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Process Damping Analytical Stability Analysis and Validation

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ABSTRACT
This paper describes an analytical solution for turning and milling stability that includes process damping effects. Comparisons between the new analytical solution, time-domain simulation, and experiment are provided. The velocity-dependent process damping model applied in the analysis relies on a single coefficient similar to the specific cutting force approach to modeling cutting force. The process damping coefficient is identified experimentally using a flexure-based machining setup for a selected tool-workpiece pair (carbide insert-AISI 1018 steel). The effects of tool wear and cutting edge relief angle are also evaluated. It is shown that a smaller relief angle or higher wear results in increased process damping and improved stability at low spindle speeds.

INTRODUCTION

The analytical stability lobe diagram offers an effective predictive capability for selecting stable chip width-spindle speed combinations in machining operations [1-4]. However, the increase in allowable chip width provided at spindle speeds near integer fractions of the system's dominant natural frequency is diminished substantially at low spindle speeds where the stability lobes are closely spaced. Fortunately, the process damping effect can serve to increase the chatter-free chip widths at these low speeds. This increased stability at low spindle speeds is particularly important for hard-to-machine materials that cannot take advantage of the higher speed stability zones due to prohibitive tool wear at high cutting speeds.

Many researchers have investigated process damping in turning and milling operations. Early studies were carried out by Wallace and Andrew [5], Sisson and Kegg [6], Peters *et al.* [7], and Tlusty [8]. More recent efforts include:

- a plowing force model based on the interference between the tool and workpiece [9]
- the application of this plowing force model to milling operations [10-13]
- a mechanistic description of the contributions of shearing and plowing forces to process damping [14]
- a first-order Fourier transform representation of the interference between the tool and workpiece [15-16]
- numerical simulation of the nonlinear process damping stability model [17-18]
- an experimental investigation of the nonlinear process damping stability model [19]
- experimental identification of the process damping model [20-21].

These studies described process damping as energy dissipation due to interference between the cutting tool clearance face and machined surface during relative vibrations between the tool and workpiece. It was shown that, given fixed system dynamics, the influence of process damping increases at low spindle speeds because the number of undulations on the machined surface between revolutions/teeth increases, which also increases the slope of the wavy surface. This, in turn, leads to increased interference and additional energy dissipation.

In this paper, an iterative, analytical stability analysis is described that incorporates the effects of process damping. The analytical stability limit is validated using time-domain simulation and experiments. The paper is organized as follows. In the first section, process damping is described and the process damping force model is defined. Next, the stability algorithm is detailed. Results are then provided followed by the conclusions.

PROCESS DAMPING DESCRIPTION

In descriptions of regenerative chatter in machining, the variable component of the instantaneous cutting force may be written as:

$$F = K_s b (Y_0 - Y), \tag{1}$$

where K_s is the specific cutting force (which depends on the tool-workpiece combination and, to a lesser extent, the cutting parameters), b is the chip width, Y_0 is the vibration amplitude in the surface normal direction, y, from the previous cutting pass, and Y is the current vibration amplitude. See Fig. 1. The underlying assumption in Eq. 1 is that there is no phase shift between the variable force and chip thickness; this is indicated by the real values of K_s and b. However, for low cutting speeds, V, it has been shown that a phase shift can occur. This behavior is captured by the phenomenon referred to as process damping. Practically speaking, the effect of process damping is to enable significantly higher chip widths at low cutting speeds than linear stability analyses predict.

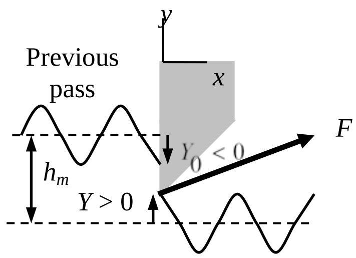

Figure 1. The variable component of the cutting force, F, depends on the instantaneous chip thickness. The chip width is measured into the page; the mean chip thickness, h_m, is also identified.

To describe the physical mechanism for process damping, consider a tool moving on a sine wave while shearing away the chip [22]; see Fig. 2. Four locations are identified: 1) the clearance angle, γ, between the flank face of the tool and the work surface tangent is equal to the nominal relief angle for the tool; 2) γ is significantly decreased and can become negative (which leads to interference between the tool's relief face and surface); 3) γ is again equal to the nominal relief angle; and 4) γ is significantly larger than the nominal value.

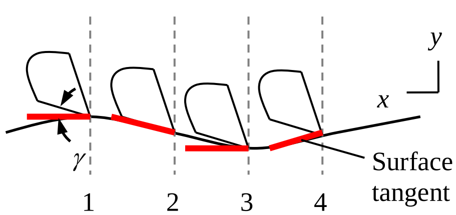

Figure 2. Physical description of process damping. The clearance angle varies with the instantaneous surface tangent as the tool removes material on the sinusoidal surface.

