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Design of a support system with a pivot mechanism for suppressing vibrations in thin-wall milling

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ABSTRACT

This paper addresses the design process of support systems for suppressing vibrations during thin-wall milling. Supports with point contact and surface contacts are examined under different preloading conditions to understand their effects on the cantilever and torsion vibrations of a cantilever wall. A pivot mechanism is presented that provides a rotation motion to the surface support and damping to the vibration modes. Both impact and milling tests showed that the surface support performs better at suppressing vibrations with the pivot mechanism.

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1. Introduction

In the aircraft industry, cost-effective and high-performance machining is required to adopt new design strategies that employ more complex geometry and lightweight and high-temperature-resistant materials for aircraft parts [1]. Typical engine parts such as airfoils, casings, and shafts include thin-walled structures, which suffer from deformation during machining. Dynamic cutting forces induce vibrations that may damage workpiece surfaces. Both forced and self-excited vibrations can cause problems and reduce productivity. Various suppression techniques have been proposed and utilized to avoid self-excited chatter [2]. A well-known method is optimizing cutting parameters by using a stability lobe diagram (SLD); this is also applied to the design of thin-wall milling [3]. For the machining of difficult-to-cut materials, however, cutting speeds have upper bound and adjacent lobes are close, which increases the prediction uncertainty.

Because thin-walled parts cannot provide enough stiffness by themselves, supporting fixtures are commonly used in actual machining fields. Aoyama and Kakinuma proposed a multi-pin support system that can adapt to a thin and compliant plate [4]. They also showed that this system can provide damping. Möhring and Wiederkehr presented a comprehensive report on the development of intelligent fixtures that reduce vibrations and distortions during the machining of flexible structures [5]. Their report covered optimization of the fixture layout, active techniques to dissipate the vibration energy, and adaptive clamping systems. Kolluru and Axinte reported the coupled dynamic interaction of a tool and workpiece during the machining of a casing [6]. They presented unique surface damping techniques that combine viscoelastic tapes, neoprene sheets, and tuned masses [7]. They also showed better vibration suppression could be obtained with torsion springs that stretch the neoprene tape against the casing [8].

Although these new distributed and smart-type supporting techniques provide better vibration suppression performance, actual machining fields still rely on low-cost and discrete-type supports. These conventional supports softly contact the work-piece surface to avoid static deformation not only from the cutting force but also from the supporting or clamping forces themselves. The present paper reports a soft-touch support system with a pivot mechanism as a potential improvement over conventional supports. For the design strategy, vibration modes during the interaction of a pivot support and plate were investigated. Cutting tests were conducted to analyze if the pivot can suppress vibrations during thin-wall milling.

2. Design concept and test support

2.1. Support design concept

The vibration modes of thin-wall plates or cylinders can be represented with sine, cosine, hyperbolic sine, and hyperbolic cosine functions. In order to suppress the vibration modes, supports are commonly placed at the most vibrant points, which are the anti-nodes of the vibration modes. Fig. 1(a) illustrates a situation where a cantilever plate is supported. Because the most vibrant point is the free end of the plate, the support should be provided here (Fig. 1(b)). A point support at the center of the plate can suppress the first vibration mode but not the second mode. This is because the second mode is a sine mode with respect to the first point support, as shown in Fig. 1(c). If a solid surface is used as a contact to suppress a sine mode, the local contact may inspire other vibration modes or distort the structure (Fig. 1(d)). This is why compliant and distributed supports are preferred for thin-walled structures. If a solid surface support is equipped with a

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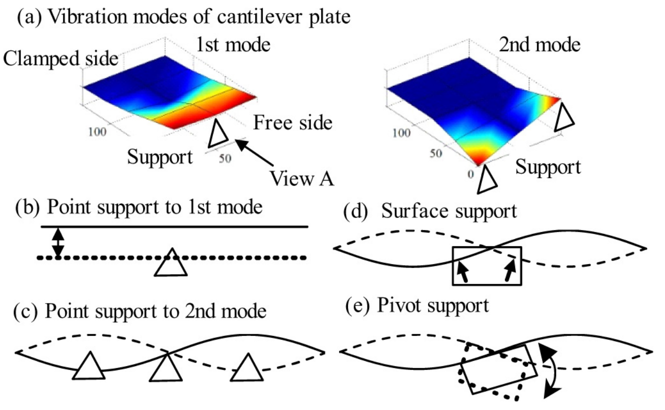

Fig. 1. Vibration modes and support concept.

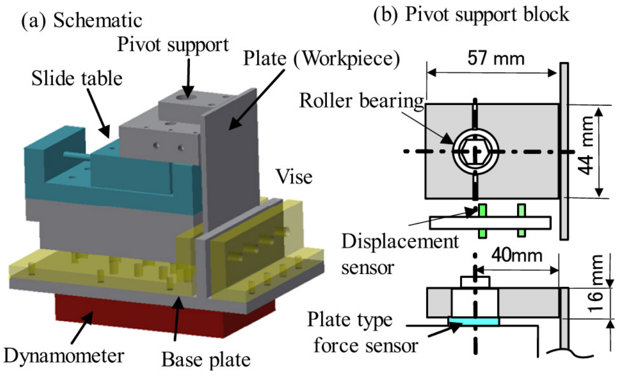

Fig. 2. Experimental device for fixing and supporting a plate.

pivot action, it may allow the contact surface to align with the sine mode while it constrains the cosine mode, as shown in Fig. 1(e).

2.2. Experimental device

In order to investigate the effect of a pivot support on vibration suppression, a test device that fixes and supports a plate was designed. Fig. 2 shows the schematic of the device. The device consists of a vise unit, support unit mounted on a slide table, and base plate. One end of the plate is clamped with the vise, and the opposite end is supported. The pivot support is rotatable with a roller bearing. Although various designs are available for the support and contact geometries, a simple block unit with a flat surface contact was selected. Two materials were selected for the support block: steel and PLA. The table slide is operated manually to give the contact force to the support and can be locked with a clamping handle. A thin-plate-type force sensor is inserted between the support and slide table and can measure the contact force. The maximum length and height of the plate that this device can fix and support are 100 and 150 mm, respectively. The entire unit can be installed on a dynamometer to measure the cutting forces.

3. Vibration test

3.1. Experimental procedure

An experimental device with a carbon steel plate (normal steel, C: 0.45%) was installed on the table of a machining center, and a hammering test was carried out. After several plates were examined, the plate dimensions were set to 100 mm (length) × 150 mm (height) × 5 mm (width) (cramped width was 50 mm). The coordinates were defined as shown in Fig. 3(a) to explain the experiment results. The X-, Y-, and Z-directions are the normal, lateral and vertical directions, respectively, relative to a free cantilever plate. Accelerometers were attached to the support side surface of the plate to detect the acceleration

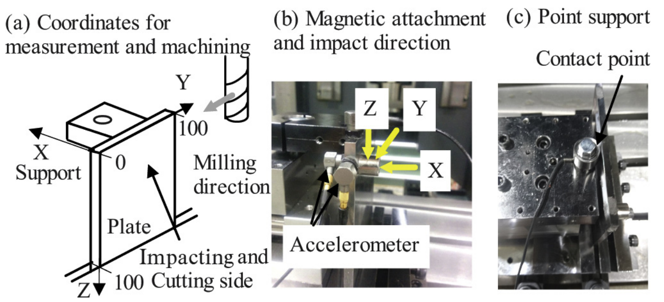

Fig. 3. Setup for measurement, vibration, and machining tests.

in the X-direction because they could show a larger vibration amplitude than those attached to the side of the wall in the Y- and Z-directions.

Impact tests are generally carried out by hitting the structure surface in the normal direction. In thin-wall milling, however, the tangential forces on the surface are important because they also induce vibration modes. For example, normal forces cannot inspire sine modes at the support points because they are anti-nodes. To provide tangential forces, a small neodymium magnet ring (mass: 3.8 g) was attached to the plate surface with glue to reinforce the attachment (Fig. 3(b)). The impact force and acceleration signals were captured with a digital signal analyzer, and frequency response functions (FRFs) were estimated with the H_1 method. The sampling time was set to 0.0416 ms.

For comparison, FRFs were also measured for the case where just the roller bearing was in contact with the plate (Fig. 3(c)). Although the roller may have had line contact in the Z-direction, it was referred to as a "point support" with respect to the Y-direction. Because the steel support showed better performance for vibration suppression in preliminary tests, this report focuses on the results for the steel support.

3.2. Measured FRFs and vibration modes

First, measured FRFs without support (i.e., free), with point support, and pivot support conditions were compared to understand how each support changed the dynamics of the plate. The contact force was set at 15 N for the point and pivot supports. Fig. 4 shows examples of the imaginary parts of the FRFs for the accelerance and corresponding coherence. In the cases of the free and point support conditions, the coherence was almost unity up to 8 kHz (Fig. 4(a)).

In the case of the pivot condition, the coherence for a Y-direction impact was better than that for an X-direction impact (Fig. 4(b)). This is because the plate and support had nonlinear interaction in the X-direction.

The natural frequencies and corresponding vibration modes were identified by referring to the imaginary parts of the FRFs, as

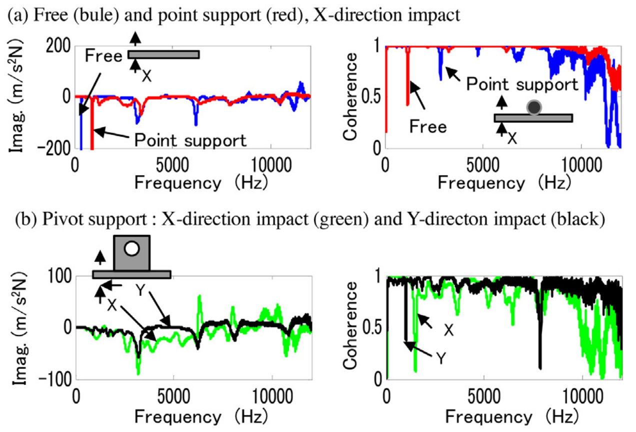

Fig. 4. Measured FRFs for the accelerance and coherence of free, point, and pivot supports measured at (Y, Z) = (10, 10) mm.

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