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Investigation of the effects of spindle unbalance induced error motion on machining accuracy in ultra-precision diamond turning

P. Huang, W.B. Lee*, C.Y. Chan

State Key Laboratory of Ultra-precision Machining Technology, Department of Industrial and Systems Engineering, The Hong Kong Polytechnic University, Hong Kong, PR China

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ABSTRACT

In ultra-precision machining, error motions of the aerostatic bearing spindle (ABS) have significant effects on the machining accuracy. Spindle unbalance is a critical factor attributing to error motions of the ABS. Much work currently has been focused on the measurement of error motions and spindle balancing. However, the unbalance induced spindle error motion (UISEM) and the corresponding effects on machining accuracy are not well understood. In this paper, a dynamics model of the ABS was established to characterize the UISEM and its dynamic behavior with consideration of the unbalance effects. A series of groove turning experiments were especially designed to investigate the UISEM. Good agreement between theoretical and experimental results was achieved, demonstrating the low frequency enveloping phenomenon of the error motions of the ABS, identified as the unique superposition effects of two motion components at high frequency in the spindle vibration. In addition, the experimental result reveals that the relative distance between the rotational axis of the ABS and the tool tip varies with respect to the different spindle speeds, significantly degrading the machining accuracy.

1. Introduction

Machining accuracy is critical for machined components, and is dependent on a variety of factors, including motion errors, cutting force induced deformations, positioning errors and thermal errors [1–3]. To improve the machining accuracy, these errors have been critically identified and compensated by certain state-of-the-art methods. For example, kinematics errors of the machine tool [4], tool de-centring [5], thermal deformation [6,7], and tool deflection of small milling tools due to the cutting force [8].

The aerostatic bearing spindle (ABS) is a key component of the machine tool and plays a key role in machining [9,10]. A large amount of attention has been focused on measuring the error motions of the ABS [11–17]. Martin et al. analyzed the effect of spindle error motion on the surface finish and form error of machined components, and they conducted a series of experiments to measure error motions in five degree-of-freedoms, based on the ASME standard [17]. To reduce the out of flatness of the machined surface in diamond turning, Gao et al. measured the angular and axial error motions of the ABS by means of an autocollimator and a set of capacitance probes. The motion errors were then compensated by employing the fast tool servo (FTS) system [18]. Kim and Kim investigated the optimal preloads at different cutting conditions to improve the running accuracy of the ABS [19]. The thermal characteristics of an ABS have an important influence on the spindle errors [21,22]. To improve the machining accuracy in grinding, Aleyaasin et al. proposed an optimization frame for determining the stiffness and damping of the ABS, with consideration of the rotation speed, the loading and the dimension features [23]. Based on the method of spindle speed variation, Sastry et al. compensated the radial run out in face milling [24].

As discussed above, the effects of spindle error motions on the machining accuracy have been widely studied. However, understanding of unbalance induced spindle error motion (UISEM) and the corresponding dynamics behavior in ultra-precision machining is far from being complete. Generally, the eccentricity error motion is a fundamental component of the error motions of the ABS. With the assumption that the eccentricity is caused by the eccentric installation of the test ball in the measurement [20], it is usually removed from the measurement results of error motions in most studies. This assumption is reasonable for fabricating axis-symmetric components with fixed tools, such as turning with a given operating spindle speed. However, the requirement of freeform or structured surfaces leads to the development of fast tool servo or slow slide servo assisted turning [28–34]. These methods are more sensitive to the eccentricity motion errors of the ABS, since significant distortions of these special surfaces will occur when the eccentricity motion error is generated. In addition,

* Corresponding author.
E-mail address: wb.lee@polyu.edu.hk (W.B. Lee).

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a rotating spindle can vibrate with bending mode pairs under excitation, including spindle unbalance and cutting force [35–37]. This can significantly affect the machining quality in ultra precision machining even though the amplitude of vibration is in dozens of nanometers.

