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Investigation of variable optimum preload for a machine tool spindle

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ABSTRACT

Angular contact ball bearings have been widely used in machine tool spindles, and the bearing preload plays an important role on the performance of the spindle. With the development of high speed machining, especially for high speed milling, heavy cutting at a low speed and light cutting at a high speed are often performed on a single machine tool spindle, thus, high stiffness at low speed and low temperature rise at high speed are required. The traditional constant pressure preload method cannot meet the technical requirement of this kind of spindle any more. The variable preload technology is systematically investigated in this paper. At high speed range, FEM method is used to analyze the temperature distribution of the spindle, and the variable spindle preload is determined according to the constraint of temperature rise of bearings. At low speed range, the spindle preload is resolved by the fatigue life of bearings. The dynamic stiffness of the variable preload spindle is analyzed utilizing the Transfer Matrix Method (TMM) and a nonlinear bearing model that includes the centrifugal force and gyroscopic effects. An experimental set-up for the variable preload spindle is developed using hydraulic pressure that can automatically adjust the bearing preload. The proposed method to determine variable preload is verified experimentally by measuring the dynamic stiffness of the spindle and the temperature rise of the test bearing. The results show that the variable preload spindle gives outstanding behavior that the temperature rise at high speed is lower than that of the constant pressure preload spindle, and the dynamic stiffness at low speed range is significantly increased.

1. Introduction

Initial preload of rolling bearings is widely applied to obtain high stiffness of machine tool spindles, restrain vibration and enhance rotational accuracy [1–3]. Proper preload of the angular ball bearing is important to the speed, rigidity, accuracy and life of the spindle. The proposed bearing load mechanisms mainly include hydraulic instruments [3,4,7] and piezoelectric actuators [8]. The controller using hydraulic pressure is a most popular method for applying an automatic variable preload due to its simple structure and maintenance.

Currently, with the development of high speed machining, especially for high speed milling, heavy cutting at low speed and light cutting at high speed are often performed on a single machine tool spindle. For conventional spindle, the constant pressure preload method is preferred because it is simple and easy to maintain. However, the constant pressure preload is not favored for a high speed machining spindle.

Literature review shows that some researchers have investigated the effect of the preload on the dynamic characteristics of the spindle system [2–7], and found that high preload of the bearing can enhance the stiffness and the natural frequency of the spindle. Other works have focused on the effects of bearing preload on the thermal behaviors of the spindle [7–11], and revealed that when the spindle speed exceeds a certain value, there exists a variable bearing preload that produces minimum temperature rise [8,9].

Hagiu [12] has studied the optimum preload for a spindle supported by a pair of angle contact bearings (type of 7207) with a rotational speed of 20,000 rpm. In the paper, the effect of axial preload on the service life and dynamic stability was analyzed, and the following conclusions were given:

(1) There are certain preload values that ensure higher service life for the tested bearings and lower vibration levels of the test spindle. Moreover, for these preload values stable functioning thermal regimes for the test spindle were obtained.
(2) The natural frequencies of test spindle, also, the corresponding vibration levels increase with the increase of bearing preload.

It should be pointed out that the optimum preload of the spindle under a certain constant speed was discussed by Hagiu, and the aim was to lengthen the service life of the bearing and improve the dynamic behavior of the spindle at this certain speed.

In general, at low speed range, due to heavy cutting, a higher preload is required to obtain high spindle stiffness and reduce the vibration of spindle; at high speed range, due to heat generation of bearing, a lower preload is required to reduce the heat amount