At points 1 and 3 in Fig. 2, the clearance angle is equal to the nominal value so there is no effect due to cutting on the sinusoidal path. However, at point 2 the clearance angle is small (or negative) and the thrust force in the surface normal direction is increased. At point 4, on the other hand, the clearance angle is larger than the nominal and the thrust force is decreased. Because the change in force caused by the sinusoidal path is 90 deg out of phase with the displacement and has the opposite sign from velocity it is considered to be a viscous damping force (i.e., a force that is proportional to velocity).Given the preceding description, the process damping force, F_d, in the y direction can be expressed as a function of velocity, chip width, cutting speed, and a constant C [21]. See Eq. 2.

$$F_d = -C\frac{b}{V}\dot{y} \tag{2}$$

As a final note regarding the sinusoidal path description in Fig. 2, the damping effect is larger for shorter vibration wavelengths, λ, because the slope of the sinusoidal surface increases and, subsequently, the variation in clearance angle increases. The wavelength equation, provided in Eq. 3, shows that lower cutting speeds or higher vibrating frequencies, f, gives shorter wavelengths and, subsequently, increased process damping.

$$\lambda = \frac{V}{f} \tag{3}$$

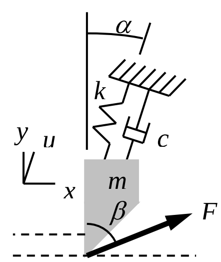

Figure 3. Single degree of freedom turning model.

STABILITY ALGORITHM

Single Degree of Freedom Turning

To describe the stability algorithm, consider the single degree of freedom turning model displayed in Fig. 3. Tlusty [22] defines the limiting stable chip width, $b_{\lim}$, for regenerative chatter using:

$$b_{\lim}=\frac{-1}{2K_s\,\mathrm{Re}(G_{or})}\,, \qquad (4)$$

where G_{or} is the oriented frequency response function, $G_{or}=\cos(\beta-\alpha)\cos(\alpha)G_u$. In this expression, β is the force angle relative to the surface normal, α is the angle between the *u* direction and the surface normal, and G_u is the frequency response function in the *u* direction. To relate the frequency-dependent $b_{\lim}$ vector to spindle speed, Ω, Eq. 5 is applied to define the relationship between Ω and the valid chatter frequencies, f_c (i.e., those frequencies where the real part of G_{or} is negative):

$$\frac{f_c}{\Omega}=N+\frac{\varepsilon}{2\pi}\,, \qquad (5)$$

where $N=0,1,2,...$ is the integer number of waves per revolution (i.e., the lobe number) and $\varepsilon=2\pi-2\tan^{-1}\left(\frac{\mathrm{Re}(G_{or})}{\mathrm{Im}(G_{or})}\right)$ (rad) is the phase between the current vibration and the previous pass.

To incorporate the process damping force (which acts in the *y* direction), it is first projected into the *u* direction:

$$F_u=F_d\cos(\alpha)=-C\frac{b}{V}\dot{y}\cos(\alpha)=-\left(C\frac{b}{V}\cos(\alpha)\right)\dot{y}\,. \qquad (6)$$

The final form of Eq. 6 emphasizes that the *u* projection of the process damping force is effectively a viscous damping term. Therefore, the force can be incorporated in the traditional regenerative chatter stability analysis by modifying the structural damping in G_u. As shown in Fig. 3, the single degree of freedom, lumped parameter dynamic model can be described using the mass, *m*, viscous damping coefficient, *c*, and spring stiffness, *k*. In the absence of process damping, the equation of motion in the *u* direction is:

$$m\ddot{u}+c\dot{u}+ku=F\cos(\beta-\alpha)\,. \qquad (7)$$

The corresponding frequency response function in the *u* direction is:

$$G_u=\frac{U}{F\cos(\beta-\alpha)}=\frac{1}{-m\omega^2+ic\omega+k}\,, \qquad (8)$$

where ω is the excitation frequency (rad/s). When process damping is included, however, the equation of motion becomes:

$$m\ddot{u}+c\dot{u}+ku=F\cos(\beta-\alpha)-\left(C\frac{b}{V}\cos(\alpha)\right)\dot{y}\,. \qquad (9)$$

Replacing $\dot{y}$ in Eq. 9 with $\cos(\alpha)\dot{u}$ gives:

$$m\ddot{u}+c\dot{u}+ku=F\cos(\beta-\alpha)-\left(C\frac{b}{V}\cos^2(\alpha)\right)\dot{u}\,. \qquad (10)$$

Rewriting Eq. 10 to combine the velocity terms yields:

$$m\ddot{u}+\left(c+C\frac{b}{V}\cos^2(\alpha)\right)\dot{u}+ku=F\cos(\beta-\alpha)\,, \qquad (11)$$

where the new viscous damping coefficient is $c_{new}=c+C\frac{b}{V}\cos^2(\alpha)$. Replacing the original damping coefficient, *c*, (from the structure dynamics only) with c_{new} enables process damping to be incorporated in the analytical stability model. The new frequency response function is:

$$G_u=\frac{U}{F\cos(\beta-\alpha)}=\frac{1}{-m\omega^2+ic_{new}\omega+k}\,. \qquad (12)$$

