Design, Control and Cutting Process for a
Three-Degree-of-Freedom Ultrasonic Vibration Tool
Holder

by

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Abstract

Ultrasonic vibration-assisted cutting is a popular unconventional manufacturing process with lower cutting forces and less heat generation. Special tools are required to excite high-frequency vibrations at the tool tip during cutting; however, there is no ultrasonic vibration actuated tool holder for general-size milling or drilling tools reported in the literature. This thesis presents the design of a novel three-degree-of-freedom (3DOF) ultrasonic vibration tool holder with a sensorless control system. In addition to proposing a mechatronics design, this thesis presents the cutting dynamics and mechanics exhibited by the developed vibration tool holder.

The 3DOF ultrasonic vibration tool holder is designed for milling and drilling operations. 3DOF vibrations are generated by the actuator consisting of three groups of piezoelectric rings actuating in the X-, Y-, and Z-directions at the natural frequencies of the structure. The vibrations excited in the XY produce an elliptical locus to assist milling process. The vibrations along Z-axis are used in drilling operations.

A sensorless method is developed to track and control the frequency and amplitude of ultrasonic vibrations produced by the 3DOF vibration tool holder during machining. A dynamic model of the actuator is first established to obtain a transfer function between the supply voltage and driving current. An observer with Kalman filters in each actuator direction is designed to estimate the vibrations during cutting to closed-loop control the amplitude and track the resonance

The dynamics of the ultrasonic elliptical vibration-assisted milling operations is analyzed to assess the system stability. The chip thickness is modeled by considering the rigid body motion of the tool, regenerative vibration and ultrasonic
vibration. The loss of contact between the tool and workpiece at the ultrasonic vibration excitation frequency is considered in evaluating the directional factors. The stability of the system is solved using the semi-discrete time-domain method and verified experimentally.

The effects of ultrasonic vibration assistance in cutting of Ti-6Al-4V are investigated. A plastic chip flow model is developed to predict the stress and temperature variations in the primary shear zone. Simulation results show that the temperature in vibration-assisted cutting is much lower than that for conventional cutting.
Lay Summary

The aerospace/aviation and biomedical industries demand high-performance cutting tools with longer tool life and better surface finish in cutting of difficult-to-cut materials such as carbon-fiber-reinforced polymer and titanium alloys. Investigations have shown that ultrasonic vibration-assisted cutting processes have lower cutting forces and temperatures, with longer expected tool life.

This thesis expands the applications of ultrasonic vibration-assisted cutting to milling and drilling by developing a novel three-degree-of-freedom ultrasonic vibration tool holder and a corresponding sensorless control system. In addition to presenting a cutting tool design, this thesis investigates the dynamics of ultrasonic vibration-assisted milling. Moreover, the effects of ultrasonic vibration in the cutting of Ti-6Al-4V are analytically studied. The research presented in this thesis will benefit the manufacturing industry in machining advanced hard-to-cut material.
Preface

This thesis presents the design, control and cutting process of a novel three-degree-of-freedom (3DOF) ultrasonic vibration tool holder. This research was performed in Manufacturing Automation Laboratory at The University of British Columbia under the supervision of Dr. Yusuf Altintas. The contributions of each chapter are as follows:

- A version of Chapter 3, which focuses on the mechatronics design of a 3DOF ultrasonic vibration tool holder, has been published in [24] [J. Gao and Y. Altintas. Development of a three-degree-of-freedom ultrasonic vibration tool holder for milling and drilling. IEEE/ASME Transactions on Mechatronics, 24(3):1238-1247, June 2019.]. The author of this thesis was the lead investigator and was responsible for the development of the tool holder and its corresponding instrumentation, including a digital controller, conditioner circuits and power amplifiers. Dr. Altintas supervised the project and edited the manuscript.

- A version of Chapter 4, which presents a sensorless control system design for the 3DOF ultrasonic vibration tool holder, has been published in [26] [J. Gao, H. Caliskan, and Y. Altintas. Sensorless control of a three-degree-of-freedom ultrasonic vibration tool holder. Precision Engineering, 58:47-56, 2019.]. The manuscript was written by the author of this thesis and was edited by Dr. Caliskan and Dr. Altintas. In addition to editing the paper, Dr. Caliskan contributed to the state-space model of periodical signals used to formulate Kalman filters for the phase estimations from the supply voltage to the transforming current and the vibration amplitudes.
A version of Chapter 5, which investigates the dynamics and stability of the elliptical vibration-assisted milling process, has been accepted by the CIRP Journal of Manufacturing Science and Technology [J. Gao, Y. Altintas. Chatter stability of synchronized elliptical vibration assisted milling.]. The author of this thesis was the lead author and was responsible for the model of the elliptical vibration assisted milling dynamics and deriving a stability solution using the semi-discrete method. Dr. Altintas supervised the project and edited the manuscript.

A version of Chapter 6, which describes the effect of ultrasonic vibration on chip formation in the cutting of Ti-6Al-4V, has been published in [25] [J. Gao and X. Jin. Effects of ultrasonic vibration assistance on chip formation mechanism in cutting of Ti-6Al-4V. Journal of Manufacturing Science and Engineering, 141(12), 10 2019.]. The author of this thesis was the lead investigator and was responsible for analytically modeling the cutting mechanics of Ti-6Al-4V under ultrasonic vibration and for conducting experiments. Dr. Jin edited the manuscript. Moreover, the cutting mechanics model of Ti-6Al-4V proposed in Chapter 6 has been applied to milling process to predict cutting forces, and a paper regarding ultrasonic vibration-assisted milling of Ti-6Al-4V has been submitted as [X. Jin, J. Gao, Y. Altintas. Mechanics of Elliptical Vibration Assisted Milling of Titanium Ti-6Al-4V. CIRP Annals-Manufacturing Technology.]. The author of this thesis was responsible for the model development and conducting experiments.
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<td>$A_0$, $B_0$</td>
<td>Vibration amplitudes</td>
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<td>$b$</td>
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<td>Static capacitance of piezo</td>
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<td>Mechanical equivalent capacitance</td>
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<tr>
<td>$c$, $c_x$, $c_y$</td>
<td>Damping coefficient</td>
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<td>$d_{33}$</td>
<td>Piezoelectric strain constant along the thickness direction</td>
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<td>$F_{ext}$</td>
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<td>$g_{1,j}, g_{2,j}, g_{3,j}, g_j$</td>
<td>Unit step function for tool-workpiece disengagement</td>
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<td>$h_{0,j}$</td>
<td>$j$th tooth static uncut chip thickness</td>
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$h_{r,j}$ Resultant uncut chip thickness of $j$th tooth
$I$ Area momentum of inertia in section 3.2, Driving current
$I_t$ Transforming current
$K_M$ Electromechanical coupling coefficient
$K_t, K_r$ Cutting force coefficients
$K_{sp}$ Indentation coefficients
$k$ Stiffness in Chapters 3, 4 and 5, thermal conductivity in Chapter 6
$k_x, k_y$ Stiffness
$L$ Length of the piezo actuator
$L_m$ Mechanical equivalent inductance
$m$ Mass
$N$ Number of flutes
$N_A$ Vibration magnification factor
$N_T$ Number of teeth
$q(t)$ State function
$R_e$ Tool edge radius
$R_m$ Mechanical equivalent resistance
$\dot{r}_j$ Velocity along radial direction for $j$th tooth
$s_{r,j}$ Resultant displacement of $j$th tooth
$T$ Tooth passing period in Chapter 5, temperature in Chapter 6
$T_0$ Room temperature
$T_m$ Melting temperature
$u(z,t)$ Displacement of bending mode
$V, V_{x0}, V_{y0}, V_0$ Voltage
$V_S$ Nominal cutting speed
$V_r$ Resultant cutting speed
$w$ Vibration displacement in $Z$
$x, \dot{x}$ Vibration displacement and velocity along $X$
$x_0$ Vibration amplitude along $X$
$\mathbf{\hat{s}}$ Model displacement vector
$Y_{33}$ Young’s modulus of piezo along the thickness direction
$\alpha$ Rake angle
$\beta_e$ Separation angle
\( \beta_k \) Wave coefficient of \( k \)th bending mode
\( \varepsilon \) Normal strain
\( \gamma, \gamma_p \) Shear strain
\( \dot{\gamma} \) Shear strain rate
\( \mu \) Friction coefficient
\( \psi \) Force factor of piezo, transforming gain
\( \psi_{xx}, \psi_{yy}, \psi_{xy}, \psi_{yx}, \psi_{z} \) Transforming gains
\( \rho \) Mass density
\( \Omega \) Spindle rotation speed
\( \omega_n \) Natural frequency
\( \omega_e \) Excitation frequency
\( \phi \) Phase from voltage to current
\( \phi \) Shear angle
\( \phi(t) \) Immersion angle
\( \phi_j \) Immersion angle of \( j \)th tooth
\( \phi_p \) Tooth spacing angle
\( \sigma \) Normal stress
\( \zeta \) Damping ratio
\( \theta, \theta_{xy} \) Phase difference between \( x \) and \( y \) vibrations
\( \tau_j \) Time delay of \( j \)th tooth
\( \tau_s \) Shear stress in the primary shear zone
List of Abbreviations

AC Alternating current
AD Amplitude detector
ADC Analog-to-digital converter
CFRP Carbon-fiber reinforced plastics
CNC Computer numerical control
DAC Digital-to-analog controller
DOF Degree of freedom
FE Finite element
FFT Fast Fourier transform
FRF Frequency response function
MRR Material removal rate
PD Phase detector
PI Proportional integral
PLL Phase lock loop
PWM Pulse-width modulation
RPM Revolution per minute
SEM Scanning electron microscope
SP Separation point
VCA Voltage controlled amplifier
Acknowledgments

First and foremost, I want to thank my supervisor, Prof. Yusuf Altintas, for his unconditional support throughout my Ph.D. years. He offered me a chance to continue my studies as a Ph.D. student at Manufacturing Automation Laboratory when I was in my low moments. Moreover, his insights and passion for manufacturing research always encourage me to pursue further achievements, and this experience is a lifetime of wealth for me.

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I thank all members of the Manufacturing Automation Laboratory and the Precision Mechatronics Laboratory. In the family-like atmosphere, I never hesitated to ask any questions, and they always offered help and provided guidance.

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Chapter 1

Introduction

Ultrasonic vibration-assisted cutting is an unconventional manufacturing process. The cutting tool is forced to vibrate at a high frequency ($\geq 15$ kHz) with a small amplitude (10-20 $\mu$m). Ultrasonic vibration-assisted cutting has been demonstrated to reduce cutting forces and heat generation in comparison to conventional machining operations. As a result, ultrasonic vibration-assisted cutting has become a popular method in industry, particularly in machining of carbon-fiber reinforced plastics (CFRP), which is widely used in light-weight, high-strength automotive and aircraft parts [49, 96]. Ultrasonic vibrations are also widely applied in precision manufacturing of hardened steel molds with mirror surface finishes [79]. However, to finish ultrasonic vibration-assisted cutting process, special tools with vibration generations at tool tip are required, which may increase the cost of the manufacturing process.

The fundamental feature of the ultrasonic vibration-assisted cutting process is to generate high frequency ($\geq 15$ kHz) vibration at the cutting tool tip in order to make additional intermittent contact between the cutting tool and workpiece. The ultrasonic vibration assistance can be generated along the tangential direction of cutting process and the normal direction, as shown in Figure 1.1 a) and b). Alternatively, the vibration assistance can be generated in both tangential and radial directions simultaneously to form an elliptical vibration locus to change the friction between chip and tool’s rake face as shown in Figure 1.1c).

Piezoelectric actuators are commonly used in ultrasonic vibration-assisted cut-
ting to deliver vibrations with desired frequencies and amplitudes. These actuators generally resonate at their natural frequencies to increase the vibration amplitude at high frequencies. Various ultrasonic vibration-assisted cutting tools have been developed for different machining processes. In the past, single-axis vibration and two-dimensional elliptical vibration systems have been investigated. Ultrasonic vibration assistance was first applied to turning operations, which require a stationary cutting tool, due to their lower-complexity vibration tool design [56, 76]. Later, two-dimensional elliptical vibration cutting was invented by Shamoto and Moriwaki [78], and then it was applied at ultrasonic vibration [58]. The elliptical vibration was applied to the turning process in which a mirror surface is applied to a hardened steel mold. Machining processes using rotary cutting tools, such as milling and drilling can reach higher productivity for mass production. Thus, machine tool companies have developed tool holders to generate ultrasonic vibrations along the rotating spindle axis for rotary cutting operations. For example, DMG MORI developed an ultrasonic vibration-assisted machining center with vibrations along the spindle axis for grinding metals and drilling CFRP materials [76]. Acoustech Systems presented a single axis ultrasonic vibration tool holder that is compatible with standard spindle mounting systems (HSK and CAT) for drilling applications [56]. Thus far, most research on two-dimensional elliptical vibration cutting has been performed with stationary cutting tools due to the limitations of vibration generation devices, and ultrasonic vibration-assisted rotary processes have been primarily studied for single-axis vibration devices.

As shown the research flow chart in Figure 1.2, the first aim of this thesis is to
develop a three-degree-of-freedom (3DOF) ultrasonic vibration cutting tool holder for both milling and drilling operations using general-purpose cutting tools. The proposed holder has a standard mounting interface (HSK or CAT) for CNC (computer numerical control) machines; the holder can deliver vibrations in the XY plane to assist milling along the tangential and radial directions and generates vibrations along the Z-axis for drilling applications. An actuator with three groups of piezoelectric rings is designed to create vibrations along the three axes, respectively. The vibration mode shapes are selected to maximize the amplitudes, and the piezoelectric components are then parametrically designed using a lumped parameter model. Finite element analysis (FEA) is used to verify the natural frequencies and the mode shapes before the holder was manufactured. The prototype of the 3DOF ultrasonic vibration tool holder shown in Figure 1.3 was tested with cutting experiments. In addition to the actuator design, a corresponding electrical system, including power amplifiers, output signal-conditioning circuits and current sensing
The ultrasonic vibrations are produced by resonating the natural modes of the tool holder structure. However, the structural dynamics of the piezo-driven tool holder structure vary under cutting loads, which must be detected to resonate the structure. The vibration amplitude at ultrasonic frequencies also decreases under cutting load disturbance, which must be maintained at the desired reference level. Conventional closed-loop control methods reported in the literature use vibration sensor feedback to track the frequency and amplitude of the ultrasonic vibrations during cutting with additional piezo stacks or strain gauges mounted on the actuator’s surface. The vibration feedback is used to track the resonance and to control the vibration amplitude; moreover, the feedback can eliminate crosstalk arising between orthogonal piezo actuators in elliptical vibration by compensating the excitation voltage [80]. However, these dedicated vibration sensors increase the cost of the tool and the complexity of signal transmission during rotary cutting applications. Therefore, sensorless control methods were implemented to track the resonance frequency and keep desired amplitudes. In contrast to existing methods, a new sensorless closed-loop control method, which can be adaptive to different cutting tool structural dynamics, is proposed for the developed 3DOF ultrasonic vibration tool holder to track the resonance frequencies and amplitudes of ultrasonic vibrations.
vibrations during machining. In the control system, the feedbacks are estimated from a dynamic model of the piezo actuator using Kalman filters, and the crosstalk arising in elliptical vibrations is eliminated. This control system was implemented during cutting using the experimental setup shown in Figure 1.4 for milling and drilling operations.

For the effective operation of the ultrasonic vibration tool, the dynamics of the vibration-assisted process must be modeled and analyzed. Chatter, which is a type of regenerative vibration that can lead to unstable cutting, should be avoided during operation. Thus, a dynamics model of elliptical vibration-assisted milling is proposed in the third part of this thesis. The chatter stability is predicted from the proposed dynamics model to select the desired spindle speed and depth of cut. In milling, the developed 3DOF ultrasonic vibration tool holder can deliver vibrations along the tangential and radial directions at the same excitation frequency to produce an elliptical locus. In the dynamics model, the tangential vibration generates high-frequency tool-workpiece separations, which change the dynamic milling co-
efficients and thus affect the milling stability. Additionally, the radial vibration affects the indentation between the tool and workpiece periodically, which alters the process damping and the milling stability.

Ultrasonic vibration assistance has been experimentally demonstrated in cutting of various materials such as hardened steel, titanium alloy, CFRP, and glass. However, a analytical physical model elucidating the mechanisms of vibration assistance has not yet been established in the literature. The mechanism of vibration assistance may vary for different materials. In this thesis, a titanium alloy, Ti-6Al-4V, is studied as an example to demonstrate the principle of ultrasonic vibration assistance. The effect of ultrasonic vibration assistance on shear band formation and chip segmentation in orthogonal cutting of Ti-6Al-4V is investigated. A chip flow model is developed to determine the shear flow mechanism in the primary shear zone and to explain the suppression of adiabatic shear bands when ultrasonic vibration assistance is applied along the tangential cutting direction. The chip geometries and cutting forces predicted by the proposed analytical model are verified by orthogonal cutting experiments.

The remainder of this thesis is structured as follows. Chapter 2 reviews previous literature pertaining to ultrasonic vibration actuator design, resonance and amplitude tracking, chatter stability of vibration-assisted cutting processes and models of Ti-6Al-4V cutting. The 3DOF ultrasonic vibration actuator design is presented in Chapter 3, with performance test results obtained for a prototype. A sensorless control system of the developed 3DOF ultrasonic vibration tool holder is presented in Chapter 4. Chapter 5 proposes a dynamics model of ultrasonic vibration-assisted milling to predict the stability of the developed tool holder. Chapter 6 presents the effects of ultrasonic vibration assistance in cutting of Ti-6Al-4V with a physical model of chip formation. The shear flow in the primary shear zone is analyzed based on the experimental observations to illustrate the advantages of vibration assistance. The conclusions obtained from this thesis are presented in Chapter 7 along with a discussion of future work.
Chapter 2

Literature Review

2.1 Overview
This thesis focuses on the design and control of a novel 3DOF ultrasonic vibration cutting tool, the dynamics of the elliptical vibration-assisted milling process, and the mechanics of vibration-assisted cutting of Ti-6Al-4V. This chapter reviews past research related to these areas. Section 2.2 presents the state of the art in ultrasonic vibration cutting tool design, and section 2.3 discusses existing control methods of ultrasonic vibration during cutting. Milling dynamics models and the studies related to the stability of vibration-assisted cutting are reviewed in section 2.4. Section 2.5 reviews cutting mechanics models of Ti-6Al-4V, with a focus on the formation of adiabatic shear bands, and presents experimental studies on vibration-assisted cutting of Ti-6Al-4V.

2.2 Ultrasonic Vibration Cutting Tool Design
Single-axis and two-dimensional elliptical vibration systems have been investigated in both academia and industry. DMG MORI developed an ultrasonic vibration-assisted machining center with vibration along the spindle axis for grinding metals and drilling CFRP materials [76]. Recently, Acoustech Systems presented a single-axis ultrasonic vibration tool holder that is compatible with standard spindle mounting systems (HSK and CAT) for drilling applications [56, 84]. The effect of
ultrasonic vibrations on machining performance has been experimentally investigated by several researchers using different vibration cutting tools. For example, Li et al. presented the drilling performance of ceramic matrix composites with a single-axis rotary ultrasonic vibration assistance [46], with an observed reduction in cutting forces and a better surface finish. Liu et al. studied single-axis rotary ultrasonic machining for drilling of brittle materials to achieve larger material removal rates (MRRs) [47]. Lucas et al. used two full ring piezos to excite the coupled longitudinal-torsional vibration mode of a drill for ultrasonic vibration-assisted drilling [3]. Jin and Murakawa developed a one-dimensional ultrasonic vibration-assisted turning tool based on the Langevin transducer, which was then used to improve the surface roughness in the turning of hardened steel [34]. Xiao et al. reported the suppression of chatter during turning by using a single-axis actuator to achieve intermittent contact between the cutting tool and the workpiece [95].

Elliptical vibration cutting was proposed by Shamoto and Moriwaki [78] for mirror surface machining of hardened steel dies and molds; in this work, they mounted four piezoelectric plates on the surface of the tool to generate an elliptical vibration locus [58]. Later, Shamoto et al. developed a controller for elliptical ultrasonic vibration cutting [80]. The fundamental objective behind imposing elliptical vibrations on the tool is to reduce the friction force, which in turn reduces the heat generation and the tool wear [82]. With this approach, carbon-rich diamond tools can cut carbon-hungry steel alloys due to the significant reduction in heat. It has been found that the main reason for the tool wear reduction is a protective oxide film formed on the newly-developed surface during the non-contact period [22]. Shamoto et al. expanded upon the elliptical ultrasonic vibration assistance to develop a 3DOF ultrasonic vibration tool to sculpture free-form surfaces with a specially designed diamond cutting tool [81][87]. This tool holder applies 3DOF ultrasonic vibration for ultra-precision shaving of hardened steel dies, where the cutting force amplitudes are much smaller (approximately 2-5 N) than those obtained in regular machining applications. Jung et al. used a stationary ultrasonic vibration transducer with ring-type piezo stacks to excite both axial and bending modes of the structure for shaping applications [38]. The elliptical vibration cutting was first applied to micro texturing of die steel [88]. Later, Guo and Ehmann
Table 2.1: Summary of ultrasonic vibration cutting tool design

<table>
<thead>
<tr>
<th>Manufacturing process</th>
<th>SDOF assistance</th>
<th>2DOF assistance</th>
</tr>
</thead>
<tbody>
<tr>
<td>Turning</td>
<td>[34, 95]</td>
<td>[58, 80, 81]</td>
</tr>
<tr>
<td>Drilling</td>
<td>[3, 46, 47, 56, 84]</td>
<td>[49]</td>
</tr>
<tr>
<td>Milling</td>
<td>[76]</td>
<td></td>
</tr>
<tr>
<td>Grinding</td>
<td>[76]</td>
<td>[27]</td>
</tr>
<tr>
<td>Shaping</td>
<td></td>
<td>[38]</td>
</tr>
<tr>
<td>Texturing</td>
<td></td>
<td>[20, 29, 98]</td>
</tr>
</tbody>
</table>

developed an elliptical vibration generator combining two Langevin transducers for high speed texturing by imposing elliptical vibrations on the tool [29]. Yang et al. analytically designed an elliptical vibration tool based on a portal frame structure using the Euler-Bernoulli beam model for surface texturing processes [98]. Dow et al. excited the axial and bending modes of a turning tool at 40 kHz with piezo plates to generate features in nanocoining processes [20]. Kim and Loh designed and implemented an elliptical vibration tool with two parallel piezoelectric actuators for micro grooving and observed a reduction in cutting forces and burrs [41]. Liu et al. developed an elliptical ultrasonic vibration device for CFRP drilling with diamond abrasive core drills and reported better surface finish with less delamination [49]. Geng et al. proposed a 2DOF ultrasonic vibration cutting tool for peripheral grinding of CFRP using half-ring piezos [27].

Table 2.1 summarizes the existing ultrasonic vibration cutting tool designs for different cutting processes. Due to the limitations of existing ultrasonic vibration generation devices, most research on two-dimensional elliptical vibration cutting has been performed with stationary tools, such as turning, texturing and shaping tools. For rotary cutting processes, studies on ultrasonic vibration assistance have focused on single-axis vibration devices, except for the works described in [49] and [27]. Moreover, the existing rotary ultrasonic vibration generation devices are designed for specific applications. In this thesis, a novel 3DOF ultrasonic vibration tool holder is presented in Chapter 3, which can be used for milling and drilling with various cutting tools.
2.3 Control for the Ultrasonic Vibration Actuator

During vibration-assisted cutting, the ultrasonic vibrations are produced by resonating the natural modes of tool holder structure. However, the structural dynamics of the piezo driven tool holder structure vary under cutting loads, which need to be detected to resonate the structure. The amplitude of the vibration at ultrasonic frequency also reduce under cutting load disturbance, which need to be maintained at the desired, reference level. The conventional close-loop control methods use sensor feedback to track the frequency and amplitude of the ultrasonic vibrations during cutting. Shamoto et al. used two more piezo stacks mounted on the actuator’s surface as feedback sensors in elliptical vibration-assisted precision diamond turning [80]. The vibration feedback was used to track the resonance and control the vibration amplitudes, and they removed the crosstalk between the orthogonal piezo actuators by compensating the excitation voltage with the sensor feedback. Babitsky et al. also developed an auto-resonant control with multiple sensors to track the resonance during ultrasonic vibration cutting [6]. Later, Suzuki et al. sculptured the hardened steel surface by adjusting the supply voltage for open loop control of the elliptical vibration amplitudes [88]. They controlled the vibration amplitude within 4 µm along the thrust direction within 300 Hz bandwidth. Some researchers focused on the control of rotary ultrasonic vibration tool in the literature. Ketelaer installed a piezoelectric ring with four segments coaxially in an ultrasonic vibration tool holder to detect vibrations in multiple directions during cutting to monitor and control the tool holder [40]. Chen et al. bonded a strain gauge on the ultrasonic vibration actuator surface to detect the vibration to track the resonance frequency, and they designed a pancake coil to transmit the sensor output from the rotary tool [15]. Industrial companies developed sensorless control methods for commercial ultrasonic vibration cutting tools. Taga Electrics developed a vibration control system to track the resonance frequency using the supply voltage and driving current through electronic circuits for general ultrasonic vibration apparatuses [30]. Later, Taga Electrics used the developed sensorless control system in the ultrasonic vibration cutting process to track the resonance frequency and monitoring the cutting process [31]. Short developed a closed-loop control system to track the resonance for ultrasonic vibration assisted drilling process by
monitoring the power consumption in power amplifiers [85].

The piezoelectric materials exhibit hysteresis and nonlinearities that depend on the driving conditions [28, 52], which affect vibration amplitude generated by the piezo actuators. Newcomb et al. showed that the linearity of a piezoelectric ceramic actuator can be improved if the applied electric charge, rather than the applied voltage, is varied to control the actuator’s displacement [60]. Ryba et al. compensated the hysteresis of piezoelectric actuator using a polynomial model [71]. Woronko et al. used a sliding mode controller to handle the nonlinear gain of the actuator [94].

The use of dedicated sensors in control systems for vibration-assisted cutting tools increase the cost of the system and the complexity of signal transmission during rotary cutting applications. Additional slip rings or coils of the rotary transformer are necessary for feedback signals transmission to the controller unit. Thus, this thesis proposes a sensorless closed loop control method in Chapter 4 to track the resonance frequency and amplitude of ultrasonic vibrations during machining. In addition to the single-axis controller, the crosstalk between two axes is compensated using the estimated feedback.

### 2.4 Ultrasonic Vibration-Assisted Milling Process and Its Dynamics

Vibration-assisted cutting has been applied to various cutting operations including turning [79], shaping [29, 38, 88], grinding [27], and drilling [56, 69]. Milling is an intermittent-contact process in which each cutting tooth touches the workpiece periodically within the entry and exit angles at the tooth-passing frequency. The externally applied ultrasonic vibrations generate additional, high-frequency tool-workpiece separations, which change the dynamics of the milling process. The vibrations are either delivered through the workpiece or the cutting tool. Jin and Xie developed a workpiece carrying piezo vibration stage to achieve vibration assistance along the feed or normal direction of milling with varying frequencies (5-11 kHz) for micro-cutting of glass and reported an improved surface finish [36]. Chen et al. used a piezo vibration stage to generate synchronized vibrations along the feed and normal directions at 8 kHz in a micro-milling process to produce micro-
texture patterns on the workpiece surface [16]. Ni et al. applied vibrations at 20 kHz along the feed direction from the workpiece side and reported an improved surface finish and reduced cutting forces in the milling of Ti-6Al-4V [61]. It is preferable to deliver vibrations from the tool side in order to machine large workpieces with ultrasonic vibration assistance. Chen et al. applied ultrasonic vibration along the axial direction in helical milling of Ti-6Al-4V and reported reductions in both the cutting forces and heat generation [14]. Elliptical vibration-assisted cutting was proposed by Shamato and Moriwaki in which vibrations along the tangential and thrust directions are synchronized in diamond turning and milling [78]. Then, it was applied to various workpieces at various cutting conditions afterwards. Moriwaki and Shamato also developed an elliptical vibration milling machine using an eccentric sleeve to deliver vibrations at 167 Hz and reported a better surface finish in the milling of hardened steel [59]. Liu et al. applied ultrasonic elliptical vibrations along the tangential and radial directions in the milling of Ti-6Al-4V [50] and demonstrated that rotary ultrasonic elliptical vibration-assisted milling significantly improved the integrity of the machined surfaces. In this thesis, Gao and Altintas developed an ultrasonic vibration tool holder that can vibrate independently in three orthogonal directions [24]. A group of half-ring piezos is used to excite the third bending mode of the holder to generate ultrasonic vibrations along the tangential and radial axes in milling operations. The peak-to-peak amplitude of vibration along the X and Y axes can reach 20 \( \mu \)m at 17 kHz, and the two vibrations share the same excitation frequency with an adjustable phase difference to form an elliptical vibration locus. The excited elliptical vibration is synchronized to the spindle rotation, and the tool holder can be installed on a CNC machining center using standard spindle interfaces such as CAT or HSK.

The dynamics of the vibration-assisted machining process must be modeled to identify chatter-free cutting conditions. Although the chatter stability of conventional machining operations has been studied extensively in the literature [5, 33], the stability of milling under ultrasonic vibrations has not been sufficiently investigated. In a previous work, Xiao et al. numerically and experimentally studied the chatter stability of a 1DOF vibration-assisted turning process and reported a larger depth of cut [95], but they did not predict the stability lobes of the process. Tabatabaei et al. applied a Fourier transform to a pulsating cutting force with a de-
lay of spindle rotation in turning and numerically predicted the stability in the time domain [89]. Ma et al. excited ultrasonic elliptical vibration on a turning tool and analyzed the chatter stability in the frequency domain using the Nyquist criteria [51], and they considered a periodic contact time of the actuator by using a Fourier series, and applied a dynamic orthogonal cutting model to determine the stability. Ko and Tan implemented ultrasonic vibration along the spindle axis during milling and observed a reduction in chatter marks on the surface due to improved chatter stability [42]. Recently, Wan et al. provided the first analytical chatter stability model of a vibration-assisted milling process using the semi-discrete method [93]. In Wan’s study, the vibration was excited along the feed or normal direction from the workpiece side, and the high-frequency tool-workpiece separations dynamically changed the time delay and hence shifted the stability lobes. Consequently, the authors concluded that the critical depth of cut was not improved.

Thus far, no dynamic model has been reported in the literature for spindle-synchronized elliptical vibration-assisted milling processes. In Chapter 5, a model of elliptical vibration-assisted milling dynamics is proposed to predict the stability, and the elliptical vibration assistance is verified to improve the depth of cut due to additional high-frequency tool-workpiece separations, achieved by using the 3DOF ultrasonic vibration tool holder developed in Chapter 3 and the corresponding control system presented in Chapter 4.

2.5 Mechanics of Vibration-Assisted Cutting Ti-6Al-4V

The titanium alloy Ti-6Al-4V is widely used in biomedical and aerospace applications due to its light weight, high temperature strength and corrosion resistance. However, the low thermal conductivity of Ti-6Al-4V limits its machinability because the heat accumulates in the primary shear zone generating adiabatic shear bands and leads to segmented chips in high-speed cutting [18]. Chip segmentation causes stress and temperature oscillations at the tool edge, leading to faster tool wear and damage to the machined surface [8].

Chip segmentation in the cutting of Ti-6Al-4V includes localized shear banding and severe fractures [83]. As Recht suggested [70], the adiabatic shear banding commonly occurs in high speed cutting of high-strength metal alloys such as hard-
ened steel, titanium, and nickel alloys. When the cutting speed exceeds a critical value, there is not sufficient time for heat to dissipate in the primary shear zone, and consequently, the thermal softening of the workpiece material overcomes the strain hardening. As a result, adiabatic shear bands form with an instantaneous reduction in shear flow stress. When new material enters the primary shear zone, the shear stress increases, and a new cycle of shear band formation occurs. This oscillation in shear stress causes variations in the material plastic flow in the primary shear zone thus generating segmented chips. Komanduri and Brown investigated machined chips and reported that the plastic instability in the primary shear zone leads to adiabatic shear bands, which forms segmented chips in the cutting of Ti-6Al-4V [43]. Later, Komanduri and Hou noted that the temperature rise that occurs during the cutting of Ti-6Al-4V causes plastic instability [44], and the relationship between the uncut chip thickness and the temperature in the shear band was determined. Burns and Davies studied the shear band instability by developing governing equations of shear stress and temperature variations in the primary shear zone based on material plasticity [11, 19], and they concluded that an increased cutting speed leads to a Hopf bifurcation; thus, the shear stress and the temperature in the primary shear zone transition from constant to periodic steady-state behavior. Molinari et al. modeled the adiabatic shear banding mechanism in the cutting of Ti-6Al-4V, and predicted the width and pitch length of the shear bands [57].

Researchers have also investigated the segmentation mechanism by considering the occurrence of fractures in cutting of Ti-6Al-4V. Vyas and Shaw proposed that chip segmentation may be caused by periodic crack propagation, based on quick-stop micrographs of machined chips [92]. FE methods have been widely used in the literature to investigate the chip segmentation mechanism related to material failure under different criteria. Chen et al. implemented the Johnson-Cook constitutive model to simulate high-speed cutting of Ti-6Al-4V while considering the ductile failure of the material [13]. Ozel et al. simulated chip segmentation in cutting of Ti-6Al-4V by using an FE method with a modified Johnson-Cook model [66], and the effect of insert coatings on the cutting process was determined. Melkote et al. considered microstructure evolution-induced flow softening in simulations of chip segmentation and grain refinement in shear bands using an FE method [54]. Childs et al. performed FE simulations of chip segmentation us-
ing a material flow stress and failure model calibrated from experiments [17], and it was concluded that flow stress models involving large strain softening are not needed for predicting chip segmentation. Zhang et al. considered stress triaxiality in the material failure criteria of Ti-6Al-4V using FE simulations to predict the temperature and strain [99]. Liu et al. studied chip segmentations in the cutting of Ti-6Al-4V by implementing the Johnson-Cook failure properties combined with Johnson-Cook constitutive model in FE simulations with different tool coatings and rake angles [48]. Other researchers demonstrated that adiabatic shear banding and fracture failure are major factors influencing the formation of segmented chips during the cutting of Ti-6Al-4V. Lee and Lin suggested that adiabatic shear banding is the precursor of a fracture locus due to a localized high strain intensity [45], with the segmentation pitch depending on the frequency of shear band formation.

Conventionally, high-strength metal alloys are machined at relatively low cutting speeds with a small uncut chip thickness in order to avoid excessive heat accumulation in the primary shear zone [97]. However, the productivity of these machining operations is consequently limited. Various techniques have been reported to improve the machinability of high-strength metal alloys. For example, Ozel et al. investigated the effect of insert coatings on cutting of Ti-6Al-4V for improving the tool life [66]. Recently, Sagapuram and Viswanathan proposed a viscous sliding mechanism to study the nucleation and evolution of shear banding [72] and investigated fluid-like behavior with small viscosity in the bands. Subsequently, Sagapuram et al. proposed a passive geometric flow control method to suppress adiabatic shear banding in the cutting of titanium and nickel alloys [73]. In addition, ultrasonic vibration-assisted cutting is a popular unconventional machining method with the advantages of lower cutting forces, better surface finishes and longer tool lifetimes [9]. Ultrasonic vibrations with peak-to-peak amplitudes of approximately 15 - 20 µm at frequencies above 15 kHz are applied at the tool tip to generate intermittent contact between the cutting tool and the workpiece, which enables periodic heat dissipation. Patil et al. modeled the effects of ultrasonic vibration in the cutting of Ti-6Al-4V using FE analysis and demonstrated that a reduction in cutting forces and temperature when ultrasonic vibration is added [67]. Experimental investigations have also been conducted on ultrasonic vibration-assisted cutting of titanium alloys. For example, Sui et al. reported that the tool life was extended
by 300%, and cutting force was reduced by approximately 50% when ultrasonic vibration assistance was applied in turning [86]. Pujana et al. studied ultrasonic vibration-assisted drilling of Ti-6Al-4V and observed a reduction in cutting forces [69]. Similar observations of reduced cutting forces and heat generation were reported by Sanda et al. in the drilling of Ti-6Al-4V stacked with CFRP when ultrasonic vibration assistance was implemented [74]. Pawar et al. experimentally studied the axial and torsional vibration-assisted tapping of Ti-6Al-4V and reported that both the cutting forces and temperature were reduced by applying vibrations [68].

Although it has been demonstrated that ultrasonic vibration assistance can enhance the cutting performance of Ti-6Al-4V, previous studies in the literature are limited to experimental observations. Thus far, there has been no report in the literature on how the tool-workpiece separation in ultrasonic vibration-assisted cutting influences the shear banding mechanism. Chapter 6 presents the effect of ultrasonic vibration assistance on shear band formation and chip segmentation in orthogonal cutting of Ti-6Al-4V. A chip flow model is developed to determine the shear flow mechanism in the primary shear zone and to explain the experimental observations of chip formation when ultrasonic vibration assistance is applied in the tangential direction. The chip geometries and cutting forces predicted by the proposed analytical model were verified by orthogonal cutting experiments.
Chapter 3

3DOF Ultrasonic Vibration Tool Holder Design

3.1 Overview

A novel three-degree-of-freedom (3DOF) ultrasonic vibration tool holder is designed to expand the application of vibration assistance in milling and drilling. The proposed tool holder can deliver the elliptical vibration in the $XY$ plane to assist milling operations along the radial and tangential directions through piezoelectric actuators, and it can generate vibration along the spindle axis ($Z$) to help drilling process.

The mechanical structure of the actuator is modeled in section 3.2 using the beam equations to select the vibration modes to resonate the structure. A lumped parameter model is implemented in section 3.3 to design dimensions of piezoelectric components. In section 3.4, a finite element model is used to verify the natural frequencies and to refine the mechanical structure. The prototype of the 3DOF ultrasonic vibration tool holder is manufactured and presented in section 3.5. The electronics to drive and to monitor the piezo actuator, including power amplifiers and conditioning circuits, are presented in section 3.6.
### 3.2 Mechanical Structure and Mode Shapes

The ultrasonic vibration tool holder generates vibrations in $X$, $Y$ and $Z$ directions using piezoelectric components. Piezoelectric stacks are excited at the natural frequencies of the tool holder structure to amplify the vibrations up to 20-30 µm range. The actuator is connected to the tool holder housing at the neutral nodes of vibration mode shapes where the modal displacements are zero as shown in Figure 3.1.

