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Development of a high bandwidth XY stage for vibration-assisted milling

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Abstract

To realize vibration assisted machining (VAM) in micro milling, a high bandwidth XY vibration stage driven by piezoelectric stack actuators was designed, analysed and tested. Firstly, a novel flexure-joint configuration was proposed by comparing several alternative XY stage configurations. The proposed configuration employs double coupling circle flexure-joint structures which can suppress the coupling displacements of the stage along X and Y directions effectively. Secondly, in order to meet the machining requirements of high spindle speed in micro milling, the structural parameters of the proposed vibration stage are optimized by the finite element method to improve the resonance while reducing the stiffness of the vibration stage. The resonance frequencies of larger than 2,000 Hz in both directions and low stiffness of 50N/µm were obtained through the parameter optimization. Finally the experiments were performed on the designed stage using two capacitance displacement sensors, and the transient vibration locus of the stage is obtained.

Vibration-assisted machining, micro milling, vibration stage design, structural optimization

1. Introduction

VAM technology has been rapidly developed in decades. It has a broad range of research and applications in vibration assisted cutting, grinding, finishing, plastic processing etc. VAM has found applications in processing hard-to-machine materials to improve machining quality and extend tool life. The vibration stage is the key element in the VAM system. There are two main types of the vibration stage. One is resonant type, which uses a piezoelectric or magnetorestrictive actuator to create reciprocating harmonic motion of high frequency but low amplitude. A shaped acoustical waveguide booster and horn (also called a "sonotrode") amplifies this ultrasonic motion. Resonant type can reach high vibration frequency (>10k Hz), but vibration frequency is limited at the resonant frequency of the vibration stage structure. The other is non-resonant type, in which sinusoidal voltage signals are applied to piezoelectric actuators causing them to extend and contract but at a frequency below the first natural frequency of the system. The frequency and amplitude of the non-resonant type stage are easy to adjust, while the frequency is usually less than 1k Hz because it is limited by the first natural frequency of the system. In this paper, a high bandwidth (>2K Hz) non-resonant XY stage is designed, analysed, and tested for vibration assisted milling.

2. Configuration design of the vibration stage

The configuration design of the platform has an important influence on the performance of vibration stage. Various configuration designs of 2D platform for vibration assisted machining have been developed.

A structure design of the platform is proposed by Zhang et al. [1] as shown in Figure 1a), two piezoelectric actuators were generating different direction vibration to the objective table. It used the planar integrated structure. It has the characteristics of compact structure, zero clearance and no mechanical friction. While, the piezoelectric actuators in X direction, improved the moving mass of the vibration stage. Thus, the vibration frequency in Y direction is restricted, usually less than 1k Hz.

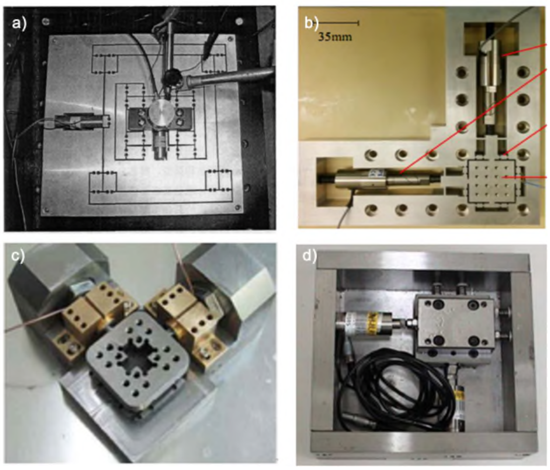

Figure 1. Typical structure of the vibration stage

An improved 2D-vibration stage was developed by Jin et al. [2], as shown in Fig.1b). The piezoelectric actuators have been set beside of the hinge part, which transforms vibration to the workpiece access the face A and B of the flexible hinges. However, it also has a disadvantage. The structure only has one layer of flexible hinge around the platform. The piezoelectric actuator can generate the vibration to the vertical direction flexible hinges normally, but in the horizontal direction, the flexible hinge will be affected by the coupling stiffness, and the displacement of the platform will be uneven. Therefore, it is difficult to control the displacement of the platform precisely. Li et al. [3] developed a 2D vibration stage by using a single flexible four bar mechanism, but when the single flexible parallel four bar mechanism vibrate at one direction, there will be a cross-coupling displacement in the vertical direction, which reduce the motion accuracy. Different to the typical 2D vibration solutions which use the flexible mechanism drive by piezo-actuators, a vibration worktable was designed by Chang et al.[4], realized by employing piezoelectric actuators in conjunction with two slideways for generating the desired two-dimensional vibration directly.

According to the design evaluations above, a novel platform structure has been proposed as shown in Figure 2 with the structure of flexible hinges zoomed in. It can be seen that the platform has two layers of flexible hinges. In order to make sure that the vibration can transmit at both directions normally. The inner layer flexible hinges guide the vibration in the actuation direction, while the outer layer flexible hinges guide the vibration in the cross actuation direction. The outer and inner parts are combined with each other so that the vibration can transmit successfully. This structure design adopts the advantages of the previous design and resolves the issues abovementioned.

