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Relative fatigue life prediction of high-speed and heavy-load ball bearing based on surface texture

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ABSTRACT

In this paper, a new calculation method of relative fatigue life considering surface texture on high-speed and heavy-load ball bearing is presented. Rolling bearing quasi-dynamics, micro-TEHL analysis, non-Gaussian surface simulating technique and stress analysis with FEM are combined to obtain the relative fatigue life. The maximum subsurface stress with the new method are compared with data in inference, the relative error is 3.9%, which caused by decreasing surface stress under high speed. The non-Gaussian surface textures effect on surface pressure and shear stress are studied, which show transverse texture perpendicular to entrainment direction is helpful to form film because of the increasing hydrodynamic effect, and decrease the pressure and shear stress, but longitudinal texture has an opposite effect. Non-Gaussian parameters effects on relative fatigue life are researched, which display relative fatigue life increase with the increase of transverse texture, and decrease with the increasing longitudinal texture. The increasing skewness and curtosis can bring the decrease of relative fatigue life, but the relative fatigue life is beneficial as skewness is negative and curtosis is small, which is helpful for bearing life. Finally, the whole relative fatigue life of inner race and balls are given.

1. Introduction

Aero-engine main-shaft bearings are typical bearings with high speed, heavy load and great heat [1,2]. With the development of bearing material, the fatigue life improved, but lots of problems related to the life prediction still need to be studied for the complicated running conditions. In the early study, L-P model [3,4] was usually employed to predict bearing fatigue life, but the predicted life was far shorter than the actual one for the improving material. Later based on L-P model, I-H model [5] was developed, which had a consistent predicting life with the actual one, but as the stress is very large, the denominator becomes 0 and the model fails. Zaretsky [6] developed the fatigue life model neglecting the depth term, which can avoid the above problem effectively. Fatigue life model is usually based on crack initiation criterion. The crack initiation generates in two ways, one is from the maximum stress in subsurface to surface, and early researchers had done lots of work [3–5]. And the other one is from maximum stress in surface with roughness peeks or defects to subsurface, which related with elastohydrodynamics lubrication (EHL) and surface roughness, recently, the correlative studies increase.

The experimental research of surface skewness effect on fatigue life was investigated by Akamatsu [7]. Zhai [8] simulated the pitting surface and carried on EHL analysis, which explained the phenomenon in Ref. [7]. Vrbka [9] had done experiments to study skewness effect on fatigue life, which stated the reasonable texture design can be helpful for lubrication and increase fatigue life. Based on developed I-H model, Deolalikar and Sadeghi [10] discussed load, velocity and RMSD effects on relative fatigue life with mixed lubrication. 2D line contact fatigue life with several fatigue failure criteria considering micro-EHL was researched by Qiao [11]. On the basis of Zaretsky fatigue life model, The mixed lubrication model and contact stress analysis model were employed to study the real machining surface roughness effects on relative fatigue life by Epstein and Keer [12]. Zhu [13] studied the 3D roughness effect on rolling fatigue life with mixed lubrication model. The non-Gaussian surface parameters effects on point contact fatigue life had been studied by Yan [14,15], and Jia [16] researched the roughness and viscosity effect on line contact fatigue life.

But the fatigue life with surface roughness in most of researches can not be applied for rolling bearing directly, because the above model all based on single rolling contact pairs and the applied contact conditions are given directly. But the contact conditions are unknown in practical engineering and can be effected greatly by system and environmental conditions, which should be obtained by bearing kinematic and mechanical analysis.

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Nomenclature

XYZ	bearing fixed coordinate system
$x_j'y_j'z_j'$	follow-up coordinate system of the jth ball
Φ_j	azimuth angle of the jth ball (deg)
ω_2, ω_m, $\omega_{o'j}$	revolution angular velocities of inner race, cage and balls (rad/s)
$\omega_{x'j}$, $\omega_{y'j}$, $\omega_{z'j}$	rotation angular velocity components of the jth ball
ω_{2m}	relative angular velocity of inner race and cage
D_1, D_2	diameters of outer race and inner race
Dr_1, Dr_2	diameters of outer raceway and inner raceway
D_m	pitch circle diameter, $D_m=0.5(Dr_1+Dr_2)$
D_m, D_w	diameters of pitch-circle and balls (m)
R_2, R_x, R_y	inner raceway radius and radii in x and y directions (m)
α_{2j}	contact angle of balls and inner race (deg)
a_{2j}, b_{2j}	semi-minor axis and semi-major axis of the contact ellipse of ball and inner race (m)
R_k	the elliptical ratio
U_{2bj}, U_{2rj}	velocities of the points in the ball and inner race in the rolling direction (m/s)
U_{2sj}, U_{2j}	sliding and entrainment velocities of the ball and inner race in the rolling direction (m/s)
Q_{2j}	contact load of the ball and inner race
S_{2xj}	slide ratios in the rolling direction
U_{o2j}	entrainment velocity at the contact center in the rolling direction (m/s)
P_{2j}, p_H	film pressure and maximum Hertzian pressure (Pa)
h_{2j}, h_0	film thickness, reference parameter for dimensionless film, $h_0=a_{2j}^2/R_x$ (m)
t, t_0	temperature of lubricant, environment temperature (°C)
x_{2jin}, x_{2jout}	inlet coordinates of calculation domain (m)
y_{2jin}, y_{2jout}	outlet coordinates of calculation domain (m)
η, η_0	viscosity of lubricant, environment viscosity (Pa s),
ρ, ρ_0	density of lubricant, environment density (kg m^{-3})
C_1, C_2	pressure–density coefficient
C_3	temperature–density coefficient
z_0	pressure–viscosity coefficient
s_0	temperature–viscosity coefficient
x_{2j}, y_{2j}, z_{2j}	coordinates along semi-minor axis, semi-major axis and film thickness directions (m)
$\bar{x}_{2j}$, $\bar{y}_{2j}$, $\bar{z}_{2j}$	dimensionless coordinates, $\bar{x}_{2j}=x_{2j}/a_{2j}$, $\bar{y}_{2j}=y_{2j}/a_{2j}$, $\bar{z}_{2j}=z_{2j}/h_0$
F_X, F_Y, F_Z	axial and radial loads of bearing (N)
M_Z, M_Y	bending moments along X and Y directions (N m)
R_{2j}	total roughness height of ball and inner race
R_q	roughness root-mean-square, $\bar{R}_q=R_q/h_0$
R_{q1}, R_{q2}	roughness root-mean-square of ball and inner race
sk, cu	roughness skewness and curtosis
sk_1, sk_2	roughness skewness of ball and inner race
cu_1, cu_2	roughness curtosis of ball and inner race
L_x, L_y	roughness autocorrelation length in x and y directions, $\bar{L}_x=L_x/\alpha_{2j}$, $\bar{L}_y=L_y/\alpha_2$
L_{x1}, L_{y1},	roughness autocorrelation length in x direction of ball
L_{x2}, L_{y2}	roughness autocorrelation length in y direction of inner race
L_{2jR}	relative fatigue life of inner race and ball

