

Condition Monitoring Tool for Angular Contact Ball Bearing

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Abstract: - This paper deals with condition monitoring of an angular contact ball bearing in high speed at applicable axial loading in rotating systems such as motors or space actuators that nowadays can be found in a variety of consumer products. Some measuring methods of torque in instrument ball bearing in small motors or similar applications have been summarized. All these methods are not suitable for the extremely low torques and depend on dimensions of the systems. Therefore a completely new diagnosis tool for angular contact ball bearing with applied applicable axial loading at high speed, in applicable range is designed and developed. This tool allows us to gauge ball bearing quality and integrity, ensuring that precision specifications are under control, and the maximum efficiency can be gained. Through this tool the user ensures maximum performance of ball bearing by physically diagnosing of irregularities in bearing prior to use. Problems such as retainer hang-up, ball or race surface problems and contamination, can be determined on a bearing through different samplings.

Key-Words: Angular contact, Ball Bearing, Frictional Torque, Condition Monitoring, Race, Ring, Measurement

1 Introduction

In recent years the behaviors of rolling element of ball bearings has become the subject of many researches, specially in high technologies. The bearing faults are the most series is causes of motor failures. Therefore bearing fault detection is the focus of this study. H. P. Poritsky et al (1947) has analyzed pivot type bearings. These researches were the same as angular contact bearings through the ratio of transverse race radius to ball radius is usually considerably higher in the pivot bearings [1].

K. L. Johnson (1958) studied the spinning of a rolling ball on a plane and demonstrated that partial slip occurs in the contact. He included hysteresis losses due to surface deformation and showed how both spin moments and hysteresis loss change with spin per roll ratio. Corrected spin moments were in good agreement with experiments [2].

G. S. Reichenbach (1960) introduced the angular contact ball bearings spinning action of the ball is the major component of rolling resistance and therefore of torque. To bring the calculated torques into line with experiment, Reichenbach had to suppose that the

effective sliding friction coefficient in the spinning per rolling ball contact varies with the spin to ball ratio [3].

J. Halling (1966) developed an analytical model for elemental strip theory to present solutions for the micro slip with spin patterns in ball thrust bearings [4].

Subsequently, J. Halling (1967) analyzed the spin and roll conditions in angular contact ball bearings and showed that the partial slip, taking place in spinning and rolling contacts, reduces the energy dissipation or torque compared with that of full slip. Halling neglected transverse slip in his treatment and suggested that thereby the true spin torque would be underestimated [5].

J. J. Kalker (1968) analyzed the ball motion in angular contact ball bearings and presented an iterative solution to the problem of non identical contacts. In this analysis transverse slips were included, but there was no comparison with experimental work [6].

B. Snare (1968) developed an analytical method for calculating rolling resistance in ball bearings with both axial and radial loads [7].

From comparison with measured torques he deduced effective friction coefficients, which were then used to calculate torque in a wide range of ball bearings. Unfortunately Snare measured torque at several hundred rpm and assumed, questionably, that hydrodynamic effects were independent of applied load. There appears to be little published since 1971 when Harris (1971)[8] considered ball motion in angular contact ball bearings with coulomb friction, and in particular, observed experimentally regimes of outer and inner raceway control.

M. J. Todd and K. L. Johnson (1986) developed a model for coulomb torque hysteresis in ball bearings. The frictional torque resisting rotation of an angular contact ball bearing builds up from rest to its steady value through a small but finite angle of rotation. A similar transient torque is observed when the direction of rotation is reversed so that, in a bearing which undergoes oscillating rotations of small amplitude, the resistance torque traces out a hysteresis loop with angle of rotation. The shape of the loop is expressed in terms of two parameters : (i) the initial and reversal gradient s and (ii) the steady-state torque T_s . They derived theoretical expressions for both s and T_s . s depends upon the elasticity of the bearing materials but not upon the frictional conditions at the contact interface whereas the opposite applies to T_s . Both depend upon the geometry of the bearing, particularly the conformity between the ball and race [9].