The final design of the tool holder with 3DOF actuators is shown in Figure 3.2. Three sets of piezo ring pairs, one set for each direction, are integrated into the rotating tool holder, which receives power from the stationary source via four slip rings. The piezoelectric stacks are full rings in the $Z$ direction and used to excite the axial mode of the structure. Two sets of two half rings, which are assembled as a single full ring by the manufacturer, are used per $X$ and $Y$ direction in the design as shown in Figure 3.2b). Each half ring has the opposite polarity, and the assembled ring set is covered by a full ring copper electrode on the upper and lower surface to supply AC voltage. Figure 3.2c) shows the connections of electrodes for $X$ piezo rings as an example. Since the polarity of each half ring is opposite along the thickness direction, one half ring set expands and the other set shrinks when voltage is applied from the three electrodes, thus creating a moment to excite the bending mode of the holder. The vibration of each direction ($X$ or $Y$) is excited...
Figure 3.2: Final design of the 3DOF ultrasonic vibration tool holder: a) 3DOF ultrasonic vibration tool holder design; b) piezoelectric rings for $X$ and $Y$ vibrations; c) side view of X piezo rings with connections of electrodes.

Independently with the same frequency to create elliptical vibrations. The masses in the front and rear are used to tune the mode shapes of the structure. Different milling and drilling tools can be clamped in the front through a standard collet, which is ER-16 in this design.

The actuator is designed to resonate the structure, hence the natural frequencies and mode shapes are parametrically designed based on beam dynamic equations. The axial modes to generate vibrations in $Z$ axis are governed by the following beam equation:

$$\frac{\partial}{\partial x} (EA(z) \frac{\partial w(z,t)}{\partial z}) = \rho A(z) \frac{\partial^2 w(z,t)}{\partial t^2}$$

(3.1)

where $E$ is Young’s modulus, $A$ is the cross-section area, $w$ is the displacement along $Z$, $\rho$ is the mass density, and $L$ is the length of the actuator. The natural frequencies in the axial direction are evaluated for a free-free beam with the boundary
The bending mode of the beam is identified by the following Euler’s beam model:

\[ \rho A \frac{\partial^2 u(z,t)}{\partial t^2} = -\frac{\partial^2}{\partial z^2}(EI \frac{\partial^2 u}{\partial z^2}) \]  

(3.3)

where \( u \) is the displacement along the radial directions (\( X \) or \( Y \)), and \( I \) is the area momentum of inertia. The natural frequencies of the bending modes are evaluated by using free-free boundary conditions (\( \frac{\partial^2 u}{\partial z^2} = 0 \) and \( \frac{\partial^3 u}{\partial z^3} = 0 \) at \( z = 0 \) and \( z = L \)) as:

\[ \omega_k = \beta_k^2 \sqrt{\frac{EI}{\rho A}}, k = 0, 1, 2... \]  

(3.4)

where \( \beta_k \) is the wave coefficient for mode \( k \). The wave coefficients for the first four modes are given as \( \beta_1 = 4.73/L \), \( \beta_2 = 7.85/L \), \( \beta_3 = 10.99/L \) and \( \beta_4 = 14.14/L \)[53].

Stainless steel 304 is used as the primary material of the 3DOF vibration actuator, and the overall length was designed to be less than 200 mm due to the cutting tool length and stiffness requirements. The diameter of the actuator was targeted to be around 30-45 mm, which is compatible with the spindle interface. The desired vibration frequencies have been set to be about 15-20 kHz in three directions; hence the first axial mode was selected to generate vibrations in the \( Z \) direction, and the third bending mode was used to produce vibrations in \( XY \) plane.

### 3.3 Lumped Parameter Model of the Actuator

A lumped parameter model is used to convert each direction of the actuator motion to an equivalent circuit in order to design the capacitance of the piezo stacks for each direction (see Figure[3.3]) [75]. The capacitance of piezoelectric components due to element’s dielectric properties is represented by \( C_0 \); \( L_m \) is the equivalent inductance corresponding to actuator’s mass; \( C_m \) is the equivalent capacitance related to the stiffness of the actuator; and \( R_m \) is the equivalent resistance due to damping of the system. The relationship between mechanical parameters and electrical pa-
Parameters are modeled as:

\[ L_m = \frac{m}{\psi^2} ; \quad C_m = \frac{\psi^2}{k} ; \quad R_m = \frac{c}{\psi^2} \]  \hspace{1cm} (3.5)

where \( m \) is the mass of the actuator, \( k \) is the stiffness, \( c \) is the damping coefficient and \( \psi \) is the force factor of the piezoelectric stacks.

![Equivalent circuit of piezoelectric actuator for each axis.](image)

**Figure 3.3:** Equivalent circuit of piezoelectric actuator for each axis.

The force factor (\( \psi \)) is related to the piezoelectric material properties and geometry, and can be expressed as:

\[ \psi = \frac{Y_{33}d_{33}A}{t} \frac{1}{N_A} \]  \hspace{1cm} (3.6)

where \( A \) is the cross-section area of piezoelectric rings, \( t \) is the thickness of rings, \( Y_{33} \) is Young’s modulus of the piezoelectric material, and \( d_{33} \) is the piezoelectric strain constant along the thickness, which is also the polarity direction. \( N_A \) is the vibration magnification factor between piezo stacks and the cutting tool tip due to the geometry of the actuator which is experimentally identified as explained in section 4.3.

The impedance of the piezo actuator for each direction is modeled by the equivalent circuit, and it leads to a resonance frequency (\( f_r \)), which corresponds to the natural frequency of the mechanical structure with the minimum impedance magnitude, and an anti-resonance frequency (\( f_a \)) with a maximum impedance magnitude.
(defined in Eq. (3.7)).

\[
f_r = \frac{1}{2\pi \sqrt{C_m L_m}}; \quad f_a = \frac{1}{2\pi} \sqrt{\frac{C_0 + C_m}{L_m C_m C_0}}
\] (3.7)

The electromechanical coupling coefficient \((K_M)\), which shows the ability of the system to store mechanical energy, is defined as:

\[
K_M = \sqrt{1 - \frac{f_r^2}{f_a^2}}
\] (3.8)

The resonance frequencies have been selected at around 16.5 kHz in \(Z\) axis, and 17.5 kHz for the \(X\) and \(Y\) directions for the actuator by considering the stiffness requirements for CFRP and Titanium finish machining and standard tool holder interface with regular CAT40/HSK63 spindle interface used in industry. Since the axial vibration along \(Z\) direction and elliptical vibration \(XY\) plane are used separately for drilling and milling, the natural frequencies for two modes are selected with 1 kHz difference to prevent coupling effects hence avoid exciting the tool in undesired directions. The piezo ring diameter range needs to be under 45 mm to fit the actuator to the targeted spindle interface. The diameter range of the tool is under 10 mm, and the stick out can be tuned to excite the natural frequencies of the actuator. The electromechanical factors \((K_M)\) for all three directions have been set to 0.25. The equivalent resistance related to damping is an estimated value for each direction. With an estimated modal mass of the actuator (around 1 kg), the equivalent inductance \(L_m\) and the capacitance of the piezoelectric material are evaluated from Eq.(3.5). Based on Eq.(3.7), the capacitance of the piezoelectric components is derived as:

\[
C_0 = \frac{1}{4\pi^2} \frac{1}{f_a^2} \frac{1}{L_m}
\] (3.9)

The capacitance of the piezoelectric components has been decided by the material properties and the geometry as:

\[
C_0 = \frac{k_e \varepsilon_0 A}{t}
\] (3.10)

where \(k_e\) is the dielectric constant of the material, \(\varepsilon_0\) is the permittivity of free
Table 3.1: Piezoelectric material parameters

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Relative dielectric constant, $k_e$</td>
<td>1375</td>
<td>N/A</td>
</tr>
<tr>
<td>Piezoelectric strain constant, $d_{33}$</td>
<td>300</td>
<td>$10^{-12}$ m/V</td>
</tr>
<tr>
<td>Young’s modulus, $Y_{33}$</td>
<td>63</td>
<td>GPa</td>
</tr>
<tr>
<td>Density, $\rho$</td>
<td>7.6</td>
<td>g/cm$^3$</td>
</tr>
<tr>
<td>Mechanical quality factor, $Q_m$</td>
<td>1400</td>
<td>N/A</td>
</tr>
</tbody>
</table>

space, and $t$ is the thickness of the piezo rings. By evaluating the required $C_0$ from Eq.(3.9), the required thickness and cross-section area of the piezo rings are evaluated from Eq.(3.10).

The selected hard piezoelectric material, which has a high mechanical quality factor and low dielectric loss, are comparable to PZT-8, and its properties are listed in Table 3.1. The selected thicknesses of piezoelectric rings are 4.5 mm in $z$ and 5.5 mm in $X$ and $Y$ directions with all having 40 mm outer and 15 mm inner diameters in order to match the designed capacitance. Table 3.2 lists the parameters used in the simulations with selected piezo rings.

Table 3.2: Lumped parameters

<table>
<thead>
<tr>
<th></th>
<th>$C_0$ [nF]</th>
<th>$C_m$ [nF]</th>
<th>$L_m$ [H]</th>
<th>$R_m$ [Ω]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Simulation in $X$</td>
<td>6.58</td>
<td>0.438</td>
<td>0.376</td>
<td>100</td>
</tr>
<tr>
<td>Experiment in $X$</td>
<td>7.4</td>
<td>0.366</td>
<td>0.454</td>
<td>325</td>
</tr>
<tr>
<td>Simulation in $Z$</td>
<td>4.71</td>
<td>0.312</td>
<td>0.588</td>
<td>50</td>
</tr>
<tr>
<td>Experiment in $Z$</td>
<td>5.5</td>
<td>0.327</td>
<td>0.561</td>
<td>113</td>
</tr>
</tbody>
</table>

The impedance frequency response functions (FRF), which are used to design and check resonance frequencies, of the electromechanical system in three directions have been predicted from the equivalent circuit and validated experimentally with an impedance analyzer (Keysight E4980AL). Figure 3.4 shows the impedance FRFs of the ultrasonic vibration actuator clamping an 8 mm dummy tool along $X$ and $Z$. The impedance FRFs of $X$ and $Y$ actuators are close to each other within 50 Hz due to symmetry. The resonance frequencies of the fabricated actuator were measured to be 17.45 kHz in $X$ and $Y$, and 16.51 kHz in $Z$ as planned. The parameters including $C_m$, $R_m$, $L_m$ and $C_0$ can be realized through the impedance FRF.
Figure 3.4: Impedance frequency response functions of piezoelectric actuator: a) $X$ actuator; b) $Z$ actuator.
as well, and the values are listed in Table 3.2. The electromechanical coupling coefficients for both actuators were measured to be about $K_M = 0.2$.

### 3.4 Finite Element Model of the Actuator

The finite element (FE) analysis of the 3DOF ultrasonic vibration actuator has been carried out to verify natural frequencies, mode shapes and neutral nodes of the modes to locate the clamping locations of the actuator with the tool holder body. A solid model of the actuator with estimated material properties determined from the lumped parameter model was used under free-free boundary conditions in the analysis. The mode shapes of the actuator were predicted using commercial software (COMSOL Multiphysics) as shown in Figure 3.5 for the radial ($X,Y$) and axial ($Z$) directions.
The neutral nodes of the mode shapes, where the modal displacements are zero, are shown in Figure 3.5. The neutral node is a circular band for Z vibration, and two neutral points per X and Y directions exist along the axis of the actuator. The actuator is clamped to the housing with four leaf springs at the neutral points of the third bending mode in order to minimize the transmission of vibrations to the tool holder body while not affecting the values of natural frequencies. The nodes of X/Y and Z vibrations have been tuned to a small region by adjusting the shapes of the front mass and the back mass. The orientations of half-ring piezos for X and Y vibrations align to the leaf springs because of excitation directions. The natural frequencies of the actuator with a carbide tool having 50 mm length (35 mm stick out of the collet) and 8 mm diameter have been predicted as 17.5 kHz in X, 17.4 kHz in Y and 16.6 kHz in Z, respectively.

3.5 Prototype Manufacturing

A prototype ultrasonic vibration tool holder has been fabricated to fit CAT40 spindle as shown in Figure 3.6. Figure 3.6(b) shows the ultrasonic vibration actuator inside the holder housing. An 8 mm carbide dummy tool with a 35 mm length out of the collet was clamped in order to measure the impedance FRFs shown in Figure 3.4. A two-flute end mill with 8 mm diameter was inserted into the actuator with a collet as shown in Figure 3.6(a), and the stick out of the tool was adjusted to match the natural frequencies.

The flexibility FRFs of the actuator assembly with the end mill have been measured by impact modal tests (shown in Figure 3.7). The natural frequencies of the actuator assembly are measured to be 16.5 kHz in Z and 17.5 kHz in X and Y directions which agree with the predicted values during the design. If the stick out length must be different from the design value in production, the natural frequencies will shift. However, the resonance tracking controller presented in Chapter 4 identifies the resonance frequency on-line and update the excitation frequency dynamically to keep the amplitudes at the desired level.

The maximum dynamic flexibilities of the system are 0.06 μm/N in radial and 0.09 μm/N in axial directions respectively to excite the ultrasonic vibrations. The static flexibility of radial direction is measured as 0.04 μm/N from the impact tests,
which is about twice of the flexibility (0.02 μm/N) of a normal tool holder clamping the same cutting tool through an ER25 collet. The main reason for the static stiffness loss is contributed by the leaf springs, which are the clamping mechanism between the ultrasonic vibration actuator and the housing. The dynamic stiffness of the tool holder with actuator is 960 kN/m, which is sufficient to resist against chatter in targeted machining of CFRP at a limit depth of cut of 2.5 mm. The reduction in the stiffness limits the application of ultrasonic vibration-assisted tools
in heavy cuts, hence they are designed for finishing operations to obtain a good surface finish.

Slip rings are used to transmit the power to the rotating ultrasonic vibration tool holder. Four rings, which assemble together as a rotor, are required to connect positive electrodes for all three groups of piezos (X, Y and Z) and the ground. The distance between two adjacent rings is determined by the maximum supply voltage of piezo actuators, which is 200 V, and it is designed to be 10 mm according to the

---

**Figure 3.7:** Structural frequency response functions: a) FRF along X; b) FRF along Z.
specifications from the vendor. The maximum current on the rings is 5 A. Four carbon brushes, which contact the rings with preload, are installed to the CNC machine statically to deliver power as shown in Figure 3.8. The maximum rotation speed of the slip rings is 3500 rev/min to prevent sparks. The slip rings limit the maximum spindle speed of ultrasonic vibration-assisted milling or drilling operation using the proposed holder. Contactless power transmissions such as rotary transformer are preferred in the future improvement to expand applications of the 3DOF ultrasonic vibration tool holder.
### 3.6 Instrumentation

The instrumentation for the 3DOF ultrasonic vibration tool holder has been designed and implemented. The instrumentation system per channel includes: 1) an input conditioner to produce appropriate signals for the voltage amplifier from the digital computer; 2) a voltage stage as a power amplifier to drive the three piezo vibration actuators; 3) a current sensor for feedback control. A digital real-time computer governs all analog instruments to control and monitor the 3DOF ultrasonic vibration tool during cutting. Figure 3.9 shows the instrumentation of X piezo actuator as an example.

The first objective of the instrumentation system is to prepare signals for the voltage power amplifiers. In order to excite the ultrasonic vibration, three sinusoidal voltages are generated as:

\[
\begin{align*}
V_x &= V_{x0} \sin(2\pi f_xt) \\
V_y &= V_{y0} \sin(2\pi f_yt + \theta_{in}) \\
V_z &= V_{z0} \sin(2\pi f_zt)
\end{align*}
\]

where \(f_x, f_z\) are the excitation frequencies in (X,Y) and Z directions, respectively, and \(\theta_{in}\) is the phase difference between the input voltages of X and Y actuators. \(V_x\) and \(V_y\) are synchronized and share the same excitation frequency to generate elliptical vibrations.

![Figure 3.9: Electronic system for X vibration.](image-url)
A dSPACE microLab is used as the digital real-time computer, which can run up to 50 kHz as a sampling rate. Figure 3.9 shows the instrumentation of X actuator as an example. The periodic square wave signals with \( f_x \) or \( f_z \) are modulated by a waveform generator embedded in the digital controller. The square wave is converted to a sinusoidal wave using an active RC, fourth-order low pass filter in order to eliminate high-frequency harmonics. Following the filter, a voltage-controlled operation amplifier (Texas Instruments VCA822) is used to adjust the amplitude of the input sinusoidal signal to the desired value \( (V_{x0}, V_{y0} \text{ and } V_{z0}) \) for each channel. The gain of the VCA is adjusted by an analog continuous voltage produced by the digital controller through a digital-analog converter (DAC). The input phase difference \( (\theta_{in}) \) is adjusted by a phase-shift integrated circuit using a potentiometer manually.

A voltage stage is used to amplify the signal from the VCA to drive the piezo actuator. The supply voltage of the actuator is designed to be 200 V peak-to-peak voltage. The bandwidth of the voltage stage needs to reach high frequency to excite the selected vibration modes of the actuator without voltage loss. MP118 from APEX, which is a linear power amplifier, is selected as the power amplifier in the voltage stage. It has maximum supply voltage of 200 V with 10 A continuous output current, and the voltage closed loop bandwidth is up to 100 kHz with the gain of 10.

The actuator can produce the ultrasonic vibration for each axis independently using the power amplifiers and the input conditioner, and Figure 3.10 shows the results of preliminary experiments for all three directions. A laser vibrometer (Polytec CLV-2534) was used to measure the vibrations in the experiments. The input voltage peak-to-peak amplitude was adjusted to be 200 V. A 8 mm diameter carbide dummy cutting tool was clamped in the holder with 30 mm overhang. The ultrasonic vibration actuator was resonated at 17.5 kHz in X- and Y-directions with measured peak-to-peak displacement amplitudes of 20 \( \mu \)m in X and 18 \( \mu \)m in Y, respectively. The Z-axis was excited at 16.5 kHz with a measured vibration amplitude of 25 \( \mu \)m.

It is important to drive the piezo actuator at its resonance frequencies to obtain enough amplitudes during cutting. The full details of the control system are presented in Chapter 4. Briefly, the phase of impedance, which can be character-
Figure 3.10: Vibrations outputs from the laser vibrometer for X, Y and Z.

ized from the current flowing through the piezoelectric actuator and the voltage output from the power amplifier, has been used to track the resonance frequency. The current flowing through each piezo actuator must be measured to obtain the phase. According to this requirement, a current sensing circuit is designed and tested. In the 3DOF ultrasonic vibration tool holder, three piezo actuators, one for each vibration freedom, are pressed by a central bolt and share the same ground when receiving supply voltages. Therefore, to measure the current individually for each channel, the current sensing resistor is placed in front of the piezo actuator, as shown in Figure 3.9. Thus, the common-mode voltage is high up to hundreds of volts when measuring the voltage drop across the sensing resistor. A high common-mode voltage difference op amplifier (Texas Instruments INA117) is selected to obtain the current signal from the sensing resistor. The current signal is collected by an analog-to-digital converter (ADC) in the digital controller as feedback. On the other hand, the supply voltage signal is fed back using a step-
down voltage amplifier to calculate the phase in order to track the resonance during cutting.