The accuracy kinematic and mechanical parameters are important prerequisite of rolling contact life prediction. So in this paper, taking 476728NQ bearing as an example, quasi-dynamics is carried on to obtain kinematic and mechanical parameters, digital filter technique is employed to simulate surface roughness, surface stress will be obtained by micro-TEHL analysis in a full film and steady state, subsurface stress can be obtained by FEM, and Zaretsky model is employed to get the relative life.

2. Governing equations

2.1. Quasi-dynamic model

In this paper, the kinematic and mechanical parameters of rolling contact pairs are obtained by bearing quasi-dynamics, and Ref. [17] gives the quasi-dynamic equations. Assuming inner race rotates and outer race is fixed, the azimuth coordinate system and loads of a ball bearing are shown in Fig. 1. Rolling bearing can be loaded with $\{F_X, F_Y, F_Z, M_Y, M_Z\}$.

Taking the contact pairs of ball and inner race as cases, Fig. 2 gives the contact motion state, only the velocity in the rolling direction is considered, the velocity perpendicular to the rolling direction is assumed as 0.

The velocity U_{2bj} on the ball and the velocity U_{2rj} on the inner race at the contact region are given in formula (1) and (2).

$$U_{2bj} = (\omega_{x'j}\cos\alpha_{2j} + \omega_{z'j}\sin\alpha_{2j}) \times \left[D_m/(2\cos\alpha_{2j}) - \left(\sqrt{R_2{}^2 - y_{2j}{}^2} - O'M\right)\right] \tag{1}$$

$$U_{2rj} = (\omega_2 - \omega_{o'j})\cos\alpha_{2j} \times \left[D_m/(2\cos\alpha_{2j}) - \left(\sqrt{R_2{}^2 - y_{2j}^2} - O'M\right)\right] \tag{2}$$

where the entrainment velocity $U_{2j}=(U_{2rj}+U_{2bj})/2$, $O'M = \sqrt{R_2{}^2 - a_{2j}{}^2} - \sqrt{(D_W/2)^2 - a_{2j}{}^2}$, the sliding velocity $U_{2sj}=U_{2bj}-U_{2rj}$, the slide ratio is defined as the ratio of sliding velocity to entrainment velocity at the contact center, $S_{2xj}=U_{2sj}/U_{o2j}$, the entrainment velocity at the contact center is written as follows:

$$U_{o_2j} = [\omega_{x'j}\cos\alpha_{2j} + \omega_{z'j}\sin\alpha_{2j} + (\omega_2 - \omega_{o'j})\cos\alpha_{2j}]\,[D_m/(2\cos\alpha_{2j}) + (R_{2j} - O'M)]/2 \tag{3}$$

2.2. Micro-TEHL model

The Reynolds equation [18] associated with thermal effect is given by the following form:

$$\frac{\partial}{\partial x_{2j}}\left[\left(\frac{\rho}{\eta}\right)_e h_{2j}{}^3 \frac{\partial p_{2j}}{\partial x_{2j}}\right] + \frac{\partial}{\partial y_{2j}}\left[\left(\frac{\rho}{\eta}\right)_e h_{2j}^3 \frac{\partial p_{2j}}{\partial y_{2j}}\right] = 12U_{2j}\frac{\partial(\rho^* h_{2j})}{\partial x_{2j}} \tag{4}$$

where $(\rho/\eta)_e$ and ρ^* are defined as:

$(\rho/\eta)_e = 12(\eta_e\rho'_e/\eta'_e - \rho''_e)$, $\rho^* = 2(\eta_e\rho'_e(U_{2bj} - U_{2rj}) + \rho_e U_{2bj})$.

where η_e, η'_e, ρ_e, ρ'_e, ρ''_e can be obtained by:

$$\eta_e = h_{2j}\Big/\int_0^{h_{2j}}\frac{dz_{2j}}{\eta};\ \eta'_e = h_{2j}^2\Big/\int_0^{h_{2j}}\frac{z_{2j}dz_{2j}}{\eta};\ \rho_e = \frac{1}{h_{2j}}\int_0^{h_{2j}}\rho dz_{2j};$$