Towards additively manufactured tool holder with cavity to mitigate chatter

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A Thesis

entitled

Towards Additively Manufactured Tool Holder with Cavity to Mitigate Chatter

by

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Submitted to the Graduate Faculty as partial fulfillment of the requirements for the

Master of Science Degree in Mechanical Engineering

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An Abstract of

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In machining, chatter vibration exist in almost any machining operation and is the central obstacle that prevents attaining maximum productivity. Chatter leads to negative results such as degradation of quality, increased tool wear, decrease in tool life, waste of materials and time. Considering the negative effects of chatter, this problem has motivated many researchers to devise various methods to counteract the undesirable effects of chatter vibration. Among the various methods to suppress chatter, the particle damping mechanism is the focus of this thesis. The objective of this research is to work toward an innovatively designed tool holder which contains internal cavities filled with metal powder that has inherent vibration damping properties. The tool holder will be fabricated using an emerging manufacturing technique called the additive manufacturing. This work approaches this goal, both in experimental and analytical aspects for the new design of this tool holder and ultimately mitigate chatter.
# Table of Contents

Abstract ........................................................................................................................................................................ iii

Table of Contents ........................................................................................................................................................... v

List of Tables ................................................................................................................................................................. vi

List of Figures ................................................................................................................................................................. vii

List of Abbreviations ....................................................................................................................................................... xii

List of Symbols ................................................................................................................................................................. xiii

1 Introduction ................................................................................................................................................................. 1

1.1 Stability Lobe Diagram ........................................................................................................................................ 3

1.2 Strategies for chatter free machining .................................................................................................................. 8

  1.2.1 Out-of-process strategies ..................................................................................................................................... 9

  1.2.2 In-process strategies .......................................................................................................................................... 12

  1.2.3 Passive strategies .............................................................................................................................................. 16

  1.2.4 Active strategies .............................................................................................................................................. 20

2 Objective and Background ........................................................................................................................................... 23

2.1 Particle damping ..................................................................................................................................................... 27

2.2 Detection of chatter in machining ....................................................................................................................... 31

2.3 Additive Manufacturing ...................................................................................................................................... 41

3 Experimental Study .................................................................................................................................................... 49

4 Results ......................................................................................................................................................................... 64

5 Analytical Study ......................................................................................................................................................... 70

6 Conclusion .................................................................................................................................................................. 82

References ..................................................................................................................................................................... 84
List of Tables

1.1 Conditions for cutting states ................................................................. 14
3.1 The first 5 natural frequencies ................................................................. 62
4.1 Natural frequencies in X, Y and Z directions for the tool holder ................. 69
4.2 Maximum amplitudes in X, Y and Z directions for the tool holder ............... 69
List of Figures

