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ORIGINAL ARTICLE

Design of a passive damper with tunable stiffness and its application in thin-walled part milling

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Abstract The optimal parameters of a passive damper (i.e., frequency ratio and damping ratio) for the machining vibration attenuation of thin-walled part are quite affected by the material removal process, and the damper becomes ineffective easily due to its narrow vibration band. A design of passive damper with tunable stiffness is proposed for being adaptive to the varying machining process, and the frequency tuning is carried out by orienting the mass block inside the damper. Modal tests are performed to verify the amplitude reduction of the target mode of the thin-walled part. It shows that the optimal vibration suppression is reached when the orientation is 20° due to a smaller frequency difference between the damper and target mode, and the amplitude of the damped frequency response function is reduced to 1.3 %. Finally, machining tests are carried out, and the machining vibration and surface quality validate the large increase of machining stability. The experimental critical depth of cut under optimal tuning is increased by 1.8-folds compared with the most ineffective tuning, and the machined surface roughness is reduced by more than 80 %.

Keywords Passive damper · Thin-walled part · Milling · Vibration

1 Introduction

Thin-walled structures are widely employed in the aerospace due to their less weight and high strength-weight ratio. During the high percentage material removal process, machining deflection and vibration are easily occurred which decrease the product quality and manufacturing efficiency.

The major methodologies for the part deflection control in the state of art are fixture layout design [1], tool path optimization [2], and cutting parameters selection [3]. The idea is either to improve the workpiece stiffness or reduce the cutting force exerted on the workpiece.

The dynamic machining vibration can be controlled by the machining process optimization and structural dynamics modification (i.e., active and passive approaches). Campa et al. investigated the chatter avoidance in the milling of flexible thin floors with a bull-nose end mill [4]. Wan et al. provided an efficient method to predict the stability lobes of milling systems with multiple modes in the time domain [5]. Herranz et al. proposed a working methodology for high-speed milling of low-rigidity parts in order to avoid static and dynamic problems [6]. Aguirre et al. proposed semi-active control for machine tool chatter suppression based on a damper device with variable stiffness [7]. Rashid et al. developed an active control system according to the piezoelectric actuator and adaptive control algorithm [8]. Lopez de Lacalle et al. utilized active damping device to damp modes of milling machine above 20 % [9]. The active control is effective, but complex hardware and software are required. On the contrary, passive control has the advantages of simple design and easy implementation.

Many scholars have made a wide research on the passive damper from the single DOF to the multi DOFs. Rashid et al. [10] and Duncan et al. [11] studied on the suppression of chatter in cutting process with a single DOF passive damper. Nakano et al. installed three passive dampers on the main shaft of the milling machine and

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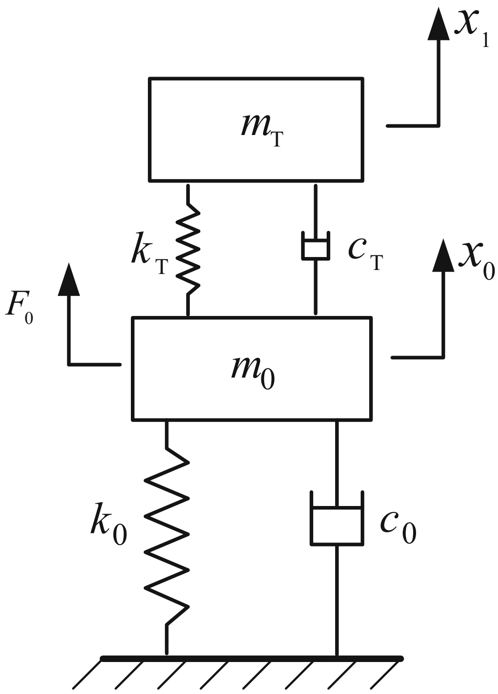


Fig. 1 SDOF passive damper damps the primary structure with single mode

studied the critical flutter stability at different speeds12 [12]. Kolluru et al. mounted six dampers in the thin-walled part around to suppress the vibration of the milling process, and the root mean square value of the vibration signal was decreased by 77 % [13]. Yang et al. investigated milling vibration attenuation of thin-walled part with multiple modes through passive approach, and a two-DOF tuned mass damper incorporated with eddy current damping is designed [14]. Kolluru et al. utilized a passive damping solution to minimize the vibration of thin wall casings while focusing on the change in coupled interaction between tool and workpiece due to added tuned dampers [15].

According to optimization criterion proposed by Den Hartog [16] and Brock [17], the frequency ratio between the passive damper and target mode must fall in certain range otherwise the optimal damping cannot be achieved. As the machining process carried out on a workpiece, its material removal leads to the change of mass and ultimately the vibration frequency then the damper is no longer able to function properly.

In present research, a design of passive damper with tunable stiffness is proposed that can suppress wide range of vibration frequencies without falling out of the optimization criterion. The effective frequency range of this damper can be varied by changing its orientation of mass block as the mass of the target mode decreases. This design then is verified by several experiments of modal test, and the undamped and damped frequency response functions (FRF) of the thin-walled part are investigated. Finally, this damper is tested under different machining tests in which different cutting parameters are implemented.

