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Stability lobes in milling including process damping and utilizing Multi-Frequency and Semi-Discretization Methods

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ABSTRACT

In this work the equivalent viscous model of process damping is integrated into the Multi-Frequency Solution and the Semi-Discretization Method to establish the stability lobes in milling. The basic formulations are presented along with the comparisons between the two approaches using examples from the literature. A non-shallow cut is employed in the comparisons. Assessing the performance of the two methods is conducted using time domain simulations. It is shown that the Semi-Discretization Method provides accurate results over the whole tested range of cutting speed, whereas higher harmonics are required to achieve the same accuracy when applying the Multi-Frequency Solution at low speeds. Stability lobes established using the Semi-Discretization Method are verified experimentally. It is shown that these lobes agree closely with cutting tests.

1. Introduction

In the early years of investigating machining chatter, negative damping was known as the reason for vibration instability, but later Tobias and Fishwick in [1] and Tlusty and Polacek in [2] identified the feedback between subsequent cuts, regeneration, as the main source of vibration instability in machining. These authors also formulated the dynamics of regenerative chatter by combining the structural dynamics of the machine tool and the mechanics of orthogonal cutting, resulting in a linear Delay Differential Equation, DDE, with constant coefficients. They solved the DDE to determine the border of stable width of cut at each spindle speed and established the ‘‘stability lobes’’.

In milling operations, the cutting force direction and chip thickness vary around the arc of cut due to tool rotation. Therefore, the dynamics of regeneration is described by a DDE with time varying coefficients [3]. In [4] the average direction was used and the single point chatter analysis approach was applied to milling. Opitz and Bemardi [5] also used the average value of the coefficients to calculate the stability borders. Recently, Altintas and Budak [6] approximated the coefficients by a finite number of their Fourier expansion coefficients. Although they formulated this Multi-Frequency Solution, MFS, they showed that only the first term of the Fourier expansion, the zero order, suffices for most of the milling operations. Insperger and Stepan [7] used the Semi-Discretization Method, SDM, to examine the stability of milling. These authors showed that neglecting the higher orders of the Fourier expansion coefficients in [6] results in the elimination of period doubling instability in highly interrupted milling operations. Merdol and Altintas [8] obtained results similar to [7] using a large number of higher harmonics in the MFS of [6].

In the above works process damping was not taken into consideration. This mechanism of damping affects process stability at low cutting speeds in a profound way. Wu [9] showed that the indentation of material under the flank face of the tool generates a great source of damping in machining. Huang and Wang [10] studied the different sources of process damping, and came to the conclusion that the indentation of material was indeed the major source of vibration energy dissipation at low cutting speed. Chiou and Liang [11] simplified Wu's indentation model to a piecewise linear viscous damper by assuming small amplitude of vibration.

While the viscous model of process damping has been implemented in several works to establish the lobes in turning, e.g. [12–14] very few cases have been reported for milling. Budak and Tunc [15] considered the effect of process damping as an additional damper in the zero order MFS. They identified the additional damping coefficients by measuring the stability limits experimentally and fitting the results to the stability model. Kurata et al. [16] identified the process damping coefficient through inverse chatter analysis of plunge turning, and then used it in the prediction of stability lobes in milling. Eynian and Altintas [17] integrated the experimentally identified viscous dampers in the zero order MFS as well, but they used rotating tool modes to alleviate the effect of neglecting higher harmonics. Bachrathy and Stepan [18] used Liang's piecewise linear damper

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in SDM to calculate the stability lobes; they linearized the damping model around the periodic solution of the vibratory system.

In this paper the equivalent viscous damper of [19] is integrated into MFS and SDM to calculate the stability lobes in milling. Unlike the work in [17] the development here will be shown for the general case of Fourier expansion coefficients in MFS. Also, the linear damping model will be used in SDM rather than the linearization approach followed in [18]. What is more important here is to put the two formulations together to allow for comparison between the two approaches to establishing the stability lobes in milling including process damping. Assessing the accuracy of the established lobes from the two methods will be conducted using time domain simulations.

The 2 Degree Of Freedom (DOF) dynamic model used in this investigation is described in the next section. Since numerical simulations are used to compare the performance of MFS and SDM, a brief description of the numerical simulation model is presented in Section 3. In this section, a more effective approach to identifying the onset of instability from the simulation results will be presented. The formulations and computation steps of MFS and SDM are given in Sections 4 and 5, respectively. Section 6 will present the results of examples conducted to compare between the two approaches to establishing the lobes. Moreover, the effect of including higher harmonics in MFS, and choosing a proper discretization size in SDM will be discussed in that section. In these examples, the cut is not highly interrupted (non-shallow immersion). It will be shown that the zero order MFS stability lobes agree well with the ones obtained from SDM at high cutting speeds. The two methods, however, disagree considerably at low speeds and high damping when the zero order is utilized. This disagreement diminishes by adding higher harmonics to MFS at the expense of longer simulation time. Experimental verifications of stability lobes established using SDM will be presented in Section 7. These experiments will show the close agreement between the lobes from SDM and the actual cutting tests.

