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Influence of contact condition between flexible plate and passive pivot support on machining vibration

Atsushi Matsubara[a]*, Hiroaki Nagai[a], Nils Knorr[b], Daisuke Kono[a]

[a]Department of Micro Engineering, Kyoto University, Kyoto 6158540, Japan
[b]Institute of Machine Tools and Manufacturing (IWF), ETH Zürich, Zürich 8092, Switzerland

*Corresponding author. Tel.: +81-75-383-3675; Fax: +81-75-383-3675. *E-mail address:* matsubara@prec.kyoto-u.ac.jp

Abstract

A passive support with a pivot mechanism was previously designed for increasing damping of the vibration modes of a flexible structure. Here, to investigate the influence of the contact conditions between the pivot support and a flexible plate, the contact area is discretised into line contacts using metal rollers. The effect of contact stiffness is also investigated using additional plastic rings. Impact hammering tests are conducted and measured dynamic compliances are compared. Machining tests are subsequently performed to confirm vibration suppression. The line contacts are found to exhibit good vibration suppression capability in both the vibration and machining tests.

Keywords: machining vibration; support; damping; contact condition

1. Introduction

Aircraft parts have light-weight structures, which incorporate thin walls and/or thin webs [1]. However, the material removal processes used to fabricate such structures suffer from static and dynamic deformations induced by cutting forces. To avoid this deformation and ensure finishing accuracy, the support systems that provide stiffness and/or damping are indispensable [2]. Damping systems can be classified as active, semi-active, and passive [3]. Recently, several sophisticated semi-active and passive systems have been proposed, which do not require continuous energy supply [4-6]. Although these new systems can offer robustness or self-tuning capability, the design of support system for the flexible workpiece is difficult. This comes from the difficulty in managing the contact conditions between the support and workpiece.

The most reliable passive supports may employ a full-contact surface, which fits the workpiece surface and suppresses the various vibration modes arising during machining. However, design and preparation of contact surfaces that precisely fit the workpiece shapes are time-consuming. To solve this problem, flexible support systems have been investigated, the contact points of which can adaptively change to fit the structure to be machined [7][8]. To achieve the design simplicity and to reduce vibration, the authors' group previously investigated a passive support with a pivot action [9], which can adapt the inclination of the support surface as required. Experiments indicated that the pivot-type support provides better vibration suppression than a fixed-type support. However, this support can only be applied to a flat surface through a face contact. To support a curved structure, the contact must be discretised.

In this paper, the influence of the contact conditions between the pivot support and a flexible plate are reported, based on discretisation of the contact area into line contacts using metal rollers. The effect of contact stiffness is also examined using plastic rings.

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2. Experiment design

2.1. Purpose of experiment

Point or line contacts are preferable for passive supports as they can easily be applied to different workpiece surfaces. However, a problem exists in that contact points (or lines) become nodes of vibration modes (referred to as 'sine modes' hereafter). At such nodes, translational motion can be minimised, but rotational motion persists, as shown in Fig. 1. In this work, the capability of a pivot support (Fig. 2) to suppress the sine modes was investigated. In a previous study [9], both vibration and machining tests indicated that a pivot support could suppress the sine modes of a flexible plate. However, the mechanism and influence of the contact and cutting conditions on the machining vibration remain unknown. If the contact surface can be reduced, support for non-flat (e.g., curved or stepped) surfaces can potentially be achieved. To this end, the influence of the contact conditions between a pivot support and a flexible plate were investigated, with the flat contact being discretised into line contacts.

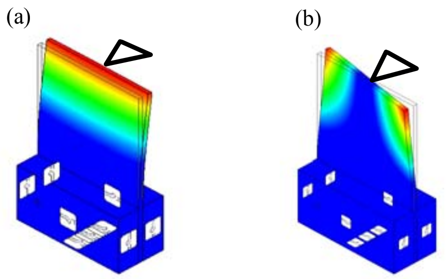

Fig. 1. Plate vibration modes and effect of support: (a) first and (b) second (sine) modes.

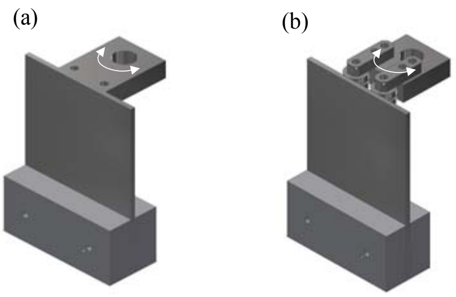

Fig. 2. Pivot support with (a) flat surface contact and (b) line contacts.