1-1 Chatter publications per year [2] .................................................................................................................. 3
1-2 Mechanism of regeneration [1] ...................................................................................................................... 4
1-3 Effect of phase delay on chip thickness [2] ..................................................................................................... 4
1-4 SDOF orthogonal turning model [1] .................................................................................................................. 5
1-5 Typical SLD showing the stability lobes for various speeds and width of cut [1] .............................................. 7
1-6 Research lines focused on chatter vibration [2] .............................................................................................. 9
1-7 A comparison of stability charts with dynamic characteristics of the cutting tool and the workpiece. The solid line represents the stability of the flexible case in terms of relative motion between the cutting tool and workpiece; the dashed line represents the stability of SDOF model in which vibrations in the cutting tool are ignored [8] ........................................................................................................................................ 10
1-8 Stability lobe with and without wear [9] .......................................................................................................... 11
1-9 Amplitudes of RMS AE from experiments and model predictions for a fresh tool cut all at the stability boundary [10] ........................................................................................................................................ 13
1-10 Amplitudes of RMS AE from experiments and model predictions for a worn tool cut at the stability boundary [10] ......................................................................................................................................... 13
1-11 Illustration of typical example of the experimentally obtained dynamic cutting forces occurred at different cutting states; continuous chips, broken chips and chatter [11] ....................................................................................................................................... 15
1-12 Geometry of endmill and four-fingered mechanical damper [12] ............................................................... 17
1-13 Stable cutting depth vs. spindle speed [13] ..................................................................................................... 17
Configuration of cutting tool system with plate inserted [14] ............................................. 18
Sound amplitude with the regular cutter [16]................................................................. 19
Sound amplitude with the variable pitch cutter [16]..................................................... 19
Map of limit of depth for sinusoidal speed variation [17].............................................. 21
Experimental results with 15% variation at 3 Hz [17].................................................. 21
Turning cutting system without piezoactuation [18]...................................................... 22
Turning cutting system with proposed piezoactuator adaptive controller [18] ....... 22
CAD models of tool holders produced by SLM: (a) solid holder (b) holder with internal powder capsules [28] ............................................................................................................. 24
Influence of different tool holders on static vibration behavior during turning of Ti6Al2Sn4Zr6Mo [28] ................................................................................................................................. 25
Confocal laser scanning micrographs of Ti6Al2Sn4Zr6Mo workpiece surfaces processed by using the (a) solid tool holder (b) tool holder with powder capsules. Influence of different tool holders on tool wear at cutting insert after machining Ti6Al2Sn4Zr6Mo with the (c) solid tool holder (d) tool holder with powder capsules [28] ................................................................................................................................. 26
Packing arrangement of glass balls [19] ........................................................................ 28
Effect of ball size on packing ratio in close-packed structure [19]................................. 28
Difference in generation mechanism of damping due to ball size (a) d=1-10 mm (b) d=12-20 mm [19] ................................................................................................................................. 29
Schematic drawing of damping wave [20]................................................................... 30
Effect of packing ratio on vibration dissipation time [20]............................................. 30
Schematic of experimental setup for facing [22]......................................................... 32
2-10 Schematic of experimental setup for turning [22] ......................................................32
2-11 Waterfall plots of force in (a) X-direction (b) Y-direction and (c) Z-direction
displaying the onset of chatter at 2.8 mm width of cut in facing [22] ....................35
2-12 Waterfall plots of acceleration in (a) Y-direction and (b) Z-direction displaying
the onset of chatter at 2.8 mm width of cut in facing [22] .......................................36
2-13 Waterfall plots of amplitude of acoustic signal from microphone displaying the
onset of chatter at 2.8 mm width of cut in facing [22] .............................................37
2-14 Waterfall plots of force in (a) X-direction (b) Y-direction and (c) Z-direction
displaying the onset of chatter at 2.4 mm width of cut in turning [22] ....................39
2-15 Waterfall plots of acceleration in (a) Y-direction and (b) Z-direction displaying
the onset of chatter at 2.4 mm width of cut in turning [22] ....................................40
2-16 Economic comparison of different manufacturing techniques [25] ..............42
2-17 Phenix Systems PXM Selective Laser Melting Machine [23] .........................43
2-18 Sequence of operations of the SLM process conducted on a Phenix Systems PXM
[23] .........................................................................................................................44
2-19 Scheme of the AM processing principle: (a) CAD model, (b) sliced CAD model
prepared for AM (c) cyclic AM procedure- melting →platform lowering →
powder deposition (d) complex Nitinol structure produced by AM [25] ...........44
2-20 Visual representation of the Staircase Effect and the principle of support
structures [23] .......................................................................................................45
2-21 Basic scan strategy showing laser trajectories all in the same direction [23] .....46
2-22 Alternating x/y scan strategy [23] ........................................................................47
2-23 Alternating x/y scan strategy with 90 degree rotation per layer [23] .............47
3-1 Cantilever beam used in the experiment ..............................................................50
3-2 Geometrical model of the beam used in the experiment ..........................51
3-3 Channel setup ........................................................................................................52
3-4 Impact Scope ...........................................................................................................53
3-5 AC Calibration completed ....................................................................................54
3-6 Impact Setup – Trigger .......................................................................................55
3-7 Impact Setup – Bandwidth ....................................................................................56
3-8 Impact Setup – Windowing ...................................................................................57
3-9 Measure ................................................................................................................59
3-10 MAC of the cantilever beam ..............................................................................61
3-11 1D beam model with 6 nodes .........................................................................62
3-12 1D beam model with 10 nodes .......................................................................62
3-13 MAC of the 6 node beam model ....................................................................63
3-14 MAC of the 10 node beam model ....................................................................63
4-1 Tool holder .............................................................................................................65
4-2 Impact test and measurement in X-direction .................................................66
4-3 Impact test and measurement in Y-direction ..................................................67
4-4 Impact test and measurement in Z-direction ..................................................68
5-1 Model of a multi-unit particle damping system [21] .....................................71
5-2 Schematic of experimental apparatus [21] ......................................................71
5-3 Relation between the cavity radius $R$ and $\zeta (r = 0.003m)$ [21] ..........75
5-4 Influence of the cavity radius on the spring constant $(r = 0.003m)$ [21] ...76
5-5 Cavity arrangement [21] .....................................................................................77

x
5-6 Comparison between experimental and calculated results (4 cavities, $\lambda=0.098$)

