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Chatter suppression of external grooving tools

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Abstract

Generating deep grooves using long blades is a challenge due to excessive chatter vibrations. In the present paper, an analytical simplified solution of the critical grooving width is obtained. It is found that a cross FRF of the tool dominates the solution. A conventional grooving tool with long overhang is tested where chatter vibrations develop. A new tool with attached Dynamic Vibration Absorber (DVA) is designed to replace the conventional one and a novel DVA tuning mechanism is presented. The cross FRF real part is optimized via modal experiments and the tuning mechanism. Machining experiments are conducted for both grooving systems in order to verify the model predictions. Grooving is unstable using the conventional blade while it is stable with the tuned new one.

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Keywords: Chatter; Stability lobes; Grooving; Dynamic vibration absorber; tuning

1. Introduction

Grooving blades are widely used in turning operations. The tools have usually a rectangular cross-section with width that is much smaller than height. During machining of deep grooves, chatter vibrations become apparent since the tool overhang becomes long and its flexibility increases.

Chatter instability of turning operations has been first investigated analytically by Tlusty and Polacek [1] and Tobias and Fishwick [2]. Tlusty and Polacek modeled the dynamic system as one-dimensional (1D) and in one direction. Marui et al. [3,4] investigated the chatter in turning operation experimentally and examined an interference model at the tool flank. Kaneko et al. [5] used 2D model for chatter-mark predictions and solved it numerically. Rao and Shin [6] modeled the stability in turning in multidirectional approach and include a cross-coupling between axial and radial vibrations. Clancy and Shin [7] added process damping to their model. Ozlu and Budak [8] proposed an analytical frequency-domain model for the prediction of stability limit for multidimensional dynamic turning systems. Axial-radial cross-coupling is considered but tangential related cross FRF's were ignored. Eyniyan and Altintas [9,10] presented a stability model for turning operations with three-dimensional regeneration and process damping. All the direct and cross-FRF's of the turning tool are included. Urbikain et al. [11] showed numerically that regenerative chatter of long turning tools can develop due to modes in the tangential direction.

Enhancing the chatter resistance of turning tools can be achieved by attaching Dynamic Vibration Absorber's (DVA's) and tuning them properly [12]. Several works have been performed to find the ideal tuning. Koegnisberger and Tlusty [13] proposed an electrical circuit equivalent to the mechanical system to optimize the FRF real part for DVA design. Rivin and Kang [14] improved the stability of a boring bar by combining a high stiffness beam and DVA. Tarng et al. [15] and Rashid et al. [16] tuned manually the DVA natural frequency to match with the one of the structure target modes. Sims [17] obtained analytically DVA parameters for optimal real part of the FRF. Yang et al. [18] optimized a Multi-DVA for one dominant mode considering the negative real part of the FRF. In these works, the tool is considered as a lumped,

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linear and 1-DOF model and the DVA's are additional linear dof systems. Saffury and Altus [19] analyzed and compared the FRF of a viscoelastic cantilever beam to an elastic one with attached DVA. Miguéleza et al. [20] obtained analytical expressions for the tuning frequency of a DVA on a beam which represents a boring bar.

To the author knowledge, chatter suppression of long external grooving blades via DVA systems has not been studied yet theoretically and experimentally.

In this paper, a newly developed grooving tool with attached DVA, covered by a patent application of Hecht et al. [21], is presented. Using Finite Element Analysis (FEA) with Sims [17] approach, the DVA is first tuned numerically by optimizing a proper FRF. The regenerative chatter stability limit is obtained following Eyniyan and Altintas [9,10] three-dimensional model where the tangential-radial cross FRF is also considered. Axial cutting forces are neglected since they are small with respect to the tangential and feed ones as also shown by Bisu et al. [23]. The numerical results are followed by experimental modal analysis and machining experiments.

