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Investigation of process damping effect for multi-mode milling systems

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Abstract

Process damping acts as a significant cause of increased stability in milling particularly at low cutting speeds, which has been studied only for single-mode systems in the literature. Chatter frequency, which depends on the component causing chatter, strongly influences process damping coefficient, which is expected to vary with modes of the system. In this paper, the effect of process damping on chatter stability is investigated considering multi-mode dynamics of the system. The process damping coefficients are simulated for the fundamental chatter frequency of each significant mode and then used in the stability solution in frequency domain. An iterative milling stability solution is used as the process damping coefficients depend on the cutting depth. The stability lobe diagram is constructed with respect to multiple mode characteristics of the system. The theoretical predictions are verified through representative experimental cases and the results are discussed.

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1. Introduction

Milling is a commonly used metal cutting process in manufacturing industry. Post-milling operations are required if the desired roughness of machined surface or dimensional accuracy is not achieved, which are mostly influenced by chatter vibration. Thus, prediction of chatter-free cutting conditions is of great importance in machining industry.

Tobias [1] studied the regenerative mechanism and represented it as a function of depth of cut and spindle speed using the stability lobe diagrams. In another early work, Tlustý [2] proposed a stability formulation for end-milling. He adapted the turning formulation to end-milling by taking average number of cutting edges in cut per revolution. Later, Minis and Yanushevsky [3] determined the stability limits using Nyquist stability criterion. The first comprehensive analytical method to predict stable cutting depth and its relation to the spindle speed in end-milling was proposed by Altintas and Budak [4]. They developed zeroth-order approximation method and showed its efficiency in obtaining

stability lobes in frequency domain. Tang et al. [5] presented a stability prediction method for high-speed finishing end milling considering multi-mode dynamics. When the system has multiple dominant modes, the stability behavior varies due to interaction among modes [6]. In a recent research, Wan et al. [7] studied the milling system stability with multiple dominant modes. It was theoretically proved that the stability border for a multi-mode system can be effectively predicted by the lowest envelop of the stability lobes constructed for each single mode separately.

Difficult to machine materials such as nickel, titanium, and stainless steel alloys need to be cut at relatively low cutting speeds, causing substantial decrease in material removal rates. However, this may be compensated by the increased stability limits at low cutting speeds with the effect of process damping [8]. Process damping mainly arises from the interaction between the flank face of the cutting edge and the undulations left on the workpiece surface.

The effect of process damping on stability has been studied by several authors in the literature. In an early study, Sisson

and Kegg [9] tried to find an explanation for chatter behavior at low speeds which is consistent with published experimental observations. Das and Tobias [10] introduced a velocity term into the equations of motion to mimic the process damping effect leading to increased stability limits. Later, there have been significant efforts on modelling of process damping effect considering the tool-workpiece interaction [8]. Chiou et al. [11] developed a model in which process damping force was assumed to be proportional to volume of deformed material beneath the flank face of the tool. In another work, Altintas et al. [12] presented a cutting force model including three dynamic cutting force coefficients related to regenerative chip thickness, velocity and acceleration terms. They used Nyquist criterion to solve stability of the dynamic process. Huang and Wang [13] extended cutting force model to include the effect of process damping and investigated process damping mechanism through time domain simulations. In a recent work, Tunc and Budak proposed a practical process damping identification method for milling using inverse stability solution approach [14]. In this method, the process damping coefficients are identified from experimental stability limits and are further used in identification of the indentation constant and modeling of process damping. They showed that, once the indentation constant is identified, stability limit for different tool geometries and cutting conditions can be predicted, as well. In another identification method [15], the process damping coefficient was predicted from frequency domain decomposition of vibration signal in stable cutting region.

In the literature, the process damping models considered only one dominant mode. However, contribution of multiple modes at distinct natural frequencies may lead to different outcomes as process damping is considered. The effects of fundamental vibration frequency, vibration amplitude, cutting conditions and tool geometry on process damping have been discussed by Tunc and Budak [14]. It was concluded that the process damping effect significantly diminishes at low frequency modes due to the decreased tool-workpiece interaction. This is an important conclusion for multi-mode milling systems, where multiple modes having comparable amplitudes may exist. In such cases, while the expected chatter at higher frequency mode is suppressed by process damping, the low frequency mode may cause chatter, as it may not be suppressed. In this study, process damping effect on stability limits for a multi-mode milling system has been investigated. It is shown that if multiple modes are ignored in prediction of process damping, the stability limits at low frequencies are not predicted accurately. In the experiments, it was observed that after a critical cutting speed region, the chatter mode might shift to the lower frequency mode contrary to theoretical expectation. Henceforth, the paper is organized as follows; the dynamics and stability of milling system with process damping is briefly given in section 2. Then the estimation of process damping coefficients is explained in section 3, which is followed by constructing stability lobe diagrams for multi-mode systems in section 4. The experimental study is presented in section 5, and conclusions are given in section 6.

Nomenclature

m	modal mass
k	modal stiffness
c^s	structural damping coefficient
c^p	process damping coefficient
a_{lim}	stability limit
Λ_I	real part of the eigenvalue
Λ_R	imaginary part of the eigenvalue
N	number of cutting tool teeth
K^d	indentation constant
$U(t)$	indentation area

2. Dynamics and stability of milling with process damping

The stability of the multi-mode milling system is considered for each mode separately, where stability diagram of each mode with the effect of process damping is simulated. This approach is valid if the modes of the system are well-separated [7]. In this section, the dynamics of single-mode milling system with the process damping term is briefly presented. The equations of motion of milling system are followed by the frequency domain solution of the stability limits [4].

2.1. Equation of Motion

The cross section of a helical end mill, which is flexible in x and y directions with N number of cutting flutes is illustrated in Fig. 1. As the cutter rotates, the cutting tooth indents into the wave left on the surface by the previous tooth. Correspondingly, indentation forces arise in tangential and radial directions on the tool flank face, creating a damping effect. For this system, the equations of motion with the effect of process damping can be written in x and y directions as:

$$\begin{aligned} m_{x,i}\ddot{x} + c_{x,i}^t\dot{x} + k_{x,i}x &= F_x; & c_{x,i}^t &= c_{x,i}^s + c_{x,i}^p \\ m_{y,i}\ddot{y} + c_{y,i}^t\dot{y} + k_{y,i}y &= F_y; & c_{y,i}^t &= c_{y,i}^s + c_{y,i}^p \end{aligned} \quad i = 1, \dots, N_t \quad (1)$$

Where m , c^s and k are the modal mass, structural damping, and stiffness of the system, and c^p indicates the average process damping coefficient in each direction, respectively. N_t stands for the number of dominant modes of the system.

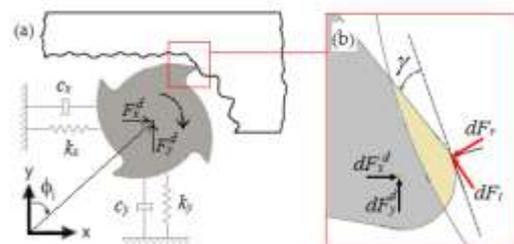


Fig. 1. Dynamic milling with process damping. (a) cross section of a helical end mill, (b) flank-workpiece interaction.

2.2. Milling stability with process damping

After organizing the equations of motion in the frequency domain and solving the eigenvalue problem [4], the stable

axial depth of cut can be obtained in terms of the frequency response function (FRF) of the system as follow:

$$a_{lim} = -\frac{2\pi\Lambda_R}{NK_t} \left(1 + \left(\frac{\Lambda_R}{\Lambda_I} \right)^2 \right) \quad (2)$$

In the above equation, K_t , N and T are the tangential cutting force coefficient, number of cutting edges, and the tooth passing period, respectively. Λ_I and Λ_R are the real and imaginary parts of the eigenvalue, which is written in terms of the FRFs [4], where the effect of the process damping term is introduced in the stability solution.

3. Simulation of process damping coefficient

In a previous study [14], the average process damping coefficients were determined through inverse stability solution, assuming that process damping is the only cause for the difference between experimentally obtained stability limit a_{lim}^{exp} and analytically calculated stability limit a_{lim}^{cal} , which is verified at high cutting speeds. The experimentally determined process damping coefficients were used to identify the indentation coefficient K^d through damping energy analysis, to simulate process damping coefficients for different cases.

3.1. Damping energy analysis

The damping forces arising due to the flank face – workpiece indentation acts against the vibration direction when the tool is moving down the undulation, leading to an additional damping effect (see Fig. 1b).

The average process damping coefficients, in x and y directions, are defined through energy balance analysis. For such a purpose, the vibration energy dissipated by the average process-damping coefficients is equated to the energy dissipated by the indentation forces over one tool rotation period, T_{sp} as illustrated in Fig. 2.