3.7 Summary
A 3-DOF ultrasonic vibration tool holder design to be used in milling and drilling operations was presented in this chapter. The proposed design includes a piezoelectric actuator, which can be activated in three directions to expand its application to various machining operations in a single tool holder. While the radial vibrations are delivered by resonating the third bending mode, the axial vibrations are created by exciting the first axial mode of the actuator assembly at around 17 kHz with a peak-to-peak amplitude range of 20-25 µm. The system has a feature of maintaining a phase shift between X and Y vibrations, which can be used in delivering elliptical vibrations. The instrumentation system was also developed to drive the proposed piezoelectric actuator.
Chapter 4

Sensorless Control of 3DOF Ultrasonic Vibration Tool Holder

4.1 Overview

The piezoelectric actuator of the vibration tool holder works at natural frequencies to enlarge the amplitudes for high-frequency vibration. The natural frequencies of ultrasonic vibration actuators vary under the varying cutting force disturbances during machining. Consequently, the amplitudes of delivered vibrations deviate from the desired reference values. As opposed to using vibration sensors, this chapter presents a sensorless digital control system for the proposed 3DOF ultrasonic vibration tool holder using a digital controller platform. The chapter starts with the fundamental principles of resonance tracking and amplitude control in section 4.2 and a dual-loop control system in section 4.3. The crosstalk compensation between the two axes is analyzed and discussed in section 4.5. Section 4.6 shows the milling and drilling experiments with the proposed control system.

4.2 Principle of Frequency and Amplitude Control

Each axis of the 3DOF ultrasonic vibration actuator can work independently using the same sensorless control strategy. The resonance frequency is tracked and the desired vibration amplitude is delivered in a closed-loop manner. A linearized
lumped parameter model is proposed to estimate the vibration characteristics from the measured supply voltage and driving current signals without having to use dedicated vibration sensors.

The piezoelectric ultrasonic vibration actuator is represented by a lumped linear model as presented in section 3.3 for each axis. The model is also used to track the varying natural frequencies under cutting loads and estimate the velocities to control the amplitudes. The piezo actuator can excite the selected structural modes of the holder to generate ultrasonic vibration as mentioned in Chapter 3. The natural frequency of the selected vibration mode shifts under the cutting load, but this shift is not wide enough to excite the neighboring modes of the system. Consequently, a single degree of freedom (SDOF) mechanical system corresponding to selected excitation mode is used to model the piezo actuator during operation. The overall tool holder is lumped into the mechanical subsystem of the piezoelectric actuator stack for each vibrating axis, and the resulting electromechanical system model is shown in Figure 4.1. The actuator is excited to deliver motion by applying a sinusoidal voltage at its resonance frequency on the terminals of the piezoelectric element. The capacitive component ($C_d$) represents the electrical subsystem of the piezoelectric stack, while the mass, spring, damper elements represent the combined mechanical subsystems of the piezo stack, and the tool holder delivers the vibration in each direction as shown in Figure 4.1.

The model is presented for $X$ axis as an example. The constitutive linear elec-
The electromechanical relationship of the piezoelectric actuator is defined as [28]:

\[ F_t = \psi V \]  \hspace{1cm} (4.1)

\[ I_t = \psi \dot{x} \]  \hspace{1cm} (4.2)

where \( F_t \) is the force transduced by the electrical energy generated by the supply voltage \( V \) on the actuator with a force factor or transforming gain \( \psi \). \( I_t \) is the transforming current to generate the vibration, and \( \dot{x} \) is the vibration velocity at the cutting tool tip. The force generated by the actuator \( (F_t) \), and the external cutting force \( (F_{\text{ext}}) \) are applied on the actuator to excite the structural mode with mass \( (m) \), stiffness \( (k) \) and damping constant \( (b) \) causing tool displacement \( (x) \):

\[ m\ddot{x} + b\dot{x} + kx = F_{\text{ext}} + F_t \]  \hspace{1cm} (4.3)

The overall driving current \( (I) \) from the power amplifier is the sum of the current flowing through capacitance \( (C_d) \) and the transforming current \( (I_t) \):

\[ I = C_d \dot{V} + I_t \]  \hspace{1cm} (4.4)

The transfer function between the transforming current \( (I_t) \) and supply voltage \( (V) \) can be evaluated by taking the Laplace transforms of (4.1) to (4.3) and letting \( F_{\text{ext}} = 0 \):

\[ G_a(s) = \frac{I_t(s)}{V(s)} = \frac{\psi^2 s}{ms^2 + bs + k} \]  \hspace{1cm} (4.5)

The frequency response function \( G_a(s = j\omega) \) (4.5) can be obtained as:

\[ G_a(j\omega) = \frac{\psi^2 s}{ms^2 + bs + k} \bigg|_{s = j\omega} = \frac{1}{k} \frac{j\psi^2 \omega_n^2 \omega}{\omega_n^2 - \omega^2 + j2\zeta \omega \omega_n} \]  \hspace{1cm} (4.6)

where the natural frequency \( (\omega_n = \sqrt{k/m}) \), damping ratio \( (\zeta = b/(2\sqrt{mk})) \) and stiffness \( (k) \) are evaluated from the modal parameters. When the input voltage frequency \( (\omega) \) matches the natural frequency of the mode \( (\omega_n) \), the phase of the FRF given in (4.6) becomes zero \( (\phi = \angle G_a(j\omega_n) = 0) \). In the proposed control system, the phase between transforming current and supply voltage \( (\phi = \angle G_a(j\omega)) \) is used
to track the resonance.

The velocity of vibration at tool tip can be derived from (4.4) and (4.2) as:

\[
\dot{x}(t) = \frac{1}{\psi} I_t = \frac{1}{\psi} \left[ I(t) - C_d \dot{V}(t) \right]
\]  

(4.7)

where driving current \( I \) and voltage \( V \) can be measured from the power amplifier, and the capacitance \( C_d \) of the piezo is measured with an LCR meter (Keysight E4980L). The transforming gain \( \psi \) in each direction is defined in (3.6). It is proposed to estimate the amplitude of vibrations from the driving current and supply voltage measurements during cutting by utilizing the model given in (4.7).

### 4.3 Dual-loop Control System

The dual loop control system is used to track the vibration resonance frequency and amplitude for each vibrating axis as shown in Figure 4.2. The inputs to the controller are the desired phase \( \varphi_r \) between the transforming current \( I_t \) and supply voltage \( V \) to deliver vibration amplitude linearly, and the desired vibration amplitude at the targeted excitation frequency. The first loop tracks the resonance of tool structure by using the phase between supply voltage and transforming current, and the phase is tracked based on the frequency response function defined in (4.6). The second loop tracks the reference vibration amplitude by using estimations based on (4.7).

The vibration observer shown in Figure 4.2 is implemented in real time to estimate the amplitude of transforming current \( I_t \) and the phase between supply voltage \( V \) and current \( I_t \). The transforming current \( I_t \) is evaluated from Eq.(4.4) using the measured current \( I \) and voltage \( V \), i.e. \( I_t = I - C_d \dot{V} \) where derivative \( \dot{V} \) is digitally evaluated. Two digital phase detectors \( (PD_1 \text{ and } PD_2) \) are used to calculate the phase difference between \( I_t \) and \( V \), and the digital amplitude detector \( (AD) \) tracks the amplitude of the transforming current \( I_t \) estimated by the observer. The current \( I_t \) is used to control the amplitude of vibration.

Various detection methods have been used to track the phase between the overall driving current \( I \) and supply voltage \( V \) in the literature, mostly using electronic circuits. Jung et al. [39] used a phase lock loop (PLL) circuit as a phase
detector to track the resonance during elliptical vibration cutting. Dow et al. [20] utilized a lock-in amplifier to detect the phase of impedance for resonance tracking during nanocoining processes. Furuya et al. [23] designed a circuit with a difference amplifier to detect the phase between supply voltage and vibration velocity whose accuracy is dependent on the physical tuning of the capacitor to match the capacitance of piezo components. As for the detection of amplitude, Juang and Gu [37] designed a circuit to recognize the amplitude of operation current for a two-phase ultrasonic motor, which also relied on a well-tuned capacitor to remove the influence of piezo capacitance. The existing circuit based methods can track the small variations in the natural frequency if the same tool geometry is used in the holder. However, when various tools are used, the variation in the natural frequency becomes too wide to be tracked by analog circuit based systems. The proposed, digital dual loop amplitude and phase detector adapts itself to the changing structural dynamics of the tool holder, which is essential in regular machining applications in production. Though there is a limitation of the adaptation because the large changing dynamics shifts neutral nodal positions, which can increase the damping during the cutting process and then reduce vibration amplitudes.

Figure 4.2: Dual-loop ultrasonic vibration control system
The dual loop vibration observer consists of two sets of dedicated Kalman filters; one detects the amplitude and phase of supply voltage on the piezo stack as shown in Figure 3.2 altered by the conditioning circuit, and the other detects the varying amplitude and phase of transforming current disturbed by the cutting load.

The measured periodic supply voltage \( V(t_k) \) and transforming current \( I_t(t_k) \) with amplitude \( A_k \) and phase \( \phi_k \) are defined as:

\[
s(t_k) = A_k \cos(\omega t_k + \phi_k) + v(t_k) \rightarrow s(t_k) = V(t_k) \text{ or } s(t_k) = I_t(t_k)
\] (4.8)

where \( t_k = kT \) is the discrete time at interval \( k \) with a sampling period \( T \). The excitation frequency is \( \omega = 2\pi f \), and \( v(t_k) \) is the measurement noise. Since the sampling frequency is just above Nyquist frequency, it is too low to track the amplitude \( A_k \) and phase \( \phi_k \) accurately. Kalman filter is used to estimate them accurately using the following two state variables defined as:

\[
q_k = \begin{bmatrix} q_{1,k} \\ q_{2,k} \end{bmatrix} = \begin{bmatrix} A_k \cos(\omega t_k + \phi_k) \\ A_k \sin(\omega t_k + \phi_k) \end{bmatrix}
\] (4.9)

At the consecutive discrete time step \( t_{k+1} = t_k + T \), the states vary due to the process noise \( w_k \):

\[
\begin{bmatrix} q_{1,k+1} \\ q_{2,k+1} \end{bmatrix} = \begin{bmatrix} A_k \cos(\omega t_k + T + \phi_k) \\ A_k \sin(\omega t_k + T + \phi_k) \end{bmatrix} + \begin{bmatrix} w_{1,k} \\ w_{2,k} \end{bmatrix}
\] (4.10)

or by using the trigonometric identities:

\[
\begin{bmatrix} q_{1,k+1} \\ q_{2,k+1} \end{bmatrix} = \begin{bmatrix} \cos(\omega T) & -\sin(\omega T) \\ \sin(\omega T) & \cos(\omega T) \end{bmatrix} \begin{bmatrix} q_{1,k} \\ q_{2,k} \end{bmatrix} + \begin{bmatrix} w_{1,k} \\ w_{2,k} \end{bmatrix}
\] (4.11)

where \( \Phi \) is the state matrix. The measurement equation defined in Eq.(4.8) for the states with noise \( v_k \) becomes as follows:

\[
s_k = \begin{bmatrix} 1 & 0 \end{bmatrix} \begin{bmatrix} q_{1,k} \\ q_{2,k} \end{bmatrix} + v_k
\] (4.12)

where \( H \) is the output matrix of the model. Combining Eq.(4.11) and Eq.(4.12)
leads to the completed state space model for the periodic voltage \( V \) and current \( I_t \) signal \( s_k \).

The state vector \( \mathbf{q}_k \) is estimated with the following Kalman filter \([10]\):

\[
\begin{align*}
\hat{\mathbf{q}}^-_k &= \Phi \hat{\mathbf{q}}^-_{k-1} \\
\mathbf{P}^-_k &= \Phi \mathbf{P}^-_{k-1} \Phi^T + \mathbf{Q} \\
\mathbf{K}_k &= \mathbf{P}^-_k \mathbf{H}^T (\mathbf{HP}^-_k \mathbf{H}^T + \mathbf{R})^{-1} \\
\hat{\mathbf{q}}_k &= \hat{\mathbf{q}}^-_k + \mathbf{K}_k (s_k - \mathbf{H} \hat{\mathbf{q}}^-_k) \\
\mathbf{P}^-_k &= (\mathbf{I} - \mathbf{K}_k \mathbf{H}) \mathbf{P}^-_k
\end{align*}
\]

where \( \mathbf{P}_k[2 \times 2] \) is the error covariance matrix and the superscript \((\cdot)^-\) denotes prior estimate. \( \mathbf{Q} = \mathbf{Q}_{\mathbf{I}} = \mathbf{E}[\mathbf{w}_{1} \mathbf{w}_{1}^T] \) is the \( 2 \times 2 \) diagonal process noise covariance matrix; \( \mathbf{R} \) is the covariance of the measurement noise \( \mathbf{E}[\mathbf{v}_{1} \mathbf{v}_{1}^T] \), where \( \mathbf{E}[\cdot] \) is the expectation operator, and it is found by taking the covariance of measured current and voltage signals in air cutting. The process noise covariance is assumed to be \( \mathbf{Q} = \lambda \mathbf{R} \), where \( \lambda \) is tuned as \( \lambda = 10^{-6} \). In the ultrasonic vibration control system, the angular excitation frequency of the periodic signal \( \omega \) is generated by the controller \( (\mathbf{C}_1) \) of phase tracking loop, and it is a known value when estimating the states. However, the frequency \( \omega \) varies during machining due to time varying cutting forces, hence the state matrix \( \Phi \), so as the Kalman gain matrix \( \mathbf{K}_k[2 \times 1] \) is updated recursively.

The amplitude \( \hat{A}_k \) and the phase \( \hat{\phi}_k \) of the signal shown by \( PD \) and \( AD \) blocks in Figure 4.2 are estimated at each sampling interval \( k \) as:

\[
\hat{A}_k = \sqrt{\hat{q}_{1,k}^2 + \hat{q}_{2,k}^2}
\]

\[
\hat{\phi}_k = \arctan \left( \frac{\hat{q}_{2,k}}{\hat{q}_{1,k}} \right) - \omega t_k
\]

The phase difference \( \varphi \) between the transforming current \( I_t \) and supply voltage \( V \) (see Eq.(4.6)) is tracked from their estimated phases from the Kalman filters (Eq.(4.15)) as:

\[
\varphi = \hat{\phi}_k - \hat{\phi}_V
\]

which is as a feedback to track the reference input phase \( \phi_r \) in the dual loop con-
controller. A digital differentiator is embedded in the vibration observer to calculate the derivative of supply voltage ($\dot{V}(k)$) for the transforming current (see Figure 4.2). The differentiator is smoothed by a Kalman filter to reduce the effect of noise in digital differentiation at a sampling frequency which is just above two times of the natural frequencies of the actuator. The transforming current ($I_t$) is calculated in real time by the differentiated supply voltage using Eq.(4.4) ($I_t(k) = I(k) - C_d \dot{V}(k)$) before detecting the phase and amplitude by Kalman filters.

In the phase tracking loop as shown in Figure 4.2, a PI controller ($C_1$) is designed to update the excitation frequency ($f$) with a feed forward input ($f_0$), which is measured natural frequency of the tool holder with impact modal tests ahead of cutting operation. The feedback of this loop is the estimated phase between supply voltage and transforming current derived by Eq.(4.16). The PI controller is defined as follows:

$$C_1(s) = \frac{\Delta f}{\Delta \phi} = K_p \left(1 + \frac{\omega_l}{s}\right)$$

(4.17)

where $K_p$ and $\omega_l$ are the proportional gain and the integrator corner frequency, respectively.

In the vibration amplitude control loop, the referenced vibration velocity amplitude ($|\dot{x}_r|$) is converted to the referenced transforming current amplitude ($|I_t|_r = \psi |\dot{x}_r|$) where $\psi$ is the transforming gain in Eq.(4.7). The amplitude of transforming current ($|I_t|$) is tracked by another PI controller ($C_2(s)$ in Figure 4.2), as Newcomb et al. [60] suggested that the piezoelectric actuator performs more linearly by controlling the current in comparison to the voltage control. PI controllers are implemented with anti-windup configuration to remove the steady state errors while preventing the integration wind-up when the outputs are saturated.

The sine wave generator in Figure 4.2 consists of a digital controller, a conditioning circuit and a power amplifier per channel as presented in section 3.6 (see Figure 3.9) to generate the supply voltage to the piezo actuator. The output signal is a sinusoidal function defined as follows:

$$V(t) = V_0 \cos(2\pi ft)$$

(4.18)

where $V_0$ is the supply voltage amplitude produced by the vibration amplitude con-
controller \( C_2(s) \), and \( f \) is the excitation frequency calculated by the phase tracking controller \( C_1(s) \).

### 4.4 Verification of Control System

The dual control system has been implemented, and multiple tests have been conducted to verify its performance. Both closed loops are designed to have at least 60° phase margins to guarantee the stability of the system with the highest bandwidth possible within the physical limits of the conditioning circuit. The corner frequency of integrator is set as tenth of the gain crossover frequency \( \omega_I = \omega_c/10 \) to minimize the steady state errors. The measured frequency response functions of the dual loop controller are shown in Figure 4.3, where the phase and amplitude tracking loops have 130 Hz and 140 Hz bandwidths, respectively. The vibration estimation is based on a linear relationship between the vibration velocity and transforming current in the implemented system. The effect of nonlinearity of the piezo actuator on the transforming gain (\( \psi \)) is analyzed using the experimental setup shown in Figure 4.4.

The proposed dual loop controller is used by setting the desired phase (\( \phi_r \)) between the transforming current and supply voltage and varying the referenced transforming current amplitude (\( |I_t|_r \)). The transforming gain (\( \psi \)) is found from the ratio of the current (\( |I_t| \)) and the vibration velocity measured with a laser vibrometer (Polytec CLV-2534). The experiments were repeated at several reference phases \(-45^\circ \leq \phi_r \leq 45^\circ \), and the identified sample transforming gains for X axis are shown in Figure 4.5. The stationary excitation frequency (\( f_0 \)) was matched with the unloaded resonance frequency of the actuator which was 15.43 kHz for X axis. The transforming gain for each reference phase is slightly different, but it is most linear at the referenced phase of \( \phi_r = -30^\circ \), where the goodness of linear fit is 0.9996, without saturating the power amplifier. Consequently, the reference phase between the transforming current and supply voltage is fixed at \( \phi_r = -30^\circ \) in the controller setup for the ultrasonic actuator in machining applications. Since the tool length and diameter will change at various operations, the transforming gain is calibrated for each cutting tool clamped to the holder.

The performance of controller has been tested in turning aluminum alloy AL
7050 workpiece attached to the spindle, while a turning tool was clamped to the stationary actuator as shown in Figure 4.4. The transforming gain was measured to be $\psi = 0.1754 \text{ A/(m/s)}$ for the tool at a fixed reference phase $\phi_r = -30^\circ$, see Figure 4.5. The measured vibration velocity with laser vibrometer is compared against the estimated velocity in Figure 4.6a. The referenced vibration velocity amplitude was set at 0.6626 m/s which was closely tracked by the controller during air cutting.
but the vibrometer measurements deviated up to around 1.0 m/s when the tool cuts the material. However, the laser vibrometer does not only measure the ultrasonic vibrations delivered by the actuator but also the vibrations of tool holder clamped on the flexible cantilevered plate which vibrates at its lower natural frequency during cutting. The natural frequency of the flexible fixture, which is 175 Hz, was measured with impact modal tests with an impulse hammer (Dytran 5800B4) and an accelerometer (Dytran 3225F) at a sampling rate of 50 kHz. The vibrations of the actuator and plate fixture are separated in frequency domain as shown in Figure 4.6 b). The amplitude of the actuator vibration velocity was 0.6413 m/s at the ultrasonic vibration excitation frequency of 15.47 kHz during cutting, which was close to the reference velocity of 0.6626 m/s. The plate vibrations contribute 0.1963 m/s at its natural frequency of 175Hz. The controller also maintained the reference phase at $\phi_r = -30^\circ$ (see Figure 4.6 c)) and adjusted the excitation frequency from unloaded natural frequency of actuator at $f_0 = 15.43$ kHz to the loaded natural fre-
Figure 4.5: Measured transforming gains ($\psi$) as a function of reference tracking phase ($\phi_r$).

