A New Test Rig For Frictional Torque Measurement In Ball Bearings

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Abstract: - This paper presents experimental results to study the effects of axial preloading of angular contact ball bearings on the frictional torque behavior of rotating systems. Selecting the right bearing or decision to utilize a special bearing, represents an effort to deal with performance requirements and operating limitations. Among the most important application considerations that must be evaluated are speed and load conditions. The present investigations are intended to provide a clear understanding of the role of the bearing in the application with respect to its design, its quality and the way in which it is mounted in the housings. The objective of this study is to provide the user with tools that enable them to efficiently predict the influence of ball bearings on the dynamic behavior of the application. With the help of these tools user can find answers to several design and manufacturing related questions.

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

1 Introduction

Ball bearing can be important generators of the most series causes of motor failures. Due to the rotation of the lubricated contacts, the stiffness in the bearing is time dependent and generates parametric excitations. 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 or 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 elasiticity 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.


1. TEST RIG

A general view of the test rig is shown in figure 1. In the test bed, an advanced motion control servo amplifier is used to drive electro craft dc motor at different switching frequencies. They are fully protected against over-voltage, over-current, overheating and short-circuits. All models interface with digital controllers or can be used as stan-alone drives. They require only a single DC power supply. Housing assembly shaft is feted on test bed plate. A single belt connects the DC motor to housing assembly shaft. LVDT is installed in mounting blocks, and its core is connected to the housing assembly by two rods.

![Figure 1. Test bed set up](image)

### NOMENCLATURE

| LVDT | Linear variable differential transducer |
| K    | Total stiffness coefficient per ball    |
| n    | Number of balls                        |
| D    | Ball diameter                           |
| d    | Pitch circle diameter                   |
| $\alpha$ | Contact angle                         |
| $F_o$ | Normal ball load                       |
| $\mu$ | Sliding friction coefficient            |
$T_m$  Torque of bearing due to micro slip

$T_h$  Torque of bearing due to hysteresis

$T_s$  Torque of bearing due to spin

$E_2$  Complete elliptic integral (2nd kind)

$a, b$  Hertiz ellipse semi-major, semi-minor

### 2. EXPERIMENTAL METOD

Frictional torque measurements are carried out as follows.

### 3. SPRING PRELOADING

Preloading is the removal of internal clearance by placing a permanent thrust load on the bearings. The elimination of radial and end play, by preloading increases rigidity of system, reduction of non-repetitive run out, limitation of change in contact angle between inner and outer ring at very high speed and prevention of ball skidding under very high acceleration.

Bearings should not be preloaded more than it is necessary to obtain the desired rigidity. Excessive preloads cause undesired heat which reduces speed capability and bearing life. To achieve an appropriate method for preloading, in this experiment we have used wave washers. This is the simplest way to preload bearings. It should be considered first, and used if all application requirements can be met. Most assemblies which do not require specific yield characteristics can be successfully spring preloaded.

Spring preloading offers several advantages. With properly selected springs, it can be more constant than other systems. It is generally less sensitive to differential expansion, and offers more accommodation to minor misalignment. It also allows greater speeds than in rigidly preloaded systems.

Angular contact bearings are loaded by spring wave washers is shown in figure 2.

### 4. TEST BED ASSEMBLY

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 bearings to the outer ring was sensed by an LVDT, the transducer output was amplified and fed to a host computer. A mechanical sketch of housing assembly is shown in figure 2.

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:

- Outer ring diameter: 35 mm
- Inner ring diameter: 12 mm
- Ball diameter: 6 mm
- Contact angle: $15^\circ$

![Fig.2. Mechanical sketch of housing assembly.](image)
The typical ball bearing geometry and contact angle after applying the loads, is shown in figure 7.
5. **Mathematical formulation**

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

Figure 8 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 \( \delta_i \) relationship between ball \( i \) and inner and outer races of angular contact ball bearing may be written as follows [8]:

\[
F_i = K \delta_i^{3/2}
\]

Where \( K \) is the total stiffness coefficient per ball.

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

\[
\delta_o = A \left( \frac{\cos \alpha_o}{\cos \alpha_p} - 1 \right)
\]

Where \( A \) is the initial distance between the centers of curvature of the inner and outer races, \( \alpha_o \), the unloaded contact angle due to interference fitting of bearings \( \Delta C_d \) calculated from the following equation:

\[
\alpha_0 = \cos^{-1} \left( 1 - \frac{C_d - \Delta C_d}{2A} \right)
\]

Where

\[
C_d = d - d_i - 2D
\]

\[
Z_0 = \frac{\delta_o \sin(\alpha_p - \alpha_o)}{\cos \alpha_p}
\]

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

\[
F_0 = K \delta_0^{3/2}
\]

### 5.1 Micro slip Torque

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

\[
T_m = \frac{F_m}{2} \cdot R_i
\]

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

\[
T_m = \frac{F_m}{2}\cdot(R_i + d)
\]

### 5.2 Hysteresis Torque

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

\[
T_h = k \frac{F_o}{2d} \alpha \left[ b_i(R_i + d) + b_o R_o \right]
\]

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:

\[
R_i = \frac{1}{2}(PD + D \cos \alpha)
\]

Where PD is the pitch circle diameter, \( D \) is the ball diameter and \( \alpha \) is the contact angle.

### 5.3 Spin Torque

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

\[
\alpha_0 = \cos^{-1} \left( 1 - \frac{C_d - \Delta C_d}{2A} \right)
\]
\[ T_s = \frac{3}{8} \mu F_{0} a E_2 \sin \alpha \]  

(13)

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

6. RESULTS AND DISCUSSIONS

The results of comparative tests show that, with a constant applied load, the peaks and valleys depend on the speed, as shown in figure 10. In this figure the output voltage of LVDT indicating the output torque of ball bearings is sketched vs. the time. As it is shown, the peaks and valleys change depending the speed.

![Time Domain Voltage and FFT results](image)

Figure 10. The results of the comparatives tests.

7. CONCLUSIONS

This study provided an experimental procedure for determining the frictional torque in high speed ball bearings. A test rig and new testing device were realized.

The bearings are tested under a controlled load and speed to meet their particular specification. This device allows the user to know how the bearings will perform under dynamic conditions.

Angular contact ball bearings from 6202 series were tested in light lubricant and pure axial load conditions. The comparative tests demonstrate that the friction torque tester device ensure to user the maximum performance of ball bearing by physically diagnosing problems in bearing prior to use. Problems such as retainer hang-up, ball or race surface problem, contamination, internal geometry and structural defects can be determined on a bearing through sampling.

References: