MODELING AND TOOL TRAJECTORY MONITORING OF AN ULTRASONIC ELLIPTICAL VIBRATION TOOL

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ABSTRACT
The vibration trajectory of an ultrasonic elliptical vibration cutting tool is critical for its performance in vibration-assisted cutting/texturing. The shape and amplitude of vibration trajectories will significantly affect the surface quality, tool life, and material removal mechanism. The tool trajectories are often unregulated in current practice due to the difficulties in direction measurement during cutting. In this paper, an analytical model based on the feedback signals from piezoelectric plate sensors has been proposed for the prediction of tool vibration trajectory. The proposed model calculates the elliptical vibration trajectory considering the mode superposition of two orthogonal resonant modes and the measured sensor amplitudes and phase angles. The relationship between the excitation voltage and frequency and the model parameters were studied and calibrated. Experiments were conducted to evaluate the model accuracy and to verify the feasibility of online prediction of the tool trajectory.

NOMENCLATURE

\( \alpha \) Tool sensor voltage ratio over the tool holder bending

\( \beta \) ratio between the Mid sensor voltage and symmetric bending deformation of the center point \( O \)

\( F_{\Omega}(\omega) \) ratio between the tool tip motion in the cutting direction and the excitation voltage in the transverse mode

\( F_{\psi}(\omega) \) ratio between the tool tip motion in the depth direction and the excitation voltage in the transverse mode

\( \gamma \) phase angle between the Mid sensor voltage and the tool tip motion

\( \theta \) phase angle between the Tool sensor voltage and the tool tip motion

\( \psi_n \) phase angle between the reference signal and the tool tip motion in the depth direction

\( \psi_t \) phase angle between the reference signal and the tool tip motion in the cutting direction

\( \phi \) input phase angle between two excitation signals

\( \omega \) excitation angular frequency

\( V_m \) detected voltage of the Mid sensor

\( V_t \) detected voltage of the Tool sensor

\( P_{N,\text{cut}} \) tip motion in the cutting direction in the normal mode

\( P_{N,\text{depth}} \) tip motion in the depth direction in the normal mode

\( P_{T,\text{cut}} \) tip motion in the cutting direction in the transverse mode

\( P_{T,\text{depth}} \) tip motion in the depth direction in the transverse mode

INTRODUCTION
Functional surfaces with sophisticated micro/nano-structures or patterns have shown great advantages in various fields in terms of their enhanced performance in biological, mechanical, and optical properties [1]. The demand for their corresponding manufacturing technologies has facilitated the development of various surface micro/nano-machining technologies, including physical/chemical deposition, laser ablation, lithography, etc. [2, 3]. These existing processes, however, still have some critical limitations due to their low efficiency for large scale production or the restriction on the specific class of workpiece materials. Elliptical vibration texturing (EVT) has previously been proposed by the authors for fast generation of micro/nano-structures [4]. In EVT, the ultrasonic tool vibration is coupled with a nominal cutting motion. Micro/nano-patterns are imposed onto the workpiece surface due to the resultant modulated tool trajectories. Compared with other processing methods, the EVT process is of low cost, high efficiency, and high flexibility. The process is suitable for rapid prototyping of customized micro/nano-patterns.

The form accuracy and consistency of generated micro/nano-structures have a huge impact on the functional performance of structured surfaces. For example, in the structural coloration application using micro grating structures [5], the uniformity of grating spacing and the aspect ratio of grating profile will determine diffraction efficiency and apparent color. The accuracy and consistency of produced surface geometry in the EVT process are largely influenced by the tool vibration trajectory, such as its amplitude, aspect ratio, and orientation. In current practice, the
vibration trajectories are often measured offline before the texturing process. It is difficult to achieve direct measurement of the tool vibration during cutting due to the interferences of electromagnetic noises, cutting fluids, chips, etc. Meanwhile, the actual tool vibration amplitudes are susceptible to process variations from cyclic cutting loads, self-heating, and input power fluctuations [6]. The change in tool vibration trajectories will deteriorate surface integrity and its functional performance, even make the process unstable.

In order to monitor and control the tool vibration during the cutting process, one of the most commonly used methods is based on a phase-locked loop (PLL). It is often adopted for resonant-type ultrasonic vibration tool to track its resonant frequency. The PLL is based on the principle that due to electromechanical coupling between the piezoelectric elements and mechanical structures, there exists a characteristic relationship between the resonant frequency and the current-voltage phase difference [7, 8]. This automatic resonant frequency tracking algorithm will drive the tool close to its resonant frequency, which greatly alleviates vibration amplitude loss due to the resonant frequency drift. This strategy only ensures the system efficiency, but it does not directly track the tool vibration amplitude. In addition, its performance is not guaranteed for a system with two very close resonant frequencies, such as coupled-resonant tools [9-11].