2.2. Experiment system

The experimental device shown in Fig. 3, which was used in the previous test [9], was employed in this work. The device consists of a vice unit, a support unit mounted on a slide table, a base plate, and a dynamometer. The plate dimensions were set to 100 mm (length) × 150 mm (height) × 5 mm (width) (cramped width: 50 mm). A new pivot support was designed, as shown in Fig. 4, with two steel rollers being attached to the pivot block to form line contacts to the plate. A thin-plate-type force sensor was inserted between the support and slide table to measure the contact force F_c.

To change the stiffness of the contact surface, plastic rings made of polyamide resin were also attached to steel rollers, as shown in Fig. 5(b). Using the plastic rings, the contact surface could be configured easily. Rings cut to make surface contacts were prepared, and the corresponding performance was compared with that for un-cut rings.

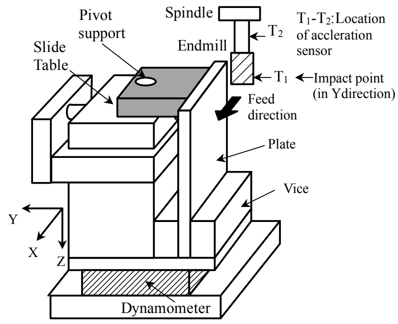

Fig. 3. Experimental device.

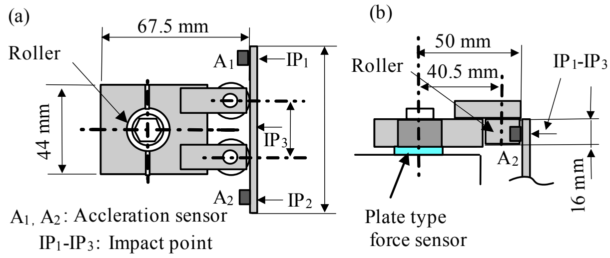

Fig. 4. Designed support unit and supported plate: (a) upper and (b) side views.

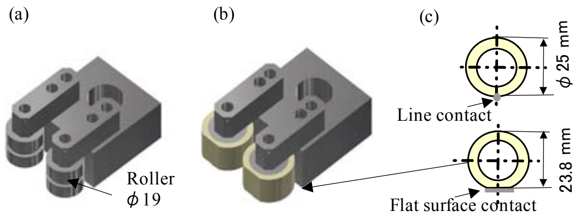

Fig. 5. Pivot support (a) with steel rollers, and (b) with steel rollers covered by (c) plastic rings (upper view).

3. Impact hammering test

The influence of the support conditions on the dynamic characteristics of the plate was investigated through a vibration test. The F_c of the support was set to 1 or 10 N, and impact hammering tests were conducted by hitting the plate in the normal direction. Accelerometers were attached to the plate to detect the acceleration in the Y-direction. The impact force and acceleration signals were captured by a digital signal analyser. Frequency response functions (FRFs) were estimated using the H_1 model [10] and transferred to compliance. The sampling frequency was set to 0.0416 ms. The FRFs for tool-side, which was used for a cutting test, were also measured. The sensors

were attached to the edge and shank of an endmill, as shown in Fig. 3. FRFs were measured under the following conditions:
(1) flat surface contact with the block only;
(2) line contacts with steel rollers;
(3) line contacts with plastic rings;
(4) flat surface contacts with plastic rings.

Fig. 6 shows examples of the measured FRFs for cases (1) and (2), and the FRF for an unsupported plate. As shown in Fig. 6(a), the magnitude plot for the unsupported plate exhibits sharp peaks at the first and second natural frequencies, which correspond to the mode shapes shown in Figs. 1(a) and (b), respectively. In contrast, the resonant peaks in the FRFs for cases (1) and (2) are very small (Figs. 6(b) and (c), respectively). The FRFs in Figs. 6(b) and (c) appear very similar, even though the nominal contact area was drastically reduced from the surface contact (case (1)) to the line contacts (case (2)). The same results were obtained, where F_c was set at 10 N. These results imply that the real contact area was quite limited under small F_c conditions.

Figs. 7 (a) and (b) show examples of the FRFs measured for cases (3) and (4), respectively. As apparent from Fig. 7(a), the resonance peak at 893 Hz was significant for the line contact with F_c = 1 N. However, this peak was suppressed for F_c = 10 N. For the flat surface contacts (Fig. 7(b)), the resonant peaks were smaller for the same F_c values, where the other peak next to the highest one also became apparent.