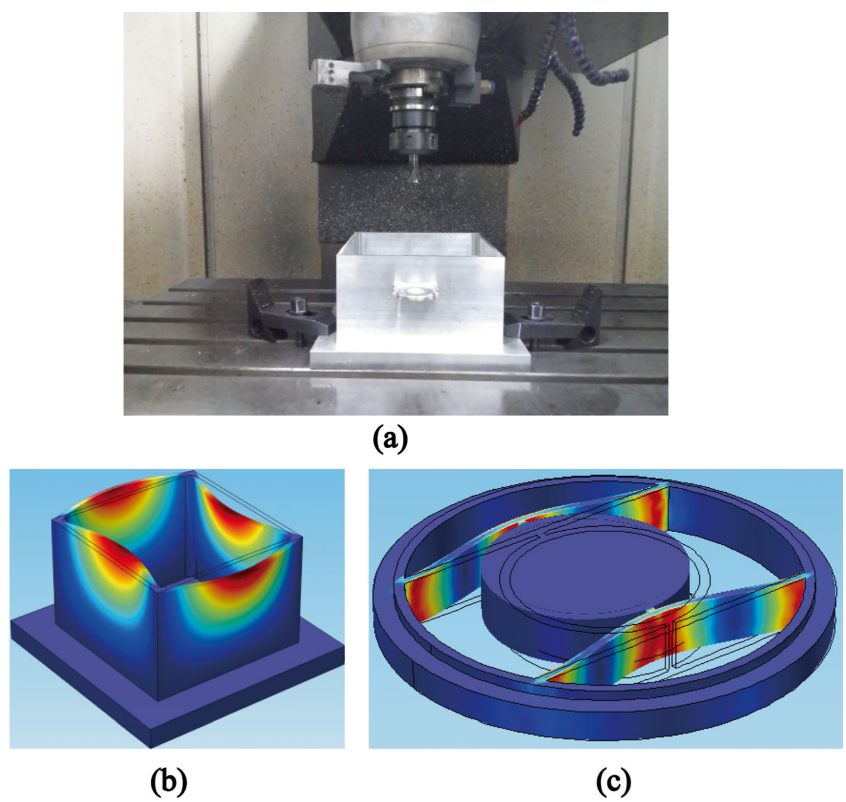


Fig. 2 Experiment setup. **a** Thin-walled part mounted with the damper, **b** dominant mode shape of thin-walled part, and **c** dominant mode shape of the damper

Table 1 Geometry of the thin-walled part

Thickness	Height	Length	Corner radius
5 mm	108 mm	190*190 mm	8 mm

2 Design of the damper

2.1 Design criterion

The primary structure with single mode damped by single degree-of-freedom (DOF) passive damper is shown in Fig. 1. The external force acting on the m_0 is assumed as harmonic force F_0, as the cutting force exerted on the workpiece is periodical. The damper mass, stiffness, and damping are m_T, k_T, and c_T. According to the equal peaks, optimization criterion proposed by Den Hartog [16] and Brock [17], the optimal parameters of the damper is calculated as follows:

$$\text{Frequency ratio}\, \beta_T = \frac{\omega_T}{\omega_0} = \frac{1}{1+\mu} \tag{1}$$

$$\text{Damping ratio}\, \xi_T = \sqrt{\frac{3\mu}{8(1+\mu)^3}} \tag{2}$$

where

ω_T	the natural frequency of the damper m_T
ω_0	the natural frequency of primary structure m_0
$\mu = \frac{m_T}{m_0}$	the mass ratio between the damper and target mode
$\xi_T = \frac{c_T}{2\sqrt{k_T m_T}}$	damping ratio of the damper

Den Hartog and Brock's tuning is accurate for the undamped structure ($c_0 = 0$). Generally, μ is less than 5 % in the industry; therefore, β_T is larger than 0.95. However, the actual optimal frequency ratio is larger than β_T in Eq. (1) considering the existence of damping ($c_0 \neq 0$). For simplicity in the engineering design, the optimal frequency ratio is approximated as 1, which means that the natural frequency of the passive damper is designed the same as the target mode.

During the machining process, the dynamics of the workpiece are altered with the removal of material, especially for the thin-walled part. Therefore, the mass ratio μ is varying and then the optimal frequency ratio β_T. It is necessary to design a damper with tunable stiffness (i.e., natural frequency) to be adaptive to the mass varying process.

A thin-walled part is targeted for damping, and the dominant vibration mode is bending (Fig. 2). Its geometries are shown in Table 1. The dynamic parameters of the thin-walled part are identified for the design of the damper. The accelerometer Kistler 8778A500, impact hammer PCB 086C03, and signal acquisition card NI 9233 are utilized in modal test. The signal processing and analysis software are Cutpro MALTF module. Through the modal test, the first natural frequency of the thin-walled part is obtained as 680 Hz.

2.2 Structural design of the damper

A damper with tunable stiffness is designed as shown in Fig. 3 by referring to the literature [7]. It is composed of the mass block, frame, bolts, and cover plates. The mass block is fixed with the frame by two bolts at the circular edge. According to the mode shape of the target mode, the

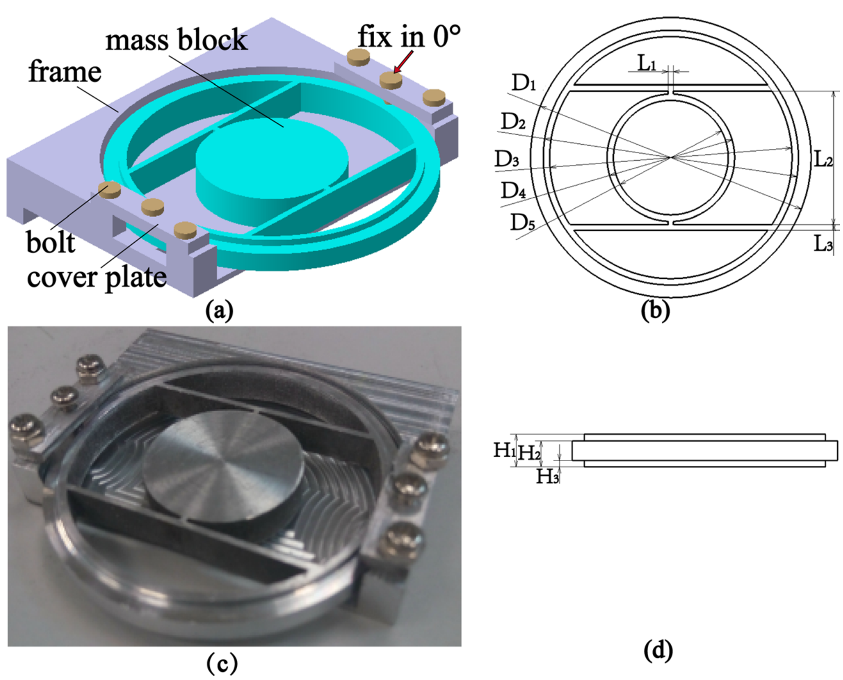


Fig. 3 Passive damper design. **a** Geometrical model when the damper orientation is 0°, **b** top view, **c** the device of damper, **d** and front view

mass block of the damper is designed to have a dominant bending motion as shown in Fig. 2c. The damper is mounted at the middle of the thin-walled part where the vibration amplitude is largest according to the finite element simulation. As the target mode is excited by the milling force, the damper mass starts vibrating to absorb the energy. Depending on the orientation angle of the mass block placed, the equivalent stiffness of the vibration mass is changed that results in the changes of the natural frequency of the damper.

The relative distance L_2 and thickness of the flexible beam L_1 and L_3 are the main factors of determining the stiffness (also frequency) of the damper. According to the geometry of the thin-walled part (Fig. 2a), the geometry of the damper is optimized after dynamics simulation in the software COMSOL (Table 2).

3 Modal tests

3.1 Natural frequency of the damper

The natural frequency of the damper is varied by placing the mass block at different orientations. The FRFs of the damper when the mass block is oriented from 0° to 90° for an interval of 10° are collected (Fig. 4), and the natural frequencies of the dominant mode around 360° are plotted symmetrically (Fig. 5). It is verified that the natural frequency of the damper varies with the orientation of the mass block. The highest natural frequency of 683 Hz is in the direction of 30°, and the lowest natural frequency 670 Hz is in the direction of 70° and 80°. The tunable frequency range of the damper is 13 Hz which shows that the damper has a certain bandwidth, and it can be applied to the varying machining process of thin-walled part.

3.2 The FRF of the workpiece

After gluing the damper on the surface of the thin-walled part (Fig. 2), the damped FRFs of workpiece are measured. Two orientations of the mass block at 20° and 80° are selected and compared with the undamped FRF (Fig. 6). It is seen that the target mode of 680 Hz is split

Table 2 Geometry of the damper

H_1	1 mm	D_4	20 mm
H_2	4 mm	D_5	18 mm
H_3	5 mm	L_1	0.8 mm
D_1	44 mm	L_2	21 mm
D_2	40 mm	L_3	0.9 mm
D_3	38 mm		

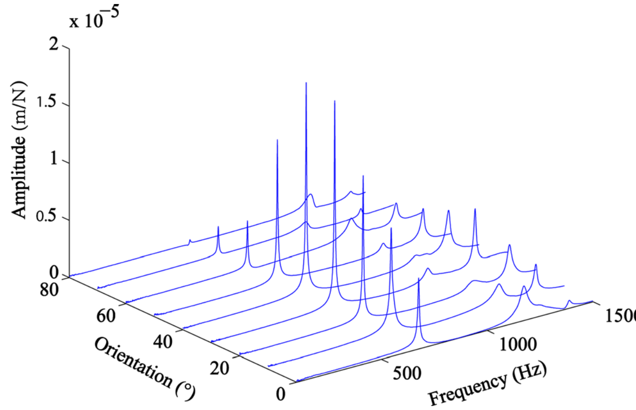


Fig. 4 Experimental FRFs of the damper when the mass block is located at different orientations

into two peaks with the amplitude greatly reduced, and the orientation of 20° achieves better damping.