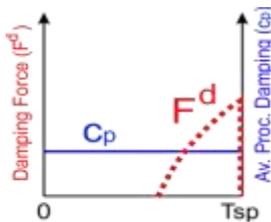


Fig. 2. Damping energy balance analysis.

The additional process damping coefficient at the expected chatter frequency, ω_c is derived as follows:

$$\frac{\int_0^{T_{sp}} F_i^d(t) \dot{u} dt}{\int_0^{T_{sp}} \dot{u}^2 dt}, i = x, y \quad (3)$$

where, $u = A \sin \omega_c t$

The time varying indentation forces, $F_i^d(t)$, acting on the tool in x , and y directions are calculated by orienting the

indentation forces in chip thickness, $F_r^d(t)$, and $F_t^d(t)$, tangential directions, which are written as function of the indentation volume, $U(t)$, and the indentation constant as follow:

$$\begin{aligned} F_r^d(t) &= K^d U(t) dz \\ F_t^d(t) &= \mu F_r^d(t) \end{aligned} \quad (4)$$

In equation (4), $U(t)$ is the indentation volume, which is calculated according to the model given in [14] and μ is the friction coefficient.

4. Process damping in multi-mode milling systems

The dynamic milling system consists of several components such as machine tool axis carriers, spindle, tool holder and the cutting tool, each of which introduces dynamic flexibility in a unique frequency range. The vibration frequency significantly affects the process damping as it arises from the indentation between the tool flank face and the undulations left on the workpiece surface. The effect of vibration frequency on the process damping coefficients was previously emphasized by Tunc and Budak [16]. The variation of cutting depth specific average process damping coefficient with the vibration frequency is shown in Fig. 3, where it is seen that as the vibration frequency decreases the amount of process damping reduces substantially.

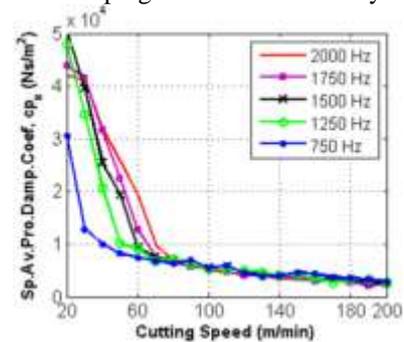


Fig. 3. Effect of vibration frequency on average process damping coefficients [16].

In dynamic milling, chatter is expected to occur at a single mode. For a milling system having multiple dominant modes at distinct frequencies, even though the higher frequency mode is suppressed by process damping, the lower frequency mode may not be suppressed as the process damping is smaller at the lower vibration frequency. Thus, theoretically, the chatter frequency may shift to low or high frequency range depending on the amount of process damping acting on each mode. On the other hand, the vibration frequency dependent process damping may cause the milling system experience higher frequency mode vibration, while it is chattering at lower frequency mode.

4.1. Constructing the multi-mode stability lobes with process damping

The amount of process damping acting on a vibrating mode depends on the cutting depth. Thus, the stability limit at a spindle speed can be calculated in an iterative manner as

proposed by Tunc and Budak [14]. Although it is a simplification of the nonlinear effect of process damping on stability, within the scope of this study, the stability lobes are calculated separately for each dominant mode with the process damping effect. Then, the lowest envelop of the stability lobes due to all dominant modes is taken as the absolute stability border. However, alternative solution approaches should be further investigated.

A representative stability diagram for down milling of AL7075, including process damping is given in Fig. 4. For this simulation, half immersion down milling and force coefficients of $K_{tc}=1600$ MPa and $K_{rc}=600$ MPa are considered. The tool is an 18 mm diameter, four fluted, solid carbide end mill. The transfer function of the system used in the experiments (section 5), which is shown in Fig. 5, is also used for this representative case. The solid lines show the stability lobes due to the two dominant modes, i.e. at low frequency and high frequency, when process damping is ignored. It is seen that, the absolute stability limit is governed by the high frequency mode if the process damping is not considered. However, as the process damping is considered, the absolute stability line of both modes shifts up, where they cross-cut each other at point B. As a result, the high frequency mode governs the absolute stability limit from point A to point B. Then, from point B on, the low frequency mode governs the stability limit. This is due to the fact that, the amount of process damping introduced by the low frequency mode is not enough to increase stability at that vibration frequency. As a result, the absolute stability limit corresponding the low frequency mode fails to shift up as much as the high frequency mode.

