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Heat generation modeling of ball bearing based on internal load distribution

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ABSTRACT

In the operation of a machine tool (MT), the frictions in ball bearings entail sudden and violent heating of the balls which dominants its thermal deformation, and subsequently results in degradation of its accuracy and performance. Modeling of the heat generation in a bearing is a quite difficult job because of the constantly changing characteristics. In this paper, an analytical approach was proposed to calculate the heat generation rate of supporting bearing in a ball–screw system of the MT, with consideration of the operating conditions, such as rotation speed and external loads of the machine tool. The influences of operating conditions to internal load distribution, contact angles and heat generation rate of ball bearings were analyzed. The friction torque due to the applied load and the sliding torque within the contact area were discussed in detail. Experiments were carried out in a high-speed ball–screw system to verify the validity of the presented analytical method. The work described in this paper can be seen as a foundation for the accuracy thermal modeling and thermal dynamic analysis of the ball–screw system in the machine tools.

1. Introduction

The demands of high-precision machine tools and three coordinate measuring machines are rapidly increasing in response to the development of precision machining technology which requires high-precision and high productivity. Research on high-speed precision MT can be approached on the main spindle and the ball–screw system. A high-speed precision ball–screw system reduces non-cutting operating time and tool replacement time, which makes machining processes more economical. As main parts of high-speed precision ball–screw system, there are many joints existing in the supporting ball bearings, such as the interfaces between the bearing and the shaft, the bearing and the bearing support and so on. When two surfaces are in contact, the presence of contact surfaces produces contact friction at the joint, no matter how much the pressure between the surfaces is. The friction in ball bearings entails a sudden and violent heating of the balls that can have very detrimental effects. The increase of temperature generated by these phenomena can involve mechanical micro-deformations and an overheating of the cooling fluid (especially when dealing with cryogenic fluids). Such a temperature heating of ball bearing plays a significant role in the thermal characteristics of the ball–screw system, causing serious thermal deformation that subsequently degrades the accuracy of MT or other mechatronics instruments where the precision ball bearing are used.

Ever since the early 1960s, the importance of thermal effects on the accuracy of machine tools has been increasingly felt [1]. It has been reported that thermal errors account for 40–70% of the total errors arising from various error sources. Significant research has since taken place in the study of machine tool thermal behavior. Researchers have considered a number of ways of reducing thermal error, including the thermally symmetric design of a structure, separation of the heat sources from the main body of the machine tool, cooling of the structure, compensation for the thermal error and so forth [2–10]. From the researches thus far, it can be seen that most of the work carried out is based on the principle of directly mapping the thermal error against the temperature of critical machine elements irrespective of the operating conditions [3–6]. The different operating parameters used were just a means to generate varying thermal states on the machine. But recently, Ramesh et al. [8], justified the point that the thermal error of a machine tool was strongly dependent upon the specific operating parameters and conditions that the machine was put through in order to generate a specific thermal condition. In that case, the theoretical accurate modeling of thermal error becomes indispensable.

In machine tool operations the major heat sources include the heat generated by the cutting process and the heat from bearings. It is assumed that the majority of cutting heat is taken away by coolant. Therefore, the heat generated by bearings is the dominant heat causing thermal deformations. The calculation of temperature in the ball bearing under friction has been of great scientific interest

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Nomenclature

D	diameter of the ball
r_b	radius of the ball
r'	radius of the raceway groove curvature
f	r'/D
B	f_i+f_o-1
A_1	axial distance between the position of inner and outer raceway groove curvature centers
A_2	radial distance between the position of inner and outer raceway groove curvature centers
α^0	free contact angle
δ	displacement of the bearing
α	contact angle of ball– raceway contact
δ	normal deformation of the raceway groove curvature
F_c	centrifugal force
M_g	gyroscopic moment
F	external loads
B	ball attitude angle
w/n	rotating speed
M	total frictional torque of the bearing
H_f	heat generation rate
Q	ball load