[21] .................................................................78

5-7 Comparison between experimental and calculated results (5 cavities, $\lambda=0.098$)

[21] .................................................................78

5-8 Response of the system amplitude vs. the frequency (3 cavities, $R = 0.038\, m$)

[21] .................................................................79

5-9 Influence of the cavity radius on the damping efficiency [21] .........................80
# List of Abbreviations

<table>
<thead>
<tr>
<th>Abbreviation</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>AC</td>
<td>Accelerometer</td>
</tr>
<tr>
<td>AE</td>
<td>Acoustic Emission</td>
</tr>
<tr>
<td>AISI</td>
<td>American Iron and Steel Institute</td>
</tr>
<tr>
<td>AM</td>
<td>Additive Manufacturing</td>
</tr>
<tr>
<td>CAD</td>
<td>Computer Aided Design</td>
</tr>
<tr>
<td>CSSV</td>
<td>Continuous Spindle Speed Variation</td>
</tr>
<tr>
<td>FFT</td>
<td>Fast Fourier Transform</td>
</tr>
<tr>
<td>FRF</td>
<td>Frequency Response Function</td>
</tr>
<tr>
<td>g</td>
<td>Gravitational acceleration</td>
</tr>
<tr>
<td>hp</td>
<td>Horsepower</td>
</tr>
<tr>
<td>Hz</td>
<td>Hertz</td>
</tr>
<tr>
<td>LMS</td>
<td>Learning Management System</td>
</tr>
<tr>
<td>MAC</td>
<td>Modal Assurance Criterion</td>
</tr>
<tr>
<td>MMRR</td>
<td>Maximum Metal Removal Rate</td>
</tr>
<tr>
<td>mm</td>
<td>Millimeter</td>
</tr>
<tr>
<td>N</td>
<td>Newton</td>
</tr>
<tr>
<td>Nitinol</td>
<td>Nickel-Titanium</td>
</tr>
<tr>
<td>PLM</td>
<td>Product Lifecycle Management</td>
</tr>
<tr>
<td>rev</td>
<td>Revolution</td>
</tr>
<tr>
<td>RMS</td>
<td>Root Mean Square</td>
</tr>
<tr>
<td>rpm</td>
<td>Revolutions per minute</td>
</tr>
<tr>
<td>SDOF</td>
<td>Single-Degree-of-Freedom</td>
</tr>
<tr>
<td>SLD</td>
<td>Stability Lobe Diagram</td>
</tr>
<tr>
<td>SLM</td>
<td>Selective Laser Melting</td>
</tr>
<tr>
<td>V</td>
<td>Volt</td>
</tr>
<tr>
<td>W</td>
<td>Watt</td>
</tr>
<tr>
<td>2DOF</td>
<td>Two-Degrees-of-Freedom</td>
</tr>
<tr>
<td>3D</td>
<td>Three-Dimensional</td>
</tr>
<tr>
<td>μm</td>
<td>Micrometer</td>
</tr>
</tbody>
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### List of Symbols

- $\beta$ : Packing ratio
- $\Gamma(s)$ : Transfer function
- $\gamma$ : Spindle speed variation frequency
- $\Delta$ : Amplitude of the spindle speed variation
- $\delta_n$ : Normal displacement
- $\delta_n^\prime$ : Normal velocity
- $\delta_t$ : Tangential velocity
- $\zeta$ : Damping ratio
- $\theta$ : Phase delay/shift
- $\lambda$ : Mass ratio
- $\mu$ : Coefficient of friction between two particles or between a particle and the cavity wall
- $\nu_p$ : Poisson’s ratio of the particle
- $\nu_0$ : Poisson’s ratio of the cavity wall
- $\varphi_i$ : Angular velocity vector of the particle $i$
- $\psi$ : Phase angle
- $\Omega$ : Instantaneous spindle speed
- $\Omega_0$ : Average spindle speed
- $\omega$ : Frequency of chatter vibration
- $\omega_n$ : Natural frequency
- $a_0$ : Initial amplitude
- $b$ : Width of cut
- $b_{lim}$ : Stability limit
- $C_1$ : Threshold values for determining the cutting states
- $C_2$ : Threshold values for determining the cutting states
- $C_3$ : Threshold values for determining the cutting states
- $c$ : Damping coefficient
- $c_n$ : Damping coefficient between two particles or between a particle and the cavity wall
- $d$ : Diameter of ball
- $E_p$ : Modulus of elasticity of the particle
- $E_0$ : Modulus of elasticity of the cavity wall
- $F$ : Component of the total force exerted by all the particles on the primary system in the direction of the excitation
- $F_i$ : Sum of the contact forces acting on the particle $i$
- $F_f(t)$ : Cutting force in the feed direction
- $f_u$ : Contact force component in each cavity
- $f_n$ : Normal component of the contact force
- $f_t$ : Tangential component of the contact force
\( G(\omega) \) 
Real part of the transfer function

