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Machining prediction of spindle–self-vibratory drilling head

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ABSTRACT

The drilling of deep holes with small diameters remains an unsatisfactory technology, since its productivity is rather limited. The main limit to an increase in productivity is directly related to the poor chip evacuation, which induces frequent tool breakage and poor surface quality. Retreat cycles and lubrication are common industrial solutions, but they induce productivity and environmental drawbacks. An alternative response to the chip evacuation problem is the use of a vibratory drilling head, which enables the chips to be fragmented thanks to the axial self-excited vibration. Contrary to conventional machining processes, axial drilling instability is sought, thanks to an adjustment of head design parameters and appropriate conditions of use. A dynamic high-speed spindle/drilling head/tool system model is elaborated on the basis of rotor dynamics predictions. In this paper, self-vibratory cutting conditions are established through a specific stability lobes diagram. Investigations are focused on the drill's torsional–axial coupling role on instability predictions. A generic accurate drilling force model is developed by taking into account the drill geometry, cutting parameters and effect of torsion on the thrust force. The model-based tool tip FRF is coupled to the proposed drilling force model into an analytical stability approach. The stability lobes are compared to experimentally determined stability boundaries for validation purposes.

1. Introduction

The drilling of deep, small-diameter holes is an unsatisfactory machining operation that results in poor surface quality and low productivity. These drawbacks are mainly related to difficulties in evacuating the chips through the drill flute during the cut. Non-productive retreat cycles and the use of high-pressure lubrication are the current industrial solutions used to evacuate chips, but present respectively productivity and environmental problems. New drilling techniques have emerged, based on the tool's axial vibration, in order to fragment the chips and enhance their evacuation without the need for lubricants and retreat cycles. The self-vibratory drilling technology uses the cutting energy to generate tool axial vibration (Guibert et al., 2008). A specific self-vibratory drilling head (SVDH) excites low-energy chatter vibration for specific process parameters by using a combination of a low-rigidity axial spring and an additional mass located between the spindle and the tool. By tuning both the stiffness of the spring located between the SVDH body and the SVDH vibrating subsystem i.e. the drill-holder, and the additional SVDH mass, the regenerative axial vibrations can be controlled for adequate cutting parameters. These self-excited vibrations must have a magnitude greater than the feed per tooth, which enables the fragmentation of the chips without external adjunction of energy. Changes brought about by controlled vibratory cutting include decreased average force and temperature. The challenge is to tune and keep operating condition in stabilized self-excited vibration at a suitable frequency and magnitude for a good quality of cutting. So, instead of many traditional manufacturing processes, the cutting parameters are chosen to be in the unstable domain.

In this paper an original approach to establishing accurate stability lobes diagram in self-excited drilling operations is proposed. The predicted speed-dependent transfer function of the overall system, composed of spindle–SVDH–twist drill, is then integrated into an analytical chatter vibration stability approach to calculate the associated dynamic stability lobes diagram.

In Section 2, the spindle–SVDH rotor dynamics model is presented. A special rotor-beam element, developed by Gagnol (Gagnol et al., 2007a,b) in a co-rotational reference frame is implemented. The rolling bearing stiffness matrices are calculated around a static function point on the basis of T.C. Lim's formulation (Lim and Singh, 1990) and then integrated into the global finite element model. The rotating system is derived using Timoshenko beam theory.

The literature on the modelling and analysis of spindle systems shows that the tool tip FRF is also greatly influenced by the contact dynamics of the spindle–holder–tool interfaces (Erturk et al., 2006; Schmitz et al., 2007). The flexibility of the aforementioned interfaces plays an important role in spindle dynamics (Ahmadian and Nourmohammadi, 2010). The identification of contact dynamics in spindle–SVDH–tool assemblies has been carried out by Forestier

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Nomenclature

$\mathbf{H}_{model}$	interface model receptance
k	interface stiffness
c	interface damping factor
$\mathbf{D}$	damping matrix
$\mathbf{M}$	mass matrix
$\mathbf{K}$	stiffness matrix
$\mathbf{G}$	gyroscopic matrix
$\mathbf{N}$	spin softening effects matrix
Ω	rotor angular velocity
$\mathbf{q}_N$	nodal displacement
$\mathbf{F}(t)$	force vector
F_Z	axial cutting forces
K_Z	axial cutting pressure
K_t	tangential cutting pressure
D	drill diameter
h	chip thickness
B	chip width
Z	axial tool tip displacement
f_z	feed rate per tooth
ω_c	chatter pulsation
N	spindle speed
p	individual lobe number (integer)

(Forestier et al., 2011) using the receptance coupling method on the basis of experimental substructure characterization. The identified interface models are then integrated into the global model. The global model is then validated by comparing the numerical and experimental Frequency response Functions.