Subscripts

A	refer to axial direction
r	refer to radial direction
i	refer to inner raceway
o	refer to outer raceway
m	refer to orbital motion
R	refer to rolling element

over the past few decades. Since the pioneering works concerning semi-infinite solids which are adiabatic outside the region heated by a moving heat source. The case of an elliptic moving heat source on an insulated semi-infinite body has been studied by Bejan [11], Tian and Kennedy [12] with asymptotic solutions. The analytical solutions are often difficult to get because of: (i) the non-homogeneous boundary conditions, (ii) the relative motion and (iii) the small size of the contact region in relation to other dimensions of solid. From available studies [13–15], it is clearly shown that accurate thermal dynamic modeling of the heat source is quite difficult because of the constantly changing characteristics during the operation processes: the thermal inertia, the operating conditions and the complexity of the machine tool structure, etc. Because the determination of the temperatures in a bearing requires a good knowledge of the thermal contact parameters, the researches so far reported have therefore been unable to provide satisfactory, convinced and precision results on the accurate modeling of the heat source. In the course of this research, the finite element method (FEM) and the finite difference method (FDM) have often been employed to estimate the thermal behavior of machine tools. However, there are also some significant differences between estimated and experimental results, which are attributed to the facts that it is difficult to establish the boundary conditions because of the complex shape of the structure and the varying heat generation rate [13–15].On the other hand, many authors have already numerically solved ball bearings problems, such as loads [16,17], stiffness matrix [18–20] and angles [21], using analytical methods. However, the computational methods used in these works are unhelpful for determination of heat generation rate of ball bearings.

In this study, we present an analytical method to calculate the heat generation rate of supporting bearing in a kind of ball–screw system which is used widely in machine tools and mechatronics instruments. The operating conditions which were not regarded by other researchers before, such as rotation speed and external loads, are taken into account for calculating heat generated by the bearings. The determination of bearing element torques is mainly focused on both of the friction torque due to applied load and the sliding torque at the contact area. In Section 2, the distribution of internal loading in a ball bearing is analyzed. Section 3 discusses calculation of the friction torque and sliding torque in ball bearing. And more, some experiments and testbed used for these test are analyzed in Section 4 and the difference between theoretic analytic method and experiment are discussed. Finally in Section 5, we conclude the results of this study and give the prospect research in the future.

2. Distribution of internal loading in a ball bearing

2.1. Deflection of ball bearing under applied load

The ball baring can be illustrated in its simple form as shown in Fig. 1. The ability of a ball bearing to carry load depends in large measure on the osculation of the rolling elements and raceway, which is the rate of radius of rolling element (r_b) to that of the raceway in a direction transverse to the direction of rolling (r_i', r_o'). Also from Fig. 1, it can be seen that the free contact angle is made by the line passing through the points of contact of the ball and both raceways and a plane perpendicular to the bearing axis of rotation. The free contact angle α^0 can be described as follows:

$$\alpha^0 = \cos^{-1}\left(1-\frac{r_o-r_i-r_b}{BD}\right) \tag{1}$$

In a ball bearing, depending on the contact angles, ball gyroscopic moments and ball centrifugal forces can be of significant magnitude such that inner raceway contact angles tend to increase and out raceway contact angles tend to decrease. Under zero load the centers of the raceway groove curvature radius are separated by a distance BD defined by (as shown in Fig. 1 a)

$$BD = (r_i'+r_o'-2r_b) = (f_i+f_o-1)D \tag{2}$$

where r_b is the radius of ball, while D is the diameter of the ball. f is defined by $f=r'/D$.

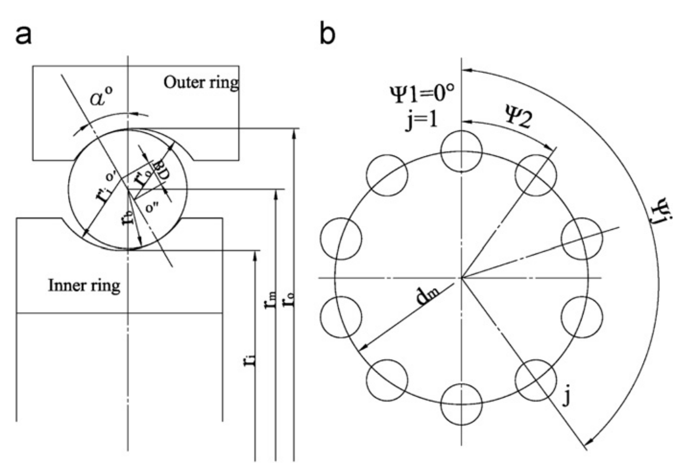

Fig. 1. Schematic of ball bearing. (a) Bearing parameters. (b) Angular position of rolling element.