\( g \) 
Acceleration vector due to gravity

\( H(\omega) \) 
Imaginary part of the transfer function

\( I \) 
Moment of inertia of the particle

\( I_1 \) 
Ratio of the average variance of the main force to feed force

\( I_2 \) 
Ratio of the average variance of the main force to thrust force

\( I_3 \) 
Ratio of the average variance of the feed force to thrust force

\( K_f \) 
Cutting coefficient in the feed direction

\( k \) 
Spring constant

\( k_n \) 
Spring constant between two particles or between a particle and the cavity wall

\( MAC \) 
Modal Assurance Criterion matrix

\( M \) 
Mass of the primary system

\( m \) 
Mass

\( N \) 
Spindle speed

\( N_u \) 
Number of cavities

\( n_{ij} \) 
Unit vector from particle \( i \) to particle \( j \)

\( p_i \) 
Position vector of the particle \( i \)

\( p_j \) 
Position vector of the particle \( j \)

\( R \) 
Cavity radius

\( R_{opt} \) 
Optimum cavity radius

\( r \) 
Radius of the particle

\( T \) 
Vibration dissipation time

\( T \) 
Spindle period

\( T_i \) 
Sum of the torque caused by the contact forces acting on the particle \( i \)

\( t \) 
Time

\( u \) 
Harmonic motion of the support point

\( V \) 
Cutting velocity

\( x_i \) 
\( x \)-coordinate of the particle \( i \)

\( y_i \) 
\( y \)-coordinate of the particle \( i \)

\( x_c \) 
\( x \)-coordinate of the center of the cylindrical cavity

\( y_c \) 
\( y \)-coordinate of the center of the cylindrical cavity

\( x(t) \) 
Wavy surface created during the current revolution

\( x(t - T) \) 
Wavy surface created during the previous revolution

\( x^{(1)} \) 
First family of mode shapes

\( x^{(2)} \) 
Second family of mode shapes

\( x_i^{(1)} \) 
Mode \( i \) of the first family of mode shapes

\( x_j^{(2)} \) 
Mode \( j \) of the second family of mode shapes

\( X \) 
Average variance of the cutting force in main direction

\( Y \) 
Average variance of the cutting force in feed direction

\( Z \) 
Average variance of the cutting force in thrust direction
Chapter 1

Introduction

Machining techniques have advanced greatly recently due to development of science and the demand for competitive markets. Understanding the dynamics of machining is important in order to improve the machining process and especially to facilitate online monitoring and automation, yet it is still complex and difficult. The chatter vibration is highly complex and difficult to understand due to various components that constitute the system: the cutting tool, tool holder and the workpiece. The chatter vibration phenomenon has always been the major hindrance in achieving excellent productivity and quality. It is imperative to mitigate chatter vibration accompanied in the machining process [1] [2].

In machining, mechanical vibration during machining can be categorized into three types of vibrations: free, forced and self-excited vibration. Free vibrations are the results of shocks. Forced vibrations are the result of rotating unbalance in parts such as bearings and spindles. Free and forced vibrations can be identified, reduced and eliminated. Self-excited vibration is categorized into primary chatter and secondary chatter [3].
Primary chatter is a result of the friction and thermo-mechanical interference between the tool and the workpiece. Secondary chatter is an unstable vibration that amplifies due to regeneration of waviness on the workpiece surface [4].

When the word ‘chatter’ is mentioned in most of the papers, it refers to the secondary chatter due to wave regeneration unless stated otherwise. This self-excited chatter is complex and not completely understood and above all, it is the most destructive and undesirable. Chatter has been investigated since the beginning of 20th century. In 1907, Taylor identified chatter as a limitation and stated that “chatter is the most obscure and delicate of all problems facing the machinist” [5]. Many researchers have endeavored to understand and alleviate this regenerative chatter.