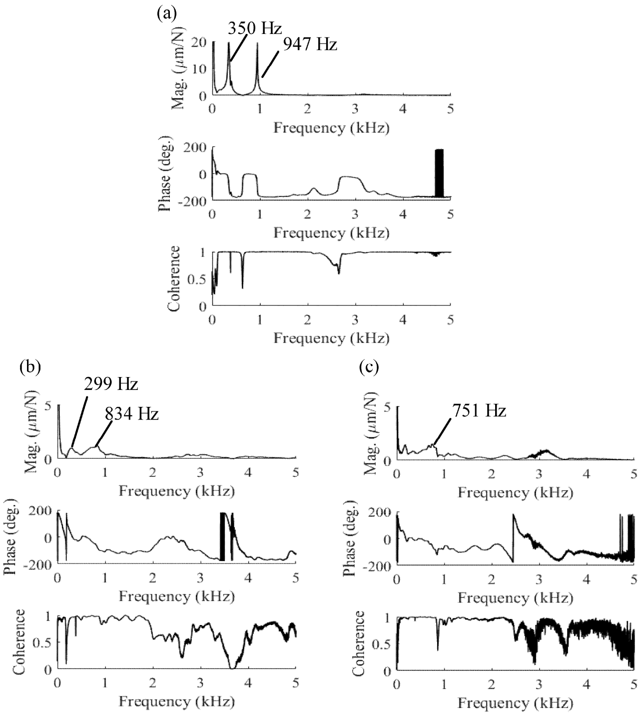

Fig. 6. Measured FRFs and coherence (IP_1 to A_1) for (a) unsupported plate, (b) flat surface contact comprised of the block only, and (c) line contacts comprised of steel rollers. (F_c: 1 N)

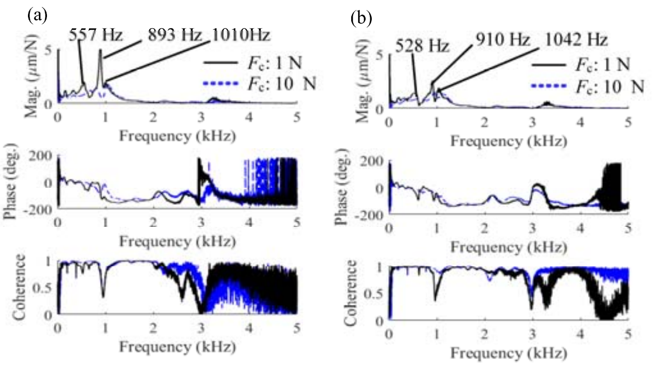

Fig. 7. Measured FRFs and coherence for plastic-ring-covered rollers with (a) line contacts and (b) flat surface contacts.

4. Machining test

4.1. Machining conditions

To investigate the influence of the contact conditions on the machining vibration, end milling tests were conducted. The device shown in Fig. 3 was set on the table of a machining centre. The cutting conditions are listed in Table 1. The spindle speed was chosen from the most critical values, i.e., those at which vibrations occurred easily in a preliminary test. As the feed per tooth was found to be critical on a small scale, two levels were chosen. Thus, the cutting was repeated to decrease the plate thickness, with the feed per tooth and F_c being adjusted to the second values listed in Table 2. The same acceleration sensors were used to monitor the vibrations during machining.

Table 1. Cutting conditions.

Tool	TiAlN-coated WC square endmill, four teeth, diameter: 14 mm, projection length: 47 mm
Workpiece material	S50C
Workpice size, W×H×t (mm)	100 × 150 × 5 (before cut)
Milling direction	Downcut
Radial depth of cut, R_d (mm)	0.2
Axial depth of cut, A_d (mm)	16
Spindle speed, S (min^{-1})	4000
Feed rate, F (mm/min)	800, 1600
(Feed per tooth: f_t [mm/tooth])	(0.05, 0.10)
Tooth passing frequency (Hz)	266.7
Contact force of support, F_c (N)	1, 10
Plate thickness, t (mm)	5.0-2.2

Table 2. Test order.

