Countermeasure Against Regenerative and Forced Chatter of Flexible Workpieces in Milling Processs Using Bi-directional Excitation

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(Received 5 December 2017; revised 14 January 2018; accepted 20 January 2018)

Abstract: A novel extended methodology for chatter suppression in milling process by applying external forced vibrations to the workpiece in two orthogonal directions which are the feed and cross-feed directions. Both the regenerative and forced chatter suppression during the milling process of flexible workpieces are investigated. Here, the workpiece is subject to a sinusoidal periodic force in the feed direction to disrupt the regenerative effect. Additionally, to minimize the forced chatter, the workpiece is subject to the periodic excitation force in cross-feed direction. This force is proportional to the magnitude of the estimated cutting force in cross-feed direction and has a phase opposite to the cutting force to minimize the vibration amplitudes. The effectiveness of the proposed method is evaluated numerically and experimentally, for the spindle speed located in both the local minima and local maxima of the stability lobe diagram. The numerical simulations indicate significant suppression effect in terms of vibration amplitudes, resulting in suppression of both the regenerative chatter and the forced chatter. Experiments were conducted by using a workpiece-mounted active stage composed of flexure hinges and driven by piezoelectric actuators. The experimental results agree qualitatively with the numerical simulations. The proposed method indicates a remarkable vibration reduction effect for both regenerative and forced chatters.

Key words: self-excited vibration; vibration control; regenerative chatter; forced chatter; milling

CLC number: TN925 **Document code**: A **Article ID**: 1005-1120(2018)01-0010-10

0 Introduction

Chatter is an undesired instability phenomenon that occurs during the machining process which drastically reduces machining efficiency, limits obtaining the required surface quality and adversely affects tool life. Since the late 1950s, prediction of chatter has been a main research interest. Tobias[1] introduced the "stability lobes diagram (SLD)", where the axial depth of cut is plotted against the spindle speed, representing the threshold of the chatter, or the stability limit. The concept was to predict the two optimal parameters of stable machining in advance, to achieve the maximum "material removal rate (MRR)". In 1995, Altintas et al.[2] proposed a semi-analytical frequency domain method for the prediction of chatter in milling, which for most cases is a fast estimation.

Prior investigations[3,4] mainly focused on chatter suppression of the tool, instead of the workpiece, as chatter easily occurs on the tool when manufacturing sufficiently stiff workpieces. However, recently, manufacture of lighter and more flexible components is in high demand, especially by the aeronautical and aerospace industries. Machining of such components and thin-walled structures such as turbine blades and impellers generally causes chatter in the workpiece instead of the tool. These flexible workpieces

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How to cite this article: Nakano Yutaka, De Silva Amal Dilanka, Takahara Hiroki, et al. Countermeasure against regenerative and forced chatter of flexible workpieces in milling process using bi-directional excitation[J]. Trans. Nanjing Univ. Aero. Astro., 2018,35(1):10-19.

http://dx.doi.org/10.16356/j.1005-1120.2018.01.010

typically have lower stability limits (resulting in lower MRR), which makes it difficult to avoid chatter only by using SLDs. Therefore, it is necessary to take countermeasures against the chatter of the workpiece and consequently increase the stability limit.

There are several chatter suppression approaches for machining of flexible workpieces. There are passive strategies[3-6] such as application of a variable helix tool[7] and a variable pitch cutter[8], tuned mass dampers[9]. However, in the case of machining of thin-walled flexible workpieces, vibrational characteristics can rapidly vary with material removal from the workpiece. Therefore, active strategies[10-13] are more popular and industrially appealing approaches for machining of flexible workpieces[14]. Prior work[15-17] by Sallese et al. has investigated chatter suppression by implementing an actuation stage (also referred to as an "active workpiece holder (AWH)" or "active fixture holder (AFH)") controlled by piezoelectric actuators. The investigations were mainly focused on tool chatter.

Further, the resonance due to the intermittent cutting forces acting on the workpiece, can easily cause forced chatter on the workpiece. Therefore, countermeasures should be taken to suppress not only the regenerative chatter, but also the forced chatter.

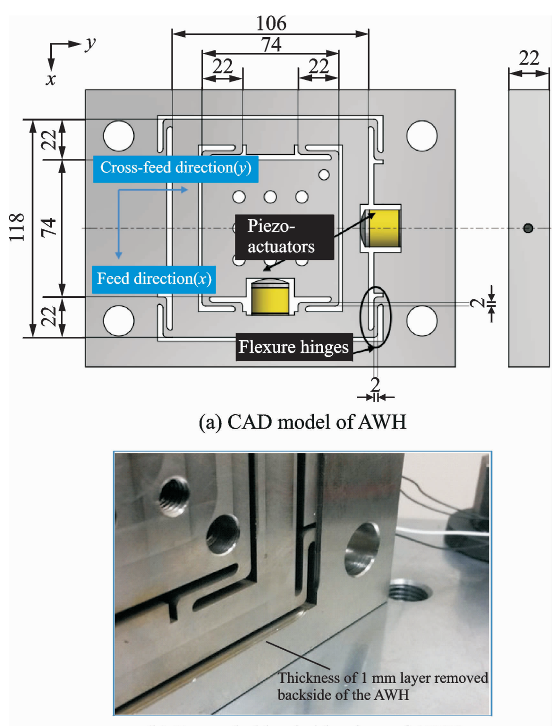
The regenerative and forced chatter suppression of flexible workpieces by exciting the workpiece in two orthogonal directions with the application of AWH focus on the up-cut milling process as chatter vibration can be easily arisen. Although, the vibrational characteristics can vary due to the material removal from the workpiece, the present investigation was conducted where the change of the vibrational characteristics is negligible. The method can be applied even when the vibrational characteristics are changing, if the vibrational characteristics can be identified and consequently optimizing the excitation parameters in real time. Additionally, the results are compared with the conventional one-directional excitation method reported by the authors[18]. Our findings suggest that the proposed excitation method has a remarkable vibration reduction effect for both the regenerative chatter and forced chatter vibrations.

1 The Proposed Excitation Method

In this section, AWH which is designed to execute bi-directional excitations, is briefly explained. In the analysis, the workpiece, AWH system, is modelled as a lumped-mass model. Consequently, the governing equations of motion required for the numerical simulations are derived. Further, the modal parameters are identified and the proposed excitation method is described.

1.1 Active workpiece holder design

The literature reports different mechanical designs for the AWH design which allow two directional excitations[19]. The monolithic nested architecture designed with flexure hinges as shown in Fig. 1(a) is more appealing, because it theoretically allows the AWH to have decoupled motion in two orthogonal directions[17]. A thin layer of thickness of 1 mm was removed from the backside of the AWH as shown in Fig. 1(b), permitting the intermediate stages of the AWH to



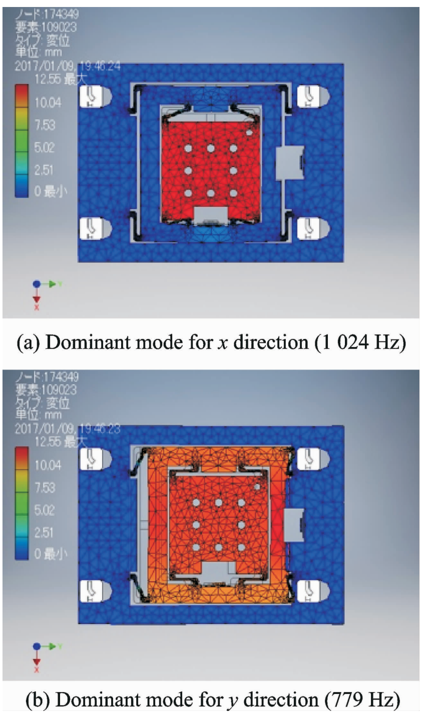
(a) CAD model of AWH

(b) Expanded backside view of AWH

Fig. 1 Active workpiece holder[20]

move freely without any frictional interference. Hereinafter, we set the coordinate system, such that x represents the feed direction and y represents the cross-feed direction.

The stage is made from stainless steel (SUS304). Figs. 2 (a, b) show the dominant modes for the AWH by FEM analysis. We obtained 1 024 Hz along the feed direction (x direction) and 779 Hz along the cross-feed direction (y direction), as the dominant modes without the workpiece mounted.



(a) Dominant mode for x direction (1 024 Hz)

(b) Dominant mode for y direction (779 Hz)

Fig. 2 Dominant modes obtained by FEM analysis for the AWH

Two ring stack piezo actuators (NAC2125-H14) are mounted to the AWH. Our calculations predict that each piezoelectric actuator can produce about 10 μm strain at the maximum allowable voltage 150 V generated by the piezo drivers (Mess-Tek M26107).

The AWH fixed to the machining center (Hitachi-Seiki VK45) is shown in Fig. 3. The workpiece was intentionally made more flexible than the designed AWH in the cross-feed direction by mounting the workpiece on two leaf springs. Thus, the chatter vibration occurs easily. The end mill (OSG EDS6) with a diameter of 6 mm, 2 flutes and a helix angle of 30° was utilized. Free-cutting brass C3713P was used as the workpiece material for easy identification of the chatter marks.

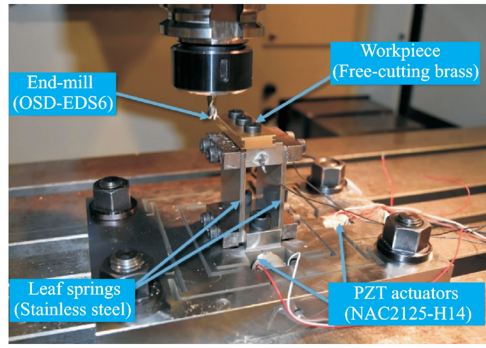


Fig. 3 AWH with workpiece fixed to the machining center

1.2 Equation of motion

The workpiece-AWH system can be modelled as two degrees of freedom subsystems in each direction (i. e. feed direction x and cross-feed direction y) as shown in Fig. 4. For the sake of simplicity, the AWH is assumed to be a lumped mass system. The AWH is capable of producing movements in x and y directions independently. Thus, the equations of motion can be derived as

$$\boldsymbol{M}\ddot{\boldsymbol{X}} + \boldsymbol{C}\dot{\boldsymbol{X}} + \boldsymbol{K}\boldsymbol{X} = \boldsymbol{f} \tag{1}$$

where

$$\begin{gathered}
\boldsymbol{M} = \mathrm{diag}(m_{xs}, m_{xw}, m_{ys}, m_{yw}) \\
\boldsymbol{f} = [F_{ex}, F_x, F_{ey}, F_y]^{\mathrm{T}} \\
\boldsymbol{C} = \begin{bmatrix} c_{xs} + c_{xw} & -c_{xw} & 0 & 0 \\ -c_{xw} & c_{xw} & 0 & 0 \\ 0 & 0 & c_{ys} + c_{yw} & -c_{yw} \\ 0 & 0 & -c_{yw} & c_{yw} \end{bmatrix} \\
\boldsymbol{K} = \begin{bmatrix} k_{xs} + k_{xw} & -k_{xw} & 0 & 0 \\ -k_{xw} & k_{xw} & 0 & 0 \\ 0 & 0 & k_{ys} + k_{yw} & -k_{yw} \\ 0 & 0 & -k_{yw} & k_{yw} \end{bmatrix}
\end{gathered} \tag{2}$$

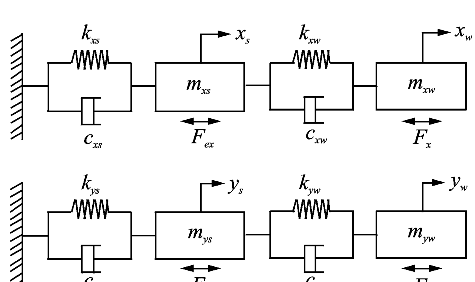


Fig. 4 Simplified model of the workpiece-AWH model

Here, the modal mass, the modal damping coefficient and the modal stiffness are denoted by m, c, k, respectively. Additionally, the subscript notations "xs, xw, ys, yw" imply the x direction of the stage (AWH), x direction of the workpiece, y direction of the stage (AWH), y direction of the workpiece, respectively. F_{ex} and F_{ey} are the external excitation forces and F_x, F_y are the cutting forces in x and y directions, respectively. Further, x_s, x_w are the coordinates of the stage and workpiece in x direction, and y_s, y_w are the coordinates of the stage and workpiece in y direction, respectively.

1.3 Hammering test

Hammering test on a workpiece mounted on the AWH was performed to identify the modal parameters of the system. The AWH was fixed to the machining table of the machining center. Then, the frequency response functions of the stage and the workpiece were obtained. Finally, curve fitting was performed using Matlab. Consequently, the modal parameters were identified (Table 1). Additionally, the hammering test was performed for the tool and the first mode natural frequency of the tool was obtained as 11.27 kHz. Thus, the vibrations in the tool can be assumed to be negligible.