Chatter is recognized by the violent noise and the chatter cut marks on the surface and the appearance of chips. Chatter vibration is unstable and results in large displacements between the tool and the workpiece. Chatter is very detrimental and has several adverse effects. Chatter leaves cut marks on the surface, resulting in poor surface finish and accuracy. Chatter yields excessive noise, tool wear and eventual breakage of tool. It hampers the machining process from attaining maximum metal removal rate (MMRR). Chatter results in tremendous excess cost in production and recycling and waste of material. Chatter results in the degradation of the product quality and reduces the lifespan of the tool. These effects definitely show that chatter is detrimental and it is imperative to research solutions to allay it. Figure 1-1 show the number of publications per year from 1966 to 2009 and increasing interest in chatter.
1.1 Stability Lobe Diagram

Figure 1-1: Chatter publications per year [2].

The mechanism of wave regeneration which is the underlying basis behind chatter is illustrated in figure 1-2 which shows the contact of tool and workpiece during machining. The tool is modelled as a mass-spring-damper vibration system with mass, stiffness and damping coefficient $m, k, c$ respectively and $V$ is the cutting velocity. During machining as vibration exists, wavy surface is left on the workpiece surface with each revolution of the spindle. $x(t)$ is the new wavy surface created during the current revolution and $x(t - T)$ is the wavy surface created during the previous revolution. In the current wave and the previous wave, a thickness difference due to vibration exists which is referred to as phase delay/shift $\theta$. This is also equal to the chip thickness by vibration. $\theta$ is the key
determinant of the occurrence of chatter. Figure 1-3 shows the effect of phase shift $\theta$ on the chip thickness. If the two waves are in phase ($\theta = 0$), the wavy surface won’t propagate and the machining process will be stable. But when the wave are out of phase ($\theta \neq 0$), the wavy surface will grow, vibration will be amplified and consequently cause unstable chatter vibration.

![Diagram of mechanism of regeneration](image1.png)

Figure 1-2: Mechanism of regeneration [1].

![Effect of phase delay on chip thickness](image2.png)

Figure 1-3: Effect of phase delay on chip thickness [2].

The SLD (Stability Lobe Diagram) is a diagram that predicts the chatter stability conditions. In order to explain the concept of SLD, it will be derived from the mathematical model in figure 1-4. The model considered here by Meritt is a simple Single-Degree-of-Freedom (SDOF) mass-spring-damper system model of turning with flexible tool holder.
and rigid workpiece [6]. Since it is a SDOF model, the force acting on the tool and the vibration is in only one direction which is the feed direction.

![SDOF orthogonal turning model](image)

**Figure 1-4: SDOF orthogonal turning model [1].**

Based on this modeling, the equation of motion can be expressed as

\[ m\ddot{x} + c\dot{x} + kx = F_f(t) \]  \hspace{1cm} (1.1)

\( F_f(t) \) is the cutting force in the feed direction.

\[ F_f(t) = K_f b [x(t - T) - x(t)] \]  \hspace{1cm} (1.2)

\( K_f \) is the cutting coefficient in the feed direction, \( b \) is the width of cut, \( T \) is the time delay between the current time and the previous time, \( x(t - T) - x(t) \) is the chip thickness due to vibration which is also equal to phase delay \( \theta \).

Taking the Laplace transform and using the relations listed in Eq (1.2) to (1.5) including the natural frequency \( \omega_n \) and the damping ratio \( \zeta \), Eq (1.1) can be re-expressed in Eq (1.7) as

\[ \omega_n^2 = \frac{k}{m} \]  \hspace{1cm} (1.3)

\[ c = 2\zeta m\omega_n \]  \hspace{1cm} (1.4)

\[ \varphi = \frac{K_f b}{k} \]  \hspace{1cm} (1.5)

\[ s^2 + 2\zeta \omega_n s + \omega_n^2 = \varphi \omega_n^2 (e^{-sT} - 1) \]  \hspace{1cm} (1.6)
The transfer function \( \Gamma(s) \) from the system is

\[
\Gamma(s) = \frac{1}{1 + (1 - e^{-sT})K_f b F(s)}
\]  

\( F(s) = ms^2 + cs + k \)  

(1.7)

Substitute \( s = j\omega \) where \( \omega \) is the frequency of chatter vibration. \( G(\omega) \) and \( H(\omega) \) are then the real part and the imaginary part of the transfer function \( \Gamma(s = j\omega) \), respectively. The phase angle \( \psi \), phase shift \( \theta \), spindle period \( T \) and spindle speed \( N \) can be expressed as

\[
\psi = \tan^{-1}\left(\frac{H(\omega)}{G(\omega)}\right)
\]  