S. F. Masri, R. K. Miller, M. I. Traina and T. K. Caughey (1991)[10] developed some bearing friction models from experimental measurements for a better understanding of the underlying physics involved in friction phenomena. A carefully controlled experimental study was presented in which the frictional forces and dynamic response of a shaft oscillating within a pair of sleeve bearings were monitored and analyzed in order to gain further insight into the basic phenomenological features of bearing friction forces. Through a qualitative review of the data, it was shown that the trajectory of the bearing force versus slip velocity exhibits an hysteretic-type loop superimposed on Coulomb and viscous actions. Parametric identification techniques are used to develop a simplified mathematical model incorporating an idealized Coulomb friction more quantitative and less obvious characteristics of the measured frictional behavior, has been presented by X. Hernot, M. Sartor and J. Guillot (2000), [11]. They

approached analytical calculation of the stiffness matrix of angular contact ball bearing.

M. O. T. Cole, P. S. Keogh, C. R. Burrowes (2002) developed the dynamic behavior of a rolling element auxiliary bearing following rotor impact [12].

2 Formulation of Static Model

2.1 PRELOAD AND AXIAL PLAY

In an application where accurate rotation is needed, the radial play must be removed from the bearing. The radial play is normally removed by using a pair of bearings which are preloaded to remove the play by pressing the races together axially until the balls are in firm contact with the raceways, shown in figure 1.

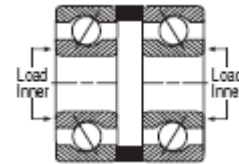


Figure 1. Type of preload

Once the preload has been applied, the balls will sit at the ball contact angle. This angle will increase as more preload is applied, and the starting value of this angle (the initial contact angle) will be larger for larger radial play values, shown in figure 2 [8].

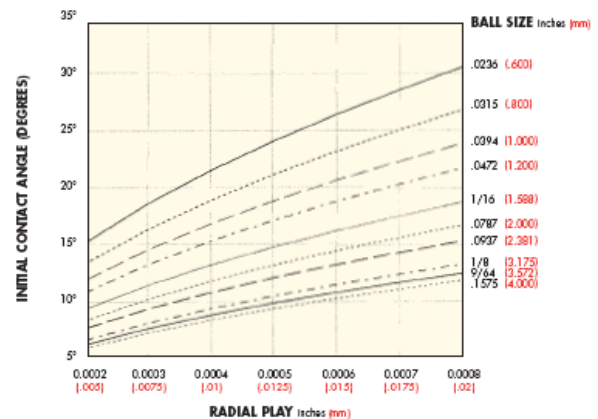


Figure 2. Initial contact angles vs. radial play

A typical ball bearing geometry and contact angle after applying load is shown in figure 3.

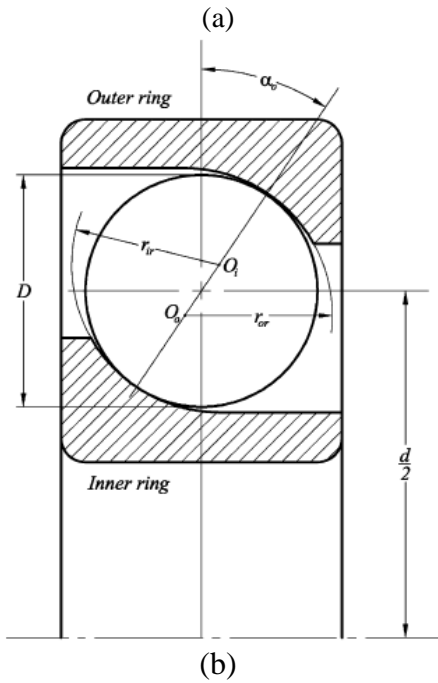
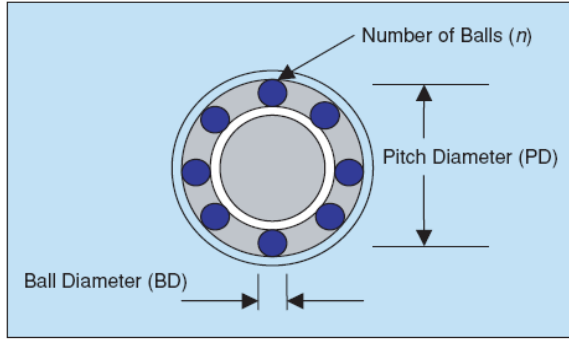