However, the new damping value is a function of both the spindle speed-dependent limiting chip width and the cutting speed. The cutting speed (m/s) depends on the spindle speed (rpm) and workpiece diameter (m) according to $V=\frac{\pi d}{60}\Omega$. Therefore, the *b* and Ω vectors must be known in order to implement the new damping value. This leads to the converging nature of the stability analysis that incorporates process damping. The following steps are completed for each lobe number, or *N* value (see Eq. 5):

1. the analytical stability boundary is calculated with no process damping to identify initial *b* and Ω vectors
2. these vectors are used to determine the corresponding c_{new} vector
3. the stability analysis is repeated with the new damping value to determine updated *b* and Ω vectors
4. the process is repeated until the stability boundary converges.

To demonstrate the approach, consider the model in Fig. 3 with $\alpha=0$, $k=6.48\times10^6$ N/m, $m=0.561$ kg, $c=145$ N-s/m, $K_s=2927\times10^6$ N/m^2, $\beta=61.79$ deg, and $d=0.035$ m. The stability boundary with no process damping ($C=0$) is shown in Fig. 4 for $N=0$ to 60. It is observed that the limiting chip width approaches the asymptotic stability limit of 0.37 mm for spindle speeds below 1000 rpm.

Results of the converging procedure with process damping for the $N=20$ stability boundary are provided in Fig. 5. Converging behavior is observed for the 10 iterations as the lobes move up and slightly to the right. Although a convergence criterion, such as a threshold percent difference between subsequent minimum values, could be implemented, a practical selection of 20 iterations was applied for the diagrams in this study to ensure convergence. Figure 6 displays the new stability diagram for $N=0$ to 60 with $C=6.11\times10^5$ N/m.

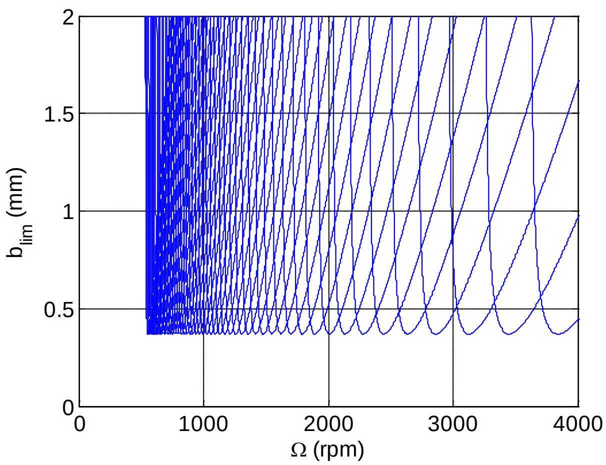

Figure 4. Stability diagram for single degree of freedom model from Fig. 3 with $\alpha=0$, $k=6.48\times10^6$ N/m, $m=0.561$ kg, $c=145$ N-s/m, $K_s=2927\times10^6$ N/m^2, $\beta=61.79$ deg, $d=0.035$ m, and $C=0$.

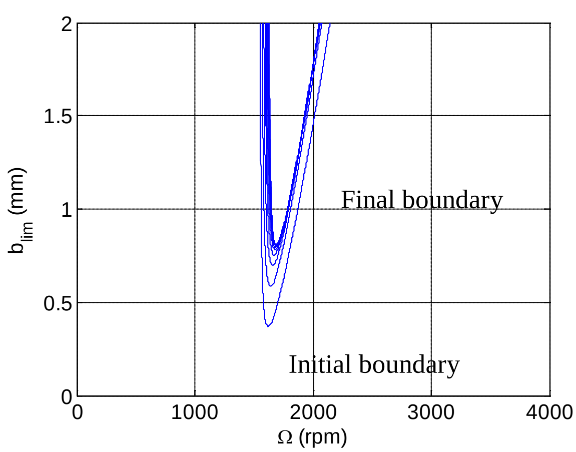

Figure 5. Convergence demonstration ($N=20$, 10 iterations) for single degree of freedom model from Fig. 3 with $\alpha=0$, $k=6.48\times10^6$ N/m, $m=0.561$ kg, $c=145$ N-s/m, $K_s=2927\times10^6$ N/m^2, $\beta=61.79$ deg, $d=0.035$ m, and $C=6.11\times10^5$ N/m.

