Precision Engineering 42 (2015) 151–164

Contents lists available at ScienceDirect

Precision Engineering

journal homepage: www.elsevier.com/locate/precision

Thermal characteristics of a CNC feed system under varying operating conditions

Chao Jin[a,b], Bo Wu[a], Youmin Hu[a], Pengxing Yi[a,*], Yao Cheng[a]

[a] State Key Lab of Digital Manufacturing Equipment and Technology, School of Mechanical Science and Engineering, Huazhong University of Science and Technology, Wuhan 430074, Hubei, PR China
[b] Research Institute of Zhejiang University—Taizhou, Taizhou 318000, Zhejiang, PR China

ARTICLE INFO

Article history:
Received 28 February 2012
Received in revised form 11 July 2014
Accepted 20 April 2015
Available online 25 April 2015

Keywords:
Thermal error
Wavelet neural network
Operating condition
Feed system

ABSTRACT

In high-speed and high-precision feed systems, thermal positioning errors are mainly caused by the non-uniform temperature variations and resulting time-varying thermal deformations under different operating conditions. The research presented here ultimately aims to develop a generic method capable of evaluating the thermal characteristics (such as temperature rise of heat sources, thermal positioning error) of the feed system induced by varying operating conditions (feed speed, cutting load and preload of ball screw). The thermal contact resistance between the balls and the inner and outer rings of supporting bearing is calculated using the Hertzian theory and JHM method. Experiments were carried out on a high-speed feed system experimental bench, and the influences of operating conditions on temperature rises of supporting bearings and ball screw nut were analyzed. Based on a WNN-NARMAL2 model, the relationship between temperature rise of supporting bearings and operating conditions was established. Furthermore, with the temperature of the ball screw nut set to be a moving heat source load, the temperature and thermal deformation distributions of the ball screw were simulated. The work described lays a solid foundation for thermal error prediction and compensation of a feed system under varying operating conditions.

1. Introduction

Positioning systems with high speed, high accuracy and long stroke become more important in precision machining. A high-speed precision feed system reduces non-cutting operating time and tool replacement time, making production more economical. However, due to the friction at the ball screw bearing and nut, a high-speed feed system generates a lot of heat, causing thermal expansion which adversely affects machining accuracy. Therefore, the thermal positioning error of a ball screw is one of the most important objects to consider for high-speed and high-precision machine tools.

Researchers have considered many ways of reducing thermal errors, including the thermally symmetric design of a structure, separation of the heat sources from the main body of a machine tool, active cooling [1]. However, the costs associated with these approaches are usually very high. In addition, there are many physical limitations, which cannot be overcome solely by design techniques. As a result, error compensation techniques to improve machine accuracy cost-effectively have received significant attention [1–3]. Some machine tool manufacturers have implemented software in their open-architecture CNC controller to compensate the thermal error in real time [4]. Position feedback systems that do not rely on the ball screw, such as scales, represent a well-established and successful approach to reducing thermal positioning errors of feed system. However, the thermal errors of a ball screw system are caused by the non-uniform temperature variations and time-varying thermal deformations in the machine structure. Therefore, accurate modeling of thermal errors remains a key challenge of error compensation.

Most current research is focused on the thermal error compensation of the whole machine tools. Thermally induced error is a time-dependent nonlinear process caused by nonuniform temperature variation in the machine structure. The interaction between the heat source location, its intensity, thermal expansion coefficient and machine system configuration creates complex thermal

* Corresponding author. Tel.: +86 27 87557415; fax: +86 27 87557415.
E-mail addresses: pxyi@mail.hust.edu.cn, pxyi@hust.edu.cn (P. Yi).

http://dx.doi.org/10.1016/j.precisioneng.2015.04.010
0141-6359/

behavior. Researchers have employed various techniques, namely, finite element methods, coordinate transformation methods, neural networks, etc, in modeling the thermal characteristics [5–9,3]. Venugopal et al. [10] carried out different types of experiments under no-load conditions. The thermal error was predicted based on the temperature of the lead screw nut by using a basic linear expansion model. Veldhuis et al. [11] conducted five different tests over a period of 10 h. The thermal error generated as a result of these tests was mapped against the temperatures measured by 17 thermocouples mounted on the machine by using a neural network model. A finite element approach to simulate the thermal behavior of the ball screw transmission was presented in [12]. When determining thermal errors arising in ball screws, it is very important to accurately identify the temperature distribution along the screw and on this basis determine its axial thermal elongation. Heisel et al. [13] used an infrared camera to measure temperatures along the screw. An experimentally determined temperature distribution and measured positioning errors for 4000 cycles were obtained.

