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Friction moment analysis of space gyroscope bearing with ribbon cage under ultra-low oscillatory motion

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Abstract This paper presents the model of calculating the total friction moment of space gyroscope ball bearings which usually work under ultra-low oscillatory motion and are very sensitive to the friction moment. The aim is to know the proportion of the friction moment caused by each frictional source in the bearing's total friction moment, which is helpful to optimize the bearing design to deduce the friction moment. In the model, the cage dynamic equations considering six degree-of-freedom and the balls dynamic equations considering two degree-of-freedom were solved. The good trends with different loads between the measured friction moments and computational results prove that the model under constant rate was validated. The computational results show that when the speed was set at 5 r/min, the bearing's maximum total friction moment when oscillation occurred was obviously larger than that occurred at a constant rate. At the onset of each oscillatory motion, the proportion of the friction moment caused by cage in the bearing's total friction moment was very high, and it increased with the increasing speed. The analyses of different cage thicknesses and different clearances between cage pocket and ball show that smaller thickness and clearance were preferred.

1. Introduction

It is a continuous goal for bearing engineers to always develop advanced bearings that provide higher efficiency, lower friction, and more reliable, etc, under adverse operating conditions.[1] There is a considerable interest in understanding the friction moments of space gyroscope ball bearings used in many sensing mechanisms, such as those used for attitude control in spacecraft, which often run under ultra-low oscillatory motion for extended periods. Usually these bearings' starting

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friction moments are tested by particular friction moment tester[2] through frequent start-stop operation. From early papers,[3–10] as a result of frequent start-stop operation, the bearing's friction moment often traces out hysteresis loops which contain two regions, the pre-rolling and the steady rolling. Dahl[3,4] developed an empirical equation to describe the hysteresis behavior. Todd and Stevens,[5] Todd and Johnson[6] analyzed the shape of hysteresis friction curves, and these curves were found to be in agreement with hysteresis loops generated using Dahl's equation. Lovell et al.[7–10] verified this hysteresis curves by three-ball experimental testing apparatus at a constant rate and sinusoidal oscillating rate, respectively. In the Lovell's study, the experimental testing apparatus was designed to incorporate three balls, rather than a full bearing (without considering the cage) so as to eliminate the additional friction elements and the hysteresis behavior could be readily determined. In the Lovell's experiments, the balls' diameters (12.7 mm) were bigger and the loads (81.6, 133.5, 185.4 N/ball) were larger. All of the above studies proved that the elastic hysteresis in relatively large ball bearings is an important source under high loads and ultra-low speed conditions. Whether the friction moment caused by elastic hysteresis under lower loads in mini-ball bearing occupies a large proportion in the total bearing's friction moment needs to be studied. In addition, the sources of friction in ball bearings are manifold[11] and other friction sources also need to be known, since the friction moment is an important factor for controlling space instruments. Some experiments[12,13] for the gyroscope bearings proved that the friction moment caused by cage was high and might not be neglected at the onset of bearing. In order to know the friction moment caused by that, a study of the ribbon cage dynamics at ultra-low speeds and its influence on the bearing's total friction moment is carried out in this paper.

More advanced analyses about dynamic behavior of cage, such as the references of Gupta,[14] Meeks,[15,16] and Houpert[17–19] have been developed at higher speeds. Because of the influences of the cage centrifugal force, the skidding phenomenon and the viscous drag of the lubricant between ball and cage, the cage analysis model at higher speed is much more complicated than that at lower speed. The difficulty in their high speed dynamic models is a considerable number of computations in a short final output time, for example, the final output time shown in Ref.[16] was 0.01 s and 2.5 s in Ref.[19]. In spite of the force model at ultra-low speeds can be relatively simply designed, the time step is much smaller than that at higher speed. The time for output results needs to be 3–5 s at least under constant rate and even longer under oscillatory motion. It needs to change the time steps to perform the numerical integration instead of the constant time step in the model. However, changing the time steps makes the program more easily divergent especially under oscillatory motion. How to make sure that the program carries out smoothly and how to improve the final output time are introduced in this paper.