Nomenclature

b	Height of the grooving blade
F_t	Force component in the tangential direction
F_f	Force component in the feed direction
FRF_{tr}	Cross Frequency Response Function (FRF) between a tangential force and a radial displacement
G_{xx}	Direct FRF in x direction at the cutting edge
G_{xy}	Cross FRF between x and y directions at the cutting edge
h	Chip thickness
K_{ct}	Specific cutting stress in tangential direction
K_{cf}	Specific cutting stress in feed direction
L	Overhang length of the grooving blade
n	Spindle speed (rpm)
τ	One spindle revolution period
w	Width of cutting edge

2. Stability model

A schematic description of a grooving tool, with a straight edge, during machining operation is shown in figure 1. The cutting forces which act on the edge in the tangential and feed (radial) directions are proportional to the uncut chip area (w·h):

$$F_t = K_{ct} \cdot w \cdot h \quad ; \quad F_f = K_{cf} \cdot w \cdot h \qquad (1)$$

where K_{ct} and K_{cf} are the specific cutting forces for the tangential and feed directions. Only the regenerative mechanism of chatter is considered. Vibrations in the tangential direction causes variation in the chip thickness as illustrated in figure 2. The cutting edge has radial displacement component in addition to the tangential one. The variations in the cutting forces (ΔF) due to variations in the chip thickness (Δh) are therefore:

$$\Delta F_t = K_{ct} \cdot w \cdot \Delta h \quad ; \quad \Delta F_f = K_{cf} \cdot w \cdot \Delta h \qquad (2)$$

The x and y components of the forces are:

$$\begin{cases} \Delta F_x = +K_{cf} \cdot w \cdot \Delta h \\ \Delta F_y = -K_{ct} \cdot w \cdot \Delta h \end{cases} \qquad (3)$$

The system is assumed flexible in all directions and linear with the changes in the cutting forces. Axial cutting forces are ignored and couplings between the axial (z) and other directions (x and y) are neglected for simplicity. Thus,

$$\begin{pmatrix} \Delta x(i\omega) \\ \Delta y(i\omega) \end{pmatrix} = \begin{bmatrix} G_{xx} & G_{xy} \\ G_{yx} & G_{yy} \end{bmatrix} \begin{pmatrix} \Delta F_x(i\omega) \\ \Delta F_y(i\omega) \end{pmatrix} \qquad (4)$$

The dynamic chip thickness in the frequency domain [10] is:

$$\Delta h(i\omega) = -(1 - e^{-i\omega\tau}) \cdot \Delta x(i\omega) \qquad (5)$$

Inserting (3) and (4) into (5) yields:

$$\Delta h(i\omega) = -(1 - e^{-i\omega\tau}) \cdot [G_{xx}K_{cf} - G_{xy}K_{ct}] \cdot w \cdot \Delta h(i\omega) \qquad (6)$$

The critical width of cut is obtained by solving the above eigenvalue problem (6):

$$w_{cr} = \frac{-1}{2K_{ct}Re[G_O]} \qquad (8)$$

where G_O is the oriented FRF defined as follows:

$$G_O \equiv G_{xx}(K_{cf}/K_{ct}) - G_{xy} \qquad (9)$$

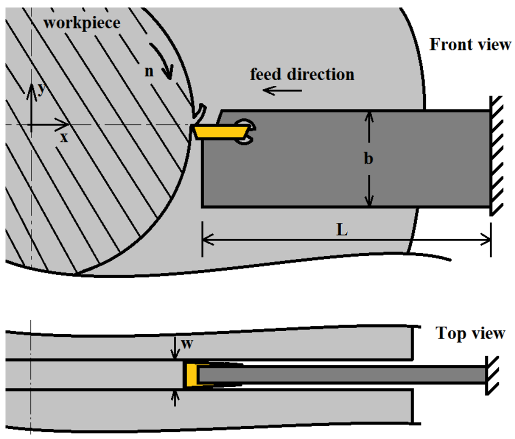

Fig. 1. Schematic description of external grooving operation.

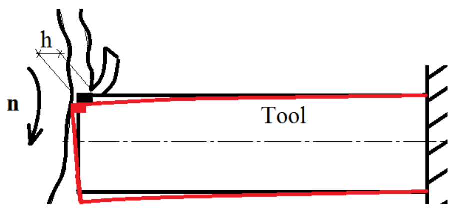

Fig. 2. Chip thickness variation due to tool bending.