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Influence of thermo-mechanical coupled behaviors on milling stability of high speed motorized spindles

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ABSTRACT

This paper presents a comprehensive model for the motorized spindles in machine tools. The proposed model consists of a thermo-mechanical dynamic model and a milling stability model. The thermo-mechanical dynamic model is establishedbased on finite element method with the consideration of the transfer function at cutting point, which is coupled with the milling stability model. The influences of thermo-mechanical coupled factors (TMCF) on the transfer function at the cutting point and the milling stability of the system are discussed and validated, and the influences of the parameters of the material properties and machining process on the milling stability are acquired. The theoretical and experimental results are useful to the dynamic analysis and design for motorized spindle machine tools.

1. Introduction

A high speed motorized spindle device commonly consists of an actuator, a transmission and an executor. This unit can achieve "nearly zero transmission" and is widely used in modern numerical control (NC) machine tools [1,2]. However, self-excited vibration is one of the most critical problems in milling processes, which can affect dimensional accuracy, surface finish, tool life and machine reliability [3]. With the use of high speed milling, studies on milling stability of high speed spindles become essential.

Due to the integration of the motor and the tool, the thermo-mechanical coupling dynamic behaviors of the motorized spindle influence the milling stability of the system, while the built-in motor brings in the difficult to predict the characteristics of the system. Kim et al. [4] presented a thermo-mechanical coupled dynamic model to study the influences of the geometrical size, assembling error, preload and thermal displacement of bearing on the dynamics of the spindle system. Chen J S et al. constructed a temperature rise-thermal displacement model of motorized spindle, which reduces the bearing friction loss and system temperature by optimizing the bearing configuration [5]. Li H Q et al. proposed a thermo-mechanical dynamic model for the motorized spindle to expatiate the influence of thermal displacement on bearing preload, and validated the temperature rise distribution and the first two natural frequencies of the system [6,7].

A milling force model is also necessary to investigate its influence on milling stability. Engin and Altintas [8] proposed a linear cutting force between the dynamic cutting force and transient cutting thickness. On the base of it, Balachandran and Zhao [9] presented a nonlinear cutting force model with the contact deficient effect between the tool and workpiece. According to the structure characteristic of a milling tool, Kline and Devon [10] advanced a dynamic cutting force model of screwed cylinder milling tool with the consideration of the mass eccentricity. Machining chatter has been studied for more than 50 years and it is accepted by most researchers that regenerative chatter is the most significant source of machining chatter [11]. Regenerative chatter theory has been used successfully in chatter prediction for facemilling [12,13] and endmilling [14–16]. Tlusty presented a time domain simulation approach to study the effect of cutting speed lobing on milling stability, and found that the gains in stability were much smaller than previously maintained [17]. Schmitz and Smith also performed a time domain simulation to predict the forces and dispalcements during milling for selected operating parameters, and concluded with a description of the experimental determination of cutting force coefficients [18]. Rubeo and Schmitz described a metric referred to as the "amplitude ratio" for evaluating the stability of milling operations via time domain simulation, which generated contour diagrams that identify stability behavior over a range of spindle speeds and axial depths of cut [19]. Based on the milling force model [8] and Fourier series methods, Altintas and Budak [20] developed a high speed milling stability model which is not influenced by the helix angle of milling tool, and its calculation accuracy is better than that of the model with the influence of the helix angle of milling tool. Based on it [20] Cao and He [21]

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considered the influence of the gyroscopic couple on the bearing-rotor system under high speed, and discussed and validated the influence of the gyroscopic couple on the milling stability of the system. Vincent G et al. proposed a high speed milling stability model based on the structure of motorized spindle, and carried out the prediction and differentiation of the milling stability [22,23] and the optimization of the structure parameters of the system [24].

Those researches above detailedly discussed the thermo-mechanical behaviors and milling stability of motorized spindle machine tools, however, the influence of the coupled dynamic behaviors on the milling stability is not investigated. Meanwhile, transfer function at the cutting point is not theoretically modeled and practically calculated. This paper presented a comprehensive model for thermo-mechanical behaviors and milling stability for motorized spindle machine tools with the influence of the transfer function at cutting point, and discussed the dynamic characteristics.

2. Comprehensive model

In motorized spindle system, the tool is directly driven by the built-in motor. In the operation of the system, the spindle-tool is supported by the bearing and the support stiffness is determined by the inner compatible geometric relationship of the bearing, and the dynamics of the spindle-tool is influenced by the support stiffness. Meanwhile, the thermal displacements caused by temperature rise change the inner compatible geometric relationship, which affects the dynamics stability of the spindle-tool system. Therefore, the motorized spindle system presents the multi-field coupling relationship, and the comprehensive model for predicting the dynamics and stability of the system is shown as Fig. 1. Hence, the comprehensive model of the system includes a thermo-mechanical dynamic model and a milling stability model.

2.1. Thermo-mechanical dynamic model

The thermo-mechanical dynamic model consists of a bearing, thermal and spindle submodels. The main coupling medium is the thermal displacements of the bearing and spindle [25].