Dong et al. [12] proposed an automatic stabilization method through adjusting the input power to compensate the additional impedance load caused by the change of resonant frequency. However, it was difficult to maintain a constant vibration amplitude because that the impedance change cannot exactly indicate the vibration amplitude, especially when the change of impedance was very small. Shamoto et al. [13, 14] developed an ultrasonic elliptical vibration controller to remove the cross talk between the vibrations in two directions, but it was unable to predict the two-dimensional vibration trajectory due to the lack of phase angle information. Zhang et al. [15] proposed a vibration locus compensation method to control the vibration amplitude for micro-structure sculpturing. The method, however, planned the vibration trajectory beforehand and worked in an open-loop manner. The research regarding real-time prediction of the tool vibration trajectory for ultrasonic elliptical vibration cutting tools is still lacking, which is very important for the industrial application of elliptical vibration cutting/texturing.

In this paper, we propose a method for the prediction of tool vibration trajectory based on an analytical model and the measured amplitude and phase information of piezoelectric sensors at critical locations on the vibration tool. The tool design based on a portal frame structure is first introduced. The analytical model is then derived to relate the tool vibration trajectory with the measured sensor signals. The effect of frequency and excitation voltage on the model parameters is discussed. The experimental results are then presented and discussed, followed by a conclusion.

TOOL DESIGN

An elliptical vibration cutting tool based on a portal frame structure was first developed by our group [10], which is shown in Fig. 1(a). A horizontal beam is connected by two vertical beams at right angles to form a portal frame structure. The tool holder is located at the center of the horizontal beam. At each side of the vertical beams, two piezo plates are attached to the beam surfaces. The piezoelectric plates are driven to generate bending excitation to the vertical beams in a resonant mode. The structure is connected to the foundation through two separate fixtures.

The working principle of the proposed tool is based on the mode superposition theory. The structure couples two orthogonal vibration modes at a similar resonant frequency to generate an elliptical vibration trajectory at the tool tip. The symmetric vibration of two vertical beams induces a normal vibration mode in the depth direction, as shown in Fig. 2(a). While the anti-symmetric vibration of two vertical beams will result in a transverse vibration in the cutting direction, which is demonstrated in Fig. 2(b). The resonant frequencies of these two modes, however, always have a discrepancy in reality due to manufacturing errors, preload conditions, variation of material properties, etc. This discrepancy will deteriorate the vibration amplitudes and the coupling of two modes. In addition, the design of separate fixtures is vulnerable to manufacturing and assembly errors, which will result in the structure instability.

Due to the above disadvantages, an improved design of an ultrasonic elliptical vibration tool is proposed, which is shown in Fig. 1(b). The new structure is changed from a pi-shape into a square shape. The square shape will make the tool more stable during operation and easy for assembly. Five screw holes are added on each side of the new tool design, which can be used to tune the resonant frequencies of the normal and transverse modes.

Figure 1: Design of the elliptical vibration tool based on a portal frame structure: (a) original design [10] and (b) modified design.
finite element simulation to guarantee that the two orthogonal vibration modes were coupled at the same frequency. The overall dimension of the frame structure is 36 mm x 30 mm x 32 mm.

**ANALYTICAL MODELING**

For a sinusoidal excitation signal whose frequency is between the resonant frequencies of the normal and transverse modes and far away from the other modes, the dynamic behavior of the structure will be dominated by the two orthogonal modes, as shown in Fig. 2. According to the superposition principle [16, 17], the coupled mode shape is the superposition of the normal and transverse modes with their respective amplitudes and phases. The resultant tool tip motion, in general an ellipse, could be estimated by a 3-by-3 matrix to relate the input and output parameters. The matrix entries are, however, non-constants and generally nonlinear functions of the excitation voltage, input phase, and frequency, which makes it difficult to get the matrix inversion for control of the tool trajectories. Therefore, we would like to get analytical expressions between the input and output parameters in order to (1) enhance the fundamental understanding of a coupled resonant vibration system and (2) extend our model for active trajectory control in the future work.