The natural frequency of the passive damper with orientation of 80° is 670 Hz, and the frequency difference between the damper and the target mode is 10 Hz. The damped natural frequencies of the split modes are 670 and 690 Hz, and the amplitudes of the FRF are 8.8e-6 and 1.7e-7 m/N which are reduced to 9.9 % of the undamped. The damping ratio of the target mode is increased by fivefolds, and the stiffness is increased by 21 folds according to the identified parameters in Table 3 (The data in bold font is selected for comparison). The frequency difference is reduced to 2 Hz when the orientation is 20°. The damped natural frequencies are 640 and 713 Hz, and the amplitude is reduced to 1.3 % of the undamped.

By implementing the passive damper with more orientations, it is found that the damping effect depends on the frequency

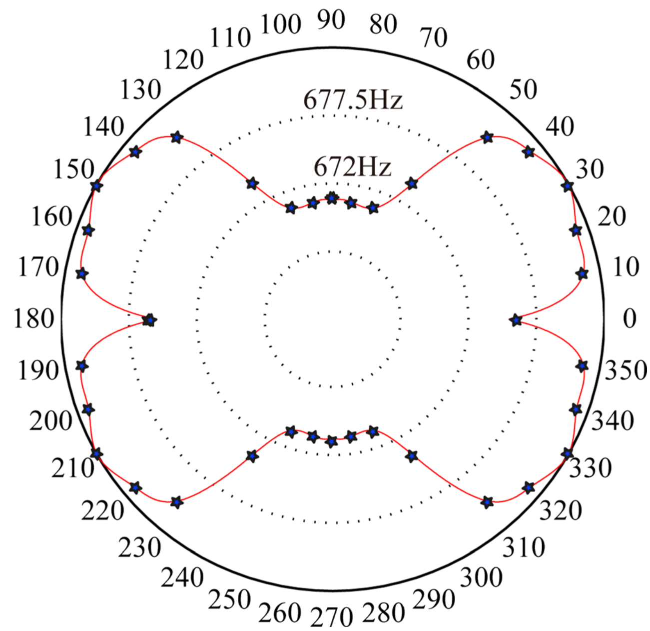


Fig. 5 Experimental natural frequency distribution of the damper when the mass block is located at different orientations

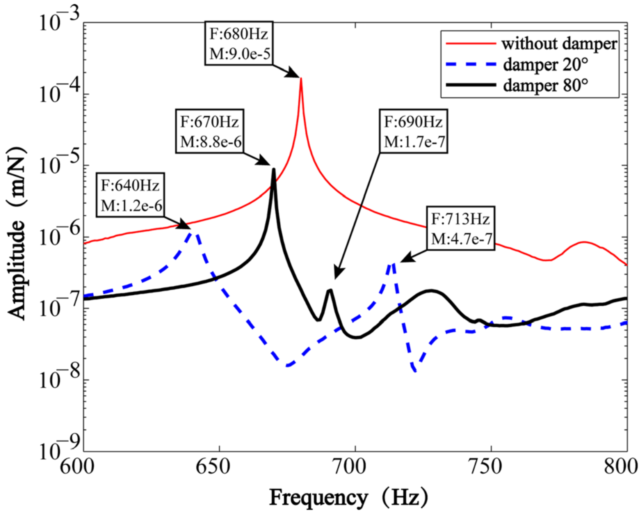


Fig. 6 FRFs of the thin-walled part without and with the damper

difference between the damper and the target mode and the orientation of 80° reaches the minimum damping. According to Fig. 5, the frequency difference of 80° is the largest.

4 Machining experiments

4.1 Bandwidth verification

The modal tests are carried out under series of cutting with depth of cut a_p = 2 mm each time, and the vibration suppression effect is observed. After the first cut, the FRFs with and without the damper in logarithmic scale are shown in Fig. 7a. It is seen that the undamped first natural frequency of thin-walled part is 702 Hz, and the corresponding amplitude of FRF is 9.17e-6 m/N. After damping, the FRF has higher peak with the amplitude of 1.87e^{-7} m/N when the mass block of the damper is adjusted to 90° and the corresponding frequency is 697 Hz. It is concluded that the damping effect is minor, and the magnitude reduction is due to the additional mass of the damper.

The FRF has the lowest peak when adjusting the mass block to 0°, and the original mode is split to two modes of 674 and 711 Hz. The amplitude is decreased to 13.9 % compared to the mass block in 90° and 28.7 % compared to without damper.