Table 1 Identified modal parameters by the hammering test

Parameter	Value
m_{xs}/kg	1.730
m_{xw}/kg	0.159
m_{ys}/kg	0.934
m_{yw}/kg	0.214
ω_{xs}/(rad · s^{-1})	2π×963
ω_{xw}/(rad · s^{-1})	2π×1 334
ω_{ys}/(rad · s^{-1})	2π×940
ω_{yw}/(rad · s^{-1})	2π×420
ζ_{xs}/%	0.73
ζ_{xw}/%	2.11
ζ_{ys}/%	0.35
ζ_{yw}/%	1.37

1.4 Excitation method

The cutting forces (F_x, F_y) in Eq. (1) are estimated by considering multi-regenerative effect and air-cutting conditions by a tool run-out[18]. In this subsection, the external forced excitations (F_{ex}, F_{ey}) are mathematically expressed. In the feed direction, the workpiece is subject to sinusoidal periodic force in order to disrupt the regenerative effect[18]. The intermittent cutting forces acting on the workpiece causes forced chatter, thus resulting in resonance. It is necessary to minimize not only the regenerative chatter but also the forced chatter. Thus, to minimize forced chatter effect, an excitation force with the magnitude of the estimated cutting force (F_y^s) is applied in the opposite direction of the cross-feed direction (i.e. y direction). It should be noted that this cutting force is merely an estimation, assuming the chip generation is purely by the static chip thickness and the dynamic chip thickness is neglected

$$F_{ex} = A_e \sin(2\pi f_e t + \psi) \tag{3}$$

$$F_{ey} = -K_{gain} F_y^s \tag{4}$$

where

$$F_y^s = \sum_{j=1}^{N} \{-F_{uj}^s \sin(\varphi_j + \alpha) + F_{vj}^s \cos(\varphi_j + \alpha)\}$$
$$F_{uj}^s = K_t a_p s_t \sin(\varphi_j + \alpha) g(\varphi_j) \gamma(\varphi_j), F_{vj}^s = K_r F_{uj}^s \tag{5}$$

where f_e, A_e, ψ are the excitation frequency, the excitation amplitude, phase shift in the feed direction, respectively. K_{gain} is a multiplier. Moreover, in the experiments we adjusted the gain by applying a voltage 100 V closer to the maximum operable values, due to the amplitude limitations of the piezoelectric actuators. Here, the effect of the phase difference of the sinusoidal signal in the feed direction, ψ is not investigated and set to zero.

The coefficients in Eq. (4), K_t, K_r, a_p, s_t are the cutting coefficients in the tangential and the radial directions[2], axial depth of cut, feed rate, respectively. Further, $g(\cdot)$ is a unit step function to evaluate whether the jth cutting edge is engaged with the workpiece or not, $\gamma(\cdot)$ is a unit step function which determines the air cutting of the tool, and φ_j represents the angle of immersion of the tooth j. In this study, a new parameter α is introduced. It denotes the phase shift between the cutting force and the actuation force.

We investigated how the suppression effect deviates with this parameter, and explained it in Chapter 2. Referring to the prior work[15], the actuation frequency (f_e) for the feed direction is determined by

$$f_e = (k + 0.5) f_{tp} \quad k = 0, 1, 2, \cdots \tag{6}$$

where f_{tp} represents the tooth passing frequency. However, in this paper, the optimal actuation frequency is selected to be the closest value to the natural frequency of the stage and it is considered to be independent of the number of flutes of the tool.

2 Numerical Simulation Results

2.1 Time history analysis without excitation

Time history analysis was performed for the equations of motion using Runge-Kutta method[21]. The multiple regenerative effect[18] count was taken up to 4 counts. The simulations are performed with 1 000 intervals per tooth passing period (i. e. $1\,000N\Omega/\pi$; where Ω denotes the spindle speed in rad/s and N denotes the number of flutes).

The stability limit was determined by identifying the chatter frequency component of the Fourier spectrum and using a dedicated algorithm (Fig. 5). The machining process is a linear system for any chatter-free cutting condition. Therefore, unless chatter vibration occurs, the dominant peak of the Fourier spectrum shows a linear increment for any linear increment of the axial depth, at any given spindle speed. Time history analysis is performed for depth of cut at ith step, a_i. Then, FFT is performed for the time history displacement data and the dominant peak P_i is obtained. Consequently, the parameters $\Delta_{2,1}$, $\Delta_{1,0}$ are calculated (i. e. $\Delta_{2,1} = P_{i-2} - P_{i-1}$, $\Delta_{1,0} = P_{i-1} - P_i$) for $i > 2$. If $\Delta_{2,1}$, $\Delta_{1,0}$ are within a preset tolerance range, the system is assumed to be linear. The critical axial depth of cut $a_{\lim}$ is defined as the first axial depth of cut value, which goes beyond the tolerance range. In this paper, the tolerance range was set to $\pm 1\%$. This stability criterion procedure was performed with an increment of an axial depth of cut, $\Delta a = 0.02$ mm