If the two excitation signals are of equal amplitude, $A$, and frequency, $\omega$, but with a phase difference, $\phi$, the mode amplitudes can be distributed according to the trigonometric relation. The resultant tool tip motion can be given as:

$$P_{\text{cut}}(t) = \text{PN}_{\text{cut}} \sin \frac{\phi}{2} \sin \omega t + \text{PT}_{\text{cut}} \cos \frac{\phi}{2} \cos \omega t$$

(1)

$$P_{\text{depth}}(t) = \text{PN}_{\text{depth}} \sin \frac{\phi}{2} \sin \omega t + \text{PT}_{\text{depth}} \cos \frac{\phi}{2} \cos \omega t$$

(2)

where $P_{\text{cut}}(t)$ and $P_{\text{depth}}(t)$ are the tool tip motion in the cutting and depth directions. $\text{PN}_{\text{cut}}$ and $\text{PN}_{\text{depth}}$ are the vibration amplitudes of the tool tip in the cutting and depth directions subject to the normal mode excitation; while $\text{PT}_{\text{cut}}$ and $\text{PT}_{\text{depth}}$ are the vibration amplitudes of the tool tip in the cutting and depth directions in the transverse mode. In general, $\text{PN}_{\text{cut}}$, $\text{PN}_{\text{depth}}$, $\text{PT}_{\text{cut}}$, $\text{PT}_{\text{depth}}$ are functions of the excitation frequency and voltage.

As shown in Fig. 2(a), the tool tip vibration amplitude in the cutting direction, $\text{PN}_{\text{cut}}$, is negligible in the normal mode, so its value can be ignored in Eq. (1). In reality, however, due to the asymmetric mass distribution, the tool holder will undergo bending vibration even in the normal mode. The additional bending of the tool holder causes extra transverse displacement of the tip $P$, which should be added to Eq. (1). Then the total transverse displacement of tip $P$ is made up of two parts: one comes from the rotation of the holder beam around the center point $O$ in the transverse mode; and the other part comes from the tool holder bending. From Fig. 2(a), it is easy to see that in the normal mode, the depth displacement of the tip $P$, is equivalent to the displacement of the point $O$, or the symmetric bending of the horizontal beam. The vibration amplitude, $\text{PN}_{\text{depth}}$, can be presented by the beam bending distance at the center point $O$. In Fig. 2(b), we observe that local curvature of the beam around the center point $O$ is almost zero in the transverse mode, which means that the total deformation of point $O$ can be represented by its amplitude in the normal mode.

In order to detect the bending deformation of the center point $O$ and the tool holder, two piezoelectric sensors are attached to the corresponding positions, as shown in Fig. 1(b). The sensor on the tool holder is named as the Tool sensor, while the sensor at center of the horizontal beam is named as the Mid sensor. The detected voltage amplitude of the sensors is proportional to the bending deformation of the tool holder and the horizontal beam around the point $O$, respectively.

The resultant tool tip motion equations then are modified as:

$$P_{\text{cut}}(t) = \text{PT}_{\text{cut}} \cos \frac{\phi}{2} \cos \omega t + \alpha V_t \sin(\omega t + \theta)$$

(3)

$$P_{\text{depth}}(t) = \text{PT}_{\text{depth}} \cos \frac{\phi}{2} \cos \omega t + \beta V_o \sin(\omega t + \gamma)$$

(4)

where $\alpha$ and $\beta$ are the coefficients between the measured sensor voltage (Tool and Mid sensors) and the actual beam deformation. $V_t$ and $V_o$ are the detected voltage amplitudes from the piezoelectric sensors. $\theta$ and $\gamma$ are the phase lags of the bending motion of the tool holder and the horizontal beam with respect to the tool tip motion. The coefficients $\alpha$ and $\beta$ are both functions of the frequency and voltage, which need to be calibrated.

The components $P_{\text{cut}}$ and $P_{\text{depth}}$ are the tool tip motion in the cutting and depth directions due to the rotation of the tool beam holder as illustrated in Fig. 2(b). They are functions of the excitation voltage amplitude and frequency as well, which can be presented as follows:

$$P_{\text{cut}} = A * F_t(\omega)$$

(5)

$$P_{\text{depth}} = A * F_w(\omega)$$

(6)

where $A$ is the amplitude of the excitation signals; $F_t(\omega)$ and $F_w(\omega)$ are the ratios between the vibration amplitude and excitation voltage, which need to be calibrated.

Since the actual tool tip motion cannot be directly measured during cutting, the phase lags, $\theta$ and $\gamma$ in Eq. (3) and Eq. (4), between the measured sensor signals and the tool tip motion can be decomposed to the phase angle measured relative to a reference signal and the intrinsic phase angle between the tool tip motion and the reference signal determined from the mode shape. If we choose the excitation signal on the left side beam as the reference signal, the phase lags can be compensated as follows:

$$\theta = \theta_s + \psi_t$$

(7)

$$\gamma = \gamma_s + \psi_n$$

(8)

where $\theta_s$ and $\gamma_s$ are the measured phase angles from the Mid and Tool sensors with respect to the reference signal. $\psi_t$ and $\psi_n$ are the phase differences between reference signal and the tool tip motion, which should be constants (according to the mode shape). They also need to be calibrated.