After the second cut, the undamped natural frequency of thin-walled workpiece rises to 714 Hz (Fig. 7b) and the corresponding amplitude of FRF is 1.37e^{-5} m/N. The damped FRF has the highest peak when the mass block is in the direction of 90°, and the amplitude is 1.36e-5 m/N. The FRF has the lowest peak when adjusting the mass block to 10°, and there are two split modes at 680 and 719 Hz. The amplitude is decreased to 13.7 % compared to the mass block in 90° and 13.6 % compared to without damper.

After the third cut (Fig. 7c), the first natural frequency of the thin-walled part is 740 Hz, which has a rise of 60 Hz compared to the initial value. The damper reaches the minimum and maximum damping in 80° and 0°, respectively. The amplitude is decreased to 39.9 % compared to the mass block in 80° and 18.9 % compared to without damper.

4.2 Machining verification

Machining tests are carried out on a three-axis vertical milling machine (VMC0850B) to validate the damping. The cutting tool is a cylindrical helical end milling cutter (SANDVIK R216.12) with a diameter D = 12 mm, number of teeth N = 2, and tool overhang L = 34 mm.

Three sets of experiments are performed. In the first case, the damping response is observed with different depths of cut, while in the second case, same experiment is verified but the spindle speed is changed. Down milling is employed in the two cases. In the third case, the damping response is observed under down and up milling modes.

Case 1 According to the chatter stability lobes, the cutting parameters are selected as the spindle speed n = 3000 rpm, width of cut a_e = 1 mm, and feed rate F = 600 mm/min. At four different depths of cut, the experiments are performed, and on every different depth, the orientation angle of the mass block is changed accordingly. The real time cutting vibration signal is collected by Cutpro MALDAQ (Fig. 8).

a Orientation 80°, a_p = 3 mm; **b** orientation 20°, a_p = 5.5 mm; **c** orientation 60°, a_p = 5.5 mm; and **d** orientation 20°, a_p = 8.5 mm.

Table 3 Identified dynamic parameters of the thin-walled part without and with the damper

	Natural frequency (Hz)	Stiffness (N/m)	Damping ratio (%)	Magnitude (m/N)
Without damper	**680**	**5.92e6**	**0.051**	**9.0e-5**
Damper orientation 20°	**640**	**1.30e8**	**0.271**	**1.2e-6**
	713	7.02e8	0.152	4.7e-7
Damper orientation 80°	**670**	**9.88e7**	**0.053**	**8.8e-6**
	690	1.80e9	0.158	1.7e-7