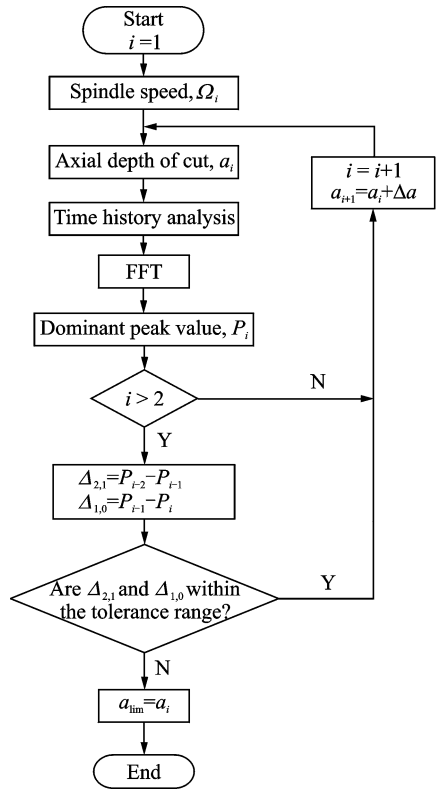


Fig. 5 Flowchart for the algorithm used in determining stability limit from time history analysis

and over the spindle speed range of 1 400—2 600 r/min with an increment of 10 r/min. The numerical simulations were validated by comparing the results with the zeroth order approximation (ZOA) method proposed by Altintas et al.[2,22] and the cutting tests without the excitation. Fig. 6 shows an excellent agreement with ZOA method and the cutting tests.

Nonetheless, the stability lobe diagram obtained by the numerical solutions are deviated at some spindle speed domains, due to the resonance caused by forced chatter.

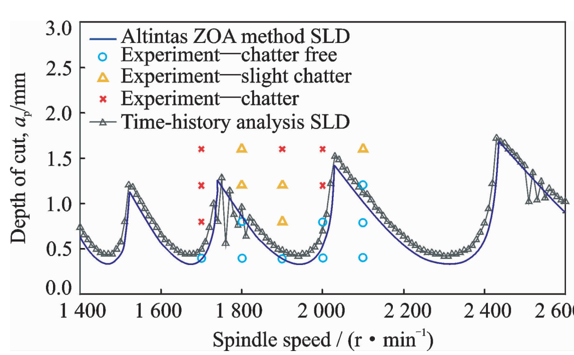


Fig. 6 Stability lobe diagram comparison without excitation for 1 400—2 600 r/min

2.2 Time history analysis with excitation

The numerical simulations were performed with excitation for both the local minima of SLD (Case-A), where the regenerative chatter is dominant (Figs. 7 (a, b)) and the local maxima of SLD (Case-B) where the forced chatter is dominant (Figs. 7(c, d)). The parameters used for the simulation are listed in Table 2.

