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Wear depth calculation and influence factor analysis for groove ball bearing

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Abstract. Based on Hertz contact theory and load distribution, the formulas for contact stress cycle times, slip distance and wear depth measurements are derived, and the influences of load, curvature coefficient, roll body diameter and friction coefficient on the contact region wear depth and distributions are thoroughly analyzed. The results show that the wear depth is zero at the pure rolling point and the long half-axle terminals of contact ellipse, and reaches maximum value near by the long half-axle terminals of the contact ellipse, and further shows that the wear depth increases with increase of the load and friction coefficient, however decreases with increase of the curvature coefficient and roll body diameter.

1 Introduction

Higher speed roll bearing in the aviation engine and its transmission system has merits of low friction coefficient, small starting moment and strong anti-off oil performance, etc. The groove ball bearing could not only suffer radial load, but also suffer the radial load and a certain amount of the axial load. Whereas, the roll body can occur direct metal contact between inner and outer race which could lead to severe friction and wear under heavy load condition at high speed, especially under starved lubrication. Consequently, the geometry and drive precision could be severely affected, and even slight geometric change would also affect the stress distribution and service life of the bearing. Moreover, the bearing clearance would also increase with increase of the wearing depth which may further result in the decrease of operation and sustaining precisions and the increase of vibration and noise.

In order to well master operation performance and structure design of the groove ball bearing, scholars have focused on the friction and wearing studies in recent years. Fitzsimmons [1] investigated the influence of impurity density, particle size and hardness in lubricants on wear of taped roller bearing. Olofsson，*et al*. [2-7] derived the wear value of spherical thrust roller bearing under boundary lubrication according to Archard wear theory, and conducted the wear experiment. The test results show better consistence with that of the theoretic calculation. Miki, *et al*. [8] studied the influence of rotation speed and lubricant viscosity on the wear value. Value. Zhang Yicheng, *et al*. [10] proposed a formula for slip wear of the high pair based upon the energy intensity theory, which well improved the wear

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of the high pair based upon the energy intensity theory, which well improved the insufficiency of the current wear calculation method under different mixed wear types. Shi Ying, *et al*. [11] established a wear digital calculation model for the cylindrical roller bearing by employing the digital simulation method.

2 Calculation of Hertz point contact

Roller element and inner/outer race of groove ball bearing is defined as point contact under empty load. With the increase of the radial load, point contact would become surface contact and further form elliptical contact.

The point contact of groove ball bearing can be resolved by applying the Hertz point contact theory, and the long half-axis a, short half-axis b, deformation δ and maximum contact stress p_0 of elliptical contact can be calculated by

$$a = a^{*}\left(\frac{3Q}{2\sum \rho E'}\right)^{1/3} \tag{1}$$

$$b = b^{*}\left(\frac{3Q}{2\sum \rho E'}\right)^{1/3} \tag{2}$$

$$\delta = \delta^{*}\left(\frac{3Q}{2\sum \rho E'}\right)^{2/3}\frac{\sum \rho}{2} \tag{3}$$

$$p_0 = \frac{3Q}{2\pi ab} \tag{4}$$

where, $\Sigma\rho$ is the sum of main curvatures, E' and Q are the comprehensive elastic module and maximum load of roller element respectively. Herein, the coefficients of a^*, b^*, δ^* could be obtained by employing the calculation method or parameter table [12].

3 Calculation method of contact stress cycle times

For the total wear, the roller element number passing on a point of the inner or outer race in unit time should be obtained in advance. Assuming the roller element number is Z in the groove ball bearing, passing on the inner and outer race is

$$J_i = Zn_{mi} = \frac{Z}{2}\left(n_o - n_i\right)\left(1 + \frac{D_b}{D_m}\right) \tag{5}$$

$$J_o = Zn_{om} = \frac{Z}{2}\left(n_o - n_i\right)\left(1 - \frac{D_b}{D_m}\right) \tag{6}$$

It is assumed that

$$K_i = \frac{1}{2}\left(1 + \frac{D_b}{D_m}\right) \tag{7}$$

$$K_o = \frac{1}{2}\left(1 - \frac{D_b}{D_m}\right) \tag{8}$$

$$J_i = Zn_{mi} = ZK_i\left(n_o - n_i\right) \tag{9}$$

$$J_o = Zn_{om} = ZK_o\left(n_o - n_i\right) \tag{10}$$

The roller element numbers passing on one point are not surely equal to the stress cycle times at this point. If the included angle of load distribution region is $2\Psi_L$ and the inner race rotation is n, the stress cycle times passing on one point of the inner and outer race per minute respectively are

$$J_i' = nZK_i \frac{2\psi_L}{360} \quad (11)$$

$$J_o' = nZK_o \quad (12)$$

The stress in the race contact region may occur a certain of variations in the cyclic process because the roll bearing suffers load distribution in Cosine function. When the race rotates to the maximum loaded area, the contact stress is the maximum one. Whereas, when the race rotates to other loaded area, the contact stress behaves in Cosine distribution. Assuming a roller bearing has 5 loaded roller elements, the inner race stress, the near two roller element stress and the other two roller element stress are p_0 , p_1 and p_2. Consequently, one point stress cycle times on the inner race is

$$J_i' = \left\lfloor \left(\frac{1}{5}\right)_{p_0} + \left(\frac{2}{5}\right)_{p_1} + \left(\frac{2}{5}\right)_{p_2} \right\rfloor J_i' \quad (13)$$