The inputs to the tool trajectory prediction model are the excitation amplitude, $A$, the driving signal phase angle, $\phi$, the excitation frequency, $\omega$, the measured voltage amplitudes of the Tool and Mid sensors, $V_t$ and $V_o$, and their phase angles, $\theta$ and $\gamma$, with respect to the reference signal. The outputs of the model are the real-time tool tip trajectory in the cutting and depth directions, $P_{\text{cut}}(t)$ and $P_{\text{depth}}(t)$. The coefficients in the model that need to be calibrated are $\alpha$, $\beta$, $F_t(\omega)$, $F_w(\omega)$, $\psi_t$, and $\psi_n$. They are in general functions of the excitation voltage and frequency.
Figure 3: Vibration tool with sensors: measurement setup of the (a) transverse and (b) normal modes.

The coefficients can be identified and calibrated by exciting the normal and transverse modes individually. It should be noted that in the calibration procedures the tool tip motion, \( P_{cut}(t) \) and \( P_{depth}(t) \), can be measured. For pure normal mode excitation, the input phase angle, \( \phi \), is set to 180°. The transverse mode amplitudes, \( P_{cut} \cos(\phi/2) \) and \( P_{depth} \cos(\phi/2) \), vanish in Eq. (3) and Eq. (4), then the coefficients, \( \alpha \) and \( \beta \), can be calculated considering the simplified Eq. (3) and Eq. (4), as:

\[
\alpha = \frac{P_{cut}(t)}{V_t} \quad (9) \\
\beta = \frac{P_{depth}(t)}{V_o} \quad (10)
\]

where \( |\cdot| \) indicates the extraction of the signal amplitude.

The phase lags, \( \psi_n \) and \( \psi_t \), are directly calculated in the normal mode using Eq. (7) and Eq. (8). For pure transverse mode excitation, the coefficients \( F_n(\omega) \) and \( F_t(\omega) \) can be calculated through the following relationships:

\[
F_n(\omega) = \frac{\mid P_{cut}(t) - a V_t \sin(\omega t + \theta) \mid}{\mid P_{cut}(t) \mid} / A \quad (11) \\
F_t(\omega) = \frac{\mid P_{depth}(t) - \beta V_o \sin(\omega t + \gamma) \mid}{\mid P_{depth}(t) \mid} / A \quad (12)
\]

EXPERIMENTAL RESULTS AND DISCUSSIONS

The assembled portal frame structure is shown in Fig. 3. A standard commercial triangular type cutting insert was fixed to the tool holder. Four piezoelectric plates (26 x 16 x 0.7 mm, PZT-4) were symmetrically attached to both sides of the two vertical beams through a conductive epoxy binder. At the center of the horizontal beam, one piezoelectric plate sensor as the Mid sensor (8 x 5 x 0.3 mm, PZT-5H) was mounted. Another piezoelectric plate sensor as the Tool sensor (5 x 5 x 0.3 mm, PZT-5H) was glued to the tool holder beam. Stainless steel screws (316L) were assembled on two sides of the frame structure to tune the resonant frequencies of the two orthogonal vibration modes. The assembled elliptical vibration tool was mounted on a vibration isolation table through a base adaptor. The sinusoidal excitation signals were programmed in LabVIEW, generated using a data output card (National Instruments USB-6363), and amplified by a piezo amplifier (TREK PZD 350). A capacitance displacement sensor (MicroSense 5501) was used to monitor the dynamic displacement of the cutting tool. The normal and transverse motions were measured respectively in two orthogonal directions. The acquired data were recorded using a mixed signal oscilloscope (Tektronix MSO2004B).

A frequency sweep test was first carried out to evaluate the resonant frequencies of the transverse and normal modes. The anti-symmetric bending excitation was first applied to identify the transverse mode. The voltage amplitude was set at 150 V and the excitation phase shift between two channels was 0. The frequency was swept from 15 kHz to 25 kHz and the vibration amplitude response was measured in the cutting direction. Similarly, the symmetric bending excitation was performed to identify the normal mode. The voltage amplitude was still kept at 150 V and the excitation phase angle was set to 180°. The frequency was changed from 15 kHz to 25 kHz and the displacement response was measured in the depth direction. The frequency response curves of the tuned structure are plotted in Fig. 4. The 1st normal mode was identified at 19,100 Hz, while the 2nd transverse mode was measured at 19,140 Hz. The two resonant modes were coupled very well in the frequency domain due to the added tuning components.