Motivated by this, the present research mainly focuses on the effect of UISEM of the ABS in ultra-precision machining. A dynamics model of the ABS is proposed to analyze the UISEM of ABS and its dynamics characteristics. Two experiments on groove cutting were carried out to investigate the effect of UISEM in ultra-precision machining.

2. Modeling of the unbalance induced error motions of the ABS

The system considered only contains a spindle shaft and a chuck for simplicity. The spindle shaft is regarded as a Timoshenko beam while the chuck is regarded as a rigid disc. The bearings of the spindle system are simplified as linear springs and dampers. A diagram of the aerostatic bearing spindle system with a chuck is shown in Fig. 1. The translation in the axial direction and torsional deformation of the spindle shaft are negligible and thus are omitted. O-*XYZ* is a fixed coordinate system, and the *Z*-axis coincides with the centerline of the spindle when the spindle shaft is not deformed. The rotation angles of a plane perpendicular to the spindle shaft centerline are denoted by ϕ_x and ϕ_y, and the translations of this plane in the x and y directions are denoted by u and v.

2.1. Dynamics of the ABS

As is mentioned above, the chuck is regarded as a rigid disc. The kinetic energy of the rigid disc can be expressed by:

$$T_d = \frac{1}{2}\begin{Bmatrix}\dot{u}_2\\ \dot{\phi}_{y2}\end{Bmatrix}^{\mathrm{T}}\begin{bmatrix}m_{\mathrm{Disc}} & 0\\ 0 & J_d\end{bmatrix}\begin{Bmatrix}\dot{u}_2\\ \dot{\phi}_{y2}\end{Bmatrix} + \frac{1}{2}\begin{Bmatrix}\dot{v}_2\\ -\dot{\phi}_{x2}\end{Bmatrix}^{\mathrm{T}} \times \begin{bmatrix}m_{\mathrm{Disc}} & 0\\ 0 & Jd\end{bmatrix}\begin{Bmatrix}\dot{v}_2\\ -\dot{\phi}_{x2}\end{Bmatrix} + \Omega\begin{Bmatrix}\dot{u}_2\\ \dot{\phi}_{y2}\end{Bmatrix}^{\mathrm{T}}\begin{bmatrix}0 & 0\\ 0 & J_p\end{bmatrix}\begin{Bmatrix}v_2\\ -\phi_{x2}\end{Bmatrix} + \frac{1}{2}J_p\Omega^2 \tag{1}$$

where J_d is the diametric moment of inertial of rigid disc, J_p is the polar moment of inertial, m_{Disc} is the mass of the rigid disc and Ω is the spindle speed.

In the present study, the spindle shaft is only considered as containing one element of Timoshenko beam and two nodes. Node 1 and Node 2 represent the left and right nodes in Fig. 1 respectively. Each node contains two translational degrees of freedom and two rotational freedoms, so the spindle has total eight degrees of freedom. The spindle shaft can be divided into a number of sliced disks, and each sliced disk is regarded as rigid. The kinetic

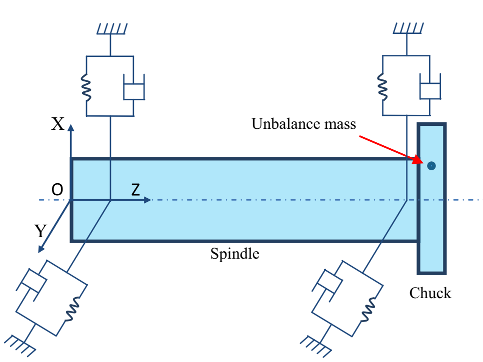

Fig. 1. Schematic of spindle system.