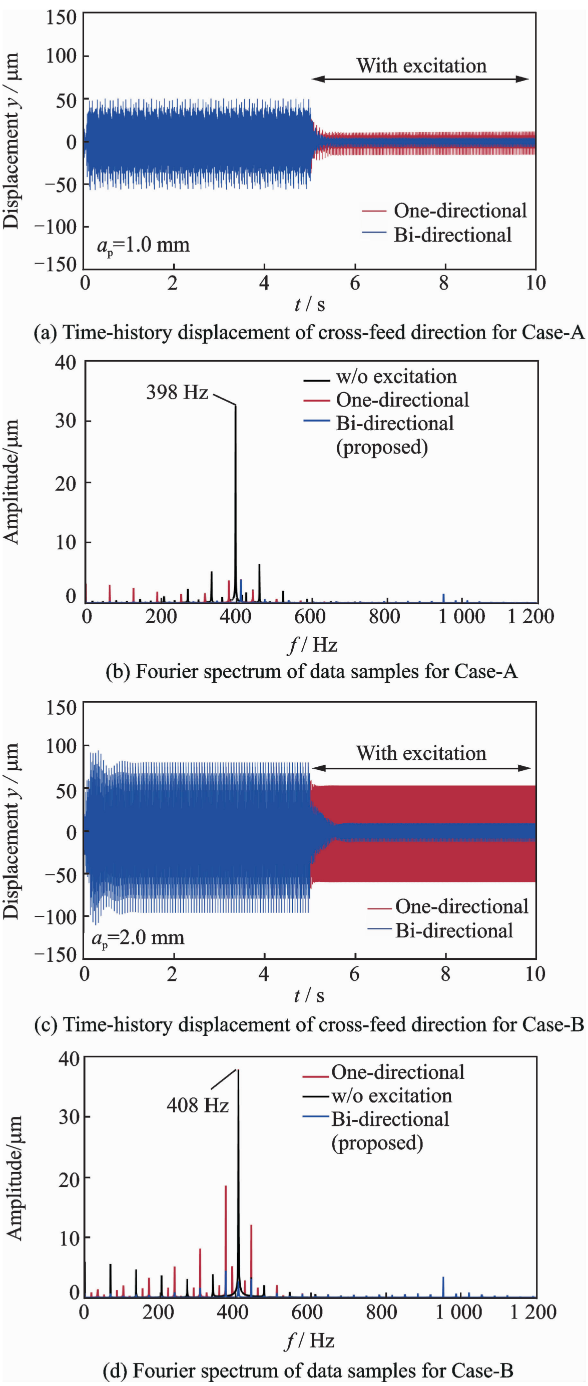


Fig. 7 Simulation results

Fig. 7 shows the vibration displacements of the cross-feed direction and their Fourier spectra

Table 2 Parameters used for numerical simulations

Parameter	Case-A	Case-B
Spindle speed/($r \cdot min^{-1}$)	1 900	2 040
Axial depth of cut a_p/mm	1.0	2.0
Radial depth of cut r_d/mm	1.5	1.5
Excitation frequency f_e/Hz	982	986
Amplitude A_e/N	240	240
Gain K_{gain}	20	20

for a chatter occurring condition at the local minima and local maxima. The results obtained by the proposed bi-directional excitation method (Blue) are compared with the conventional one-directional (i. e. feed direction only) excitation method (Red)[18]. It can be evaluated that the proposed method has a better suppression effect for the local minima of the stability lobe. Furthermore, significantly larger suppression can be observed at the local maxima of the stability lobe, where the forced vibrations acting on the workpiece are dominant. The proposed method has a significant impact on not only regenerative chatter but also the forced vibration suppression which results in a major amplitude reduction (Figs. 7(c, d)). The proposed bi-directional excitation method provides chatter-free machining conditions for both Case-A and Case-B. Although, the same periodic excitation force is applied in the feed direction, the axial depth of cut for the local maxima is two times that of local minima, implying twice MRR in the proposed method for the local maxima.

Further, the vibration amplitude suppression can be verified by the Fourier spectra (Figs. 7 (b, d)). The amplitudes without the excitation (Black), have been reduced by one-directional excitation method (Red) and significantly reduced by the proposed bi-directional excitation method (Blue).

2.3 Effect of phase shift

The phase shift α in Eq. (4) between the cutting force and excitation force in the cross-feed direction can significantly affect the suppression effect, especially for the local maxima. Thus, the relationship between the phase shift and the displacement has been numerically investigated for the local maxima (Fig. 8). The minimum and maximum values (Blue) and the root mean square values (Red) of the displacement in the cross-feed direction, the maximum peak value (Yellow) of the Fourier spectrum of the cross-feed directional displacement after the bi-directional excitation method applied was computed numerically. The aforementioned numerical results were normalized by the displacements without the excitation and plotted over the normalized phase shift, which is the phase shift, α normalized by the pitch angle (i. e. pitch angle for 2 flute end-mill is 180°). The analysis was performed for each 2.5° over one complete pitch angle.

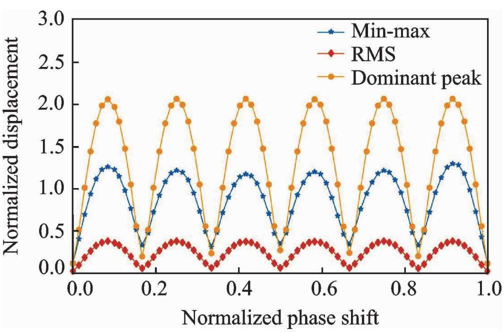


Fig. 8 Relationship between the normalized phase shift and the displacement normalized by displacements without excitation for Case B

The minimum displacement/amplitude or the maximum suppression effect has been achieved at phase shifts, $\alpha = 0°$ and $\alpha = 180°$. Fig. 8 shows that the normalized displacements vary over the phase shift. The proposed method has improved suppression effect for the phase shifts where the normalized displacement is less than 1.

3 Experimental Apparatus and Experimental Results

Fig. 9 shows a schematic representation of the experiment apparatus.

For the signal processing and control purposes, the CompactRIO platform by National Instruments with a maximum clock speed of 400 MHz and powered by an onboard field programming gate array (FPGA) was utilized along with the I/O modules (NI 9411, NI 9263). A laser proximity sensor (Keyence LV-N10) was implemented at the tool holder to estimate the starting angle of immersion, using the rotational pulse signal input. A tri-axial accelerometer (Ono-Sokki NP3572, maximum measurable acceleration = 4 000 m/s^2) was fixed to the stage and two single-axial accelerometers (Ono-Sokki NP2106, maximum measurable acceleration=100 000 m/s^2) were fixed to the jig located just below the workpiece as shown in Fig. 9. The data acquisition was performed by using the FFT Analyzer (Ono-Sokki DS-3200) with an integrated 24 bit resolution A/D converter.