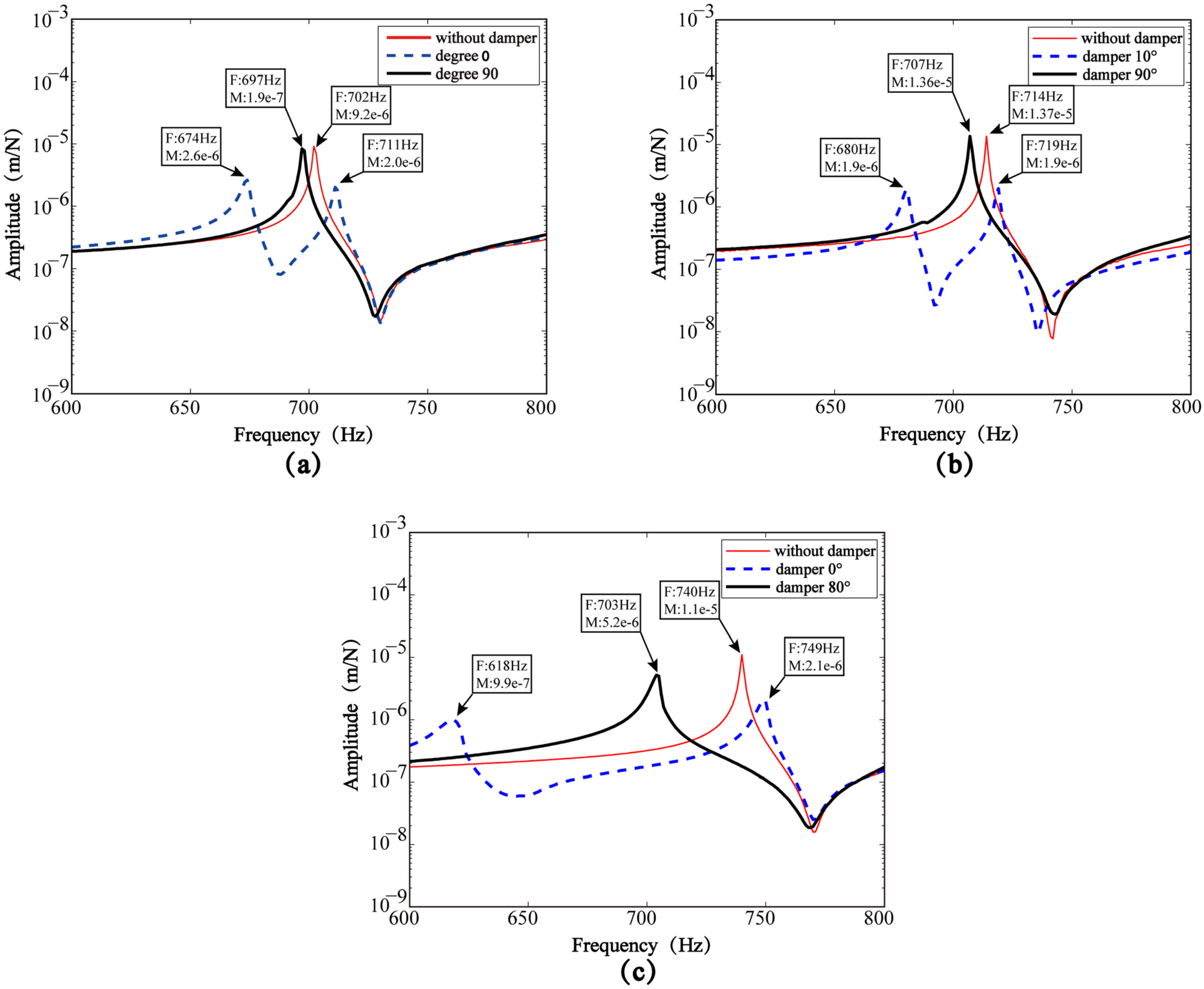


Fig. 7 Logarithmic FRFs of the thin-walled part under series of cutting with a_p = 2 mm. **a** The first cut, **b** the second cut, and **c** the third cut

Figure 8 shows that there is a large vibration when the mass block is adjusted to 80°, and the cutting depth is 3 mm. The vibration acceleration is 121.6 g which is caused by the first mode 672 Hz of the thin-walled part by checking the frequency spectrum, and the cutting surface is imprinted with clear vibration waves. After adjusting the mass block to 20°, the cutting process is transformed to be stable with a larger cutting depth of 5.5 mm. The vibration acceleration is 35.6 g