The sensitivity of transforming gain to different cutting loads was also tested by changing the feed rate in turning tests between 0.01 mm/rev and 0.1 mm/rev, which corresponded to cutting force range of 12 N to 145 N. The transforming gain changed from $\psi = 0.1754$ A/(m/s) in air cutting to $\psi = 0.1548$ A/(m/s) (see Figure 4.7) at the highest cutting load of 145 N obtained at 0.1 mm/rev feed rate. The variation in vibration velocity caused by the transforming gain fluctuation under the cutting load around 150 N is within 12%, which is acceptable in ultrasonic vibration-assisted milling and drilling of targeted CFRP materials. The transforming gain can either be calibrated at the operational cutting load or the deviation can be neglected in ultra-precision machining applications where the disturbance cutting force is negligibly small.
Figure 4.6: Single-axis ultrasonic vibration during turning: a) estimated and measured ultrasonic vibration velocity; b) FFT of vibration velocity during cutting; c) phase between supply voltage and transforming current; d) excitation frequency generated by phase tracking controller; e) turning force along tangential direction. The spindle speed of turning is 1500 RPM, the feed rate is 0.05 mm/rev, and the width of cut is 1.5 mm.
Figure 4.7: Transforming gains with different feed rates. The turning spindle speed is 1500 RPM, and the width of cutting is 1.5 mm.

4.5 Two-axis Ultrasonic Vibration Control

The actuator’s two-axis \((X,Y)\) ultrasonic vibration generation capability has been tested by delivering elliptical vibration loci for milling applications based on the dual-loop control system proposed in section 4.3. The vibrations in \(X\) and \(Y\) directions are excited by the supply voltages having amplitudes \(V_{0x}\) and \(V_{0y}\) at the identical excitation frequency \((f)\) as,

\[
\begin{align*}
V_x(t) &= V_{0x} \cos(2\pi ft) \\
V_y(t) &= V_{0y} \cos(2\pi ft + \theta_{in})
\end{align*}
\] (4.19)

where \(V_{0x}\) and \(V_{0y}\) are the amplitudes of supply voltages for \(X\) and \(Y\) vibrations respectively, \(\theta_{in}\) is the input phase difference between two supply voltages and it is adjusted through a potentionmeter.

The frequency response functions (FRFs) in \(X\) and \(Y\) directions of the tool holder are measured by impact modal tests using a miniature impulse hammer (PCB 086C80) and a laser vibrometer. A sample FRF measurement for 8 mm diameter end mill with four flutes are shown in Figure 4.8. The measured natural frequencies are \(f_x = 16.58\) kHz and \(f_y = 16.61\) kHz which differ only by about 31 Hz due to symmetric tool geometry.
There is crosstalk arising between $X$ and $Y$ directions of the actuator. The crosstalk is removed from the sensorless estimation of the vibration velocities to control the elliptical vibration locus using the observer shown in Figure 4.9. The supply voltage ($V_x, V_y$) and driving current ($I_x, I_y$) for each axis are collected in real time. The vibration observers designed in section 4.3 are used to estimate the transforming currents ($I_{tx}, I_{ty}$) flowing through piezo stacks ($X, Y$) and their phases ($\phi_x, \phi_y$). Since the phase controller is used only to track the resonance frequency, and the actuators in $X$ and $Y$ directions have almost identical natural frequencies, only the phase in one direction ($\phi_x$) is closed loop controlled. The resulting resonance frequency ($f$) is then used to excite both $X$ and $Y$ actuators. The phase controller for $y$ axis is deactivated for applications with elliptical vibrations.

The relationship between transforming currents ($I_{tx}$ and $I_{ty}$) and the vibration velocities ($\dot{x}$ and $\dot{y}$) with crosstalk is expressed as:

$$
\begin{bmatrix}
\dot{x} \\
\dot{y}
\end{bmatrix} =
\begin{bmatrix}
\frac{1}{\psi_{xx}} & \frac{1}{\psi_{xy}} \\
\frac{1}{\psi_{yx}} & \frac{1}{\psi_{yy}}
\end{bmatrix}
\begin{bmatrix}
I_{tx} \\
I_{ty}
\end{bmatrix}
$$

(4.20)

where ($\psi_{xx}$, $\psi_{yy}$) are direct and ($\psi_{yx}$, $\psi_{xy}$) are crosstalk transforming gains, re-
spectively. The estimated vibration velocity amplitude considers the crosstalk, and it is fed back to closed loop controller for on-line compensation. The transforming gains are calibrated from the transforming current and the vibration velocity measured with a laser vibrometer. It is observed that the crosstalk has range of $10^{-30}\%$ depending on tool geometries.

The phase difference ($\theta_{xy}$) between the vibrations in two directions are evaluated from the estimated phases ($\hat{\theta}_x$ and $\hat{\theta}_y$) in the crosstalk observer shown in Figure 4.9 as:

$$\theta_{xy} = \hat{\theta}_x - \hat{\theta}_y \quad (4.21)$$

The input supply voltage phase $\theta_{in}$ in (4.19) is tuned using a potentiometer until the controller delivers the desired phase $\theta_{xy}$ between the vibrations in $X$ and $Y$ directions. For example, a circular locus is generated by $\theta_{xy} = 90^\circ$. The proposed two-axis ultrasonic vibration control system has been verified by generating a circular ($\theta_{xy} = 90^\circ$) and elliptical ($\theta_{xy} = 120^\circ$) loci when the ultrasonic vibration tool holder was stationary as shown in Figure 4.10. The estimations from the proposed crosstalk observer can track the referenced elliptical loci. The estimated tool tip loci match with the laser vibrometer measurements with a maximum error of 13%
as shown in Figure 4.10.

![Figure 4.10: Elliptical loci generated by two-axis ultrasonic vibration controller with phase differences of a) $\theta_{xy} = 90^\circ$, b) $\theta_{xy} = 120^\circ$. Tracked referenced vibration velocities are set to: $|\dot{x}|_r = |\dot{y}|_r = 0.45$ m/s.](image)

### 4.6 Cutting Experiments

The sensorless control system for the 3DOF ultrasonic vibration tool holder has been tested in several milling and drilling operations on a CNC machining center shown in Figure 1.4. A stationary table dynamometer (Kistler 9256C1) with 3 kHz bandwidth was used to measure the cutting forces in three directions. The bandwidth of dynamometer measurements is much lower than the ultrasonic vibration frequencies (16-17 kHz) generated by the tool holder, hence the cutting forces can be measured only at tooth passing frequencies and their harmonics under 3 kHz. Sample drilling and milling test results are presented to illustrate the performance of ultrasonic actuator controller performance during machining.

#### 4.6.1 Drilling tests

A twist drill with 6 mm diameter was used in drilling aluminum alloy AL 7050 while exciting the actuator in Z axis at its 17.95 kHz natural frequency. The transforming gain was identified as $\psi_z = 0.1333$ (m/s)/A. The referenced vibration velocity amplitude was set to $|\dot{z}|_r = 1.5$ m/s. Figure 4.11 shows amplitudes of vibra-
Figure 4.11: Amplitudes of vibration velocities in drilling tests. The spindle speed is 1500 RPM, and the feed rate is 0.1 mm/rev. The workpiece material is aluminum alloy 7050.

tion velocities. In the first test, the actuator’s Z axis was excited at a fixed input frequency (17.95 kHz) while the ultrasonic vibration controller was off. The vibration velocity during cutting was attenuated from the desired 1.5 m/s to about 1.32 m/s with oscillations caused by periodic cutting forces at spindle periods. When the controller was turned on, the vibration velocity was tracked at the desired velocity (1.5 m/s) with small amplitude oscillations due to disturbance force compensation of the system as shown in Figure 4.11.

The effect of ultrasonic vibrations on the cutting forces is also demonstrated in Figure 4.12, where the drilling forces were reduced by about 30% relative to the normal drilling without vibration assistance and reduced by 10% relative to the drilling with uncontrolled ultrasonic vibration. When the frequency and amplitude of the ultrasonic vibration are delivered without the controller, the vibration velocity amplitude is attenuated which affects the drilling force amplitude.

4.6.2 Milling tests

An 8 mm diameter end mill with 4 flutes was used in up milling of aluminum alloy AL 7050 material. Measured FRFs of the end mill set up is shown in Figure
Figure 4.12: Thrust forces along axial direction during drilling. The spindle speed is 1500 RPM, and the feed rate is 0.1 mm/rev. The workpiece material is aluminum alloy 7050.

4.8 with the natural frequencies 16.61 kHz and 16.58 kHz, in X and Y directions, respectively. The transforming gains were $\psi_{xx} = 0.1359 \text{ A/(m/s)}$, $\psi_{yy} = 0.1373 \text{ A/(m/s)}$, $\psi_{yx} = 1.400 \text{ A/(m/s)}$ and $\psi_{xy} = 0.7003 \text{ A/(m/s)}$. The two-axis ultrasonic vibration control system was tested to generate elliptical locus in $xy$ plane as shown in Figure 4.13. The elliptical vibration with a circular locus was excited with the sensorless closed loop controller in period I, where the reference amplitudes of vibration velocities were set to 0.45 m/s in both X and Y directions with a phase angle difference of $\theta_{xy} = 90^\circ$ to generate a circular path. The controller was switched off in period II, and vibrations were excited at a fixed frequency and a constant supply voltage, which resulted in a distorted shape deviated from the circle. Without the controller, the $\theta_{xy} = 90^\circ$ phase difference to generate the circular path cannot be delivered. In period III, the vibration actuator was turned off completely which led to the increased cutting forces. The cutting force amplitudes were reduced by about 30% in Period I with a controller relative to Period III where the ultrasonic vibration actuator was off, and the cutting forces were reduced by about 10% relative to Period II where uncontrolled ultrasonic vibrations were excited.
4.7 Summary

This chapter presented a sensorless closed loop control system for a 3DOF ultrasonic vibration tool holder. A linearized lumped parameter model of the piezoelectric actuator for each axis is used to track the resonance and to estimate the vibration amplitudes during cutting. A Kalman filter based observer is used to estimate the vibration amplitude and phase from the measured driving currents and supply voltages without using vibration sensors. The proposed closed loop control system has been demonstrated to maintain the vibration amplitude at the desired level while adjusting the excitation frequency to match the varying resonance frequency of the actuator during machining. The proposed method ensures the delivery of different vibration amplitudes in each direction with the desired phase, which allows circular or elliptical vibration paths during machining. The system has been demonstrated in turning and drilling with unidirectional vibrations, as well in two-axis milling with circular vibration loci. The ultrasonic vibrations with a sensorless controller reduced the cutting forces by about 30% compared to the conventional cutting in sample drilling and milling tests.
Figure 4.13: Milling tests with 3DOF ultrasonic vibration tool holder: a) Feed ($F_x$) and normal ($F_y$) cutting forces during up milling of Al7050. b) Circular vibration locus during milling generated by the sensorless control system; c) vibration locus during milling at a fixed excitation frequency and supply voltage without controller. Cutting conditions: Spindle speed=1000 rev/min, feed rate=0.1 mm/rev, depth of cut=1.5 mm, the width of cut=2 mm. The workpiece material is aluminum alloy 7050.
Chapter 5

Dynamics of Vibration-Assisted Milling

5.1 Overview

To understand the vibration assistance and to use the proposed 3DOF ultrasonic vibration tool holder properly, this chapter presents the dynamics of elliptical vibration-assisted milling and predicts the chatter stability. Section 5.2 models the dynamics of elliptical vibration-assisted milling process by investigating the uncut chip thickness, the process time delay, tool-workpiece separations, and the process damping effect. Section 5.3 presents the stability prediction of vibration-assisted milling using the semi-discrete method, followed by simulations and milling experiments in section 5.4 for two materials.

5.2 Milling Process with Elliptical Vibration

5.2.1 Chip generation with elliptical vibration assistance

The elliptical vibrations are delivered in radial and tangential directions by the piezo actuators embedded in the tool holder at its resonance frequency which depends on the diameter and length of the end mill (i.e. $f_e \approx 16 - 17$ kHz). The vibrations along the tangential and radial directions are excited at the same ultra-
sonic vibration frequency \( f_e \) but with an adjustable phase shift of \( \theta \) to generate an elliptical path (Figure 5.1b)). The generated vibrations by tooth \( j \) in the tangential \((a_j(t))\) and radial \((b_j(t))\) directions are:

\[
\begin{align*}
  a_j(t) &= A_0 \cos \left( \omega_e t + (j - 1) \frac{2\pi}{N} \right) \\
  b_j(t) &= B_0 \cos \left( \omega_e t + (j - 1) \frac{2\pi}{N} + \theta \right)
\end{align*}
\]

where \( N \) is the number of flutes on the end mill, \( \omega_e = 2\pi f_e \) [rad/s] is the ultrasonic excitation frequency, and \( A_0 \) and \( B_0 \) are vibration amplitudes which can be adjusted between 5-10 \( \mu \)m by using the closed loop controller developed in Chapter 4. The vibration locus at the tool tip is an ellipse with harmonically varying amplitude and speed. The tooth spacing angle is \( \phi_p = \frac{2\pi}{N} \) for end mill with uniform pitch cutters. The maximum velocity of the tangential vibration \((da_j(t)/dt)\) is higher than the nominal cutting speed \((V_s)\) in order to force the cutting edge to produce high-frequency intermittent contact with the workpiece surface. The tangential vibration \((a_j(t))\) leads the radial vibration \((b_j(t))\), i.e. \( \theta > 0 \). As shown in Figure 5.1b), the tool first moves into the workpiece at point \( A \), and moves away radially towards \( B \) to lose the contact with the cut surface. The cutting edge also loses the contact with the chip because the motion in direction \( u \) has a velocity in the opposite direction to the nominal cutting speed \( V_s \) (see point \( B \) in Figure 5.1b)).

The immersion angle \( \phi_j(t) \) at tooth \( j \) is influenced by both the spindle speed \((\Omega [\text{rad/s}])\) and the tangential vibrations \((a_j(t))\) delivered by the ultrasonic actuator:

\[
\phi_j(t) = \Omega t + \frac{a_j(t)}{R} + (j - 1)\phi_p
\]

where \( R \) is the radius of the milling tool.

The cutting edge motion is governed by the rigid body rotation and linear feed of the tool; self excited, regenerative vibrations between the tool and workpiece which are contributed by the dynamic flexibilities of their structures; and the ultrasonic vibrations imposed at a constant frequency with fixed amplitudes \((A_0, B_0)\) in tangential and radial directions by the piezo actuator. The dynamic chip thickness \((h_j(t))\) removed by the tooth \( j \) at time instant \( t \) is measured in the radial direction with the following three components (Figure 5.2):
The quasi-static chip thickness \( h_{0,j}(t) \) removed by the rigid body motion of milling tool at the instantaneous immersion angle \( \phi_j(t) \) is

\[
h_{0,j}(t) = c \sin \phi_j(t).
\]  

where \( c \) is the feed rate in [m/rev/tooth]. The regenerative chip generated by structural vibrations \( v_j \) between the present and previous in-cut teeth can be expressed as:

\[
\Delta v_j(t) = -[v_j(t) - v_j(t - \tau_j)]
\]  

where \( \tau_j \) is the time delay between the present and the previous teeth which are in contact with the work material. If two adjacent teeth are in cut, the time delay \( \tau_j \) will be the tooth passing period \( T = 2\pi \Omega / N \). The structural vibrations contributed by the machine and the workpiece are modeled in feed \( x \) and normal \( y \) directions and projected to the cutting edge \( j \) location in the radial direction as:

\[
v_j(t) = x(t) \sin \phi_j(t) + y(t) \cos \phi_j(t)
\]  

Figure 5.1: Elliptical vibration-assisted milling process.
The chip thickness produced by the ultrasonic vibrations along the radial direction is modeled as the difference between the marks left by the present tooth $j$ at time $t$ ($b_j(t)$) and the previous in-cut tooth $q$ in the radial direction with $\tau_j$ delay ($b_q(t-\tau_j)$):

$$\Delta b_j(t) = -[b_j(t) - b_q(t-\tau_j)]$$  \hspace{1cm} (5.7)

The excited radial vibration ($b_j(t)$) changes the dynamic chip thickness, but it has a fixed frequency and amplitude which contributes to the forced vibrations only. However, the tool loses its contact with the cut surface when the amplitude of the ultrasonic vibration becomes larger than the chip thickness.

The tool-workpiece contact is indicated by the step function $g_j(t)$ in Eq.(5.3) as

$$ \begin{cases} 
  g_j(t) = 1, \text{ tool is in contact} \\
  g_j(t) = 0, \text{ tool is not in contact} 
\end{cases} $$ \hspace{1cm} (5.8)

Tool loses the contact with the workpiece under several conditions. When the tooth is out of the immersion as in regular milling,

$$ \begin{cases} 
  g_{1,j}(t) = 1, \phi_{st} \leq \phi_j \leq \phi_{ex} \\
  g_{1,j}(t) = 0, \phi_j < \phi_{st} \text{ or } \phi_j > \phi_{ex} 
\end{cases} $$ \hspace{1cm} (5.9)
where $\phi_{st}$ and $\phi_{ex}$ are the entry and exit angles of the cutter relative to the part, respectively.