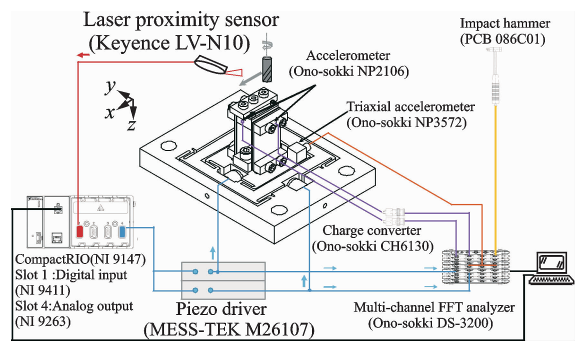


Fig. 9 Schematic representation of the experiment apparatus

Fig. 10 shows the experimental data for the local minima (1 900 r/min) where the axial depth of cut is 2.2 mm. Figs. 10(a,c) show the time-history for the one-directional and the proposed bi-directional excitation methods, respectively. Similarly, Figs. 10(b,d) show the Fourier spectra for the one-directional and the proposed bi-directional excitation methods, respectively. The proposed bi-directional excitation method (Blue) demonstrates larger suppression effect compared to the one-directional method (Red) in the local minima even at an axial depth of cut which is larger than critical depth of cut.

It can be seen before the excitation for both one-directional and the bi-directional excitation methods had similar dominant peak amplitude of nearly 60 μm at 387—389 Hz. With one-dimensional method, the peak was reduced to 25 μm, on the other hand, for the bi-directional excitation method, the dominant peak appears at 382 Hz which is about 5 μm. It should be noted that the 982 Hz (i. e. 63.3 Hz×15.5) was the excitation

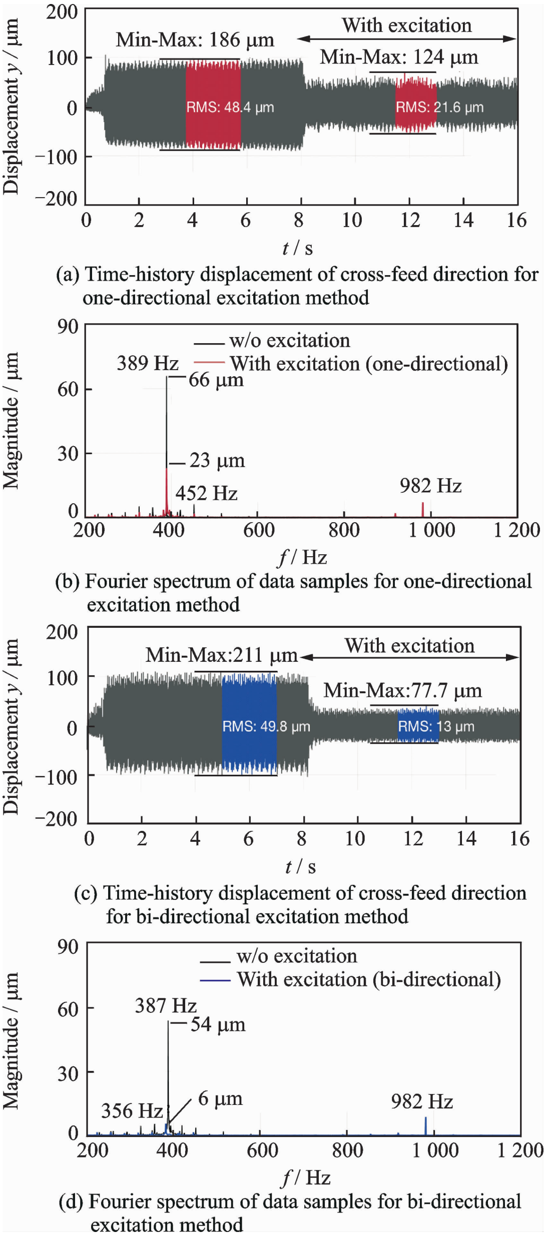


Fig. 10 Experimental results for local-minima ($1\ 900$ r/min, $a_p = 2.2$ mm)

frequency for the feed direction.