2.1.1. Bearing model

With the characteristics of low friction and high rotational speed, angular ball bearings are commonly used in motorized spindles. During the operation, the temperature rise induces the thermal displacement in radial direction of the bearing, which changes the initial inner compatible geometric relationship of the bearing. The parameter varieties are shown as Fig. 2.

The thermal displacements of inner and outer rings and ball are respectively:

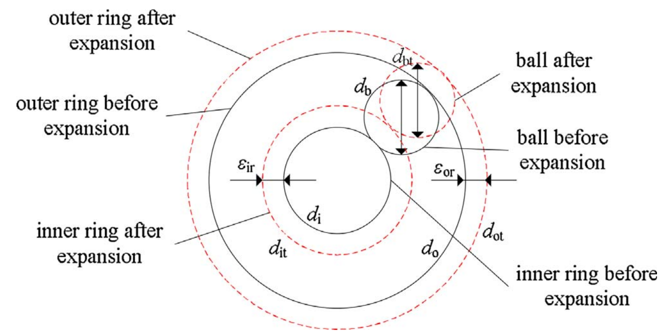

Fig. 2. Radial thermal expansion of bearing geometric parameters.

$$\begin{aligned}\varepsilon_{ir} &= (d_{it} - d_i)/2\\ \varepsilon_{or} &= (d_{ot} - d_o)/2\\ \varepsilon_b &= d_{bt} - d_b\end{aligned} \tag{1}$$

where d_i and d_{it} are the diameters of the inner ring before and after thermal expansion; d_o and d_{ot} are the diameters of outer ring before and after thermal expansion; d_b and d_{bt} are the diameters of ball before and after thermal expansion.

Meanwhile, the axial thermal displacements of housing ε_h and spindle ε_s influence the preload state of the bearing, and ε_a is the relative displacement between the inner and the outer rings of each bearing which derives from ε_h and ε_s distributed to the axial direction of each bearing based on the bearing configuration. Considering the thermal displacements, the compatible geometric relationship inside bearing is shown as Fig. 3.

Suppose the outer ring holds still, o is the center of the outer ring, i and i_t are the centers of the inner ring before and after thermal expansion, b and b_t are the ball centers before and after thermal expansion, δ_a, δ_r and θ are the relative axial, radial and angular displacements between the inner ring and the outer ring. Therefore, the coordinates of the center of inner ring after thermal expansion at the azimuth angle Ψ_j are:

$$\begin{aligned}A_{aj} &= (f_i + f_o - 1)d_{bt}\sin\alpha + \delta_a + \varepsilon_a + R_{ij}\theta\cos\psi_j\\ A_{rj} &= (f_i + f_o - 1)d_{bt}\cos\alpha + \delta_r\cos\psi_j + \varepsilon_{ir} - \varepsilon_{or}\end{aligned} \tag{2}$$

The compatible equations can be obtained by the compatible geometry relationship:

$$\begin{aligned}&[(f_i - 0.5)d_{bt} + \delta_{ij}]^2 = V_{aj}^2 + V_{rj}^2\\ &[(f_o - 0.5)d_{bt} + \delta_{oj}]^2 = (A_{aj} - V_{aj})^2 + (A_{rj} - V_{rj})^2\end{aligned} \tag{3}$$

where f_i and f_o are the curvature coefficients of the radius of inner and outer ring, respectively; R_i is the radius of the central circle of curvature of inner ring; α is the initial contact angular; δ_{ij} and δ_{oj} are the contact deformations between ball j and inner and outer rings, respectively; V_{aj}

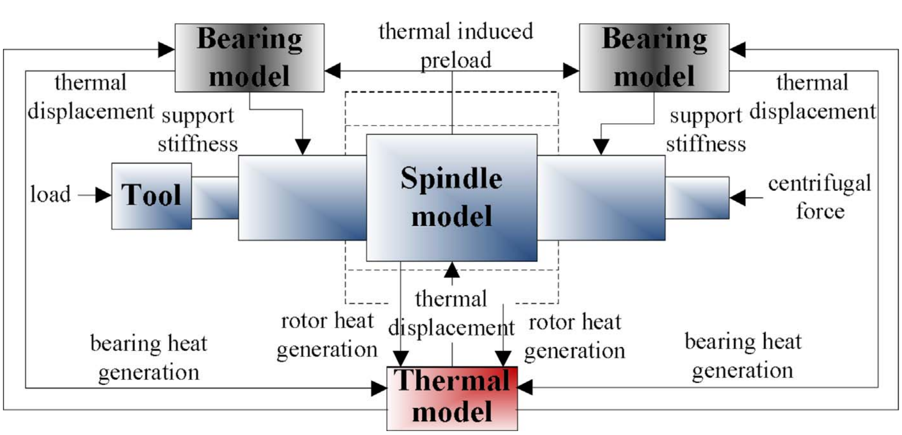

Fig. 1. Comprehensive model of the motorized spindle.