Figure 3.(a) Geometry of a typical ball bearing, (b) Contact angle after applying load.

2.2 DISPLACEMENT DUE TO AXIAL PRELOAD

A calculation can be made to determine the displacement vs. preload curve for a given bearing, shown in figure 4 [14].

$$stiffness = \Delta F_a / \Delta \delta \quad (1)$$

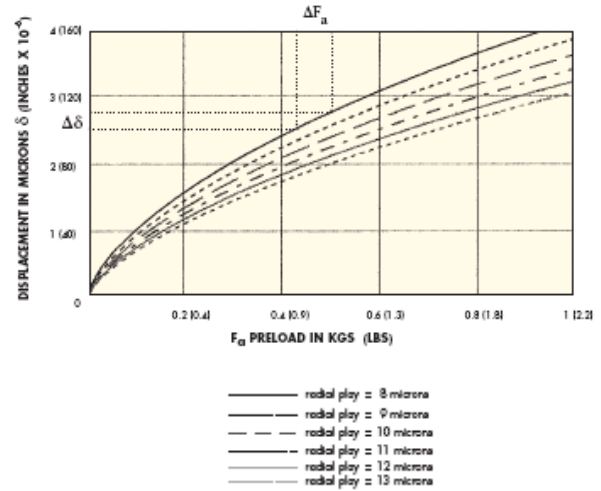


Figure 4. Stiffness of assembly is given by inverse slope of this curve.

2.3 Mathematical formulation

The bearings are identical and have a back-to-back configuration. The cage of ball bearings assumed to be constant.

Figure 5 shows the geometry of the i th ball of an angular contact ball bearing which is in contact with the inner and outer races. The local Hertzian contact force F_i and deflection δ_i relationship between ball i and inner and outer races of angular contact ball bearing may be written as follows [8]:

$$F_i = K \delta_i^{3/2} \quad (2)$$

Where K is the total stiffness coefficient per ball. The total deflection δ_i of the i th ball-race contact is shown in figure 3b. In this figure the center of curvature of the outer race O_o is assumed to be the fixed origin. It is assumed that the initial preload F_r will cause an axial displacement from O_i to $(O_i)_1$ for the free center of curvature of the inner race. Furthermore, the external radial loading due to the radial process (F_x and F_y) will move $(O_i)_1$ to a final position $(O_i)_2$. The preload contact angle α_p and initial contact deformation δ_o due to the elasticity of the bearings for a given preload F_r can be determined by solving the following equations simultaneously using the Newton-Raphson defined as [8]

$$n(K\delta_o^{3/2})\sin\alpha_p = F_r \quad (3)$$

$$\delta_o = A \left(\frac{\cos\alpha_0}{\cos\alpha_p} - 1 \right) \quad (4)$$

Where A is the initial distance between the centers of curvature of the inner and outer races, α_0 the unloaded contact angle due to interference fitting of bearings ΔC_d calculated from the following equation:

$$\alpha_0 = \cos^{-1} \left(1 - \frac{C_d - \Delta C_d}{2A} \right) \quad (5)$$

Where

$$C_d = d - d_i - 2D \quad (6)$$

The applied axial preload will cause initial displacement Z_0 in the axial direction, as shown in fig. 5.