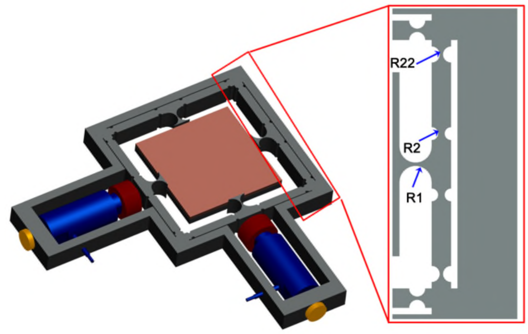

Figure 2. Proposed XY vibration stage

3. Structure optimization

According to the requirement of the vibration parameters and the properties of the piezoelectric actuators, the design requirements of the vibration stage are given as follows: the natural frequency of the platform needs to be larger than 2k Hz, and the stiffness of the platform needs to be smaller than 50N/μm. In order to achieve the design requirements, the parameters of the structure should be optimized. It can be seen from Figure 2 that the structure is symmetrical, and only three radii of the notches of the flexible hinges mainly influence the performance of the platform, which are marked as R1, R2, and R22. The ranges of them are present in equation (1):

$$\begin{cases} 2 < \mathrm{R1} < 5 \\ 0.5 < \mathrm{R2} < 2.9 \\ 0.5 < \mathrm{R1} < 2.9 \end{cases} \qquad (1)$$

Based on the goal of the optimization illustrated before, the objective functions of the optimization can be expressed as equation (2):

$$\begin{cases} \mathrm{Max}\, f = f(R1, R2, R22) \\ \mathrm{Min}\, k = k(R1, R2, R22) \end{cases} \qquad (2)$$

Where f and k are the frequency and stiffness respectively.

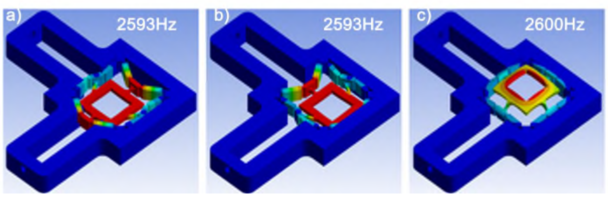

Figure 3. First three vibration modes of the stage

After optimization, the optimal value of R1, R2 and R22 are determined as 5mm, 2mm and 2.5mm respectively. The stiffness of the stage is calculated as 48N/μm. Figure 3 shows the first three vibration modes of the designed stage after optimization, and the frequencies are 2593Hz, 2593Hz and 2600Hz.

4. Experiment

The designed vibration stage and the experimental set-up are shown in Fig.4, the vibration stage is driven by two piezoceramic actuators, and the displacement is measured by two capacitance displacement sensors (Micro-epsilon, CS005).

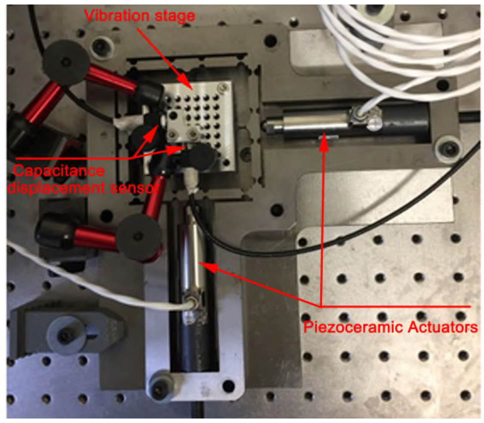

Figure 4. Experimental set-up

Figure 5 shows an example of the test result of the displacement of the stage in *x* and *y* direction. Two channel sinusoidal voltage inputs π/2 phase angle apart were applied to two piezo-actuators. Circular trajectories stage of the stage was plot with 2.3 μm amplitude at a frequency of 2000Hz. The vibration stage was tested in open loop control, hence, a small kinematic error of 0.25 μm on the stage was measured as expected, although a small deviation from the desired trajectory is believed to have little effect on VAM. The preliminary test results confirm the performance of the XY stage predicted from the simulation.

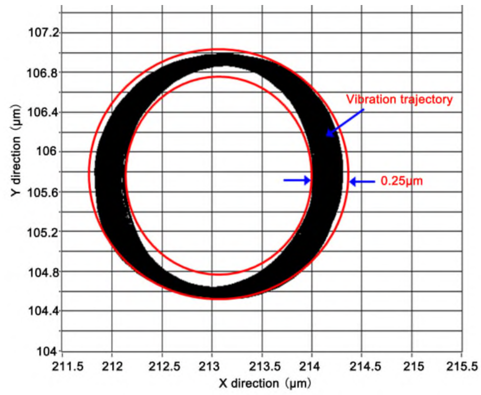

Figure 5. Vibration trajectory of the stage

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