No.	f_t (mm)	F_c (N)	t (mm)	No.	f_t (mm)	F_c (N)	t (mm)
1	0.1	10	5.0	7	0.1	1	3.6
2	0.05	10	4.8	8	0..05	1	3.4
3	0.05	1	4.6	9	0.1	1	3.0
4	0.1	1	4.4	10	0..05	1	2.8
5	0.1	10	4.0	11	0.1	10	2.6
6	0.05	10	3.8	12	0.05	10	2.4

4.2. Test results

Figs. 8 and 9 show the measured accelerations and their power spectra for the pivot support for both flat surface contact with the block (case (1)) and steel-roller line contacts (cases (2)), respectively. In the earlier machining steps (Nos. 1–3, Table 2), vibrations with frequencies of approximately 4 kHz occurred (Figs. 8(a) and 9(a)), but decreased as the plate thickness was reduced (Figs. 8(b) and 9 (b)). After No. 3, the machining process was very stable regarding vibration. In contrast, severe vibrations occurred when plastic-ring-covered rollers were employed, as shown in Fig. 10 (for flat surface contact, i.e., case (4)). For this reason, that test could not proceed beyond condition No. 2. Under these critical conditions, the vibrations with frequencies distributed around 1 kHz were large. Further, although the second mode appeared to be suppressed in the FRFs (Fig. 7 (b)), it appeared prominently in the machining test. This reason may be that the natural frequency was close to a multiple of the tool passing frequency and the resonance was not perfectly suppressed.

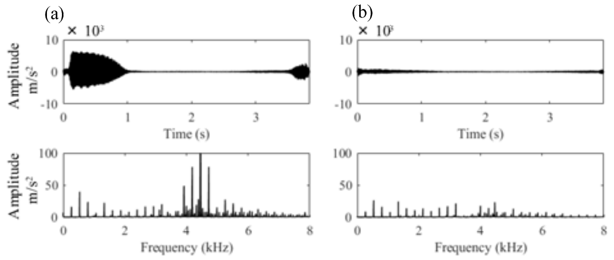

Fig. 8. Measured accelerations (upper) and power spectra (lower) for flat surface contacts: (a) exp. No. 1, t = 5.0 mm; (b) exp. No. 9, t = 3.0 mm

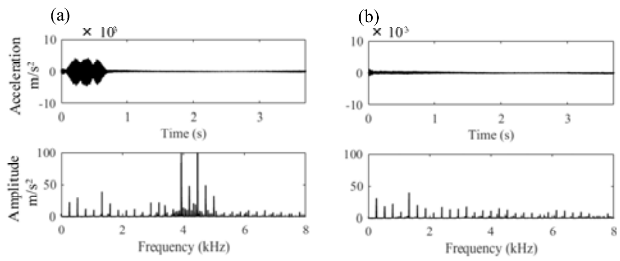

Fig. 9. Measured accelerations (upper) and power spectra (lower) for steel-roller line contacts: (a) exp. No. 1, t = 5.0 mm; (b) exp. No. 9, t = 3.0 mm.

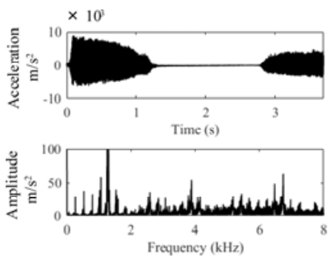

Fig. 10. Measured accelerations (upper) and power spectra (lower) for plastic-ring-covered rollers with flat surface contacts: exp. No. 1, t = 5.0 mm.

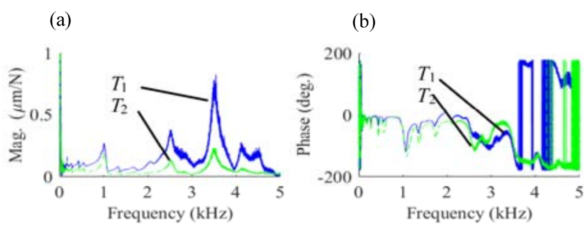

Fig. 11. Measured FRFs for spindle: (a) magnitude plot; (b) phase plot.

To confirm the source of the vibrations at approximately 4 kHz, FRFs measured for spindle impact hammering are shown in Fig. 11. A dominant resonant peak near 3.6 kHz is apparent, while the resonant peaks over 4 kHz were less apparent. Note that the resonant peaks over 4 kHz were not apparent either in the FRFs measured for the unsupported plate either (Fig. 6(a)). The reason why the machining vibration over 4 kHz became dominant is unclear.

5. Conclusion

A simple pivot support was redesigned to avoid vibrations during plate milling. To extend the application of the support to a curved structure, line contacts were introduced with metal rollers. The results of impact hammering and milling tests indicate comparable vibration suppression performance to that of a conventional flat surface pivot. In the case of a plastic contact, the vibrations were reduced in the impact hammering tests, but not in the machining tests.

Acknowledgments

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