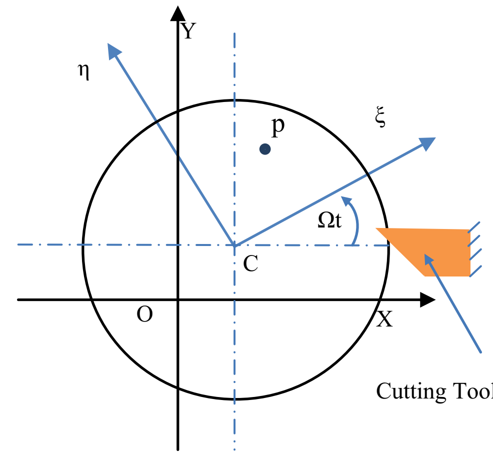

Fig. 2. Relation between the inertial and the fixed coordinates.

Table 1
Parameters used for simulation.

Damping ratio c_j	0.025
Radial stiffness k_j	22 N/μm
Shear factor χ	1.11
Spindle mass density $\rho_{spindle}$	7650 kg/m^3
Disk mass density ρ_{disk}	2710 kg/m^3
Elasticity modulus E	200*10^9 Pa
Poisson's ratio ν	0.27
Spindle length $l_{spindle}$	264.9 mm
Spindle diameter $d_{spindle}$	89 mm
Disk thickness t_{disk}	47.3 mm
Disk diameter d_{disk}	203.2 mm
Eccentric distance e	0.5 μm

energy of each sliced disk can be obtained from:

$$\mathrm{d}T = \frac{1}{2}\rho A\left(\dot{q}_x^{\,T}N_1^TN_1\dot{q}_x + \dot{q}_y^{\,T}N_1^TN_1\dot{q}_y\right)\mathrm{d}z + \frac{1}{2}\rho I_y\left(\dot{q}_x^{\,T}N_2^TN_2\dot{q}_x + \dot{q}_y^{\,T}N_2^TN_2\dot{q}_y\right) + \rho I_y\left(\Omega^2 - 2\Omega\dot{q}_y^{\,T}N_2^TN_2\mathrm{q}_x\right)\mathrm{d}z \tag{2}$$

where q_x and q_y are the coordinates of the spindle shaft, $q_x = \begin{bmatrix}u_1\\ \phi_{y1}\\ u_2\\ \phi_{y2}\end{bmatrix}, q_y = \begin{bmatrix}v_1\\ -\phi_{x1}\\ v_2\\ -\phi_{x2}\end{bmatrix}$, N_1 and N_2 are the shape functions of Timoshenko beam.

The potential energy of a sliced disk is:

$$\mathrm{d}U = \frac{1}{2}\mathrm{E}I_y\left[\left(\frac{\mathrm{d}\phi_y}{\mathrm{d}z}\right)^2 + \left(\frac{\mathrm{d}\phi_x}{\mathrm{d}z}\right)^2\right]\mathrm{d}z + \frac{1}{2}\frac{\mathrm{GA}}{\chi}\left(\gamma_{xz}^2 + \gamma_{yz}^2\right)\mathrm{d}z \tag{3}$$

where $\gamma_{xz} = \phi_y - \frac{du}{dz}, \gamma_{yz} = -\phi_y - \frac{dv}{dz}, \chi$ is the shear factor.

The kinetic energy and potential energy of the spindle shaft can be obtained by integrating all the sliced disks.

The thrust aerostatic bearing is simplified with linear stiffness and damping, so the reaction forces from the two bearing at the ends of spindle can be obtained:

$$\begin{bmatrix}F_{bx1}\\ F_{by1}\end{bmatrix} = -\begin{bmatrix}c_{xx1} & c_{xy1}\\ c_{yx1} & c_{yy1}\end{bmatrix}\begin{bmatrix}\dot{u}_1\\ \dot{v}_1\end{bmatrix} - \begin{bmatrix}k_{xx1} & k_{xy1}\\ k_{yx1} & k_{yy1}\end{bmatrix}\begin{bmatrix}u_1\\ v_1\end{bmatrix} \tag{4}$$