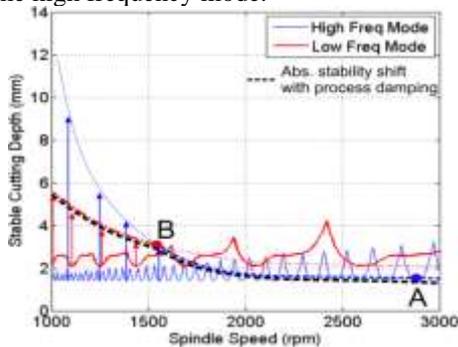


Fig. 4. Change of the mode governing the absolute stability.

5. Experimental investigation

In this section, the effect of process damping on the dynamics of the multi-mode milling system is experimentally investigated to verify the simulation results given in the previous section.

5.1. FRF measurements

All the experiments were conducted on DECKEL MAHO 5-axis milling center using an 18 mm diameter, four fluted, solid carbide end mill. The tool was clamped to the holder SK40 ER32C 160G. The tool length was kept at 60 mm. The frequency response function (FRF) at the tool tip in x and y directions were measured through impact hammer test as

shown in Fig. 5. It is seen that there are two dominant modes with close amplitudes located around 630 Hz and 3350 Hz, belonging to the tool holder and the tool, respectively. The modal parameters of these modes are given in Table 1. The workpiece was rigid compared to the tool.

Table 1. Modal parameters of the system

	k_x (N/m)	f_x (Hz)	ζ_x (%)	k_y (N/m)	f_y (Hz)	ζ_y (%)
1 st mode	4.6e7	636	1.8	2.0e7	634	5.4
2 nd mode	3.4e7	3352	1.5	4.4e7	3353	1.1

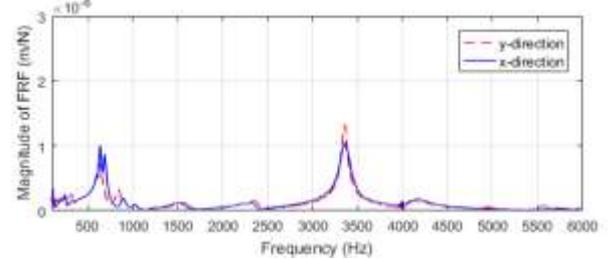
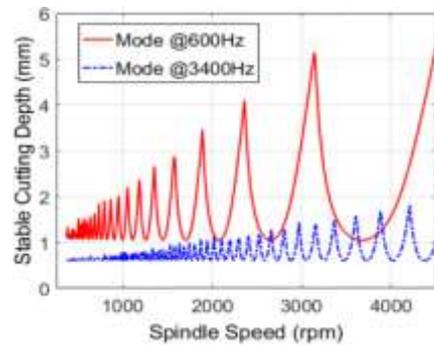


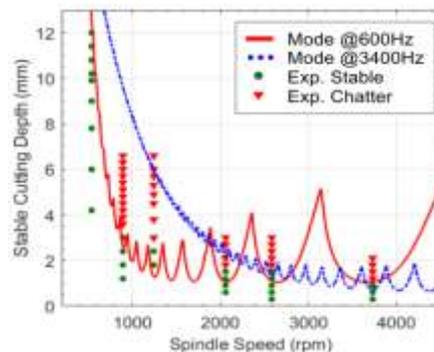
Fig. 5. Frequency response functions at the tool-tip.

5.2. Stability simulations

The stability diagrams are generated for 50% radial immersion, down milling of AISI1050 material, where the tangential and radial cutting force coefficients were taken as $K_{tc}=1600$ MPa and $K_{rc}=600$ MPa. Indentation constant K_d was $70,000$ N/mm³ as reported in [14]. In order to select the spindle speeds according to the both modes, the stability diagrams are calculated with and without the process damping effect as shown in Fig. 6.



(a) without process damping



(b) with process damping

Fig. 6. Stability lobes (a) without process damping (b) with process damping.

The mode around 600 Hz leads to absolute stability limit of 1.1mm, whereas the mode around 3400 Hz leads to absolute stability limit of 0.6mm if the process damping is ignored. In Fig. 6a, it is seen that when the effect of process damping is ignored, the tool mode (3400 Hz) governs the stability throughout the whole spindle speed range. However, when the process damping is considered, one of the two modes determines the stability limit depending on the amount of process damping.