(1.9)

\[
T = \frac{1}{\omega}(2n\pi + \theta)
\]  

(1.10)

\[
\theta = 3\pi + 2\psi
\]  

(1.11)

\[
N = \frac{60}{T}
\]  

(1.12)

The characteristic equation from the transfer function \( \Gamma(s) \) which determines the chatter stability condition is

\[
1 + (1 - e^{-j\omega T})K_f b F(j\omega) = 0
\]  

(1.13)

Depending on the roots of the characteristic equation in Eq (1.13), the system will be stable or unstable. The transition from stable to unstable happens when the real part of the root is zero. The critical parameter that decides whether the state is stable or unstable is the width of cut \( b \). The limiting width of cut \( b_{lim} \), also called the stability limit where the real part of the root becomes zero is where the transition from stable state to unstable state occurs. The stability limit \( b_{lim} \) which satisfies this condition in the characteristic equation can be found as [7]
The stability equation for positive limiting width of cut leads to the condition that the real part $G(\omega)$ of the transfer function $\Gamma(s)$ is negative ($G(\omega) < 0$). The process is stable until the width of cut $b$ exceeds the stability limit $b_{lim}$. When $b > b_{lim}$, the machining process becomes unstable and leads chatter. From equations (1.9) to (1.14), the SLD can be derived and drawn to show the relationship between the limiting width of cut $b_{lim}$ and the spindle speed $N$. By searching for chatter frequency values $\omega$ where $G(\omega) < 0$, the SLD can be constructed. Figure 1-5 displays a typical SLD where the limiting width of cut $b_{lim}$ is plotted against the spindle speed $N$. In figure 1-5, multiple lobes appear in the curve hence the name stability lobe diagram. This curve serves as a border that divides the stable chatter-free region and the unstable chatter region. Any combination of spindle speed and width of cut below the lobe curve is stable state and above the lobe curve is unstable state where chatter occurs. SLD helps the user identify proper spindle speed and width of cut value where stable operation without chatter is carried out.

\[
b_{lim} = -\frac{1}{2K_f G(\omega)}
\]  

(1.14)

Figure 1-5: Typical SLD showing the stability lobes for various speeds and width of cut [1].
1.2 Strategies for chatter free machining

Researchers have discussed various solutions and approaches regarding the chatter problem. They can be classified into two groups. The first group aims to select machining parameters in the stable region of SLD. The second group attempts to forestall chatter by modifying the behavior of the machining system itself.

The first group of methods can be categorized again into out-of-process and in-process strategies. The out-of-process strategies aim to analytically identify and predict the SLD so that it can ensure manufacturing in the stable zone and prevent chatter. The in-process strategies try to detect chatter and regulate the parameters during the machining process to stop chatter. In out-of-process strategies, the mathematical modeling is crucial for calculating the SLD. In in-process strategies, it is necessary to identify and detect chatter as soon as it occurs with sensors so that it can be readily controlled. The second group can be categorized again into passive strategies and active strategies. Passive methods focus on modifying or attaching certain passive elements in order to improve the performance of the system. Active methods use elements that could automatically calibrate itself as soon as chatter occurs in order to actively modify the behavior of the system.

In total, the methods dealing with chatter can be categorized into four types: out-of-process, in-process, passive and active as illustrated in figure 1-6. Each types of method will be described in more detail in the subsequent sections.
1.2.1 Out-of-process strategies

Out-of-process strategies focus on preventing chatter not by modifying the behavior of the system, but by identifying the SLD from the system model and selecting parameters of the stable zone. The dynamic model of the machining system enables prediction of the SLD. During many years of research, numerous analytical models for chatter have been proposed. The aforementioned analytic model by Meritt [6] where SLD was derived, is one example of the analytical models.

The model proposed by Vela-Martinez [8] is a two-degrees-of-freedom (2DOF) model and considers the flexibility of tool and workpiece. Usually, the flexibility between the tool and the workpiece is ignored. But this effect is included to seek a better dynamic
Analytic comparison is made between this 2DOF model which includes the tool-workpiece flexibility and a SDOF model which ignores this flexibility aspect. The comparison of SLD of these two models is displayed in figure 1-7. At low spindle speed, the difference between the two SLDs are small. But at high spindle speed, the difference is more prominent and the 2DOF model with the flexibility predicts a greater area of stable zone compared to model without the flexibility. Experimental validation is still the remaining work for this research. Models that take the tool-workpiece flexibility into account in general, are not as common.

