference, which is a nonlinear relation. The cross section clearance has no effect on the basic dynamic capability, but the effect on the dynamic equivalent thrust load and the service life cannot be neglected [Fig. 4(b)].

3.2 Effect of ratio of curvatures (t)

Most companies manufacture slewing bearings with t=1.04, however, there is no reasonable explanation for this fact. Fig. 5(a) shows an interesting result, wherein the maximal contact load decreases while the maximal contact stress increases as the ratio of curvatures varies from t=1.02 to t=1.08. Thus, the larger the maximal contact load, the higher the carrying capability, a notion which does not agree with commonly known



(a) (b) (c)

Fig. 6. Different contact status of various ratios of curvature: (a) t=1, (b) $1<t<\infty$ and (c) $t=\infty$.

maximal contact loads. The result indicates that the contact area (πab) grew faster than maximal contact load as the ratio of curvatures increased. This interesting result can be explained qualitatively in Fig. 6. Surface contact occurs when t=1, while $t=\infty$ leads to point contact (Fig. 6). The basic dynamic capability is independent of the ratio of curvatures [Eq. (19)], and little increase in the dynamic equivalent thrust load is found. Thus, the service life decreases slightly when the ratio of curvatures increases [Fig. 5(b)].

It should be noted that the contact between a rolling ball and a raceway is usually a conforming contact for the concave surface of the raceway (Fig. 6). The relation between contact load and stress can not be solved directly by Hertz's theory. Thus, the improvement can be found in references [2,3], in which the curvature of the concave raceway is negative.

3.3 Effect of the rolling ball diameter (d)

The rolling ball diameter (d) is a significant parameter of a slewing bearing. A large ball diameter leads to a small number of rolling balls. When the number of rolling balls applied to bear the external loads is decreased, the maximal contact load grows sharply, while the maximal contact stress decreases [Fig. 7(a)]. Less loaded balls would lead to a large contact load, large contact stress, and low carrying capability. However, the results obtained in this study indicate that the area of contact surface increased sharply and that there was an increase of the carrying capability. Similar to Fig. 6, if the ratio of curvatures is fixed, an infinitely large ball diameter means a surface contact; on the contrary, a point contact occurs if the ball diameter becomes infinitely smaller. Both the basic dynamic capability and the dynamic equivalent thrust load increased as the rolling ball diameter grew, but the basic dynamic capability increased faster than the dynamic equivalent thrust load. Thus, large rolling balls lead to long service life [Fig. 7(b)].

It should be noted that large rolling balls indicate less material of bearing rings, which implies less rigidity. The contact load and stress distribution depends significantly on supporting structures for a less rigid slewing bearing, in which case, the model established in the paper may not be applicable, and FEM is usually available. On the other hand, an extremely uneven load distribution over raceways may occur if the rigidity is not sufficient. Furthermore, an extremely uneven load distribution may lead to a sharp decrease in both carrying capability and service life of the slewing bearing.

3.4 Effect of initial contact angle (α_0)

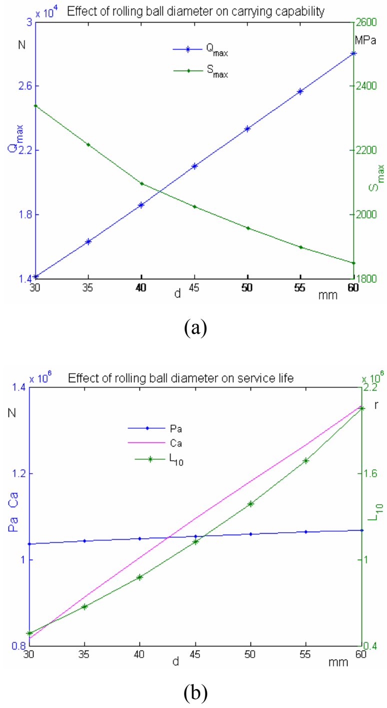


Fig. 7. (a) Effect of rolling ball diameter on the carrying capability; (b) Effect of rolling ball diameter on the service life.

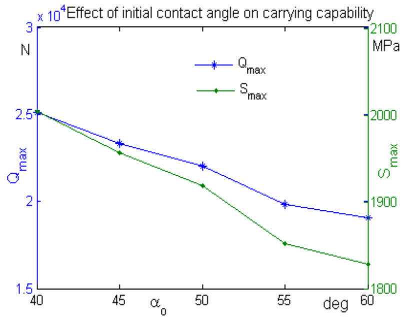
The initial contact angle (α_0) is considered conventionally as an important parameter, which influences the carrying capability of a slewing bearing significantly, whereas the large initial contact angle brings high carrying capability and long service life. Results of various initial contact angles are shown in Fig. 8. Both the maximal contact load and maximal contact stress decreased as the initial contact angle increased, this in agreement with conventional results, however, the magnitude of such decrease is small [Fig. 8(a)] and the influence is not as great as conventionally believed. The basic dynamic capability grows greatly as the initial contact angle increases, while a slight effect on the dynamic equivalent thrust load is observed. In addition, a sharp growth of the service life can also be observed [Fig. 8(b)].