### 5.2.2 Time delay in elliptical vibration-assisted milling

The time delay ($\tau_j$) in Eq.(5.7) may not always be equal to the tooth passing period depending on the instantaneous chip thickness and ultrasonic vibration amplitude of the tooth. By ignoring the regenerative term ($\Delta v_j$) in the resultant chip thickness in Eq.(5.3), the chip thickness becomes:

$$h_{r0,j} = c \sin \phi_j(t) - b_j(t) + b_q(t - \tau_j)$$

(5.10)

which can be zero or negative for small feed rates ($c$) or the immersion angles ($\phi_j$). Hence the tool disengages from the workpiece along the radial direction when:

$$\begin{cases} 
  g_{2,j}(t) = 1, h_{r0,j}(t) \geq 0 \\
  g_{2,j}(t) = 0, h_{r0,j}(t) < 0 
\end{cases}$$

(5.11)

The time delay ($\tau_j$) is determined by numerically searching the previous in-cut tooth at the present immersion angle $\phi_j(t)$ in time domain as described in Algorithm 1, where $t_i$ is the time at discrete interval $i$, $n$ is the number of tooth passing periods to search last tooth in cut, and $q$ is the index of the tooth. The index $q$ is derived based on the present tooth $j$ and the number of teeth $N$ as described in Algorithm 1 Lines 4-7. The searching procedure starts from $n = 1$, which corresponds to the last tooth present at the immersion angle $\phi_j(t_i)$. If the last tooth is not in cut, where $g_{2,q}(t_q) = 0$, then the tooth in the previous two cycles is examined until a tooth contacts the material (i.e. $g_{2,q}(t_q) = 1$). Once the time of last in-cut tooth ($t_q$) is determined, the time delay is evaluated as:

$$\tau_j(t_i) = t_i - t_q$$

(5.12)

Figure 5.3 shows two simulations at the same spindle speed but with different feed rates. If the feed rate is much larger than the amplitude of the radial vibration such as the conditions given in Figure 5.3 a), the tooth will only jump out of the workpiece along radial direction around the entry angle for up milling and around
the exit angle for down milling where sin$\phi_j$ is small. The time delay will be equal to the tooth passing period ($T$) except at the entry region for up milling and the exit region for down milling. If the feed rate per tooth is comparable to the amplitude of radial vibration such as in Figure 5.3(b), multiple tool-workpiece separations will occur within the immersion but the time delay will be integer multiples of tooth passing periods. In particular, when the tooth passing period ($T$) is selected based on the excitation vibration period as:

$$\frac{\omega_e}{N\Omega} = \frac{T}{T_e} = k$$

(5.13)
where $k$ is an integer number and $T_e = 2\pi/\omega_e$ is the ultrasonic excitation vibration period, the time delay is a constant and equals to one tooth passing period ($\tau_j = T$). Hence the selection of tooth passing frequencies (i.e. spindle speed times number of teeth) as the integer division of ultrasonic vibration frequency cancels its contribution ($\Delta b_j(t) = 0$) to the dynamic chip thickness in Eq.(5.3).

### 5.2.3 Tangential tool-workpiece separation

The resultant tangential speed ($V_{r,j}$) of the tooth $j$ is the summation of cutting speed and tangential vibration velocity:

$$V_{r,j}(t) = V_s + \ddot{a}_j(t) = V_s - A_0 \omega_e \sin \left[ \omega_e t + \left( j - 1 \right) \frac{2\pi}{N} \right]$$  \hspace{1cm} (5.14)

The cutting edge periodically disengages from the cut surface when the actuator imposed oscillating tangential vibration velocity ($\dot{a}_j(t)/dt$) exceeds the nominal cutting speed ($V_s$). The amplitude of the vibration velocity is always set greater than the nominal cutting velocity ($V_s < A_0 \omega_e$) to produce periodically intermittent contacts along the tangential direction. Figure 5.4 shows one cycle of the tool-workpiece separation. When the harmonically varying vibration velocity ($\dot{a}_j(t)$) becomes negative, and the resultant velocity at the cutting edge ($V_{r,j}$) reaches zero ($V_{r,j} = 0$, $t = t_A$ in Figure 5.4), and the tooth begins to leave the workpiece. When $t_A < t < t_B$, the cutting edge travels back until the resultant velocity ($V_{r,j}$) reaches zero again.
zero again at time \( t_B \) in Figure 5.4. Then, the tangential vibration velocity increases the resultant cutting speed and reaches the nominal cutting speed \( (V_S) \) at \( t = t_C \).

The resultant displacement \( (s_{r,j}) \) generated at the tool tip by the nominal cutting speed \( V_s \) and tangential vibration \( a_j(t) \) is:

\[
s_{r,j} = V_st + A_0 \cos \left[ \omega_e t + (j - 1) \frac{2\pi}{N} \right]
\]

(5.15)

The resultant displacement \( (s_{r,j}) \) oscillates about the nominal displacement \( (V_s t) \) due to the tangential vibration. As shown in Figure 5.4 for one cycle separation, when the tool leaves the workpiece at \( t_A \), the resultant displacement \( (s_{r,j}) \) reaches a local maximum point \( (s = s_{max}) \). As the tool travels back after \( t = t_A \), the resultant displacement is decreased until \( t = t_B \) because of the negative tangential
velocity. The resultant displacement rises after $t_B$ when the resultant tangential velocity becomes positive, and then it reaches the local maximum point $s_{\text{max}}$ again at $t = t_C$. Meanwhile, the cutting tool touches the workpiece. The local maximum ($s_{\text{max}}$) keeps updating until the resultant velocity reaches zero again after $t_C$. The tool contact along the tangential direction is considered in the step function as:

$$
\begin{cases}
g_{3,j}(t) = 1, & s_{r,j}(t) \geq s_{\text{max}} \\
g_{3,j}(t) = 0, & \text{else}
\end{cases}
$$

(5.16)

The resultant displacement ($s_{r,j}$) is recorded, and the local maximum point ($s_{\text{max}}$) is calculated and updated using the recorded displacements at each sampling period in the simulation.

The overall tool-workpiece contact ($g_j(t)$) is determined by combining the three types of tool-workpiece separations as:

$$
g_j(t) = g_{1,j}(t)g_{2,j}(t)g_{3,j}(t)
$$

(5.17)

where $g_{1,j}$, $g_{2,j}$ and $g_{3,j}$ are defined in (5.9), (5.11) and (5.16) respectively.

### 5.2.4 Process damping effect

Contacts between the round cutting edge of cutting tool and the wavy workpiece surface produce process damping in dynamic cutting [35]. In the elliptical vibration-assisted cutting, there exists additional ploughing process due to the radial vibrations as explained in section 5.2.1, which change the process damping. The process damping forces are modeled as a function of elastic contact volume between the cutting tool and workpiece [2], and the instantaneous process damping forces along the radial and tangential directions are defined as:

$$
F_{r,j}^{pd}(t) = K_{sp}V_{j}^{pd}(t) ; \quad F_{t,j}^{pd}(t) = \mu F_{r,j}^{pd}
$$

(5.18)

where $K_{sp}$ is the material dependent indentation coefficient which can be identified from cutting experiments, and $\mu$ is the Coulomb friction constant between cutting tool and workpiece. $V_{j}^{pd}$ is the dynamically indented volume ($V_{j}^{pd}$) for tooth $j$, and it can be evaluated from the cross-section area ($A_{j}^{pd}$ in Figure 5.5) and depth of cut
$w$, i.e. $V_{pd}^j = wA_{pd}^j$. The area $(A_{pd}^j)$ indented by the flank of tool with an equivalent wear length $(L)$ at cutting velocity $(V_r)$ is expressed as [2]:

$$A_{pd}^j = \frac{L^2}{2V_r} \dot{r}_j$$  \hspace{1cm} (5.19)

where $\dot{r}_j$ is the velocity along the radial direction for tooth $j$. The equivalent wear length $(L)$ can be modeled as (see Figure 5.5):

$$L = R_e \left[ \sin \beta_e + \sin \gamma + \left( \cos \gamma - \cos \beta_e \right) / \tan \gamma \right]$$  \hspace{1cm} (5.20)

where $R_e$ is the edge radius of the cutting tool, $\gamma$ is the clearance angle of the tool, and $\beta_e$ is the angle to define the separation point (SP) as shown in Figure 5.5.

If the process is conventional milling and there is no vibration assistance, the separation angle $(\beta_e)$ is assumed to be a constant so as the equivalent wear length $(L)$. However, when the elliptical vibrations are present, the separation point dynamically changes so as the separation angle $(\beta_e)$ and equivalent wear length $(L)$. The dynamic separation angle for tooth $j$ can be evaluated from the geometry given in Figure 5.5 as:

$$\beta_e(t) = \arccos \left[ \frac{R_e \cos \beta_0 + b_j(t)}{R_e} \right]$$  \hspace{1cm} (5.21)

where $\beta_0$ is the separation angle without elliptical vibration assistance, and $b_j(t)$ is
the excited vibration displacement along the radial direction defined in Eq.(5.1).

The radial velocity ($\dot{r}_j$) is contributed by the excited ultrasonic vibration ($b_j(t)$) and the regenerative vibration ($v_j(t)$) as:

$$\dot{r}_j(t) = \dot{v}_j(t) + \dot{b}_j(t)$$

(5.22)

The instantaneous indented area ($A_{pd}^j$) in (5.19) can be evaluated from (5.22). The ultrasonic vibration ($b_j(t)$) is a forced vibration with the known amplitude and frequency, thus it doesn’t contribute to the chatter stability of milling process.

### 5.3 Chatter Stability Analysis

#### 5.3.1 Dynamic milling force

The quasi-static components ($c\sin \phi_j(t) + \Delta b_j(t)$) of the dynamic chip thickness (Eq.(5.3)) cause only forced vibrations and do not affect the chatter stability of the process, hence they are dropped. Only regenerative term with the step function $g_j(t)$ which reflects the tool disengagements contributed by the ultrasonic vibrations is considered as follows:

$$h_{w,j}(t) = g_j(t) \left[-(v_j(t) + v_j(t - \tau_j))\right] = g_j(t) \left[\sin \phi_j \cos \phi_j \right] \left[ \begin{array}{c} x(t) - x(t - \tau_j) \\ y(t) - y(t - \tau_j) \end{array} \right]$$

(5.23)

where $\phi_j$ is the immersion angle of tooth $j$, and $[x(t), y(t)]^T$ are the structural vibrations in feed and normal directions, respectively. Dynamic tangential ($F_{t,j}$) and radial ($F_{r,j}$) cutting forces generated by tooth $j$ are expressed as:

$$\begin{bmatrix} F_{t,j} \\ F_{r,j} \end{bmatrix} = K_i w h_{w,j}(t) \begin{bmatrix} 1 \\ K_r \end{bmatrix} + g_j(t) C_{eq} \begin{bmatrix} \mu \\ 1 \end{bmatrix} \dot{v}_j$$

(5.24)

where $w$ is the axial depth of cut and $C_{eq} = L^2 w/(2V_r)$ is the equivalent process damping coefficient. The milling forces contributed by $N$ number of teeth are
projected to feed and normal directions:
\[
\begin{bmatrix}
F_x \\
F_y
\end{bmatrix} = \sum_{j=1}^{N} \begin{bmatrix}
-\cos \phi_j & -\sin \phi_j \\
\sin \phi_j & -\cos \phi_j
\end{bmatrix} \begin{bmatrix}
F_{t,j} \\
F_{r,j}
\end{bmatrix}
\] (5.25)

By substituting Eqs.(5.23) and (5.24) into Eq.(5.25), and the dynamic milling forces in \(xy\) coordinates are expressed as:
\[
\begin{bmatrix}
F_x \\
F_y
\end{bmatrix} = \frac{1}{2} K_t w \sum_{j=1}^{N} g_j(t) \begin{bmatrix}
\alpha_{xx,j} & \alpha_{xy,j} \\
\alpha_{yx,j} & \alpha_{yy,j}
\end{bmatrix} \begin{bmatrix}
x(t) - x(t - \tau_j) \\
y(t) - y(t - \tau_j)
\end{bmatrix} + C_{eq} \begin{bmatrix}
\beta_{xx,j} & \beta_{xy,j} \\
\beta_{yx,j} & \beta_{yy,j}
\end{bmatrix} \begin{bmatrix}
\dot{x}(t) \\
\dot{y}(t)
\end{bmatrix}
\] (5.26)

where the directional factors \((\alpha_{xx,j}, \alpha_{xy,j}, \alpha_{yx,j}, \alpha_{yy,j})\) and process damping coefficients \((\beta_{xx,j}, \beta_{xy,j}, \beta_{yx,j}, \beta_{yy,j})\) are:
\[
A_j(t) = \begin{bmatrix}
\alpha_{xx,j} & \alpha_{xy,j} \\
\alpha_{yx,j} & \alpha_{yy,j}
\end{bmatrix} = \begin{bmatrix}
-\cos \phi_j \sin \phi_j - K_r \sin^2 \phi_j & -\cos^2 \phi_j - K_r \sin \phi_j \cos \phi_j \\
-\sin^2 \phi_j - K_r \cos \phi_j \sin \phi_j & \sin \phi_j \cos \phi_j - K_r \cos^2 \phi_j
\end{bmatrix}
\] (5.27)
\[
B_j(t) = C_{eq} \begin{bmatrix}
\beta_{xx,j} & \beta_{xy,j} \\
\beta_{yx,j} & \beta_{yy,j}
\end{bmatrix} = C_{eq} \begin{bmatrix}
-\mu \cos \phi_j \sin \phi_j - \sin^2 \phi_j & -\mu \cos^2 \phi_j - \sin \phi_j \cos \phi_j \\
-\mu \sin^2 \phi_j - \cos \phi_j \sin \phi_j & \mu \sin \phi_j \cos \phi_j - \cos^2 \phi_j
\end{bmatrix}
\]

which are periodic and \(g_j(t)\) is time varying as the spindle rotates \((\phi_j = \Omega t)\) because it is dependent on ultrasonic vibrations \((a_j, b_j)\) and feed rate \((c)\).

### 5.3.2 Stability of milling with ultrasonic vibrations

The milling dynamics with multiple modes are expressed in modal coordinate space as:
\[
\ddot{\mathbf{x}} + \mathbf{C}\dot{\mathbf{x}} + \mathbf{D}\mathbf{x} = \mathbf{U}^T \mathbf{F}
\] (5.28)
where \( U \) is the mode shape matrix, 
\[
\mathbf{C} = \begin{bmatrix}
2\zeta_1\omega_{n,1} & \cdots & 0 \\
\vdots & \ddots & \vdots \\
0 & \cdots & 2\zeta_n\omega_{n,n}
\end{bmatrix},
\mathbf{D} = \begin{bmatrix}
\omega_{n,1}^2 & \cdots & 0 \\
\vdots & \ddots & \vdots \\
0 & \cdots & \omega_{n,n}^2
\end{bmatrix},
\]
\( \omega_{n,k} \) and \( \zeta_k \) are the natural frequency and modal damping ratio for mode \( k \), and \( n \) is the number of dominant modes of the spindle-holder-tool assembly. \( \mathbf{F} = [F_x \ F_y]^T \) is the dynamic milling forces acting on the milling tool. The modal displacements (\( \mathbf{\hat{x}} = [\hat{x}_1 \ \hat{x}_2 \ \cdots \ \hat{x}_n]^T \)) can be transformed to physical vibrations with:
\[
[\begin{bmatrix} x \\ y \end{bmatrix}] = \mathbf{U}\mathbf{\hat{x}} \quad \text{(5.29)}
\]

Substituting Eq.(5.26) into Eq.(5.28), the system is represented by a set of delayed differential equations with periodic coefficients:
\[
\ddot{\mathbf{\hat{x}}} + \mathbf{C}\dot{\mathbf{\hat{x}}} + \mathbf{D}\mathbf{\hat{x}} = \mathbf{\delta} \sum_{j=1}^{N} g_j(t) \mathbf{U}^T \mathbf{A}_j(t) \mathbf{U} [\mathbf{\hat{x}}(t) - \mathbf{\hat{x}}(t - \tau_j)] + \mathbf{U}^T \mathbf{B}\mathbf{\dot{U}}\mathbf{\hat{x}} \quad \text{(5.30)}
\]
where \( \mathbf{\delta} = \frac{1}{2} \mathbf{K}_a \) and \( \mathbf{B} = \sum_{j=1}^{N} \mathbf{B}_j \). By letting \( \mathbf{q}(t) = [\mathbf{\hat{x}}(t)^T \quad \dot{\mathbf{\hat{x}}}(t)^T]^T \), Eq. (5.30) is rearranged into a set of first-order equations as [32]:
\[
\dot{\mathbf{q}}(t) = \mathbf{L}\mathbf{q}(t) + \sum_{j=1}^{N} \mathbf{R}_j\mathbf{q}(t - \tau_j) \quad \text{(5.31)}
\]
where \( \mathbf{L}(t) = \begin{bmatrix} \mathbf{0} & \mathbf{I} \\ \delta \sum_{j=1}^{N} g_j(t) \mathbf{U}^T \mathbf{A}_j(t) \mathbf{U} - \mathbf{D} & \mathbf{U}^T \mathbf{B}\mathbf{U} - \mathbf{C} \end{bmatrix} \) and \( \mathbf{R}_j(t) = \begin{bmatrix} \mathbf{0} & \mathbf{0} \\ -\delta g_j(t) \mathbf{U}^T \mathbf{A}_j(t) \mathbf{U} & \mathbf{0} \end{bmatrix} \). The matrices \( \mathbf{L} \) and \( \mathbf{R}_j \) are time-varying due to the time-varying factors \( \mathbf{A}_j \) and \( \mathbf{B}_j \).

The Semi-discrete method proposed by Insperger and Stepan[32] is used to solve the stability of elliptical vibration-assisted milling process governed by Eq.(5.31). The simulation time length \( (\tau_m) \) is determined by the maximum time delay, where \( \tau_m = \max(\tau_j(t)) \). Let \( \Delta t \) be the sampling period, and the simulation period is divided into \( m \) number of time discrete intervals, i.e. \( \tau_m = m\Delta t \). At current time
\( t = t_i \), for tooth \( j \), the time delay can be expressed in as:

\[
I_{i,j} = \frac{\tau_j(t_i) + \Delta t/2}{\Delta t}
\]

(5.32)

Let \( q(t_i) = q_i \) and the state variable \( (q) \) at \( t = t_{i+1} \) is:

\[
q_{i+1} = e^{L_{i}\Delta t}q_i + \sum_{j=1}^{N} \frac{1}{2} \left( e^{L_{i}\Delta t} - I \right) L^{-1}R_{i,j} \left( q_{i-l_{i,j}+1} + q_{i-l_{i,j}} \right)
\]

(5.33)

where \( L_i = L(t_i) \) and \( R_{i,j} = R_j(t_i) \). The solution requires the states between \( q_i \) and \( q_{i-m} \), and the state function in the discrete time domain is expressed as follows:

\[
Q_{i+1} = H_iQ_i
\]

(5.34)

where \( Q_i = \begin{bmatrix} q_i^T & q_{i-1}^T & \ldots & q_{i-m+1}^T \end{bmatrix}^T \), and \( H_i \) is defined as:

\[
H_i = \begin{bmatrix}
0 & \ldots & 0 & 0 \\
I & 0 & \ldots & 0 \\
0 & I & \ldots & 0 \\
\vdots & \vdots & \ddots & \vdots \\
0 & 0 & \ldots & I
\end{bmatrix}_{2n(m+1) \times 2n(m+1)} + \sum_{j=1}^{N} \begin{bmatrix}
0 & \ldots & \frac{1}{2} \left( e^{L_{i}\Delta t} - I \right) L^{-1}R_{i,j} & \frac{1}{2} \left( e^{L_{i}\Delta t} - I \right) L^{-1}R_{i,j} & \ldots & 0 \\
0 & \ldots & 0 & 0 & \ldots & 0
\end{bmatrix}_{2n(m+1) \times 2n(m+1)}
\]

(5.35)

Thus, within a simulation period \( \tau_m \), the transition equation of the states becomes:

\[
Q_m = H_0Q_0 = H_m \cdots H_2H_1Q_0
\]

(5.36)

According to the Floquet theory, the system in (5.36) is stable if all the eigenvalues of \( H \) are less than unity, and the critical stability corresponds to unity eigenvalues.
5.4 Simulations and Experiments

The stability of milling with elliptical vibration assistance is simulated using the semi-discrete time domain method as presented in Section 5.3 and compared against experimental measurements using the setup shown in Figure 5.6. The helical end mill had 2 flutes with 6 mm diameter and 25 mm overhang. The edge radius of the cutting tool was $15 \mu m$ with a clearance angle of $5^\circ$.