In Section 3, a generic accurate drilling force model is developed by taking into account the drill geometry, cutting parameters and effect of torsion on the thrust force.

Section 4 is dedicated to the prediction of adequate drilling conditions based on controlled self-excited drill vibration. A specific instability lobes diagram is elaborated by integrating into an analytical stability analysis the overall structural model-based tool tip FRF of the system associated with the proposed drilling force model. The torsional–axial coupling of the twist drill is investigated on the basis of Bayly's model (Bayly and Metzler, 2001) and consequences on drilling instability are established.

In Section 5, the stability lobes are compared to experimentally determined stability boundaries for validation purposes. By taking into account the drill torsional–axial coupling, the model-based stability lobes are modified and correspond better with the experimentally determined stability limits.

Finally, a conclusion is presented.

2. Model building

The vibratory drilling system is composed of a SVDH body clamped to the spindle by a standard HSK63A tool-holder interface. A SVDH vibrating subsystem is jointed to the SVDH body using a specific spring, and axially guided by a ball retainer. Finally, a long drill is held in the SVDH vibrating subsystem with a standard ER25 collet chuck. The SVDH system is mounted on a spindle capable of speeds up to 15,500 rpm. The spindle has four angular bearings in overall back-to-back configuration (Fig. 1).

The spindle–SVDH–tool system is composed of four structural subsystems: the drill, the SVDH vibrating subsystem, the SVDH body and the spindle.

Table 1
Identified stiffness and damping factor of the interface.

Interface	Stiffness (N/m)	Damping factor (N s/m)
Spring and ball retainer	1.13×10^6	24
Collet chuck	14.8×10^6	2
HSK 63	Infinite	0

2.1. Structural elements

The model for the spindle–SVDH–tool system is restricted to the rotating structure composed of the spindle shaft, the SVDH and the drill. This hypothesis was established by Gagnol et al. (2007a,b) through an experimental modal identification procedure carried out on spindle substructure elements. Dynamic equations were obtained using *Lagrange* formulation associated with a finite element method. Due to the size of the rotor sections, shear deformations had to be taken into account. Then the rotating substructure was built using Timoshenko beam theory. The relevant shape functions were cubic in order to avoid shear-locking. A special three-dimensional rotor-beam element with two nodes and six degrees of freedom per node was developed in the co-rotational reference frame. The damping model used draws on *Rayleigh* viscous equivalent damping, which makes it possible to regard the damping matrix $\mathbf{D}$ as a linear combination of the mass matrix $\mathbf{M}$ and the spindle rigidity matrix $\mathbf{K}$:

$$\mathbf{D} = \alpha\mathbf{K} + \beta\mathbf{M} \tag{1}$$

where α and β are damping coefficients. The set of differential equations are detailed in Gagnol (Gagnol et al., 2007a,b) and can be written as:

$$\mathbf{M}\ddot{\mathbf{q}}_N + (2\Omega\mathbf{G} + \mathbf{D})\dot{\mathbf{q}}_N + (\mathbf{K} - \Omega^2\mathbf{N})\mathbf{q}_N = \mathbf{F}(t) \tag{2}$$

where $\mathbf{M}$ and $\mathbf{K}$ are the mass and stiffness matrices. $\mathbf{K}$ results from the assembly of the spindle rigidity matrix $\mathbf{K}_{spindle}$ and the rolling bearing rigidity matrix $\mathbf{K}_{bearings.}$ $\mathbf{D}$ is the viscous equivalent damping matrix, $\mathbf{q}_\mathrm{N}$ and $\mathbf{F}(t)$ are the nodal displacement and force vectors. $\mathbf{G}$ and $\mathbf{N}$ are respectively representative of gyroscopic and spin softening effects. Ω is the rotor's angular velocity.

2.2. Modelling of spindle–SVDH–tool interfaces

The dynamic behaviour of the interfaces represented by the HSK63 taper, spring and ball retainer, and collet chuck are taken into account (Fig. 2). The identification procedure of the interface models was carried out by Forestier et al. (2012) based on the receptance coupling method and then integrated into the model as illustrated in Fig. 2. The axial dynamic behaviour of the interface are modeled by a spring-damper element whose transfer function is

$$H_{\mathrm{interface}}(\omega) = \frac{1}{k + ic\omega} \tag{3}$$

The rigidity k and damping c values were determined by minimizing the gap between the measured and the modeled tool tip node frequency response function for a non-rotating spindle, using an optimization routine and a least-squares type object function.

The identification of the interface model is carried out by comparing the reconstructed assembly receptance with the measured receptance value. Identifications results for the spring and ball retainer, the collet chuck interface and the HSK 63 interfaces are given in Table 1. The experimental modal analyses carried out on the spindle/SVDH body system in the axial direction allow the HSK 63 interface to be considered as a rigid connection.