$$Z_0 = \frac{\delta_0 \sin(\alpha_p - \alpha_0)}{\cos\alpha_p} \quad (7)$$

The initial contact load per ball F_0 due to initial axial preloading F_r can be obtained from

$$F_0 = K\delta_0^{3/2} \quad (8)$$

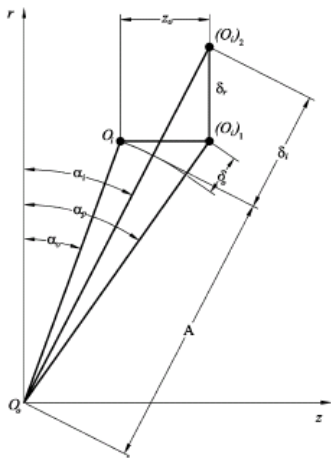


Fig. 5. deflection of the i th ball-race contact

3 Formulation of Dynamic Model

3.1 Mathematical Formulation

3.1.1 Micro slip Torque

The micro slip torque at the outer race in ball bearing given by

$$T_m = \frac{F_m}{2} \cdot R_i \quad (9)$$

Where T_m is micro slip torque at the outer race and F_m is the force acting at the ball center which is required to cause rolling [4],[5].

For micro slip torque at the inner race is

$$T_m = \frac{F_m}{2} \cdot (R_i + d) \quad (10)$$

3.1.2 Hysteresis Torque

The associated bearing torque from the hysteresis per loaded ball along is:

$$T_h = \frac{k}{2} \frac{F_o}{d} \alpha [b_i(R_i + d) + b_o R_i] \quad (11)$$

Where i and o refer to inner and outer race respectively, and R_i is the distance of the inner race contact from the bearing axis. The related bearing dimension as:

$$R_i = \frac{1}{2} (PD + D \cos \alpha) \quad (12)$$

Where PD is the pitch circle diameter, D is the ball diameter and α is the contact angle.

3.1.3 Spin Torque

The torque caused by spin per loaded ball is [5]:

$$T_s = \frac{3}{8} \mu F_o a E_2 \sin \alpha \quad (13)$$

E_2 is the complete elliptic integral of the second kind and may be found from tables or by numerical integration.!

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4 TEST RIG

A general view of the test rig containing a test bed and a motion control servo amplifiers to drive electro

craft dc motor at different switching frequencies is shown in figure 6. The electronic devices are fully protected against over-voltage, over-current, over-heating and short - circuits. The set will interface with digital controllers or can be used as stand - alone drives. It requires only a single unregulated DC power supply. Housing assembly shaft is feted on test bed plate. A belt connects motor to the housing assembly shaft. The Velocity Encoder monitors the speed of the shaft at any time. LVDT is installed in mounting blocks, and connects rod between the LVDT core and housing assembly whose motion is being measured.

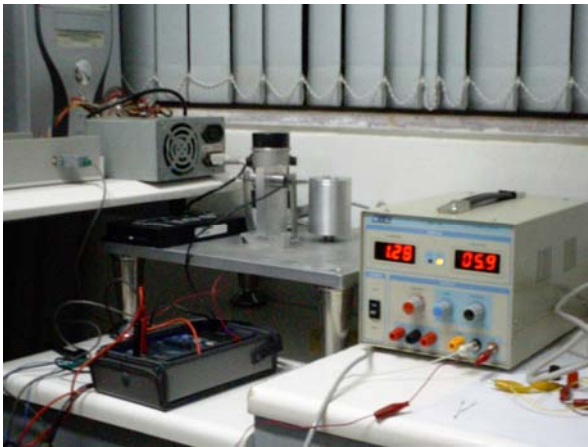


Figure 6. Test rig set up

4.1 Test Rig Assembly

Before to assemble housing unit to the test bed, with applicable torque watch range, shown in figure 7, we checked starting torque to make sure that the set up are with over specifications.



Fig. 7. Torque watch.

