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Chatter milling modeling of active magnetic bearing spindle in high-speed domain

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ABSTRACT

A new dynamical modeling of Active Magnetic Bearing Spindle (AMBS) to identify machining stability of High Speed Milling (HSM) is presented. This original modeling includes all the minimum required parameters for stability analysis of AMBS machining. The stability diagram generated with this new model is compared to classical stability lobes theory. Thus, behavior's specificities are highlighted, especially the major importance of forced vibrations for AMBS. Then a sensitivity study shows impacts of several parameters of the controller. For example, gain adjustment shows improvements on stability. Side milling ramp test is used to quickly evaluate the stability. Finally, the simulation results are then validated by HSM cutting tests on a 5 axis machining center with AMBS.

1. Introduction

Currently, parts produced by machining still represent an important proportion of mechanical industrial production. Unfortunately, the productivity of machining operations at high speed is still severely limited by machining vibrations, so called chatter. It degrades the surface roughness of the part, increases the tool wear and reduces the spindle life span.

The early work of Tobias [1] in the 50s, have presented the phenomenon of regenerative effect as the main cause of chatter. The modeling of this phenomenon induces Delay Differential Equation (DDE). A first approach is to study the asymptotic stability of this equation. The stability diagram – well known as stability lobes – obtained make it possible to choose the maximum axial depth of cut for a given spindle speed associated with a chatter free machining. This approach initially dedicated to turning process [2] has been widely extended and democratized by the work of Altintas and Budak for milling process [3–5]. This method is interesting because, it leads to an analytic expression of the stability lobes. Recently, improved methods have been developed with a more detailed stability analysis, see for example [6–9]. Thus, for high-speed milling with low radial depth of cut and low helix angle, a new kind of unstable zones has been detected, called period-doubling or flip bifurcation [10].

In addition to these frequency approaches, Time Domain Simulation (TDS) was also developed with increasing computing capacity. In this case, the equation of motion is integrated step by step, in order to obtain more detailed information about the process such as the amplitude of the vibrations, the chip thickness, or the cutting forces during the tool's rotation [11–13]. The improvements of these approaches allows even simulation of the milled surface roughness [14,15]. These approaches are very powerful and can take into account all aspects of machining, even the non-linear effect of ploughing [14] or the non-linearity when the tool leave the cut during strong vibrations [11].

However, in many realistic cases, such for thin walled part machining, it is very difficult or impossible to select stable cutting conditions (spindle speed and depth of cut) for all the machining operation [16]. Classical solutions are based on machining strategies that maximize the dynamical stiffness of the mechanical components during the machining [17]. Others solutions are based on the damping effect obtained by reducing the cutting speed or, better by adding specific damper devices [18]. Tools with variable pitches [19] or with variable helix angles [20] can also be used to suppress chatter. A similar technique is to disturb the regenerative effect by spindle speed variation [21–23]. The idea of these last techniques is that the tooth pass frequency is varying; in this way, the regenerative effect is disturbed and this may significantly reduce the self-excited vibrations for specific spindle speeds. However, despite numerous academic

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studies on this topic, only the variable helix angle is widely used in industry.

During the 2000s, the development of Active Magnetic Bearing Spindles (AMBS) became very fast with several manufacturers in competition. A quite large bibliography is also available, see for example [24–27]. A major advantage of AMBS was supposed to be their life span, much larger than the conventional roller bearing spindle [27] and also their robustness to accidental force impact. This contactless technology, can also achieve very high speeds, but not yet very high power. In recent work on a micro milling machine with specific controllers, the spindle speed reached is over 150000 rpm [28]. In addition to these advantages, it is also very easy to use all the included sensors and feedback currents, for position and force measurement. For example, Auchet et al. [29] developed a method for indirect cutting force measurement by analyzing the command voltage of AMBS. Chen and Knospe developed approaches to maximize damping, using a supplementary active magnetic bearing on the spindle, and some approaches to actively control chatter on dedicated simplified test bench [27,30,31]. Kyung and Lee [32] have studied the stability of AMBS machining, but only for conventional spindle speeds. To the best knowledge of the authors, all the work made on stability analysis of AMBS was made only for low spindle speeds, corresponding to conventional cutting speed.