2. Total friction moment of bearing

The total friction moment includes five principal sources[11] of friction in space gyroscope ball bearings. They are the friction moment due to the elastic hysteresis in rolling M_E, the friction moment due to the geometry of the contacting surfaces M_D, the friction moment due to the pivoting on contact ellipse M_S, the friction moment due to the viscous drag of lubricant M_L, and the friction moment due to the sliding between the cage and balls M_C. M_E, M_D, M_S M_L have to be deduced maturely by static analysis and M_C needs to be analyzed with more complicated dynamic models. According to the analyses by Houpert,[20] the total friction moment M acting on the outer race of bearing is

$$\begin{aligned} M &= \sum^{z}(\mathrm{d}M_L+\mathrm{d}M_E+\mathrm{d}M_D+\mathrm{d}M_S+\mathrm{d}M_C) \\ &= M_L+\sum^{z}\left(\begin{array}{l}\dfrac{\mathrm{d}M_{E\mathrm{o}}R_\mathrm{i}+\mathrm{d}M_{E\mathrm{i}}R_\mathrm{o}}{D_w}+\dfrac{\mathrm{d}M_{D\mathrm{o}}R_\mathrm{i}+\mathrm{d}M_{D\mathrm{i}}R_\mathrm{o}}{D_w} \\ +\dfrac{\mathrm{d}M_{S\mathrm{i}}+\mathrm{d}M_{S\mathrm{o}}}{2}\sin\beta+\dfrac{(F^p_\mu \pm F^p_{pbz})R_\mathrm{o}}{2}\end{array}\right) \end{aligned} \quad (1)$$

where z is the number of balls, D_w is the ball diameter, β is the contact angle, and $\mathrm{d}M_L$, $\mathrm{d}M_E$, $\mathrm{d}M_D$, $\mathrm{d}M_S$, and $\mathrm{d}M_C$ are the individual one ball contribution to the friction moment of lubricant, elastic hysteresis, the geometry of the contacting surfaces, pivoting effects and ball-cage contact, respectively. R_i, R_o are the radii to initial contact point between the ball and inner and outer race, respectively. The formulas of M_L, $\mathrm{d}M_{E\mathrm{i}}$, $\mathrm{d}M_{E\mathrm{o}}$, $\mathrm{d}M_{D\mathrm{i}}$, $\mathrm{d}M_{D\mathrm{o}}$, $\mathrm{d}M_{S\mathrm{i}}$, $\mathrm{d}M_{S\mathrm{o}}$ and their references are deduced by following Eq. (2). The subscripts i, o represents the inner and outer race, respectively.

$$\begin{cases} M_L = 160 \times 10^{-7} f d_m^3 \\ \mathrm{d}M_E = \frac{3}{16} Q_\mathrm{n}(i) b \alpha_r \\ \mathrm{d}M_D = \dfrac{\mu_b Q_\mathrm{n}(i) a^2}{R_a}\left(\dfrac{1}{10}+\dfrac{X_2^3-X_1^3}{4}+\dfrac{3X_1^5-3X_2^5}{20}\right) \\ \mathrm{d}M_S = \dfrac{3\mu_b Q_\mathrm{n}(i) a}{8}\left[2(X_2^2-X_1^2)+X_1^4-X_2^4\right] \end{cases} \quad (2)$$

In Eq. (2), f is a factor depending upon type of bearing and method of lubrication. d_m is the bearing's pitch diameter. $Q_\mathrm{n}(i)$ is the normal force between the ith ball and race. The losing energy caused by the hysteresis is thought to be the energy caused by the friction, and the losing energy due to the elastic hysteresis is a small proportion α_r (for metal, usually $0.7\% \leqslant \alpha_r \leqslant 1.0\%$, here $\alpha_r = 0.8\%$). a and b are semimajor and semiminor axes of the contact ellipse respectively. R_a is the Hertz contact radius. $X_1 = x_1/a$, $X_2 = x_2/a$ $(x_2 > 0)$, where x_1, x_2 is the locations of two pure rolling lines respectively on the contact ellipse in Ref.[20] F^p_{pbz} in Eq. (1) is the z_p component in Frame p (introduced later) of the normal ball-cage contact force and $F^p_\mu = \mu F^p_{pbz}$, of which μ is the friction coefficient between the ball and cage and F^p_{pbz} is positive or negative depending upon whether the ball is driving the cage

Table 1 Structure parameters.

Structure parameter	Value
Inner diameter of inner race, d (mm)	4.00
Outer diameter of outer race, D (mm)	9.00
Ball diameter, D_w (mm)	1.30
Number of balls, z	8.00
Contact angular, β (°)	0.00
Inner race conformity, f_i	0.55
Outer race conformity, f_o	0.55
Depth of ribbon cage pocket, k (mm)	0.67
Diameter of cage pocket, d_p (mm)	1.40
Thickness of cage, s (mm)	0.15
Clearance between cage pocket and ball, Λ (mm)	0.02