In a high-speed feed system, bearings are considered to be the main heat sources, and the thermal properties of the bearings need to be carefully studied. For a bearing, the thermal resistances for conduction through the bearing elements themselves and for radiation can be calculated using the dimensions, the thermal conductivities, the thermal-optical properties, and the temperatures of the elements. However, it ca be said that the thermal contact resistances between the balls and the rings, which are most closely related to the temperature differences across the bearings, are difficult to predict because few useful calculation method have been proposed yet. From the above, it can be seen that most of the work carried out thus far is based on the principle of directly mapping the thermal error against the temperature of critical machine elements irrespective of the operating conditions. The different operating parameters used were just a means to generate varying thermal states on the machine. But researchers [1,14] discovered 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. Accordingly, research on changing rule and dynamics characteristics of heat sources, temperature field, and structure thermal deformation must be taken into account to reduce the error of thermal deformation and improve the working accuracy of a machine tool.

The research presented here ultimately aims at the development of a comprehensive model that can predict the temperature rises of heat sources and thermal characteristics in a ball screw CNC feed system under different operating conditions (feed speed, load and preload of the ball screw). The thermal contact resistance between the balls and the inner and outer rings of supporting bearing is calculated in Section 2. The experimental setup is described in Section 3. Based on an orthogonal experimental design, the relationship between temperature rise of bearings and operating conditions is established based on a WNN-NARMAL2 neural network in Section 4. Furthermore, with the temperature of the ball screw nut as a moving heat source load, the temperature and thermal deformation distributions of the ball screw under bearings and ball screw nut heat sources were simulated in Section 5.

2. Expressions for thermal contact resistance

2.1. Contact resistance

The contact resistances between the balls and the inner and outer rings may be treated in the same manner as constriction resistance since both resistances result from the restriction of the heat flow due to small contact arrears. In the ellipsoidal coordinate system the Laplace's equation is:

$$\nabla^2 T = \frac{\partial}{\partial u}\left(\sqrt{f(u)}\frac{\partial T}{\partial u}\right) \tag{1}$$

where

$$\sqrt{f(u)} = \sqrt{(a^2+u)(b^2+u)u} \tag{2}$$

And α, b are the semi-major and semi-minor axes of the elliptic contact area, respectively; while μ is the variable along an axis normal to the contact plane.

The boundary conditions are:

$$u = 0, \quad T = T_0 \tag{3}$$

$$u \to \infty, \quad T = 0 \tag{4}$$

With Eqs. (1), (3) and (4), the temperature distribution can be obtained: [6]

$$T = \frac{Q}{4\pi k}\int_u^\infty \frac{du}{\sqrt{f(u)}} \tag{5}$$

where Q is all the heat leaving the elliptic contact area. And by the definition of the thermal contact resistance

$$R = \frac{T_0 - T_{u\to\infty}}{Q} = \frac{1}{4\pi k}\int_0^\infty \frac{du}{\sqrt{f(u)}} = \frac{1}{4\pi k}\int_0^\infty \frac{du}{\sqrt{(a^2+u)(b^2+u)u}} \tag{6}$$

Using the complete elliptic integral of the first kind, Eq. (6) can be written in the following form as

$$R = \frac{\Psi(a/b)}{4ka}, \quad \Psi(a/b) = \frac{2}{\pi}F\left(e, \frac{\pi}{2}\right) \tag{7}$$

Then, the contact thermal resistance between the ball and the inner or outer ring can be determined by using Eq. (7). For most bearing, whose ball and both rings are made from the same material, we can write the contact thermal resistance per ball as:

$$R = \frac{1}{2k}\left[\frac{\Psi(a_i/b_i)}{a_i} + \frac{\Psi(a_o/b_o)}{a_o}\right] \tag{8}$$

These expressions permit us to predict the total contact resistance resulting from the contact of an arbitrary number of balls with both the inner and outer rings by connecting the thermal resistances in parallel.

2.2. Contact areas in a ball bearing

When two elastic bodies having smooth round surface are press against each other, the contact area becomes elliptic. Assumed that the angle between the two planes containing the principal radii of curvature of the bodies are perpendicular as in the case of balls contacting the inner or outer ring of a bearing, the following expressions can be derived:

$$\begin{aligned} a &= a^*\left[\frac{3}{4}\frac{P}{A+B}\left(\frac{1-\nu_1^2}{E_1}+\frac{1-\nu_2^2}{E_2}\right)\right]^{1/3} \\ b &= b^*\left[\frac{3}{4}\frac{P}{A+B}\left(\frac{1-\nu_1^2}{E_1}+\frac{1-\nu_2^2}{E_2}\right)\right]^{1/3} \end{aligned} \tag{9}$$

$$A = \frac{1}{2}\left(\frac{1}{r_1}+\frac{1}{r_2}\right), \quad B = \frac{1}{2}\left(\frac{1}{r_1'}+\frac{1}{r_2'}\right) \tag{10}$$

In which r_1, r_1' are the radius of curvature for inner or outer race and groove, respectively. And r_2, r_2' are the radii of rolling ball. Considering the bearing model shown in Fig. 1, for the contact at inner ring side, the radius of curvature r_1' of the inner groove must be