If a slewing bearing has large initial contact angle and cross section clearance, the contact surface at the position of the maximal contact load is likely to cross the edge of the raceway in operation; however, this must be avoided.

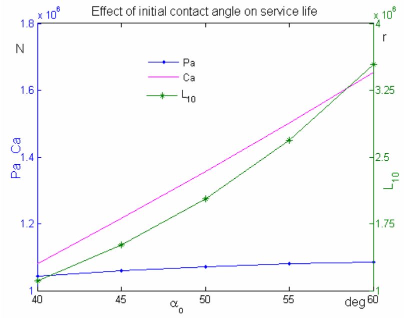
4. Conclusion

The results of the model developed in the paper are consistent with the results of other empirical models (Eq. (1) and the NREL model), demonstrating the validity of the model. The model can be applied to check the carrying capability and the service life of a slewing bearing and provide a reference for a better design.

The cross section clearance (P_r) has a significant effect on both the carrying capability and service life. Most slewing



(a)



(b)

Fig. 8. (a) Effect of the initial contact angle on the carrying capability; (b) Effect of the initial contact angle on the service life.

bearings are positive clearance bearings; thus, a slewing bearing clearance should be designed to be as small as machining accuracy and cost could allow. A negative clearance is required in some fields; in these instances, interference magnitude should be considered carefully because it could cause a sharp decrease in the carrying capability and service life of the bearing. The clearance also influences the load and stress distribution significantly, which has an important effect on the reliability of the slewing bearing. The results for the clearance will be published in our future works.

Meanwhile, the ratio of curvatures (t) is always neglected in empirical models. The results of this paper showed that the ratio of curvatures had a great effect on the carrying capability of a slewing bearing, but had little effect on service life. To improve the load capability of a slewing bearing, the ratio of curvatures should be designed to be as small as machining accuracy would allow.

A large rolling ball diameter (d) indicates large carrying capability and long service life, as well as decreases the rigidity of bearing rings. However, the rolling ball diameter of a slewing bearing of a specified average diameter is always standardized.

Compared with cross section clearance, the ratio of curvatures and rolling ball diameter, the initial contact angle (α_o) had much less effect on the carrying capability and the greatest effect on service life, which is quite different from the traditional view. If the contact surface does not cross the edge of the raceway, increasing the initial contact angle will significantly increase the service life of a slewing bearing.

Table 1. Variation of carrying capability and service life.

Geometry parameter	ΔQ_{max} (kN)	ΔS_{max} (MPa)	ΔCa_{max} (kN)	ΔPa_{max} (kN)	ΔL_{10max} (10^6r)
$-0.03 < P_r < 0.1$	11	300	0	400	1.2
$1.02 < t < 1.08$	0.3	500	0	40	0.18
$30 < d < 60$	14	500	540	35	1.6
$40^o < \alpha < 60^o$	0.5	178	574	40	2.4

The variations in carrying capability and service life with different geometry parameters is shown in Table 1, which also shows the sensitivity of each parameter influencing the carrying capability and service life of a slewing bearing.

As discussed above, carrying capability is influenced greatly by the area of contact surface. Therefore, the practice of checking or selecting a slewing bearing using the maximal contact load is not correct, although it is usually applied in some engineering fields.

On the one hand, if a high carrying capability is the main requirement, the slewing bearing should be designed with large ratio of curvature and rolling ball diameter. From Table 1 and the discussions above, there are certain differences in terms of the influences of the two parameters influence the carrying capability. The ratio of curvatures has a significant effect on the area of contact surface, but only a slight effect on the maximal contact load. The rolling ball diameter influenced both the maximal contact load and area of contact surface greatly.

On the other hand, if service life is the main requirement, more focus should be given on the initial contact angle, the cross section clearance, and the rolling ball diameter. Large initial contact angle, large rolling ball diameter and small cross section clearance can all lead to long service life as long as contact surface does not cross the edge of the raceway and the rigidity of the bearing rings is sufficient. Both the initial contact angle and the rolling ball diameter mainly influence the basic dynamic capability, which is a natural characteristic of a slewing bearing, and does not vary with external load conditions. Finally, the cross section clearance has a significant effect on the dynamic equivalent thrust load, which depends greatly on the external loads applied on the slewing bearing, but had no effect on the basic dynamic capability.

Acknowledgements

The authors would like to thank the Plan for the Industrialization of College Scientific Research Achievements in Jiangsu (Grant number: JH09-12), the Hi-tech Program of Jiangsu Key Laboratory of Digital Manufacturing for Industrial Equipment and Control Technology (Grant number: BM2007201) and Discipline Funds of Nanjing University of Technology (Grant number: NJUT2009101) for funding this research.