In the experiments presented in this paper, only axial loading was used. The inner ring was rotated at applicable speed and the frictional torque transmitted across the bearing to the outer ring was sensed by an LVDT. The schematic diagram of the set is shown in figure 8. The transducer output was amplified and fed to a host computer.

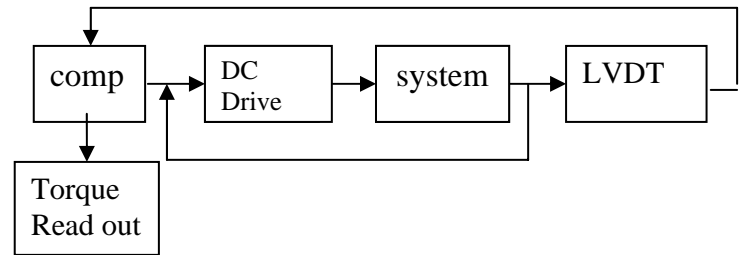


Fig. 8. Schematic diagram of measurement system configuration for ball bearing damage detection

All experiments were conducted in control laboratory. The test sampling bearings were degreased and lubricated with applicable amount, and were applied equally and gently in specific area.

Some work was done on full ball –complement, caged bearings. Most measurements were made with 6202 series, with the following specifications:

Outer ring diameter	35 mm
Inner ring diameter	12 mm
Ball diameter	6 mm
Contact angle	15°

Several tests were carried out with different loads with fixed speed or with fixed load with different speeds. Contact angle were determined by the usual method of counting the number of ring revolutions compared with the cage revolutions.

The measured frictional torque at speed of 2500 rpm is plotted in figure 9.

5 Condition Monitoring

Condition monitoring is an important issue in many fields, including commercial or space applications. Condition monitoring can be defined as a technique or process of monitoring the operating characteristics of a machine so that changes and trends of the monitored signal can be used to predict the need for maintenance before a breakdown or serious deterioration occurs, or to estimate the current condition of the machine.

Condition monitoring has become increasingly important. For example, any static overload or severe

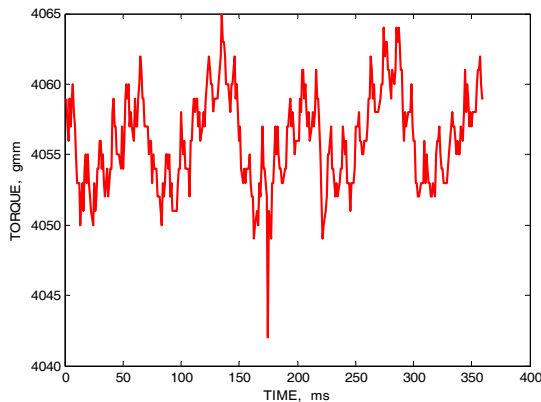


Figure 9. Frictional sampling torque at 2500 rpm.

impact can cause brinelling on the inner ring, outer ring, or balls of the operating machine. An unexpected fault or shutdown can result in a serious accident and financial loss for the company. Companies must find ways to avoid failures, minimize downtime, reduce maintenance costs, and lengthen the lifetime of their equipment. Machines can be utilized in more optimal fashion by using reliable condition monitoring [13].

6 Conclusion

Ball bearing is a widely studied subject in condition monitoring [13], [14]. There are several different methods for recognizing failures. The most widely studied methods in ball bearing condition monitoring are based on measurements of vibration, acoustic noise, or temperature, where vibration and stator current based methods seems to be the most popular methods. During the monitoring ball bearing damage in induction motor, the characteristic frequencies of ball bearing damage are often used to monitor certain frequency components in either vibration or stator current signals.

In this paper, a tool for condition diagnosis was proposed. The tool aims to be generic and not tied to a specific application; it can automatically find frictional measured torque that discriminate between any signal sets. The most advantage of this tool is the verifying behavior and physical causes of mechanical failures.

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