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Nomenclature

A	cylindrical outside surface area of outer ring
a	semi-major axe
a_1	failure probability life adjustment factor
a_2	material life adjustment factor
a_3	operation condition life adjustment factor
b	semi- minor axe
C_r	basic rating dynamic radial load
C_0	basic rating static load
c_{air}	specific heat capacitance of air
d	diameter of the flow cross-section
d_m	bearing pitch diameter
F_a	axial load
F_r	radial load
H_f	heat generation of bearing
h_r	thickness of the outer ring
h_0	initial clearance fit
k_{air}	thermal conductivity of air
L_{10}	fatigue life that 90% of a group of bearings will endure
L_{10a}	adjusted basic rating life
M	amount of bearing friction torques
M_1	load torque
M_2	viscous friction torque
N	number of balls
n	spindle speed
P_r	equivalent dynamic radial load
P_0	equivalent static radial load
Q	ball load
R	thermal contact resistance
r_h	inner diameter of housing mating bearing
T_h	temperature of housing inner face
T_r	temperature of outer ring
T_0	initial temperature
X	radial load factor of P_0
Y	axial load factor of P_0
X_s	radial load factor of P_r
Y_s	axial load factor of P_r
π	thermal permissivity
α_h	linear thermal expansion coefficient of the housing
α_r	linear thermal expansion coefficient of the outer ring
η	convection coefficient
λ	thermal conductivity of the half space
λ_a	thermal conductivity of air
λ_r	thermal conductivity of outer ring
μ_{air}	dynamic viscosity of air
v	viscosity of lubricant
v_{air}	kinematic viscosity of air
v_0	kinematic viscosity of lubricant
v_1	standard viscosity of lubricant
ξ	complete elliptic integral of the second kind

Subscripts

air	refers to air
h	refers to housing
r	refers to bearing outer ring

and lengthen the use life, and meanwhile to maintain necessary stiffness for light cutting at a high speed. Therefore, for a high speed machining spindle, a variable preload is needed to enhance the performance of the entire speed range.

The main objective of this paper is to propose a method to determine the variable preload of high speed machine tool spindle at the entire speed range, so as to ensure that the spindle has outstanding thermal and dynamic performances. In order to obtain the optimum preload effectively, this study suggests that the entire range of speed should be divided into low and high speed sections; a lower preload is applied for the high speed section within the limit of the temperature rise by the machine tool industry standard, and a higher preload is applied for the low speed section within the limit of bearing fatigue life by the general requirement of the user.

Specifically, in high speed section, the bearing preload can be determined according to the constraint of temperature rise of bearings, and the temperature distribution of the spindle has been predicted by aid of FEM. In the low speed section, the preload can be resolved by the bearing fatigue life. The dynamic stiffness of the variable preload spindle has been studied by utilizing the TMM and a nonlinear bearing model. A variable bearing preload control system has been developed by using hydraulic transmission, which on-line supplies different preload levels by adjusting the hydraulic pressure. An experimental set-up for the variable preload spindle has been developed to verify the proposed method by measuring the temperature rise of the bearing and the dynamic stiffness of the spindle.

2. Variable preload spindle system

Fig. 1 shows a schematic of the overall concept of the variable preload spindle system developed in this study. The test spindle is driven by a motorized high speed spindle. The shaft of the spindle is supported by two pairs of angular contact ball bearings, which are preloaded by a hydraulic chamber. The hydraulic pressure induces displacement in the axial direction, then pushing the outer ring of the rear bearings, and eventually the displacement will be transferred to the inner ring of front bearings through the shaft of the spindle. Therefore, the displacement caused by hydraulic pressure is converted into an axial force, which increases the preload of bearings, and the preload of each bearing is equal to one half of the hydraulic pressure. The bearings are grease lubricated in this study, while the oil–air lubrication nozzle is devised in advance for further study. A disk with an unbalanced mass is mounted at the location of spindle nose to generate a self-exciting force when it rotates, which can be used to measure the dynamic stiffness of spindle.

The operating principle of the variable preload spindle system is as follows. The rotational speed of the spindle is detected through a revolution transmitter. According to the detected spindle speed, a control signal is sent to the D/A conversion. Subsequently, the current of proportional electromagnetic is adjusted to change the hydraulic pressure in the chamber. Finally, the variable preload is applied to the bearings for different spindle speeds. As shown in Fig. 1, the temperature rise of the front bearings is measured by thermocouple, the displacement of the spindle nose is detected by an eddy current sensor and a signal acquisition unit.

3. Analysis of bearing preload

3.1. Preload for high speed

The Finite Element Method has been successfully applied to analyze the temperature distribution of the integrated spindle