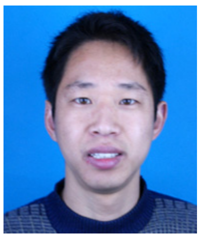
Nomenclature

A : Distance between diagonally opposed curvature centers when contact begins
A_0 : Initial distance between diagonally opposed curvature centers
A_{1F} : Final distance between $C_{id}C_{eu}$ finally (under a set of loads)
A_{2F} : Final distance between $C_{iu}C_{ed}$ finally (under a set of loads)
Ca : Basic dynamic capability
C_{ed} : Curvature center of lower raceway of the outer ring
C_{eu} : Curvature center of upper raceway of the outer ring
C_{id} : Curvature center of lower raceway of the inner ring
C_{iu} : Curvature center of upper raceway of the inner ring
D : Average diameter of a slewing bearing
F_a : Axial force
F_r : Radial force
H : Correction factor for the axial force
I : Correction factor for the turnover moment
J : Correction factor for the radial force
K : Contact stiffness factor
L_{10} : Service life with 10% probability of failure
M : Turnover moment
Pa : Dynamic equivalent thrust load
P_r : Cross section clearance
Q : Contact load
Q_1 : Contact load along $C_{id}C_{eu}$
Q_2 : Contact load along $C_{iu}C_{ed}$
Q_{max} : Maximal contact load
R : Radius of raceway curvature
S_{max} : Maximal contact stress
Z : Number of rolling balls in a slewing bearing
d : Diameter of rolling balls
fcm : Material factor
t : Ratio of curvatures
α_o : Initial contact angle
α_{1F} : Final (actual) contact angle when contact occurs along $C_{id}C_{eu}$
α_{2F} : Final (actual) contact angle when contact occurs along $C_{iu}C_{ed}$
δ_r : Radial relative displacement
δ_Z : Axial relative displacement
θ : Relative rotation
ϕ : Position of rolling balls
πab : Area of contact surface

References

[1] C. Y. Hong, Inferential reasoning formula for a slewing bearing, *Journal of Chongqing Architecture University*, 2 (1) (1980) 82-108.
[2] T. Harris, J. H. Rumbarger and C. P. Butterfield, Guideline DG03 Wind turbine design yaw & pitch rolling bearing life, NREL, USA (2000).
[3] T. A. Harris and M.N. Kotzalas, Rolling bearing analysis, Fifth Ed. Taylor & Francis Group (2006).
[4] J. I. Amasorrain, X. Sagartzazu and J. Damián, Load distribution in a four contact-point slewing bearing, *Mechanism and Machine Theory*, 38 (6) (2003) 479-496.
[5] S. Zupan and I. Prebil, Carrying angle and carrying capacity of a large single row ball bearing as a function of geometry parameters of the rolling contact and the supporting structure stiffness, *Mechanism and Machine Theory*, 36 (10) (2001) 1087-1103.
[6] A. Daidié, Z. Chaib and A. Ghosn, 3D simplified finite elements analysis of load and contact angle in a slewing ball bearing, *Journal of Mechanical Design*, 130 (8) (2008) 082601-1.
[7] T. Smolnicki, D. Derlukiewicz and M. Stańco, Evaluation of load distribution in the superstructure rotation joint of single-bucket caterpillar excavators, *Automation in Construction*, 17 (3) (2008) 218-223.
[8] S. Zupan, R. Kunc and I. Prebil, Experimental determination of damage to bearing raceways in rolling rotational connections, *Experimental Techniques*, 30 (2) (2006) 31-36.
[9] R. Kunc, A. Zerovnik and I. Prebil, Verification of numerical determination of carrying capacity of large rolling bearing with hardened raceway, *International Journal of Fatigue*, 29 (9-11) (2007) 1913-1919.
[10] R. Kunc and I. Prebil, Numerical determination of carrying capacity of large rolling bearings, *Journal of Material Processing Technology*, 155-156 (1-3) (2004) 1696-1703.
[11] International Standard ISO 281, Rolling bearing-dynamic load ratings and rating life, 1990(E).
[12] American National Standard, ANSI/AFBMA Std 9-1990, Load ratings and fatigue life for ball bearings.

Xiaodiao Huang received his MS degree from Harbin Engineering University, China, in 1987. He is currently a Professor at the Nanjing University of Technology. His research interests are in the areas of mechanical manufacturing and automation, NC technology, and technology on huge bearings and gears.



Xuehai Gao received his BS degree from the Nanjing University of Technology, China, in 2007. He is currently working towards his PhD degree in the School of Mechanical and Power Engineering, Nanjing University of Technology. His current research interests are on reliability of wind turbine and technology on huge bearings.