Furthermore, when the loaded roller element is in condition of odd pressure, and the loaded roller element number is z_0, as well as $k_n=(z_0-1)/2$, then, we obtain

$$J_i' = \left\lfloor \left(\frac{1}{2k_n+1}\right)_{p_0} + \left(\frac{2}{2k_n+1}\right)_{p_1} + \left(\frac{2}{2k_n+1}\right)_{p_2} + \ldots\ldots + \left(\frac{2}{2k_n+1}\right)_{p_{kn}} \right\rfloor J_i' \quad (14)$$

If the loaded roller element is in condition of even pressure, as well as $k_n=z_0/2$, such that

$$J_i' = \left\lfloor \left(\frac{1}{k_n}\right)_{p_0} + \left(\frac{1}{k_n}\right)_{p_1} + \left(\frac{1}{k_n}\right)_{p_2} + \ldots\ldots + \left(\frac{1}{k_n}\right)_{p_{kn}} \right\rfloor J_i' \quad (15)$$

4 Calculation of wear measurement

4.1 Calculation of slip distance

Figure 1 shows the contact and slip analysis region between ball and race, in process of the ball bearing rotation. For calculating wear depth at point (x, y) of the contact region, the point slip distance L_x should be determined when the point is acted by the contact stress at each time

$$L_x = \int_0^t V_x dt \quad (16)$$

where, V_x is the sliding speed at point (x, y) of elliptical contact with respect to the pure rolling point, and t is the time yielding relative sliding at each contact,

$$t = \frac{2b_x}{V_0} \quad (17)$$

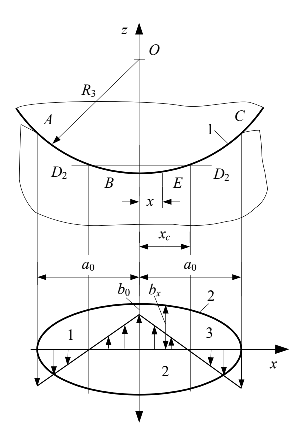

Fig.1 Slip region between ball and race

where, b_x is the half length of point (x, y) in perpendicular direction with respect to the x axis, and V_0 is the sliding speed of contact region, i. e., the relative speed of roller element with respect to race along the short half-axis. According to distribution law of the elliptical contact and speed, V_x is formulated as

$$V_x \approx \frac{V_0}{2}\left[\left(\frac{x_c}{R_3}\right)^2 - \left(\frac{x}{R_3}\right)^2\right] \tag{18}$$

where， $R_3 = D_b/2$

$$L_x = V_x t \approx b_x\left[\left(\frac{x_c}{R_3}\right)^2 - \left(\frac{x}{R_3}\right)^2\right] \tag{19}$$

where, x_c is the coordinate of pure rolling point in the long half x axis of elliptical contact, and its position can be calculated by

$$0.25Q = \int_0^{x_c}\int_{-b_0}^{b_0} p_{xy}dxdy \tag{20}$$

4.2 Calculation of wear depth

According to [10], the wear rate of ball bearing can be obtained by

$$I_h = k\left(\frac{p\xi(f)}{HB\varepsilon_0}\right)^m\left(\frac{R_{\max}}{r\psi^{1/\chi}}\right)^c \tag{21}$$

where, HB is the hardness of material, ε_0 is the extension rate when the material is broken, p is the contact stress, and r, R_{max} are the classic curvature radius and maximum height of micro-bulge part on the contact surface, ψ, χ are the curvature coefficients of support surfaces, k, m, c are the experiential values, respectively. Considering of the influence of lubricant on the contact region, $\xi(f)$ f is taken values of f, $1+f$ or $(1+f^2)^{1/2}$.

Setting

$$G_0 = k\left(\frac{\xi(f)}{\varepsilon_0}\right)^m\left(\frac{R_{\max}}{r\psi^{1/\chi}}\right)^c \tag{22}$$

where, G_0 is the wear constant.