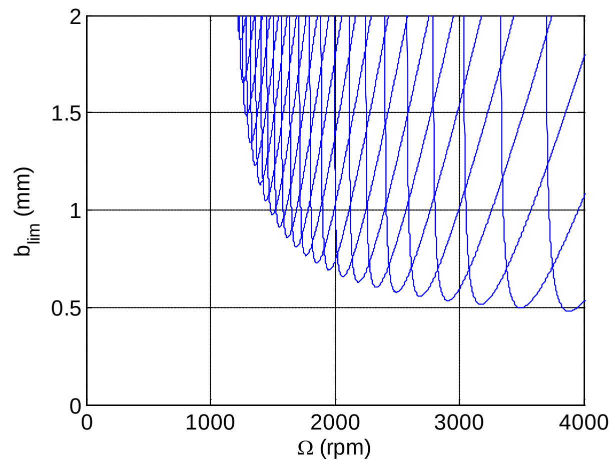

Figure 6. Stability diagram for single degree of freedom model from Fig. 3 with $\alpha=0$, $k=6.48\times10^6$ N/m, $m=0.561$ kg, $c=145$ N-s/m, $K_s=2927\times10^6$ N/m^2, $\beta=61.79$ deg, $d=0.035$ m, and $C=6.11\times10^5$ N/m.

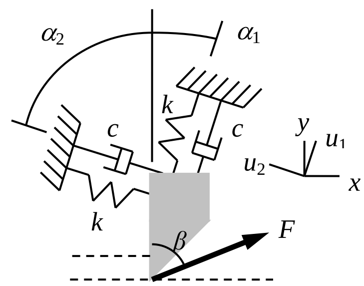

Figure 7. Two degree of freedom turning model.

Two Degree of Freedom Turning

The process damping model can be extended to consider vibration modes in two orthogonal directions as shown in Fig. 7. The analysis procedure is similar, but there are now two new damping values to be calculated: $c_{new,1}=c_1+C\frac{b}{V}\cos^2(\alpha_1)$ for the u_1 direction and $c_{new,2}=c_2+C\frac{b}{V}\cos^2(\alpha_2)$ for the u_2 direction. These two damping values are used to update the G_{u_1} and G_{u_2} frequency response functions; see Eq. 12. The oriented frequency response function for this case is $G_{or}=\cos(\beta-\alpha_1)\cos(\alpha_1)G_{u_1}+\cos(\beta+\alpha_2)\cos(\alpha_2)G_{u_2}$.

Milling

Tlusty modified the previously described turning analysis to accommodate the milling process [3]. A primary obstacle to defining an analytical solution for milling stability (aside from the inherent time delay) is the time dependence of the cutting force direction. Tlusty solved this problem by assuming an average angle of the tooth in the cut, ϕ_{ave}, and, therefore, an average force direction. This produced an autonomous, or time invariant, system. He then made use of directional orientation factors, μ_x and μ_y, to first project this force into the x and y mode directions and, second, project these results onto the surface normal (in the direction of ϕ_{ave}). The new $b_{\lim}$ and Ω expressions for milling are provided in Eqs. 13 and 14, where N_t is the number of teeth on the cutter and N_t^* is the average number of teeth in the cut; see Eq. 15, where ϕ_s and ϕ_e (deg) are the start and exit angles defined by the radial depth of cut. The ε equation remains the same as before.

$$b_{\lim}=\frac{-1}{2K_s\,\mathrm{Re}[G_{or}]N_t^*} \qquad (13)$$

$$\frac{f_c}{\Omega N_t}=N+\frac{\varepsilon}{2\pi} \qquad (14)$$

$$N_t^*=\frac{\varphi_e-\varphi_s}{\frac{360}{N_t}} \qquad (15)$$

Up Milling. The process damping force model defined in Eq. 2 was again applied, but the surface normal direction now depends on ϕ_{ave}. The geometry is shown in Fig. 8a, where n is the surface normal direction. The projection of the process damping force from the n direction onto the x direction is:

$$F_x = F_d\cos(90-\phi_{ave}) = -\left(C\frac{b}{V}\cos(90-\phi_{ave})\right)\dot{n}\,. \qquad (16)$$

Note that the velocity term is now $\dot{n}$. Substituting $\dot{n}=\cos(90-\phi_{ave})\dot{x}$ in Eq. 16 gives:

$$F_x=-\left(C\frac{b}{V}\cos^2(90-\phi_{ave})\right)\dot{x}\,. \qquad (17)$$

The new damping in the converging stability calculation for the x direction frequency response function, G_x, is therefore:

$$c_{new,x}=c_x+C\frac{b}{V}\cos^2(90-\phi_{ave})\,. \qquad (18)$$

The new y direction damping is:

$$c_{new,y}=c_y+C\frac{b}{V}\cos^2(180-\phi_{ave})\,. \qquad (19)$$

The oriented frequency response function for this case is $G_{or}=\mu_x G_x+\mu_y G_y$, where $\mu_x=\cos(\beta-(90-\phi_{ave}))\cos(90-\phi_{ave})$ and $\mu_y=\cos(180-\phi_{ave}-\beta)\cos(180-\phi_{ave})$.

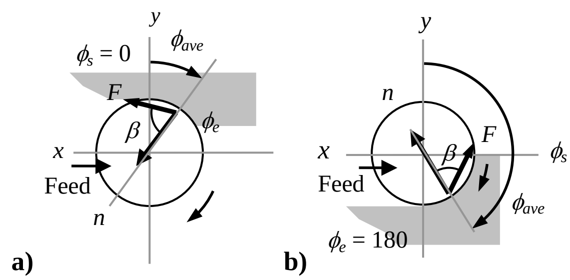

Figure 8. a) Geometry for up milling using the average tooth angle stability analysis (a 25% radial immersion cut is shown for illustrative purposes). b) Model for down milling (a 50% radial immersion cut is shown). The vector n defines the average surface normal direction.