In this paper, the modeling of AMBS is developed and analyzed in the high-speed domain, up to 40000 rpm, corresponding to the first Hopf and flip lobes. A new original AMBS machining modeling is proposed, and the results are confirmed by experiments. The structure of the paper is as follows. First, the model is presented in Section 2, then, the stability properties are predicted in Section 3. Experimental verifications are provided in Section 4. Finally, the paper is concluded in Section 5.

2. Modeling of Active Magnetic Bearing Spindle machining

2.1. Mechanical modeling of the spindle

The analysis of the machining stability at high spindle speed requires taking into account the spindle modal behavior. Analytical models of the spindle and measurements have shown that gyroscopic effect on modal frequencies is less that 1 Hz, for the resonant frequency around 1 kHz up to 40000 rpm. Thus, gyroscopic effect will not be taken into account in this study. Also, the unbalance is not taken into account because a part of the control algorithm, not defined here, is designed to center the rotation axis of the spindle at his inertia axis and thus avoid unbalance force compensation.

The x displacement of a point M of the center of the rotor located at height z is defined by $u(z, t)$ (see Fig. 1).

The point $M(0)$ corresponds to the end of the tool, the points $M(A)$ and $M(B)$ are at the level of the A and B magnetic bearings. The modal base is used to represent the vibrations of the spindle

$$u(z,t) = \sum_{i=1}^{n} \varphi_i(z) q_i(t) \tag{1}$$

$\varphi_i(z)$ is the ith modal shape, associated to the natural pulsation ω_i and the modal displacement coefficient $q_i(t)$, with

$$\omega_i = \sqrt{\frac{k_i}{m_i}} \tag{2}$$

Modal displacement $q_i(t)$ is obtained by solving the Newton equation

$$\{m_i\ddot{q}_i(t) + c_i\dot{q}_i(t) + k_i q_i(t) = f_i(t)\}_i \tag{3}$$

where m_i, c_i and k_i are the modal mass, damping and stiffness of the ith mode. The damping factor ξ_i is defined as

$$c_i = 2\xi_i\sqrt{k_i m_i} \tag{4}$$

The projection of the external forces $F(t)$ on the ith mode is $f_i(t)$. This projection takes into account the fact that the displacement at the end of the tool is different from that on the magnetic bearing level

$$f_i(t) = \varphi_i(0)F(t) \tag{5}$$

The resulting system of equations is the following:

$$\{m_i\ddot{q}_i(t) + c_i\dot{q}_i(t) + k_i q_i(t) = \varphi_i(0)F(t)\}_i \tag{6}$$

In addition to the flexible modes, it is necessary to take into account the static rigid mode of the rotor, because the rotor is free. Indeed there is a static component of the excitation force that will be compensated by the feedback control loop. As a simplification, we will consider that the spindle is mainly moving at the A bearing level and that B bearing can be considered as a hinge (see Fig. 2).

The mass of the rotor is m_r and length L. As a rough approximation we will consider that the rotor is a uniform bar. The θ angle is considered very small ($L \gg x$) and the Newton equation is

$$\frac{m_r}{3}\ddot{q}_0 \approx F(t) \tag{7}$$

The spindle dynamics will be modelized in the xz plane, mechanically defined by its mass and the two first flexible modes. A residual stiffness was tried to be added to compensate modal truncation, but this term was removed due to numerical problems during simulation. The modal parameters were identified by hammer impact at the tip

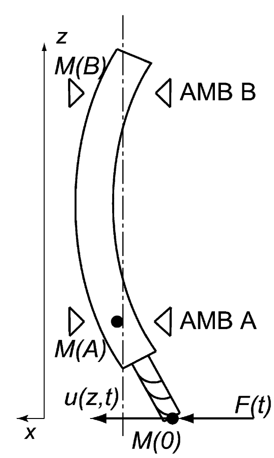

Fig. 1. Mechanical modeling of the spindle.

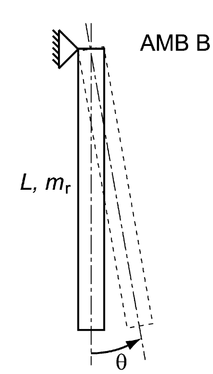

Fig. 2. Rigid mode.