5.3. Cutting tests

In the cutting tests the spindle speeds are selected such that the effect of process damping on the absolute stability due to both modes can be observed (see Fig. 6a). The cutting test conditions are given in Table 2. The feed rate was set to 0.05 mm/rev/tooth. In order to perform the cutting tests effectively, the workpiece was designed as a staggered part having stepped height increments of 1/3 of the absolute stability limit (see Fig. 7a). The length of each level was selected as 1.2 times of the tool diameter to provide enough time for chatter to develop. After each cutting level, the machine was stopped to let the tool stabilize before the next cutting level. The experiment setup is shown in Fig. 7b.

Table 2. Cutting tests conditions.

Test number	Spindle speed (rpm)	Cutting speed (m/min)
1	3730	211
2	2585	146
3	2060	116
4	1245	70
5	895	51
6	540	31

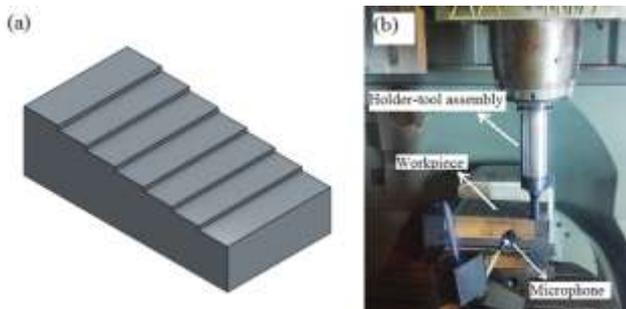


Fig. 7. Cutting experiments. (a) designed workpiece, (b) experiment setup

When chatter develops, the surface quality will significantly deteriorate due to chatter marks and there will be a significant increase in the amplitude of the sound [17]. Thus in this work, to identify chatter, the cutting sound was recorded by a directional microphone and the cut surfaces were visually examined after each test. In Fig. 8, the representative cases extracted from test series 3 (2060 rpm) are shown to demonstrate the chatter detection. In Fig. 8a, although very light marks due to tool vibration are visible, the feed marks can be seen. Also the sound spectrum shows no significant spikes around the natural frequencies of the system, except the harmonics of the tooth passing frequency. Thus, such a case is considered as stable. In Fig. 8b, the sound spectrum indicates a significant increase around the tool mode

and the surface quality is also extremely deteriorated by chatter marks where it is really difficult to identify the feed marks. It means that energy dissipation due to process damping is no longer sufficient to suppress the regenerative component of the vibration and its amplitude has started growing.

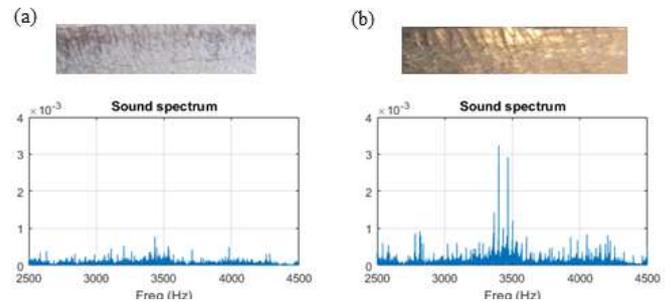
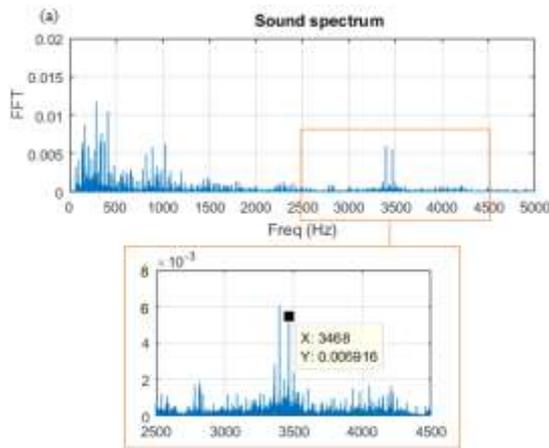


Fig. 8. Representative cases for chatter identification (at 2060 rpm). (a) stable case at depth of 1.5 mm, (b) unstable case at depth of 2.1 mm