Down Milling. The geometry for the down milling case is shown in Fig. 8b. Using the same approach as described in the up milling case, the x and y direction damping values are provided in Eqs. 20 and 21.

$$c_{new,x}=c_x+C\frac{b}{V}\cos^2(\phi_{ave}-90) \qquad (20)$$

$$c_{new,y}=c_y+C\frac{b}{V}\cos^2(180-\phi_{ave}) \qquad (21)$$

The oriented frequency response function for this case is $G_{or}=\mu_x G_x+\mu_y G_y$, where $\mu_x=\cos(\beta+\phi_{ave}-90)\cos(\phi_{ave}-90)$, and $\mu_y=\cos(\beta-(180-\phi_{ave}))\cos(180-\phi_{ave})$.

COMPARISON WITH SIMULATION

Both analytical analyses and time-domain simulations were completed to determine the process damping effects on turning and milling stability. The time-domain milling simulations were based on the 'Regenerative Force, Dynamic Deflection Model' described by Smith and Tlusty [23] where the damping force was included directly in the numerical integration of the system equations of motion; the simulation details are provided in [24]. The results are compared in the following sections.

Single Degree of Freedom Turning

The model shown in Fig. 3 was considered. A comparison between the analytical stability lobes and time-domain simulation results are provided in Fig. 9. In this case $\alpha=0$, $k=6.48\times10^6$ N/m, $m=0.561$ kg, and $c=145$ N-s/m, $K_s=2927\times10^6$ N/m^2, $\beta=61.8$ deg, and $C=6.11\times10^5$ N/m. The workpiece diameter was 35 mm. In all instances, the time-domain results agree with the analytical stability limit.

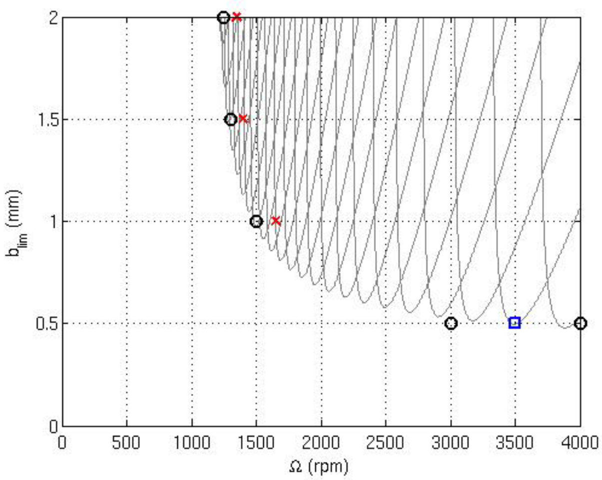

Figure 9. Single degree of freedom turning model with $\alpha=0$. The time domain simulation results are identified as: (circle) stable; (cross) unstable; and (square) marginally stable.

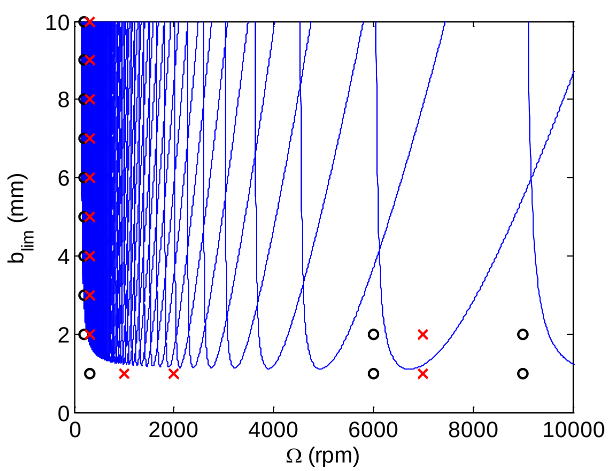

Figure 10. Up milling results for 50% radial immersion. The time domain simulation results are identified as: (circle) stable and (cross) unstable.

Up Milling

The model parameters for the three-tooth cutter were: $\phi_s=0$ and $\phi_e=90$ deg (50% radial immersion), the dynamics in both the *x* and *y* directions were described by $k=9\times10^6$ N/m, $f_n=900$ Hz, and $\zeta=0.03$, and the force constants were $K_s=2000\times10^6$ N/m^2, $\beta=70$ deg, and $C=2\times10^4$ N/m. The comparison between the analytical stability lobes and time-domain simulation is provided in Fig. 10. The results match well.