Then the model-specific coefficients \( \alpha, \beta, F_n(\omega), F_t(\omega) \), and the phase lags, \( \psi_n \) and \( \psi_t \), were calibrated. If the linear model assumption (superposition theory) is valid, these coefficients should be either a constant or a function of the frequency only. Here, we assumed a general case that they could be functions of both the excitation voltage and frequency to study their influences on the model coefficients.

The pure normal mode was first excited by setting the input phase angle to 180°. The excitation frequency was varied from 18,700 Hz to 19,500 Hz. The excitation voltage amplitude was varied from 130 V to 170 V at an increment of 10 V for each excitation frequency. The relationship between the input variables (frequency and voltage) and the model coefficients, \( \alpha, \beta, \psi_n, \psi_t \), are plotted in Fig. 5 and Fig. 6. The influence of voltage amplitude on the parameter value is represented by the error bar in the figures. The actual data point is taken as the average of the measured results from 130 V to 170 V.

In Fig. 5, the coefficients, \( \alpha \) and \( \beta \), are insensitive to the input voltage, but varied with the input frequency, which is consistent with our linear model assumption. The frequency-dependent variations are resulted from the change of mode amplitudes, or the difference between the excitation frequency and the resonant frequency of the normal model. The slight variations of the coefficients corresponding to the change of input voltage, shown as the error bars, are mostly due to the measurement noise. The coefficients, \( \alpha \) and \( \beta \), were then fitted to a 4th order polynomial function of the frequency by averaging all the curves under different input voltages. Figure 6 shows the effects of voltage and frequency on the phase lags, \( \psi_n \) and \( \psi_t \). According to their
definitions, they should be the phase shifts between the reference signal and the tool tip motion in the normal and transverse modes, which are constants and determined by the mode shapes. The results have verified this assumption, as their values are insensitive to both the frequency and voltage inputs. Similarly, the pure transverse model was excited by setting the input phase angle to 0°. The model coefficients $F_U(\omega)$ and $F_W(\omega)$ were calculated according Eq. (11) and Eq. (12). They were both functions of the excitation frequency only, as shown in Fig. 7. The relationship was fitted to a 4th order polynomial function.

After all the model coefficients were determined, the model accuracy was verified. The given input parameters were the excitation voltage, input phase angle, and driving frequency. The measured input variables were the measured voltage amplitudes of the Tool and Mid sensors, $V_T$ and $V_M$, and their phase angles, $\theta_T$ and $\gamma_T$, with respect to the reference signal. The tool trajectories were calculated according to Eq. (3) and Eq. (4).

In the first set of experiment, two arbitrary input phase angles (30° and 105°), and two excitation frequencies (19,000 Hz and 19,300 Hz), were chosen. The voltage amplitude was kept at 150 V. The tool trajectories were measured using two capacitance sensors and compared with the calculated ones from our model, as shown in Fig. 8. In the second set of experiment, the input voltage amplitude and frequency was kept at 150 V and 19,140 Hz, while the input phase angle was varied from 15° to 165°. The measured and calculated trajectories are plotted in Fig. 9. From the results, the predicted 2D tool trajectories match the measured ones with a high degree of accuracy. The angular error was within 5°, while the area error percentage was also moderate. The changes of vibration amplitude, orientation, and the aspect ratio were all captured by the model, which has verified our model efficacy. The two sets of verification experiment represent two types of common scenarios during operation. The model could work along with a PLL algorithm to track the tool trajectory even with the change of excitation frequency as shown in Fig. 8. In addition, the tool trajectory shape is usually controlled by changing the input phase angle, which is well predicted by our model, as shown in Fig. 9.

CONCLUSIONS

In summary, an analytical model for the online prediction of tool trajectory of an elliptical vibration cutting tool has been proposed. The tool utilized the coupled resonant modes of a normal and a traverse vibration modes. It was based on a portal frame structure with adjustable screws for tuning of the specific
resonant frequencies. Two piezoelectric sensors were installed on the tool holder and the horizontal center beam. Considering the mode superposition and the additional rotation and bending of the tool holder, we developed the relationship between the tool tip motion and the measured sensor information as well as the excitation signals. The effects of excitation frequency and voltage on the model coefficients were studied. The model was then verified by experimental results. The tool trajectory shape, including the vibration amplitude, orientation, and aspect ratio, was well captured by the model. The 2D tool trajectory could be estimated without the direction measurement of the tool tip motion.

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