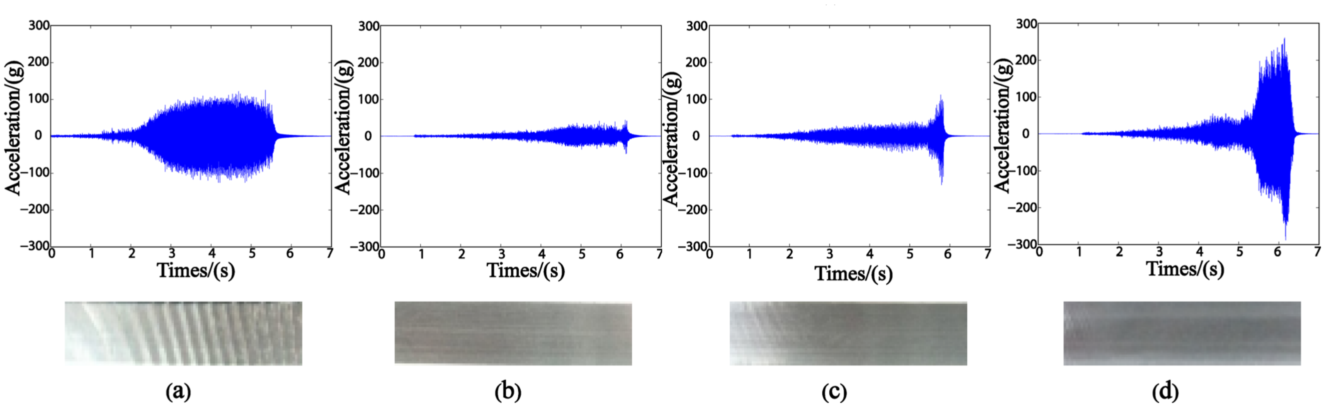


Fig. 8 Machining vibration and machined surface quality

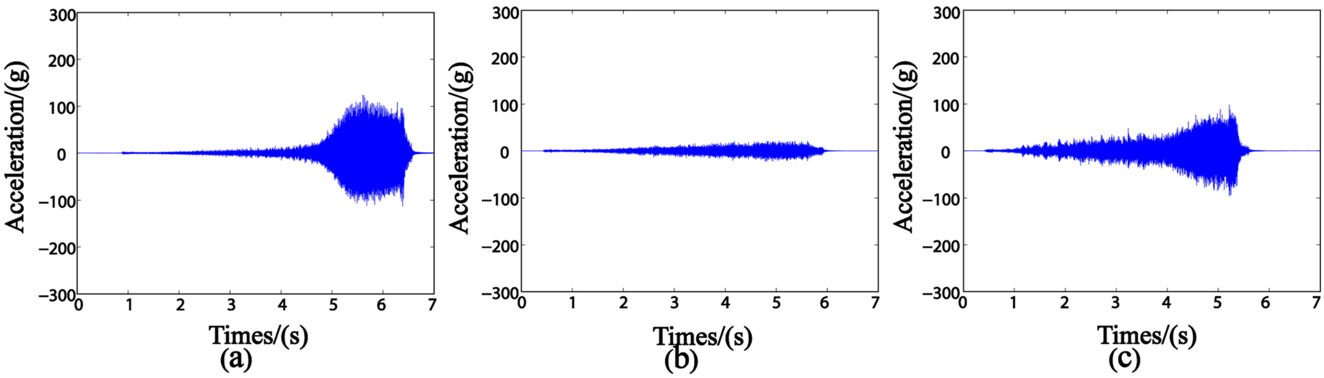


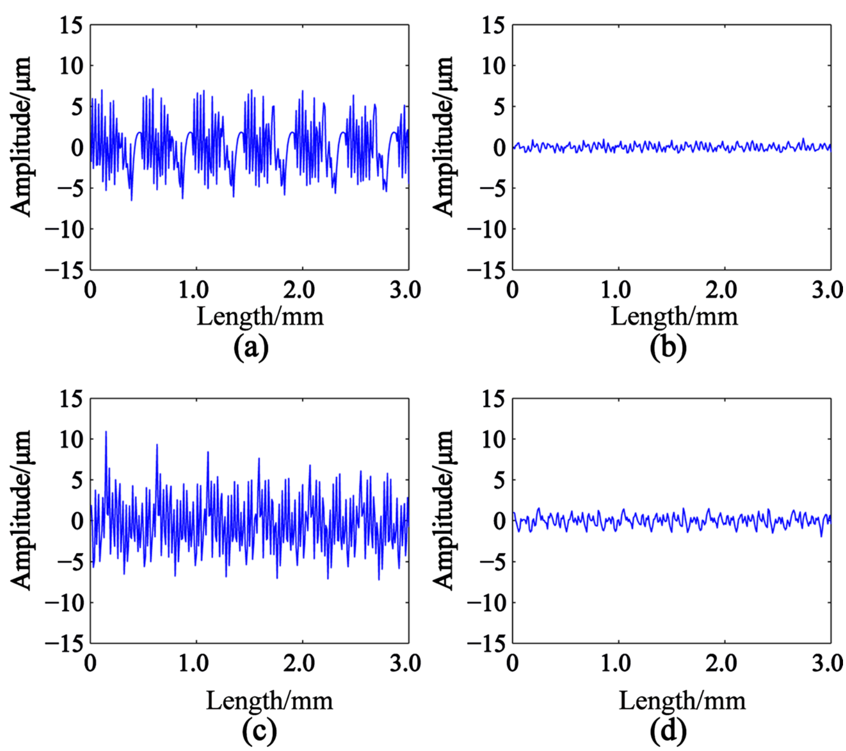
Fig. 9 Machining results. **a** Orientation 90°, $a_p = 4$ mm; **b** orientation 0°, $a_p = 4$ mm; and **c** orientation 0°, $a_p = 8$ mm

which is 29.3 % of the mass block oriented in 80°, and the cutting surface is smooth. The cutting process is kept stable when rotating the mass block to 60° and keeping the cutting depth unchanged, but acceleration reaches 112 g and is higher than the mass block in 20°. Meanwhile, the machined surface is worse. Finally, for the mass block in 20°, vibration acceleration is up to 260.3 g when the cutting depth increases to 8.5 mm. According to the modal test, the vibration is caused by the fourth mode 1685 Hz of the thin-walled part, but the first mode of the thin-walled part remains suppressed. The results show that the orientation of the damper mass has an impact on the cutting stability limit of the thin-walled part. The damper achieves best vibration suppression when adjusting the mass block to 20°. The corresponding critical depth of cut can be increased by 1.8 times compared with the mass block in 80°.

Case 2 Removing 2-mm height of thin-walled part and keeping the feed rate and width of cut same as CASE 1, the spindle speed n is changed as 3100 rpm. The first natural frequency of thin-walled part is 702 Hz which has a rise of 22 Hz compared to the original mode because of mass reduction. At three different depths of cut, the experiments are performed, and on every different depth, the orientation angle of the damper is changed accordingly.

Figure 9 shows that there is large vibration when the mass block is adjusted to 90° when the cutting depth is 4 mm. The vibration acceleration reaches 124 g which is caused by the first mode, and the cutting surface of the thin-walled part has obvious vibration. The cutting process is evolved to be stable when adjusting mass block to 0°. After increasing the depth of cut to 8 mm, the

Fig. 10 Machined surface profile. **a** Without damper, up milling; **b** with damper, up milling; **c** without damper, down milling; and **d** with damper, down milling



vibration rises up to 98 g and the unstable cutting is caused by the third mode 1085 Hz of the thin-walled part.

Case 3 The vibration reduction effect of the damper under up and down milling is analyzed. The cutting parameters are selected as the spindle speed n = 4000 rpm, width of cut a_e = 1 mm, a_p = 4 mm, and feed rate F = 800 mm/min. The machined surface profile without and with damper is measured as shown in Fig. 10.