EXPERIMENTAL IDENTIFICATION OF PROCESS DAMPING COEFFICIENT

The process damping coefficient, *C*, for a selected tool-workpiece pair was identified through a series of cutting tests. A single-tooth indexable end mill was used to mill AISI 1018 steel workpieces secured to the top of single degree-of-freedom leaf-type flexure. The stability limit was identified over a grid of axial depths of cut and spindle speeds. Using the experimental stability boundary, the process damping coefficient was identified. The effects of insert relief angle and tool wear were examined. The flexure dynamics were also adjusted to determine the sensitivity of the process damping coefficient to changes in the system dynamics.

Setup Description

In order provide convenient control of the system dynamics, a single degree-of-freedom, parallelogram leaf-type flexure was constructed to provide a flexible foundation for individual AISI 1018 steel workpieces; see Fig 11. Because the flexure compliance was much higher than the tool-holder-spindle-machine, the stability analysis was completed using only the flexure's dynamic properties. A radial immersion of 50% and a feed per tooth of 0.05 mm/tooth was used for all conventional (up) milling tests.

In order to observe the sensitivity of the process damping coefficient to changes in the system dynamics, mass was added to the flexure in order to reduce the natural frequency; the added mass decreased the natural frequency by approximately 32%.The modal parameters for both cases are provided in Table 1. The *x* and *y* directions correspond to the flexible and stiff directions of the flexure, respectively, where *y* is the feed direction.

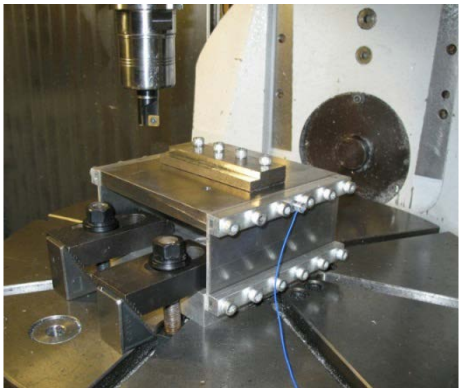

Figure 11. Setup for milling stability tests. An accelerometer was used to measure the vibration signal during cutting.

Table 1. Modal parameters for flexure with and without added mass.

	Direction	Viscous damping ratio	Modal stiffness (MN/m)	Natural frequency (Hz)
No mass	*x*	0.063	2.77	228
	y	0.037	174	1482
Added mass	*x*	0.018	4.37	156
	y	0.028	276	1137

An accelerometer (PCB Piezotronics model 352B10) was used to measure the vibration during cutting. The frequency content of the accelerometer signal was used in combination with the machined surface finish to establish stable/unstable performance, i.e., cuts that exhibited significant frequency content at the flexure's *x* direction natural frequency, rather than the tooth passing frequency, were considered to be unstable.

In order to study the influence of relief angle under milling conditions, two single-tooth indexable square end mills of similar diameter were used: 1) 18.54 mm diameter with a 15 deg relief angle (Kennametal model KICR-0.73-SD3-033.3C); and 2) 19.05 mm diameter with an 11 deg relief angle (Cutting Tool Technologies model DRM-03). Both cutting tools had a 0 deg rake angle and the inserts had no edge preparation.

The cutting force coefficients were identified under stable cutting conditions using a cutting force dynamometer (Kistler model 9257B). For the 18.54 mm diameter cutter, the specific cutting force, K_s, and cutting force direction, β, were determined to be 2359.1 N/mm^2 and 63.5 deg, respectively. For the 19.05 mm diameter cutter, the values were 2531.0 N/mm^2 and 62.0 deg. A linear regression to the mean cutting force over a series of tests at various feed per tooth values was used to identify the cutting force model values [24].

Process Damping Coefficient Identification

Conventional linear stability analysis (i.e., C = 0 N/m) was first used to validate the stability behavior at higher speeds for the flexure setup. As seen in Fig. 12, the predicted behavior was observed experimentally. Additionally, the critical limiting chip width, $b_{\text{lim,cr}}$, was identified to be approximately 1 mm for the 228 Hz setup; this result also agreed with the analytical prediction. A similar approach was used to validate the stability boundary for the 156 Hz setup. The critical stability limit was approximately 0.4 mm for this system; see Fig. 13.

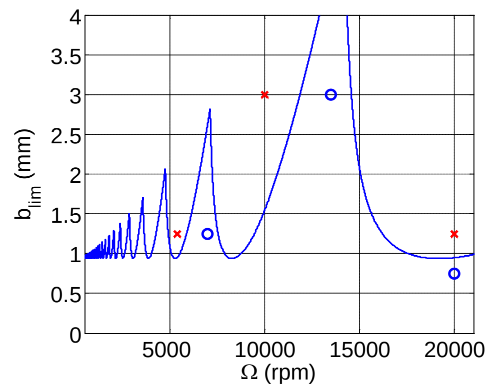

Figure 12. Stability lobe validation for the 228 Hz setup.