The frequency response function (FRF) of the tool at its tip was measured through impulse tests using a small (Dytran 5800B2) and miniature (PCB086E80) hammers to cover both low ($< 2,000$ Hz) and high frequency ($< 20,000$ Hz) modes of the end mill. A laser Doppler vibrometer (Polytec CLV-2534) was used to capture the displacements. The measured FRFs in feed ($x$) and normal ($y$) directions are given in Figure 5.7 with the curve fitted modal parameters listed in Table 5.1. The actuator’s third bending mode at $f_e = 16.35$ kHz is used to excite the ultrasonic elliptical vibrations with $A_0 = 10 \mu m$ in tangential and $B_0 = 5 \mu m$ in radial directions with a phase difference of $\theta = 90^\circ$. The stability of the system is analyzed for
the first bending dominant modes of the holder (369 Hz, 411 Hz) and the end mill (1146 Hz, 1069 Hz) since the higher modes beyond 4 kHz are process damped out [35].

Aluminum alloy 7050 and AISI 1045 steel alloys have been used as workpiece materials with the experimentally identified cutting force coefficients ($K_t, K_r$) and computed indentation ($K_{sp}$) coefficients [21]. The separation angle of plastic chip flow under the edge radius is set to $\beta_e = 50^\circ$ [35]. The predicted stability lobes and sample measurements during half immersion up milling of Al 7050 are shown in Figure 5.8. The minimum stable depth of cut increases from 1.3 mm to 2.00 mm up to the spindle speed of 2300 rev/min under ultrasonic vibrations. The effect of ultrasonic vibrations on the stability diminishes at higher speeds because the cutting
speed becomes higher than the vibration velocity, hence the intermittent contact between the tool and workpiece is lost. A sample sound measurements corresponding to point A in Figure 5.8 at the axial depth of cut of 1.5 mm and spindle speed of 1624 rev/min are shown in Figure 5.9. When the ultrasonic vibration was turned on in the beginning of the cut, the process was stable, the surface was smooth and the sound spectrum was dominated by the actuator’s excitation frequency at 16.37 kHz, which was a forced vibration. When the vibration actuator was turned off, the chatter developed quickly at the first bending mode of the tool at 1107 Hz which led to poor surface finish. The presence of chatter was decided by checking the amplitude and side bands which occur at tooth passing frequency (54 Hz) away [55]. Similar phenomenon was observed in all marked experimental results.

The predicted stability lobes for half immersion down milling of AISI 1045 steel is shown in Figure 5.10. The minimum depth of cut was increased from 0.2 mm to 0.35 mm under ultrasonic vibrations. Similar to milling Al 7050, the effect of ultrasonic vibrations diminished again after 2300 rev/min because the tool does not lose the contact with the workpiece beyond this cutting speed. A
Figure 5.9: Microphone outputs in half-immersion up milling of AL 7050 with a two-tooth cutter. Spindle speed: 1624 rev/min; axial depth of cut: 1.5 mm. a) Microphone data in time domain; b) FFT of microphone output with ultrasonic vibration assistance; c) FFT of microphone output when ultrasonic vibration turned off.

A sample of sound measurements at 1770 rev/min with 0.4 mm depth of cut (B in Figure 5.10) is given in Figure 5.11. When the ultrasonic vibration was turned on, the vibration was dominated by the frequency of actuator’s ultrasonic excitation.
Figure 5.10: Stability lobes for half immersion down milling of Steel AISI 1045 with a feed rate of \( c = 0.1 \) mm/rev/tooth. Tool: Two fluted, 6 mm diameter helical end mill with a helix angle of 30°. The modal parameters are given in Table 5.1. Cutting coefficients: \( K_t = 3850 \) MPa, \( K_r = 0.19 \), \( K_{sp} = 47000 \) N/mm³.

frequency (16.1 kHz). When the actuator was turned off, the system chattered at the holder’s bending mode of 393 Hz with having side-bands at 59 Hz away.

5.5 Summary

The chatter stability model of the synchronized elliptical vibration-assisted milling process has been presented in this chapter. It is shown that the chatter is dominated at low-frequency modes of the end mill at low speeds, and the excitation of actuator’s ultrasonic mode contributes to forced vibrations while improving the chatter stability. The ultrasonic vibrations can improve the chatter stability only if the tool has periodic contacts with the workpiece at the ultrasonic vibration frequency, which occurs if the tangential vibration velocity is higher than the nominal cutting speed of the tool or the vibration amplitude is greater than the static chip thickness governed by the feed rate and immersion. The process must be designed to select the feed rate, cutting speed, and the excited vibration amplitudes to avoid chatter without saturating the amplifier of the piezoelectric actuator. The proposed model has been experimentally validated in milling aluminum alloy 7050 and AISI 1045 steel alloys.
Figure 5.11: Microphone outputs in half-immersion down milling of AISI 1045 steel with a two-tooth cutter. Spindle speed: 1770 rev/min; axial depth of cut: 0.4 mm. a) Microphone data in time domain; b) FFT of microphone output with ultrasonic vibration assistance; c) FFT of microphone output when ultrasonic vibration turned off.
Chapter 6

Vibration Assistance on Chip Formation of Ti-6Al-4V

6.1 Overview

The high-frequency tool-workpiece separation due to ultrasonic vibration assistance changes the strain rate and heat transfer conditions in the primary shear zone, further affect the chip formation mechanism. This chapter selects a widely used difficult-to-cut titanium alloy, Ti-6Al-4V, as the target to investigate the effects of ultrasonic vibration on chip formation. Experimental observations are presented first in section 6.2, followed by a plastic flow model in the primary zone to explain the observations (section 6.3). Simulations and experimental results are presented in section 6.4 to verify the proposed plastic flow model.

6.2 Experimental Observations

Orthogonal cutting experiments of Ti-6Al-4V with ultrasonic vibration assistance were carried out, with the turning setup shown in Figure 6.1. The Ti-6Al-4V workpiece clamped into the spindle was a tube with 6 mm diameter and 1mm thickness. A turning insert was mounted in a stationary ultrasonic vibration actuator, and the tool edge, which was wider than the tube thickness, was oriented along Y-axis to achieve the orthogonal cutting setup. In the cutting experiments, the cutting tool...
Figure 6.1: Turning experimental setup with an ultrasonic vibration cutting tool.

was fed to the workpiece along Z-axis, and the ultrasonic vibration was generated along X-axis (tangential direction) in the cutting experiment, as shown in Figure 6.1. The material of the turning insert is uncoated tungsten carbide, and the rake angle ($\alpha$) is 5°. The cutting tool was fixed on a tool holder driven by a stacking piezoelectric actuator. The bending vibration mode of the tool holder structure is excited at 15.5 kHz, and the vibration assistance at the turning insert is along the tangential direction at the cutting point. The piezo vibration actuator can produce vibrations at the amplitude of 10 $\mu$m. Multiple cutting experiments with different uncut chip thicknesses and cutting speeds listed in Table 6.1 were conducted. For each condition, both conventional and vibration-assisted cutting experiments were conducted. The cutting forces for each experiment were measured by a dynamometer (Kistler 9256C1) as shown in Figure 6.1. The bandwidth of the dynamometer is 2 kHz, which is much lower than the excitation frequency of the ultrasonic vibration.

The machined chips under each cutting condition with and without vibration
Table 6.1: Conditions of orthogonal cutting experiments

<table>
<thead>
<tr>
<th>No.</th>
<th>( V_c ) (m/min)</th>
<th>Uncut chip thickness (mm)</th>
<th>Vibrations</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>40</td>
<td>0.15</td>
<td>no</td>
</tr>
<tr>
<td>2</td>
<td>40</td>
<td>0.15</td>
<td>yes</td>
</tr>
<tr>
<td>3</td>
<td>50</td>
<td>0.15</td>
<td>no</td>
</tr>
<tr>
<td>4</td>
<td>50</td>
<td>0.15</td>
<td>yes</td>
</tr>
<tr>
<td>5</td>
<td>40</td>
<td>0.2</td>
<td>no</td>
</tr>
<tr>
<td>6</td>
<td>40</td>
<td>0.2</td>
<td>yes</td>
</tr>
<tr>
<td>7</td>
<td>50</td>
<td>0.2</td>
<td>no</td>
</tr>
<tr>
<td>8</td>
<td>50</td>
<td>0.2</td>
<td>yes</td>
</tr>
</tbody>
</table>

assistance were collected and polished. They were etched by Kroll’s reagent to reveal the microstructure, and then were examined by scanning electron microscopy (SEM) imaging (Hitachi SU3500). Figure 6.2 shows the SEM images of the chips corresponding to 40 m/min cutting speed and 0.15 mm uncut chip thickness, at the magnifications of 200X, 500X, and 3000X, respectively. When no vibration assistance is applied, the chip segmentations were formed in a saw-tooth shape, as shown in Figures 6.2a) - c). Adiabatic shear bands are clearly observed between two neighboring segmented chips, including highly distorted grain structures (see Figure 6.2c)) compared to other locations. The width of the shear band is about 6 - 8 \( \mu \text{m} \) from the SEM measurement. Figures 6.2d) - f) show the morphology and microstructure of the chip in ultrasonic vibration-assisted cutting. The chips formed with ultrasonic vibration assistance also has segmentations (Figure 6.2d)), however, compared to the no-vibration result, there is no obvious shear band formation, and no severe distorted grain structures were observed between the segmented chips (see Figures 6.2e) and f)). In addition, the pitch length of the chip segmentation with ultrasonic vibration (about 130 \( \mu \text{m} \)) is larger than that in conventional cutting (about 90 \( \mu \text{m} \)) by comparing Figures 6.2a) and d). Therefore, the experimental results demonstrate that the ultrasonic vibration assistance plays a significant role on the shear band formation and chip segmentation in cutting Ti-6Al-4V. Figure 6.3 shows the SEM micrographs of the machined chips cor-
responding to tests No. 3-8 in Table 6.1, and similar observations on the effect of vibration assistance on adiabatic shear bands were obtained at different cutting conditions.

The studies in [18, 44] report that the chip segmentations in cutting Ti-6Al-4V at high cutting speed (larger than 30 m/min) is caused by heat accumulation within the primary shear zone to generate shear bands. Thus, in this chapter, it is hypothesized that the disappearance of adiabatic shear bands caused by the ultrasonic vibration assistance is due to less heat accumulation, which results in lower temperature in the primary shear zone. Therefore, ultrasonic vibration assistance is able to improve the machinability of Ti-6Al-4V with lower temperature as the experimental observations suggest. Furthermore, the tool life is expected to be elongated without reducing productivity. However, no physical model of this cutting process was reported in the literature. Section 6.3 is to model the stress and temperature variations as well as plastic shear flow in primary shear zone in cutting of Ti-6Al-4V with ultrasonic vibration assistance, in order to quantitatively...
determine the effect of vibration assistance on the chip formation mechanism.

### 6.3 Plastic Flow in Primary Shear Zone

The process model focuses on the material deformation in primary shear zone, which determines the chip segmentation, further to explain the observations in section 6.2. The model is based on plasticity theory to characterize the material flow in forming the chip. The heat transfer in the shear zone is considered to determine
the effect of temperature increasing on the plastic flow. In the literature, multiple researchers contributed to modeling the chip formation analytically. Burns and Davies established governing equations to solve the shear stress and temperature in the primary shear zone [11]. Bai et al. studied the chip formation of Ti-6Al-4V analytically using governing equations in the primary shear zone and the tool-chip interface, and they predicted the pitch of segmented chip caused by adiabatic shear bands [7]. Ning et al. modeled the cutting process of ultra-fine-grained titanium analytically using Johnson-Cook constitutive relationship [64]. Ning and Liang predicted the temperature in the primary shear zone and tool-chip interface by using a material constitutive model [63].

The vibration assistance changes the cutting tool kinematics, therefore influences the stress and temperature variations in the primary shear zone. In this model, the governing equations of shear stress and temperature including the effect of vibration assistance are established, and the material flow is predicted from the material constitutive law. By solving the governing and the constitutive equations, the shear stress and the temperature in the primary shear zone are simulated in time domain.

When vibration assistance is applied along the tangential direction (X-axis in Figure 6.1) in orthogonal cutting, the cutting speed changes periodically. The assisted ultrasonic vibration is a sinusoidal function at the excitation frequency \( f \), then the cutting tool displacement \( x(t) \) due to the assisted vibration is defined as:

\[
x(t) = x_0 \cos(2\pi ft) \tag{6.1}
\]

where \( x_0 \) is the amplitude of the ultrasonic vibration. The overall cutting speed \( V \) is the summation of original cutting speed along tangential direction \( V_c \) and vibration velocity \( V_f \), defined as:

\[
V(t) = V_c + V_f(t) = V_c - V_0 \sin(2\pi ft) \tag{6.2}
\]

where \( V_0 \) is the amplitude of the vibration velocity defined as \( V_0 = 2\pi f x_0 \).

With vibration assistance, the workpiece and the cutting tool contact intermittently if \( V_0 > V_c \). Otherwise, the tool keeps in contact with the workpiece with vary-
Figure 6.4: Orthogonal cutting with ultrasonic vibration along tangential direction: a) tool-workpiece in-contact period with cutting motion; b) chip elastic recovery period; c) tool-workpiece separation period.

Stage a): Cutting tool in contact with workpiece with cutting motion

Under this condition, the workpiece material has plastic deformation in primary shear zone due to the cutting tool motion, and forms the machined chip. Figure 6.5 shows the deformation zone in 2-D orthogonal cutting process. The primary shear zone is assumed to have a small thickness ($h$). Meanwhile, it is assumed that the shear stress and temperature are distributed uniformly along the shear plane. According to Burns and Davies [11], although the shear stress in the tool-chip contact interface causes plastic deformation of the chip material, the normal stress applied from the rake face to the chip generates local elastic deformation of the chip material because the back side of the machined chip is free surface, shown in Figure 6.5.

Based on the force equilibrium principle of the chip, the resultant force applied
Figure 6.5: Deformation zone in 2D orthogonal cutting.

to the chip in the shear plane direction is zero, that is:

\[ F_n \cos(\phi - \alpha) - \mu F_n \sin(\phi - \alpha) = F_s \]  \hspace{1cm} (6.3)

where \( F_n \) is the normal force on the rake face, \( F_s \) is the shear force in the primary shear zone, \( \mu \) is the average friction coefficient between the tool and the workpiece, \( \phi \) is the shear angle, and \( \alpha \) is the rake angle of the cutting tool. Based on equation (6.3), the relationship between the average normal stress along the chip contact region and the shear stress in the primary shear zone is expressed as:

\[ [\sigma \cos(\phi - \alpha) - \mu \sigma \sin(\phi - \alpha)] L b = \tau b \frac{d}{\sin \phi} \]  \hspace{1cm} (6.4)

where \( \sigma \) is the average normal stress on the contact length \( L \), \( \tau \) is the shear stress inside the primary shear zone, \( d \) is the uncut chip thickness, and \( b \) is the width of cut.

When chip segmentation occurs, the chip has a displacement \( \Delta u \) perpendicular to the tool rake face, causing unloading on the cutting tool. The displacement \( \Delta u \)
causes a normal strain in the local elastic deformation region (the green area in Figure 6.5), expressed as $\varepsilon = \Delta u/(d/\sin \phi \cos(\phi - \alpha))$.

The chip segmentation causes time-varying chip displacement $\Delta u$. As a result, the time-differentiation of normal stress due to the unloading from the chip segmentation is:

$$\dot{\sigma} = E \frac{\sin \phi}{d \cos(\phi - \alpha)} \Delta \dot{u}$$  \hspace{1cm} (6.5)

where $E$ is the elastic modulus of the chip material. Combined with (6.4), the variation of shear stress with respect to time is expressed as:

$$\dot{\tau} = \frac{EL \sin^2 \phi}{d^2} [1 - \mu \tan(\phi - \alpha)] \Delta \dot{u}$$  \hspace{1cm} (6.6)

where $\Delta \dot{u}$ is the speed difference between the tool and the segmented chip along the direction perpendicular to tool rake face, which is:

$$\Delta \dot{u} = (V_1 - V_2) \cos(\phi - \alpha)$$  \hspace{1cm} (6.7)

where $V_1$ and $V_2$ are the tool and formed chip speeds along the shear plane, respectively.

The shear speed due to cutting tool motion $V_1$ is a function of cutting speed $V$, shear angle $\phi$, and tool rake angle $\alpha$, expressed as:

$$V_1 = \frac{V \cos \alpha}{\cos(\phi - \alpha)}$$  \hspace{1cm} (6.8)

Meanwhile, the chip flow speed corresponding to chip segmentation $V_2$ is related to the actual shear strain rate of the plastic flow in the primary shear zone, defined as:

$$V_2 = \dot{\gamma}_p h$$  \hspace{1cm} (6.9)

where $\dot{\gamma}_p$ is plastic shear strain rate of material in primary shear zone due to the chip segmentation. Substitute (6.8) and (6.9) into (6.7), and based on the cutting speed equation (6.2), the shear stress variation with time in (6.6) is updated as:

$$\dot{\tau} = \frac{ELV_c \sin^2 \phi}{d^2} [1 - \mu \tan(\phi - \alpha)] \cos \alpha \left[ 1 - \frac{V_0}{V_c} \sin(2\pi ft) - \frac{\dot{\gamma}_p}{\dot{\gamma}_0} \right]$$  \hspace{1cm} (6.10)
where \( \dot{\gamma}_0 = \frac{V_c \cos \alpha}{\cos(\phi - \alpha)/h} \) refers to the shear strain rate in the primary shear zone corresponding to the original cutting speed without vibration assistance \((V_c)\). In equation (6.10), \( \dot{\gamma}_0 \) is a constant determined by the tool geometry and the original cutting speed, while \( \dot{\gamma}_p \) is a time-varying parameter associated with the chip segmentation.

Heat is also generated in primary shear zone during the cutting motion due to the plastic deformation of the workpiece material. By taking half of the primary shear zone as a controlled volume, the energy conservation condition is satisfied in the volume [11]. The overall stored energy is balanced by the heat generation, the convection of mass inflow and the heat transferred by conduction. The fundamental energy conservative relationship within the controlled volume is defined as follows:

\[
\dot{E}_{st} = \Delta \dot{E}_{conv} + \Delta \dot{E}_{cond} + \dot{E}_{gen}
\tag{6.11}
\]

where \( E_{st} \) is the stored energy inside the controlled volume, \( \Delta \dot{E}_{conv} \) is the heat convection variation due to the mass flow rate related to cutting speed inside the primary shear zone, \( \Delta \dot{E}_{cond} \) is the heat conduction, and heat generation \((E_{gen})\) is from strain energy caused by material deformation of the workpiece material.

The temperature at both boundaries of the primary shear zone is assumed as room temperature \((T_0)\), and the temperature at the centerline of the primary shear zone is \(T\). Therefore, the governing equation of temperature variation with respect to time is derived through (6.11) as:

\[
\rho c \dot{T} = \rho c \frac{2(T_0 - T)}{h} V \sin \phi + \frac{4k}{h^2} T_0 - T + \tau \dot{\gamma}_p
\tag{6.12}
\]

where \( k \) is the thermal conductivity, \( \rho \) is the mass density, and \( c \) is the specific heat capacity of the workpiece material.