Next, in order to evaluate the effectiveness of the forced chatter suppression at the local maxima, axial depth of cut was set to 1. 2 mm, slightly above from the critical depth of cut at 2 040 r/min. Fig. 11 shows experiment data for the local maxima, where the forced chatter is dominant. The subfigure notations, (a), (b), (c), (d) are similar to those of Fig. 10. For both cases, it can be observed that the dominant frequency, 410 Hz is the forced chatter component which represents the natural frequency of the workpiece. The frequency component of 376 Hz is the regenerative chatter frequency. The optimum excitation frequency in the feed direction 986 Hz (i. e. 14. 5 $\times$ 68. 0 Hz) can be seen in Fig. 11.

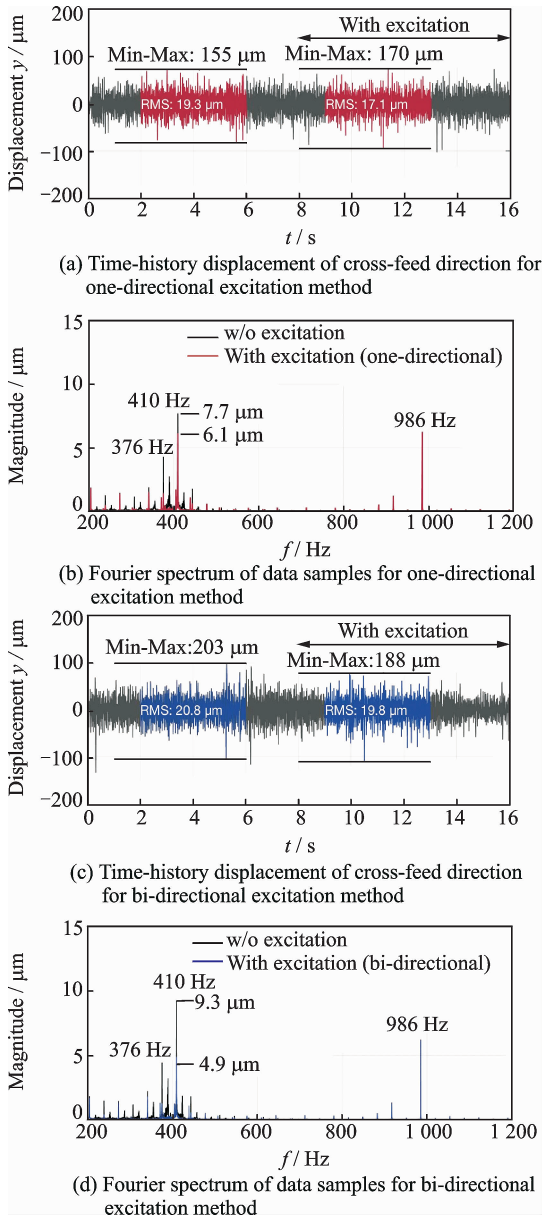


Fig. 11 Experimental results for local-minima ($2\ 040$ r/min, $a_p = 1.2$ mm)

Although, significant suppression of the vibration amplitudes (Figs. 11(a, c)) in the time history results was not visible, the Fourier spectrum for the proposed bi-directional excitation method (Fig. 11(d)) shows a larger suppression effect for the dominant peak at 410Hz, than the Fourier spectrum for the one-directional excitation method (Fig. 11(b)). Our results suggest 20. 7% reduction of the dominant peak for the one-directional excitation method and 47.3% of reduction of the dominant peak for the proposed bi-directional excitation method.

In the analysis, phase shift α was set to zero. However, in the experiment, the phase shift α could not be zero due to the estimation error of the starting angle of the cutting edge immersion estimated by using the rotational pulse signal. The deviation of the phase shift α can affect suppression effect significantly, as explained in Fig. 8. Thus, the experimental accuracy depends on the accuracy of the phase shift estimation.

4 Conclusions

The effectiveness of the regenerative and the forced chatter suppression using bi-directional external excitations applied to the workpiece is investigated. The proposed method is effective for the local minima of the stability lobe, where the regenerative chatter is dominant. Furthermore, the numerical simulations showed a remarkable vibrational amplitude reduction for the local maxima of the stability lobe, where the forced chatter is dominant. Our experiment results also confirmed of a qualitative agreement with the simulation results.

Future work is recognized as achieving higher excitation amplitudes by modifying the AWH. Additionally, it is necessary to perform wider range of experimental investigations for different combinations of spindle speeds and axial depth of cuts, including end-mills with higher number of flutes, for further validation.

Acknowledgement

This work was supported by JSPS KAKENHI (No. JP15K05856).

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(Production Editor: Zhang Tong)