The surface profile can be evaluated by the surface roughness $Ra = \sum_{i=1}^{N} |y|/N$. N is the number of the sampling points, and y is the sampling height. It is shown that the damper is effective for both up and down milling. The Ra is 2.68 and 0.28 μm for the undamped and damped up milling, while it is 2.84 and 0.53 μm for the undamped and damped down milling. The surface roughness is reduced by 89.6 % for up milling and 81.3 % for down milling.

5 Conclusions

The dynamics alteration during the thin-walled part milling leads to the invalidity of passive damper, which needs to follow the optimal parameters strictly. A passive damper with tunable stiffness is designed by rotating the mass block inside. The damper is glued on the workpiece, and the vibration mass is absorbing the energy as the target mode is excited during the machining process.

Modal tests show that the tunable frequency range of the damper is 13 Hz. The damper successfully damps the target mode of 680 Hz by locating the damper mass at different orientations. And it is found that at a proper orientation when the frequency difference between the damper and target mode is minimum, maximum damping can be achieved. After several paths of machining the workpiece with equal depth of cut, the undamped dominant frequency of the thin-walled part is shifted from 680 Hz to 740 Hz. The damper demonstrates the ability of tuning adaptability by setting a proper orientation of the mass block, and the amplitude of the damped FRF at the target mode is reduced by 40 % at most. The machining tests under multiple configurations validate the damping, as the machining vibration and surface roughness are quite reduced.

The mass block is made of permanent magnet, and the damping element design based on eddy current damping is not discussed in the paper as the details can be found in the literature [14]. The damper has a simple geometry design and is easy to be applied. As an accurate tuning is difficult to implement for rapid changing machining process, a passive damper with tunable frequency range is more practical.

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References

1. Vasundara M, Padmanaban KP (2014) Recent developments on machining fixture layout design, analysis, and optimization using finite element method and evolutionary techniques. Int J Adv Manuf Technol 70(1–4):79–96
2. Ratchev S, Liu S, Becker AA (2005) Error compensation strategy in milling flexible thin-wall parts. J Mater Process Technol 162-163:673–681
3. Wan M, Zhang WH, Qin GH, Wang ZP (2008) Strategies for error prediction and error control in peripheral milling of thin-walled workpiece. Int J Mach Tools Manuf 48(12–13):1366–1374
4. Campa FJ, De Lacalle LNL, Celaya A (2011) Chatter avoidance in the milling of thin floors with bull-nose end mills: model and stability diagrams. Int J Mach Tools Manuf 51(1):43–53
5. Wan M, Ma YC, Zhang WH, Yang Y (2015) Study on the construction mechanism of stability lobes in milling process with multiple modes. Int J Adv Manuf Technol 79(1–4):589–603
6. Herranz S, Campa FJ, Lopez de Lacalle LN, Rivero A, Lamikiz A, Ukar E, Sanchez JA, Bravo U (2005) The milling of airframe components with low rigidity: a general approach to avoid static and dynamic problems. Proc Inst Mech Eng Part B-J Eng Manuf 219(11):789–801
7. Aguirre G, Gorostiaga M, Porchez T, Muñoa J (2012) Self-tuning semi-active tuned-mass damper for machine tool chatter suppression. ISMA2012-USD2012:109–124
8. Rashid A, Nicolescu CM (2006) Active vibration control in palletised workholding system for milling. Int J Mach Tools Manuf 46(12–13):1626–1636
9. Lopez de Lacalle LN, Lamikiz A (2009) Machine tools for high performance machining. Springer-Verlag, London Limited
10. Rashid A, Nicolescu C (2006) Active vibration control in palletised workholding system for milling. Int J Mach Tools Manuf 46(12–13):1626–1636
11. Duncan G, Tummond M, Schmitz T (2005) An investigation of the dynamic absorber effect in high speed machining. Int J Mach Tools Manuf 45(4–5):497–507
12. Nakano Y, Takahara H, Kondo E (2013) Countermeasure against chatter in end milling operations using multiple dynamic absorbers. J Sound Vib 332(6):1626–1638
13. Kolluru K, Axinte D, Becker A (2013) A solution for minimising vibrations in milling of thin walled casings by applying dampers to workpiece surface. CIRP Ann-Manuf Technol 62(1):415–418
14. Yang Y, Dai W, Liu Q (2016) Design and machining application of a two-DOF magnetic tuned mass damper. Int J Adv Manuf Technol. doi:10.1007/s00170-016-9176-1
15. Kolluru KV, Axinte DA, Raffles MH, Becker AA (2014) Vibration suppression and coupled interaction study in milling of thin wall casings in the presence of tuned mass dampers. Proc Inst Mech Eng Part B-J Eng Manuf 228(6):826–836
16. Den Hartog JP (1934) Mechanical vibrations. McGraw-Hill Book Company, New York and London
17. Brock JA (1946) Note on the damped vibration absorber. J Appl Mech-T Asme 13(4):A284