2. Dynamic model

The 2 DOF system, shown in Fig. 1, is used to describe the dynamics of the vibratory model. The modal stiffness, mass, and damping in the cutting, Y, and feed, X, directions, K_y, K_x, M_y, M_x, C_y, and C_x, are usually obtained from experimental modal analysis. The tool has N cutting edges and the immersion angle of the jth tooth, φ_j $j=1\ldots N$, is measured clockwise from the Y direction. One can express the immersion angle as a function of time and tool rotational speed, Ω, as

$$\varphi_j = \Omega t + (j-1)\frac{2\pi}{N} \tag{1}$$

The cutting force components acting on the jth tooth are F_r and F_t in Fig. 1 in the radial and tangential directions, respectively. The forces arise due to the shearing and ploughing mechanisms. Therefore, the tangential and radial cutting forces are composed of the shearing, with subscript s, and ploughing with subscript p, according to

$$F_t(t) = F_{ts}(t) + F_{tp}(t);\ F_r(t) = F_{rs}(t) + F_{rp}(t), \tag{2}$$

The shear forces are computed from

$$F_{ts} = K_t g(\varphi_j) bh,\ F_{rs} = K_r F_{ts}; \quad g(\varphi_j) = \begin{cases} 0 & \varphi_j > \varphi_{ex} \text{ or } \varphi_j < \varphi_{st} \\ 1 & \varphi_{st} < \varphi_j < \varphi_{ex} \end{cases} \tag{3}$$

where b is the axial depth of cut and K_t and K_r are the tangential and radial cutting force coefficients, respectively, measured experimentally. In Eq. (3), $g(\varphi_j)$ is a window function between the tool engagement start and exit angles: φ_{st} , and φ_{ex}. In Fig. 1(b), the uncut chip thickness consists of the part produced by the tool rotation and feed motion, and the other part by the regeneration of surface waves:

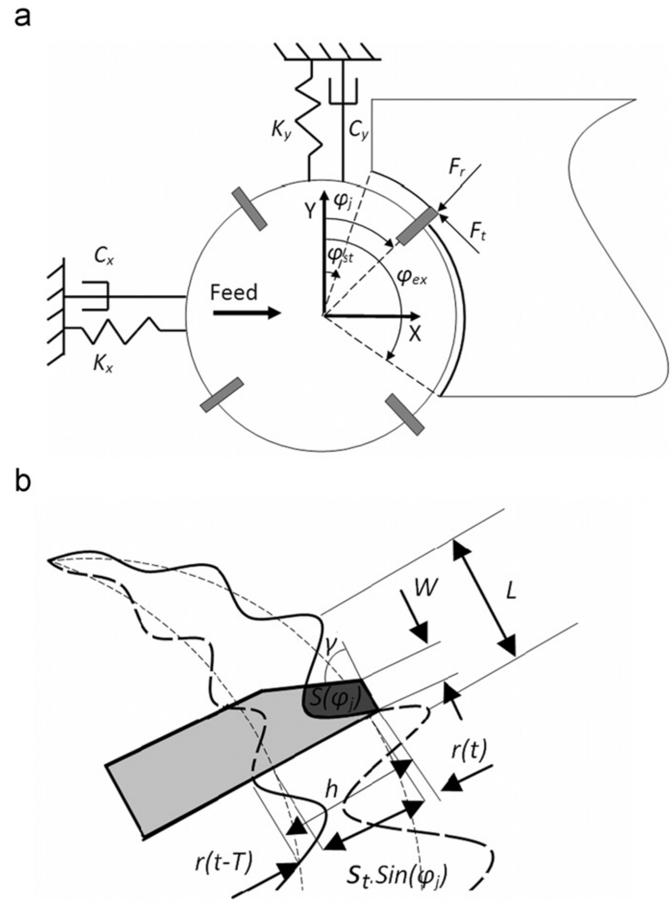

Fig. 1. (a) 2DOF vibratory model, (b) uncut chip thickness, and indentation of undulations under the flank face of the tool.

$$h(t) = s_t \sin\varphi_j + (r(t) - r(t-T)) \tag{4}$$

where $r(t)$ is the tool displacement in the radial direction, and T is the tooth passing period:

$T = 2\pi/N\Omega$.

In Eq. (4), negative chip thickness implies disengagement of the tool from the workpiece due to the vibration amplitude reaching high values. In this case the chip thickness is actually zero. The tangential and radial forces are divided into "harmonic", F^H, and "regenerative", F^{reg}, parts as follows:

$$F_{ts} = {F_{ts}}^H + {F_{ts}}^{reg}$$
$$F_{ts}^H = g(\varphi_j) K_t b s_t \sin\varphi_j;\ F_{ts}^{reg} = g(\varphi_j) K_t b (r(t) - r(t-T)) \tag{5}$$

The regenerative part is associated with the vibration in subsequent cuts, while the harmonic part is associated with the rotational angle and the feed per tooth. It is called "harmonic" because the static force "pulse" could contain a significant number of harmonics, depending on the cut geometry.

The ploughing force is calculated using Wu's indentation model [9], where the radial ploughing force is assumed to be proportional to the volume, V, of the material extruded underneath the flank face:

$$F_{rp} = g(\varphi_j) K_{sp}.V;\ V = bS \tag{6}$$