As a result, the contact ellipse wear depth, at a distance x with respect to the half axle, can be calculated as

$$\Delta h = G_0\left(\frac{p}{HB}\right)^m L_x \tag{23}$$

If the inner circle speed of bearing is n and the outer circle is in static state, and further assuming each slip distances are L_{xp_o}, L_{xp_1} , L_{xp_2} …… L_{xp_n} with respect to the acted stresses p_0, p_1, p_2……p_{kn} , respectively. The bearing wear depth measurement at rotation time t_1 minutes can be obtained as

$$\Sigma\Delta h = \left[\left(\frac{1}{2k_n+1}\right)_{p_0} p_o^m L_{xp_o} + \left(\frac{2}{2k_n+1}\right)_{p_1} p_{1o}^m L_{xp_1} + \left(\frac{2}{2k_n+1}\right)_{p_2} p_2^m L_{xp_2} + \ldots.. + \left(\frac{2}{2k_n+1}\right)_{p_{kn}} p_{kn}^m L_{xp_{n=}}\right] \times nZK_i \frac{2\psi_L}{360} G'\left(\frac{1}{HB}\right)^m t_1 \tag{24}$$

5 Analysis of influence factor on wear depth

It is found that the wear depth is zero at the pure rolling point x=0.348a and at the terminals of ellipse long half-axis from the Figs.2-5, in which the wear depth initially increases to maximum value and then decreases to zero at the terminals, as well as the wear depth shows nearly parabolic distribution between two pure rolling points.

Fig.2-4 further show the wear depth affected by the load, curvature coefficient and roller element diameter when the friction coefficient is 0.007, respectively. Amongst, the wear depth is strongly affected by the load and rapidly increases with the increase of load, as shown in Fig.2, because the load affects the wear depth with m power and also affects the slip distance. Fig.3 illustrates that the roller element diameter yields slight influence on the wear depth, which decreases with the increase of roller element diameter, and Fig.4 shows that the wear depth is also slightly affected by the curvature coefficient and decreases with the increase of curvature coefficient, however, the curvature coefficient affects the wear depth more heavy than that of the roller element diameter. Fig.5 further shows the influence of friction coefficient on the wear depth, from which it is found that the wear depth is strongly affected by the friction coefficient with m power of friction coefficient.

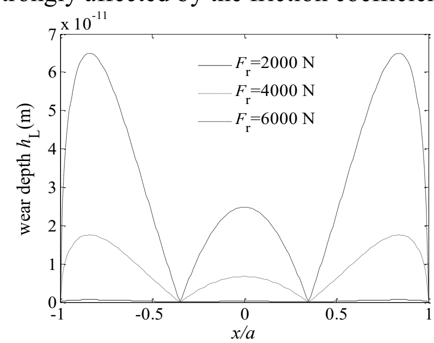

Fig.2 Load influence

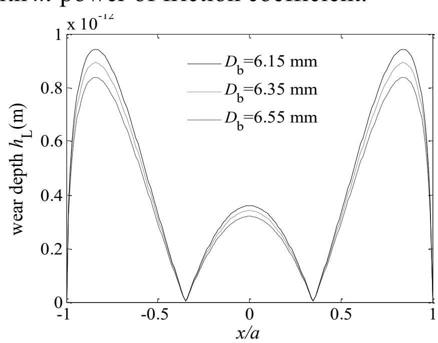

Fig.3 Roller element diameter influence

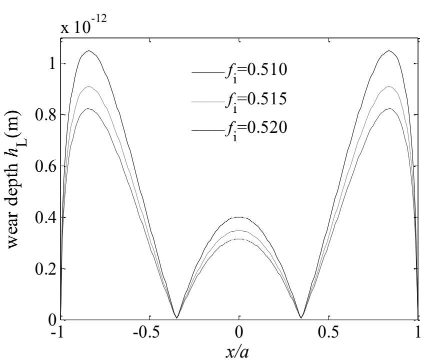

Fig.4 Curvature coefficient influence

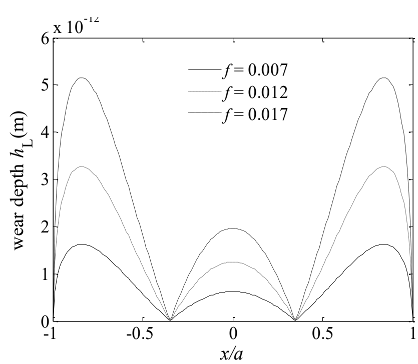

Fig.5 Friction coefficient influence

6 Conclusions

(1) The formulas are derived for calculating the stress cycle times for one point of race passing through different loaded regions, and for calculating the slip distance and total wear depth under a single stress action.

(2) It is concluded that wear depth holds a smallest value of zero at the pure rolling point and at the ellipse long half-axis terminals, and the maximum wear depth does not locate in the center of elliptical contact but nearby the long half-axis terminals.

(3) The wear depth increases with increase of the load and friction coefficient, and decreases with increase of the roller element diameter and curvature coefficient, as well as the load and friction coefficient strongly affect the wear depth, while the roller element diameter and curvature coefficient affect the wear depth in a slight way.

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