The plastic strain rate of the material in the primary shear zone is influenced by the shear stress and the temperature, and it determines the plastic flow for chip formation. Since material plastic deformation occurs in the primary shear zone, Johnson-Cook (J-C) constitutive model which is used to describe the flow stress under large strain rate and high temperature is used in this model, described as
follows[45]:

$$
\tau = \left[ \tau_A + \tau_B \left( \frac{\gamma_p}{\sqrt{3}} \right)^n \right] \left[ 1 + C \ln \left( \frac{\dot{\gamma}_p}{\dot{\gamma}_{ref}} \right) \right] \left[ 1 - \left( \frac{T - T_0}{T_m - T_0} \right)^m \right]
$$

(6.13)

where $\gamma_p$ is the shear strain in the plastic region, which is given by the average value of the plastic strain in this model, $\dot{\gamma}_p$ is the plastic strain rate, $\tau_A$, $\tau_B$, $T_m$, $C$, $\dot{\gamma}_{ref}$, $n$, $m$ are material property related constants. To simplify the proposed model so that the governing and constitutive equations are analytically solvable, in the simulations, the average plastic strain ($\gamma_p$) is used for the strain hardening term based on constant material flow assumption, and it is obtained by $\gamma_p = \cos \alpha / (\sin \phi \cos (\phi - \alpha))$, where $\phi$ is experimentally calibrated shear angle from average thickness of the machined chip. By letting $\tau_0 = \tau_A + \tau_B (\gamma_p / \sqrt{3})$, the strain rate of the material is obtained from the Johnson-Cook constitutive law as a function of stress, strain and temperature, expressed as:

$$
\dot{\gamma}_p = \dot{\gamma}_{ref} \exp \left\{ \frac{1}{C} \left[ \frac{\tau}{\tau_0} \frac{1}{1 - \left( \frac{T - T_0}{T_m - T_0} \right)^m} - 1 \right] \right\}
$$

(6.14)

The equations (6.10), (6.12) and (6.14) describe the variations of shear stress, temperature and shear strain rate of the workpiece material in the primary shear zone. They form the coupled governing equations in describing the mechanism of segmented chip formation during cutting motion with vibration assistance in the tangential direction.

**Stage b): Tool retraction when tool and chip are in contact (elastic recovery)**

As the overall cutting tool velocity including the vibration assistance changes direction, the tool retracts and moves away from the workpiece material. Since the chip is deformed elastically at the tool-chip contact area as shown in Figure 6.5, it will bounce back when the cutting tool retracts, until complete elastic recovery is achieved. During this period, the cutting tool and the chip are still in contact, therefore sharing the same velocity, as shown in Figure 6.4 b). The normal stress ($\sigma$) in the tool-chip contact area reduces based on the cutting tool's displacement, expressed as (6.15), and the shear stress ($\tau$) in the primary shear zone is calculated
from the chip equilibrium condition by (6.16):

\[
\sigma = E \frac{1}{\cos(\phi - \alpha)} \cdot \frac{\Delta u_{sep} - u(t)}{d / \sin \phi} \tag{6.15}
\]

\[
\tau = \sigma [\cos(\phi - \alpha) - \mu \sin(\phi - \alpha)] \frac{\sin \phi}{d} \tag{6.16}
\]

where \(\Delta u_{sep}\) is the relative displacement between the tool and the chip when the cutting tool retraction just starts to occur, and \(V(t)\) is zero at this moment; \(u(t)\) is the displacement of the cutting tool during retraction period, and it is calculated by the integration of the cutting tool velocity.

During the tool retraction period, no further plastic deformation occurs in the primary shear zone, therefore, the heat generation and mass inflow components in (6.12) become zero. However, heat conduction still occurs, which transfers the heat out of the primary shear zone. As a result, the temperature variation is governed by:

\[
\dot{T} = 4 k T_0 - T \frac{\rho c}{h^2} \tag{6.17}
\]

**Stage c): Tool-chip separation period**

After the complete elastic recovery of the chip material, the cutting tool is separate from the workpiece material, shown in Figure 6.4 c). There is no mechanical loading on the workpiece material from the cutting tool, hence the shear stress in the primary zone is zero:

\[
\tau = 0 \tag{6.18}
\]

Only heat conduction occurs in the primary shear zone, and it is also governed by (6.17) in the separation period. Based on the governing equations in each tool-workpiece interaction period, the stress and temperature variations in the primary shear zone can be solved in time domain using finite difference method, with the simulations and experimental validations presented in the next section.
Table 6.2: Ti-6Al-4V material properties

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mass density $\rho$</td>
<td>4430</td>
<td>kg/m$^3$</td>
</tr>
<tr>
<td>Young’s modulus $E$</td>
<td>105</td>
<td>GPa</td>
</tr>
<tr>
<td>Specific heat conductivity $k$</td>
<td>16</td>
<td>W/(m·K)</td>
</tr>
<tr>
<td>Melting temperature $T_m$</td>
<td>1610</td>
<td>°C</td>
</tr>
</tbody>
</table>

6.4 Simulations and Experiments

Time-domain simulations are performed to predict the stress and temperature variations in the primary shear zone in cutting of Ti-6Al-4V with and without ultrasonic vibration assistance.

As discussed in section 6.3, Johnson-Cook constitutive law is used in this model to determine the relationship between the strain rate, shear stress, shear strain, and temperature. Different methods in determining the Johnson-Cook constitutive parameters have been reported in the literature. Split-Hopkinson pressure bar tests were widely used to experimentally calibrate the parameters under different strain, strain rate and temperature of the material [45, 77]. In addition, inverse identification of J-C parameters from orthogonal cutting tests was investigated [1, 90]. Numerical approaches were also used to adjust J-C constants to be used in cutting process with better accuracy [91]. Moreover, analytical methods for obtaining J-C constants in cutting simulations were reported in the literature. Ozel and Zeren identified the J-C parameters based on Oxley’s analytical cutting process model[65]. Recently, Ning and Liang developed an inverse identification method to obtain J-C constants based on chip formation model and an iterative gradient searching algorithm [62].

The mechanical properties of Ti-6Al-4V are listed in Table 6.2, which are used in solving the governing equations from the developed model. The parameters of Johnson-Cook constitutive property for Ti-6Al-4V are obtained from [45], which uses SHPB tests to calibrate the parameters, with the material constants ($\tau_A$, $\tau_B$, $C$, $m$, $n$) defined in J-C equation (6.13) listed in Table 6.3.
Table 6.3: Ti-6Al-4V Johnson-Cook parameters

<table>
<thead>
<tr>
<th>$\tau_A$ (MPa)</th>
<th>$\tau_B$ (MPa)</th>
<th>C</th>
<th>n</th>
<th>m</th>
<th>$\dot{\gamma}_{ref}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>724.7</td>
<td>683.1</td>
<td>0.035</td>
<td>0.47</td>
<td>1.0</td>
<td>$10^{-5}$</td>
</tr>
</tbody>
</table>

In the simulations, the rake angle ($\alpha$) of the cutting tool is $5^\circ$. The thickness of the primary shear zone is assumed as 1/10 of the uncut chip thickness, and the contact length is assumed as twice of the uncut chip thickness\[4\]. The friction coefficient between the cutting tool and and the machined chip is assumed to be 0.3 \[12\]. The shear angle ($\phi$) was calibrated through orthogonal cutting experiments by measuring the average chip thickness under different cutting conditions listed in Table 6.1 and the shear angle was calculated by assuming constant material flow using (6.19), and was used to determine the average strain in J-C constitutive equation (6.13)\[4\].

$$\phi = \arctan \frac{r \cos \alpha}{1 - r \sin \alpha}$$

(6.19)

where $r$ is the compression ratio of the chip defined as $r = d/d_c$, $d_c$ is the average thickness of the machined chip. Based on the measured chip compression ratio, the average shear angle $\phi$ is calculated as $40^\circ$.

The proposed physical model predicts the shear stress and temperature variations in the primary shear zone given the cutting speed and the uncut chip thickness values. Figure 6.6 shows the comparison of the simulated shear stress and temperature without and with ultrasonic vibration assistance for two seconds to show the steady state response from 1.9 s. When the cutting speed is 40 m/min and the uncut chip thickness reaches 0.15 mm under no vibration assistance condition, the shear strain rate increases and the heat accumulation is localized in the primary shear zone. The adiabatic shear bands are formed to generate the segmented chip, as shown in the SEM image in Figure 6.2 a). In the simulations, the cutting motion occurs in the whole time period, therefore, the governing equations (6.10), (6.12) combining with the constitutive relationship (6.14) are solved with constant cutting speed, and the results show periodic variations of shear stress and temperature.
in Figure 6.6 a), which corresponds to the shear band formation. The maximum temperature within the primary shear zone reaches above 800 °C. The oscillation frequency of the shear stress and temperature is about 27 kHz from the simulation results.

In the simulations of ultrasonic vibration-assisted cutting, the assisted vibration parameters are based on the experimental conditions corresponding to Figure 6.2 d)-f) where the frequency is 15.5 kHz, and the amplitude is 10 µm. Intermittent tool-chip contact and speed variations are considered in the simulations, with the results of shear stress and temperature variations shown in Figure 6.6 b). Oscillations of shear stress and temperature still exist, however, the oscillation frequency is determined by the excitation frequency of the ultrasonic vibration, which is 15.5 kHz. Furthermore, the shear stress drops to zero periodically due to the tool-chip separation, and the maximum of temperature in the primary shear zone is about 300 - 350 °C, which is much lower than the temperature in cutting without vibration assistance. In vibration-assisted cutting, when the tool is separated from the workpiece, there is no heat generation from the plastic deformation of the material, and the heat is transferred out of primary shear zone due to conduction. Furthermore, when the tool and the workpiece start to be in contact, the shear stress increases from zero, and the tool-workpiece contact period is smaller compared to continuous contact without vibration assistance. Thus, the temperature is lower compared to conventional cutting, which shows the benefit of vibration assistance. This proves the SEM results in Figure 6.2b) that, the application of ultrasonic vibration assistance suppresses the generation of adiabatic shear zones between the chip segments.

For more quantitative verifications of the proposed chip flow model, the pitch lengths of the chip segmentations between the predictions and the experimental measurements are compared. From the simulations, the relative displacement of the chip to the cutting tool is derived by integrating the speed difference between the chip and the cutting tool defined in (6.7). The relative displacement ($\Delta u$) represents the chip thickness in the primary shear zone in the cutting process, and it is calculated by:

$$\Delta u = \int_0^t \frac{V_c \cos \alpha}{\cos(\phi - \alpha)} dt - h \int_0^t \dot{\gamma}_p dt$$  \hspace{1cm} (6.20)
Therefore, the chip geometry can be predicted in spatial domain. The pitch length of the chip segmentations, which is the distance between the two adjacent peak values of $\Delta u$, is then calculated from the material speed along the tool rake face direction and the oscillation frequencies of the shear stress. In orthogonal cutting experiments, the chips from every cutting condition were collected, and the pitches were measured using an optical microscope. The measured and predicted pitch lengths of the machined chips under different cutting conditions are presented and compared in Figure [6.7] It is found that the simulated results match with the experimental values within 20% errors for both without and with vibration assistance situations. The error might be from the difference of the material properties used in the simulations and the experiments, and the assumptions of primary shear zone thickness used in the model. Without ultrasonic vibration assistance,
Figure 6.7: Comparison of pitch lengths between simulations and experiments with different cutting speeds and uncut chip thicknesses ($d$).

the pitch length increases as the uncut chip thickness increases for different cutting speeds (50 m/min and 40 m/min). In contrast, when applying ultrasonic vibration, the pitch lengths with different uncut chip thicknesses ($d = 0.2$ mm and $d = 0.15$ mm) at the same cutting speed (50 m/min or 40 m/min) are close to each other, as shown in Figure 6.7, where the pitch length is around 130 $\mu$m at $V_c = 40$ m/min, and around 140 $\mu$m at $V_c = 50$ m/min. Thus, it is concluded that the segmentation pitch is only determined by the cutting speed and vibration frequency when ultrasonic vibration assistance is applied, and it does not depend on the uncut chip thickness. This verifies the simulation results in Figure 6.6, that the frequency of stress and temperature variations is determined by the excitation frequency of the applied ultrasonic vibration assistance.

In addition, the cutting forces from the simulations and the experiments are compared. The cutting forces along tangential and feed directions were measured using a small-size dynamometer (Kistler 9256C1) with 2 kHz bandwidth. The dynamometer is not able to measure the instantaneous cutting forces due to the bandwidth limit. Therefore, only average cutting forces are evaluated in experiments. The simulated tangential cutting force ($F_t$) is determined by the shear stress
in the primary shear zone as follows:

\[
F_s = \tau b \frac{d}{\sin \phi} 
\]

\[
F_t = \frac{F_s \cos (\beta_a - \alpha)}{\cos (\phi + \beta_a - \alpha)}
\]

where \(\phi\) is the shear angle, and \(\beta_a\) is the friction angle derived from the friction coefficient on rake face \((\beta_a = \arctan \mu)\). From the experiments, the average tangential cutting forces \((F_t)\) in the orthogonal cutting process were measured by the dynamometer along X-direction shown in Figure 6.1.

The edge radius of the turning insert is measured to be 10 \(\mu m\). Since it is smaller than 10\% of the uncut chip thickness value, the effect of ploughing force due to the tool round edge is neglected. Figure 6.8 shows the comparison of the average cutting forces with different uncut chip thicknesses and cutting speeds given in Table 6.1. The results show that the ultrasonic vibration reduces the average cutting force in the primary zone, which is due to the intermittent tool-workpiece contact. About 15\% difference exists between the predicted and the measured values, which may be due to the inaccurate material property parameters and the assumptions of shear zone thickness in the chip flow model. When the ultrasonic vibration is applied, the simulated cutting forces are smaller than the measured values for different uncut chip thicknesses \((d = 0.15 \text{ mm} \text{ and } d = 0.2 \text{ mm})\) and cutting speeds \((V_c = 40 \text{ m/min} \text{ and } V_c = 50 \text{ m/min})\). The reason for larger measurement values is that the chip flow model neglects the friction between the bottom of the tool edge and the workpiece along the tangential direction during cutting motion and tool retraction, while the friction also contributes to the average cutting forces along the tangential direction in the experiments, which leads to larger values than the simulated results.

### 6.5 Summary

The effect of ultrasonic vibration assistance on the chip formation mechanism in cutting Ti-6Al-4V is investigated in this chapter. Examinations of the machined chips using SEM show that the adiabatic shear bands disappear in the segmented
chips when ultrasonic vibration assistance is applied. An analytical plastic flow model in the primary shear zone is developed to simulate the shear stress and the temperature variations associated with the shear banding and chip segmentation. Governing equations on the thermal-mechanical behavior of the material are developed in cutting, tool retraction with chip elastic recovery, and tool-chip separation periods in vibration-assisted cutting. It is shown that the temperature in the primary zone shear is much lower compared to that in conventional cutting when ultrasonic vibration assistance is applied in orthogonal cutting of Ti-6Al-4V, due to periodic tool-workpiece separation which allows heat dissipation and decrease of shear stress. The predicted pitch lengths of the segmented chips and the average cutting forces are verified by orthogonal turning experiments with different uncut chip thicknesses and cutting speeds.

Figure 6.8: Comparison of tangential cutting forces between simulations and experiments with different cutting speeds and uncut chip thicknesses \((d)\). a) \(d = 0.15\) mm; b) \(d = 0.2\) mm.
Chapter 7

Conclusion

7.1 Contributions

In this thesis, a novel 3DOF ultrasonic vibration tool holder was developed, in which a piezoelectric resonance actuator is used to excite vibrations along all three axes. While radial vibrations are delivered by resonating the third bending mode, axial vibrations are created by exciting the first axial mode of the actuator assembly at approximately 17 kHz with an amplitude range of 20-25 µm. A power amplifier with conditioning circuits was also developed to drive and monitor the vibration cutting tool. The proposed design expands ultrasonic vibration assistance to various machining operations (milling, drilling, and turning) in a single tool holder.

In addition to the design of the vibration tool holder, this thesis presents a sensorless closed loop control system to generate the desired vibrations during cutting. A linearized lumped parameter model of the piezoelectric actuator is used for each axis to track the resonance and to estimate the vibration amplitude during cutting. A Kalman filter based observer is used to estimate the vibration amplitude and phase from the measured driving currents and supply voltages without the use of vibration sensors. The proposed closed loop control system has been demonstrated to maintain the vibration amplitude at the desired level while adjusting the excitation frequency to match the varying resonance frequency of the actuator during machining. The system has been demonstrated in turning and drilling with unidirectional vibrations, as well as in two-axis milling with elliptical vibration loci.
The ultrasonic vibrations applied with the sensorless controller reduced the cutting forces by approximately 30% compared with conventional cutting in sample drilling and milling experiments. The experimental results also showed that the cutting forces of the proposed control system were 10% lower than those for the uncontrolled ultrasonic vibration-assisted cutting.

The 3DOF ultrasonic vibration tool holder can deliver spindle-synchronized elliptical vibrations in the tangential and radial directions to assist in milling operations. The chatter stability must be accurately predicted to select the appropriate spindle speed and depth of cut for the milling process. This thesis proposed a dynamic model of elliptical vibration-assisted milling, including an analysis of the dynamic chip thickness and the process damping effect. The stability was derived from the proposed dynamics model using the semi-discrete method, and milling experiments were conducted to verify the predictions. It was concluded that elliptical vibrations can suppress chatter under certain cutting conditions.

As experimentally demonstrated, the ultrasonic vibration assistance alters the chip formation mechanism in the cutting of Ti-6Al-4V. Adiabatic shear bands are suppressed due to the high-frequency tool-workpiece separation. An analytical model of plastic flow in the primary shear zone was proposed in this thesis by combining the governing equations of shear stress and heat transfer. Simulations based on the proposed model showed a significant reduction in temperature in the primary zone when ultrasonic vibration is applied along the tangential cutting direction. The model was also verified in orthogonal cutting tests with force measurements.

7.2 Future Research Directions

This thesis proposed a novel 3DOF ultrasonic vibration tool holder with a sensorless control system. Furthermore, the dynamics of vibration-assisted milling and the effects on chip formation were investigated. Future work is suggested for the following aspects of this system:

- The power transmission of the 3DOF ultrasonic vibration can be improved to replace the slip rings. A rotary transformer can be designed to wirelessly transmit power to the tool holder (rotor), thus the spindle can reach much
higher speed (i.e. 20,000 rev/min).

- The power amplifier for the piezoelectric actuator can be improved. A switch-type (PWM) power stage can be designed to replace the linear amplifiers in the current system for higher efficiency, improving the compatibility of the vibration tool holder for industrial applications. In addition, higher supply voltages can be applied to broaden the adjustable range of vibration amplitudes.

- In the presented sensorless control system, the vibration amplitudes are estimated using a linearized electromechanical model of the piezo actuator, which ignores the nonlinear effects of the piezo material. Therefore, a nonlinear electromechanical model can be implemented to improve the tracking accuracy of the vibration loci.

- The proposed digital sensorless control system is adaptive to different ultrasonic vibration cutting tool designs, but the adaptation is limited in a certain range of structural design. Therefore, it is valuable to investigate the controller performance from the aspect of the changing structural dynamics of the tool holder, which is important to industrial applications.

- The effects of ultrasonic vibration assistance on chip formation have been investigated in this thesis, but only for vibration along the tangential cutting direction. In the future, it will be useful to investigate the effects of elliptical vibrations occurring simultaneously along the tangential and radial directions simultaneously, on the chip formation mechanism.
Bibliography