Figure 1-7: A comparison of stability charts with dynamic characteristics of the cutting tool and the workpiece. The solid line represents the stability of the flexible case in terms of relative motion between the cutting tool and workpiece; the dashed line represents the stability of SDOF model in which vibrations in the cutting tool are ignored [8].

Clancy and Shin [9] proposed a three-dimensional chatter model that takes the tool wear effect into account. As wear develops on the tool flank, the stability limit begins to
increase compared to when there is no wear. The greater the flank wear area, the greater the increase in the stability limit. At low speed range, there is a greater increase in the stability limit due to wear while at high speed range this escalation effect is negligible. This increase effect of SLD due to the presence of wear is demonstrated in figure 1-8.

![Graph showing stability lobe with and without wear](image)

Figure 1-8: Stability lobe with and without wear [9].

Out-of-process strategies enable analytic prediction of the chatter stability of the machining model. However, it is worth noting that it is challenging to create a more realistic model which encompasses all the geometrical, dynamic and nonlinear characteristics.
1.2.2 In-process strategies

In out-of-process strategies, an accurate calculation of SLD enables preclusion of chatter and selection of parameters in the stable area. However, an accurate calculation of SLD requires exact knowledge of the analytic model which is not easy. In contrast to out-of-process strategies that predict chatter, in-process strategies try to monitor and detect chatter immediately when it occurs. Manufacturing industries in general prefer unmanned machining for cost effective operation. It is therefore essential to automatically detect chatter.

Detection of chatter is possible through various sensors. Acquisition of force and vibration signals are the most common methods that can detect chatter. The acquisition of force and vibration signals will be used as an experimental approach in this work and will be explained in more detail, later in Section 2.2. Force and vibration measurements use sensors such as dynamometer and accelerometer and they are expensive and sometimes difficult to install, but it is still the commonly used technique.

Another possible signal that can be used to detect chatter is the acoustic signal. Chiou [10] used acoustic signal to detect chatter and also took tool wear into account. Figures 1-9 & 1-10 compare chatter amplitude of RMS (Root Mean Square) AE (Acoustic Emission) signal for cases without and with tool wear. The chatter amplitude of RMS AE signal was larger for tool with wear than fresh tool. The drawback of using the acoustic
signals is evident in an industrial environment due to the disruption caused by other noises in the environment.

Figure 1-9: Amplitudes of RMS AE from experiments and model predictions for a fresh tool cut at the stability boundary [10].

Figure 1-10: Amplitudes of RMS AE from experiments and model predictions for a worn tool cut at the stability boundary [10].
Some researchers have analyzed the chip produced during the process for chatter detection. Tangjitsitcharoen [11] monitored dynamics force components and chip formation to sense chatter. Tangjitsitcharoen classified chip formation into three types: continuous chip formation, broken chip formation and chatter. Figure 1-11 shows a typical data of force components at the three different cutting states including chatter. \( X, Y, Z \) are defined as average variances of dynamic components of three forces in main, feed and thrust directions respectively. Also new parameters \( I_1, I_2, I_3 \) are defined and used to determine and classify the three chip formation states as shown below in Eq (1.16)–(1.18) and in Table 1.1. \( C_1, C_2, C_3 \) are threshold values for determining the states and in this particular case, 1.5, 2.5 and 1.3 respectively.

\[
I_1 = \frac{X}{Y} \quad (1.16)
\]

\[
I_2 = \frac{X}{Z} \quad (1.17)
\]

\[
I_3 = \frac{Y}{Z} \quad (1.18)
\]

Table 1.1: Conditions for cutting states

<table>
<thead>
<tr>
<th>Chip formation</th>
<th>Condition</th>
</tr>
</thead>
<tbody>
<tr>
<td>Continuous</td>
<td>( I_1 &lt; C_1, I_2 &lt; C_2, I_3 &lt; C_3 )</td>
</tr>
<tr>
<td>Broken</td>
<td>( I_1 &lt; C_1, I_2 &lt; C_2, I_3 &gt; C_3 )</td>
</tr>
<tr>
<td>Chatter</td>
<td>( I_1 &gt; C_1, I_2 &gt; C_2 )</td>
</tr>
</tbody>
</table>
Figure 1-11: Illustration of typical example of the experimentally obtained dynamic cutting forces occurred at different cutting states; continuous chips, broken chips and chatter [11].