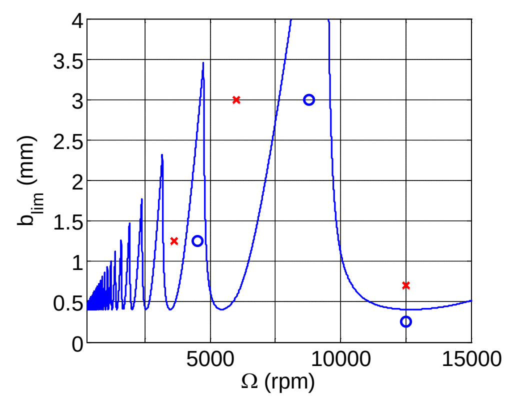

Figure 13. Stability lobe validation for the 156 Hz setup.

A grid of test points at low spindle speed was next selected to investigate the process damping behavior. Based on the stable/unstable cutting test results, a single variable residual sum of squares (*RSS*) estimation was applied to identify the process damping coefficient that best represented the experimental stability boundary; see Fig. 14. The spindle-speed dependent experimental stability limit, b_i, was selected to be the midpoint between the stable and unstable points at the selected spindle speed. The sum of squares of residuals is given by Eq. 20, where $f(\Omega_i)$ is the analytical stability boundary and n is the number of test points. A range of process damping coefficients was selected and the *RSS* value was calculated for each corresponding stability limit. The C value that corresponded to the minimum *RSS* value was selected to identify the final stability boundary for all test conditions.

$$RSS = \sum_{i=1}^{n} \left(b_i - f(\Omega_i)\right)^2 \qquad (20)$$

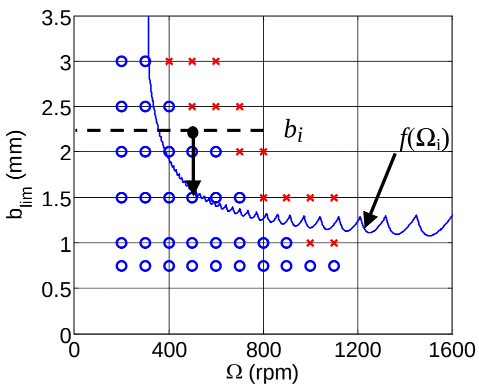

Figure 14. Description of variables for *RSS* estimate of process damping coefficient.

Using the minimum *RSS* method, stability testing was performed for the 18.54 mm diameter, 15 deg relief angle end mill. Because flank wear can affect the process damping behavior, the flank wear width (FWW) was limited to less than 100 µm for these tests. A process damping coefficient of C = 2.5×10^5 N/m was found to best fit the data for the 228 Hz system (50% radial immersion up milling). The corresponding stability boundary is provided in Fig. 15. The procedure was repeated for the 156 Hz setup and a process damping coefficient of C = 2.6 × 10^5 N/m was identified. These results are displayed in Fig. 16.

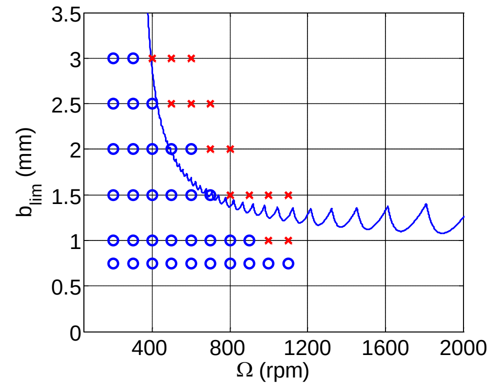

Figure 15. Up milling stability boundary for 50% radial immersion, 15 deg relief angle, low wear milling tests using the 228 Hz flexure setup (C = 2.5×10^5 N/m).

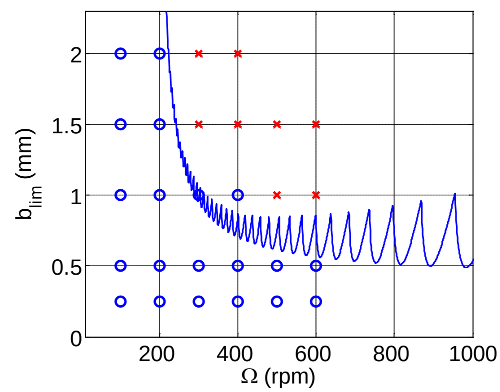

Figure 16. Up milling stability boundary for 50% radial immersion**,** 15 deg relief angle, low wear milling tests using the 156 Hz flexure setup (C = 2.6×10^5 N/m).

Tests were then performed using the 19.05 mm diameter, 11 deg relief angle end mill. The same procedure was following and the FWW was again limited to be less than 100 µm for all cuts. The process damping coefficient for both the 228 Hz and 156 Hz setups was 3.3×10^5 N/m. See Figs. 17 and 18.

The low wear stability test results are summarized in Table 2. The process damping coefficient for the 228 Hz setup increased by 32% for the 11 deg relief angle tool relative to the 15 deg relief angle tool. A 27% increase was observed for the 156 Hz setup.