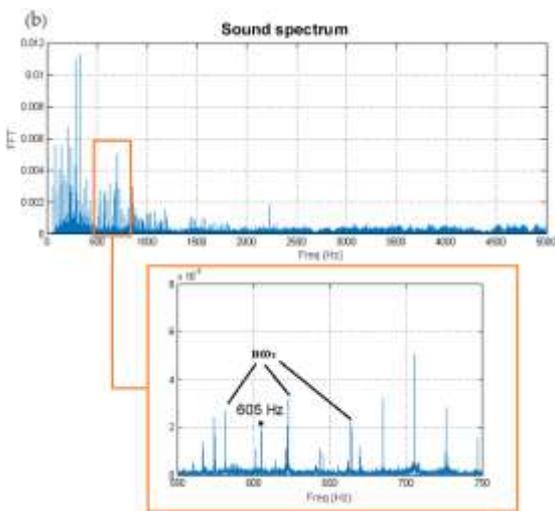
In Fig. 6 it is seen that the stability limits significantly increase with the effect of process damping, especially at speeds lower than 2500 rpm. The stability limits at high cutting speeds were also verified. The tool mode governs the stability limits for the spindle speed range higher than 2000 rpm, while the tool holder mode becomes active for the spindle speeds less than 2000 rpm, where the tool mode is suppressed by process damping acting on the tool mode. Consequently, if any vibration starts to arise around the tool mode it is expected to be suppressed and the chatter frequency would shift to the tool holder mode. The experimentally identified stability limits are plotted on top of the stability lobes in Fig. 6b. Although, there are some discrepancies at the transition regions within a spindle speed range of 2000 to 2500 rpm, the experimental results are in a good agreement with the predictions. According to the experimental results, the sound spectrum shows an increase in terms of amplitude at 3468 Hz (see Fig. 9a) at cutting speeds higher than 1500 rpm (85 m/min). This implies that chatter occurred around the tool mode. However, at a lower cutting speed of 70 m/min (1245 rpm), chatter frequency was observed at 605 Hz (see Fig. 9b) which is around the tool holder mode. Thus, as expected, at low cutting speeds, the tool mode has almost been damped out completely due to high process damping effect of the high frequency vibrations around 3400 Hz. However, the mode around 605 Hz was not suppressed as the process damping was not enough to stabilize the cut at such high cutting depths. This reveals the fact that, although the stability lobe diagram simulated for the high frequency mode of the tool offers high process damping effect, chatter can occur due to lower frequency modes of system. Note that in Fig. 9a and Fig. 9b, the other peaks except the chatter frequency are the harmonics of tooth passing frequency (ω_T).

6. Conclusions

Process damping is highly influenced by chatter frequency, according to the modes of the milling system. In this study, process damping effect was investigated for



(a) sound spectrum 2060rpm, 2mm



(b) sound spectrum 1245 rpm, 3 mm

Fig. 9: Sound spectrum of representative tests.

a multi-mode milling system. The theoretical simulations were verified by cutting experiments. The effect of process damping on the responsible chatter modes was demonstrated. In general, the absolute stability limit is expected to be governed by the most flexible mode. However, it was shown that under the effect of process damping, the mode governing the absolute stability may shift to other modes. In a certain cutting speed zone, due to the high process damping caused by the high-frequency modes of the milling system, chatter may develop at the low-frequency modes at a depth of cut lower than the predicted one considering only the high frequency mode. This is mainly due to the fact that the amount of process damping generated by low frequency modes is not enough to stabilize the process. Such information may be used to identify the cross cutting point, where the mode governing the absolute stability shifts from high frequency mode to the low frequency mode with the effect of process damping. Such a point would show the cutting speed after which the absolute stability limit may not further increase with decreasing cutting speed.

The experimental results showed that the absolute stability limit may be governed by the low frequency modes rather

than the high frequency modes at low cutting speeds with the effect of process damping. Under such circumstances if the amount of process damping introduced by the lower frequency mode is not enough to stabilize chatter vibration at that mode, it will limit the chatter-free material removal rate, leading to less productivity. This may be considered as a typical case when a heavy duty milling operation is aimed to be performed at a machine tool with flexible column, axis carriages, spindle or tool holder, which introduces relatively low frequency modes.

Although some discrepancies were observed between the experimental results and the calculated stability limits, the general behaviour of the system under the effect of process damping was well predicted. The discrepancies reveal that the current model for prediction of stability limits with process damping needs further improvements for the multi-mode milling systems, which is considered to be a future work.

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