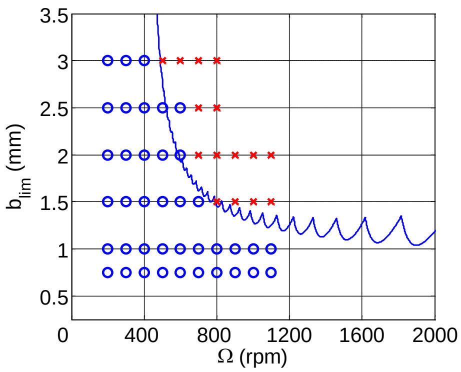

Figure 17. Up milling stability boundary for 50% radial immersion**,** 11 deg relief angle, low wear milling tests using the 228 Hz flexure setup ($C = 3.3\times10^5$ N/m).

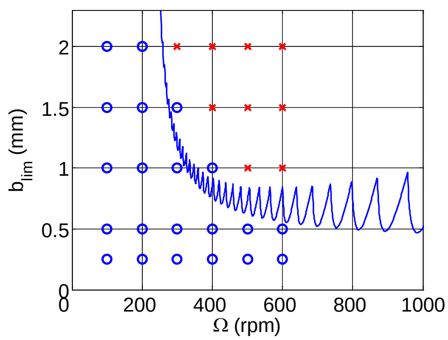

Figure 18. Up milling stability boundary for 50% radial immersion**,** 11 deg relief angle, low wear milling tests using the 156 Hz flexure setup ($C = 3.3\times10^5$ N/m).

Table 2. Comparison of process damping coefficients for low wear tests.

Relief angle (deg)	C (N/m) for the 228 Hz setup	C (N/m) for the 156 Hz setup
15	2.5×10^5	2.6×10^5
11	3.3×10^5	3.3×10^5

Tool Wear Effects on Process Damping Coefficient

In order to explore the effect of tool wear on the process damping performance, tests were completed using worn tools where the FWW was maintained at a level of 200 μm. For the 15 deg relief angle tool, the specific cutting force and cutting force direction were 2441.0 N/mm^2 and 63.5 deg, respectively; this represents a 3.5% increase in the specific cutting force relative to the unworn tool tests. However, the process damping coefficient was found to increase from the unworn tool tests by 20% for the 228 Hz setup and 31% for the 156 Hz setup. Similarly, for the 11 deg cutter, the cutting force parameters experienced only a slight change (K_s = 2550.2 N/mm^2 and β = 62.0 deg). However, the process damping coefficient increased by 15.2% for both flexure setups.

Table 3. Comparison of process damping coefficients for moderate wear tests.

Relief angle (deg)	C (N/m) for the 228 Hz setup	C (N/m) for the 156 Hz setup
15	3.0×10^5	3.4×10^5
11	4.0×10^5	3.8×10^5

Repeatability

Repeat testing was performed using the 19.05 mm diameter, 11 deg relief angle cutting tool in order to observe the variability in the process damping coefficient. A series of three additional cutting tests were performed on the 228 Hz system with an unworn insert. The three process damping coefficients were: 3.3×10^5 N/m, 3.3×10^5 N/m, and 2.9×10^5 N/m. Assuming a normal distribution, a two-sided 90% confidence level was computed for this small sample size. The confidence interval for the population mean was: $C = (3.2 \pm 0.15)\times10^5$ N/m. Figure 19 illustrates the corresponding confidence region.

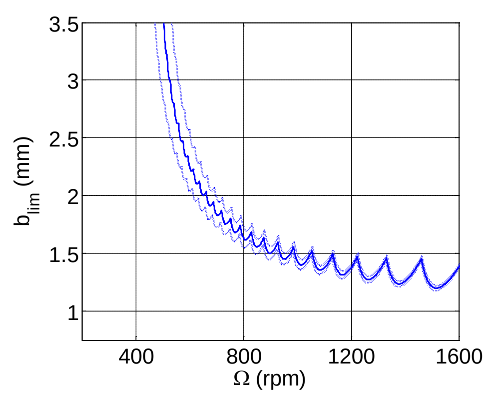

Figure 19. Up milling stability confidence region for 50% radial immersion**,** 11 deg relief angle milling tests using the 228 Hz flexure setup with an unworn cutting edge ($C = (3.2 \pm 0.15) \times 10^5$ N/m).

CONCLUSIONS

An analytical solution for machining stability while considering process damping was provided, where the process damping model relied on a single coefficient, C.

This is analogous to the specific cutting force, K_s, approach to modeling cutting force. Stability testing was completed using a single degree-of freedom flexure to identify the process damping coefficient for low-speed milling of AISI 1018 steel under various conditions. It was demonstrated that a reduction in the relief angle and an increase in flank wear of the cutting edge increased the process damping coefficient.

Process damping is particularly important for hard-to-machine materials, such as titanium, nickel super alloys, and hardened steels. In these instances, tool wear generally prohibits higher surface speeds and the use of the large stable zones available at high spindle speeds. This limits the spindle speed to low values, which decreases the material removal rate. However, by exploiting process damping, higher stable axial depths and material removal rates can be achieved.

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