The main disadvantage of in-process method is that although chatter is detected damage is already done.
1.2.3 Passive strategies

Passive strategies are another line of strategies that seek to mitigate chatter by modifying the behavior of the system. Usually the tool design is changed or an additional device is attached to damp the chatter vibration.

There are a wide variety of passive methods to suppress chatter. One example of a method is to employ dampers to reduce chatter. Ziegert et al. introduced a damper to reduce vibration in high speed milling [12] [13]. The damper is a multi-fingered hollow cylinder shape inserted directly inside the rotating spindle as shown in figure 1-12. The multiple fingers are made by cutting axial slits along the length. During vibration of the tool, relative sliding and friction between the adjacent surfaces of the tool and the damper causes dissipation of the vibration energy and subsequently minimizes chatter. Compared to normal solid tool and hollow tool without damper, it achieves more effective damping and greater depth of cut and saves great amount of time and cost which is demonstrated in figure 1-13.
Figure 1-12: Geometry of endmill and four-fingered mechanical damper [12].

Figure 1-13: Stable cutting depth vs. spindle speed [13].

Marui et al. devised a similar damping mechanism [14]. A rectangular hole is cut in the tool shank and then a plate slightly thicker than the height of the hole is inserted into
the hole as displayed in figure 1-14. The friction during vibration acting between the inner wall of the hole and the inserted plate surface induces damping.

Another passive method is to improve and modify the design of the tool cutting. Budak et al. designed milling cutters with non-constant pitch, also known as variable pitch cutter [15] [16]. They are different from regular cutters which have constant pitch angles. They can enhance the performance of milling and result in more productivity. For example, the regular cutter has 6 flutes of equal pitch angles. For the variable pitch cutter, the pitch angles are 55°, 57°, 59°, 61°, 63°, 65°. The amplitude of the sound spectrum for variable pitch cutter is lower compared to that of the regular pitch cutter. This comparison is visible in figures 1-15 & 1-16.
Passive strategies are easy to implement, low in cost and do not require any external energy source. Passive strategies can ensure stability for the controlled system. But, passive strategies require very accurate tuning in order to be effective.
1.2.4 Active strategies

The active strategies are different from passive strategies in that they continuously monitor the machining condition, sense the occurrence of chatter, actively and immediately carry out a chatter suppression mechanism. The recent advent of advanced computers, sensors and actuators have cultivated active strategies.

Zatarain et al. used Continuous Spindle Speed Variation (CSSV) which is another method to suppress chatter in milling [17]. Instead of using a constant speed, it uses a speed that varies as a function of time. The sinusoidal speed variation can be expressed as

$$\Omega = \Omega_0 + \Delta \cos \gamma t$$  \hspace{1cm} (1.19)

where $\Omega$ is the instantaneous speed, $\Omega_0$ is the average speed, $\Delta$ is the amplitude of the speed variation and $\gamma$ is the speed variation frequency. In this experiment, the spindle speed variation amplitude $\Delta$ is more significant parameter than the spindle speed variation frequency $\gamma$ as shown in figure 1-17. Figure 1-18 is the result of the experimental tests at speed of 525 rpm. At constant speed, there is a great deal of vibration while by application of CSSV ($\Delta=15\%$, $\gamma=3\text{Hz}$) the vibration is reduced and stabilized. CSSV is useful in reducing the vibration in milling process. However, it is a challenge to induce a high amount of torque to overcome the inertia of the rotation speed in CSSV.
The application of smart materials is another method for actively suppressing chatter. Smart materials have the ability to convert mechanical energy into other forms of energy and vice versa. Examples of smart materials include piezoelectric materials, magnetorheological fluids and shape memory alloys. Pan and Su [18] used piezoelectric actuator based adaptive control to counteract chatter. Piezoelectric materials can exert push or pull when it receives electric voltage and it possesses fast response and precise control characteristics. The simulation results with and without piezoactuation are shown in figures 1-19 & 1-20, respectively. It is clear that oscillation due to chatter is abated with piezoelectric adaptive control compared to the case when it is absent.
Active suppression methods do not require exact knowledge of the model and only a robust control technique. However, most of the active systems are limited to the low speed regions, require an external energy source and are more expensive and complex compared to passive systems.
Chapter 2

Objective and Background

The ultimate goal of this research is to enhance the productivity and quality in machining and reduce the time and cost in machining. The excessive vibration called chatter that occurs during the machining process is the central challenge in machining which hampers the performance. Chatter is a major challenge in machining and is detrimental with respect to productivity, workpiece quality, tool wear and tool life. Therefore, mitigating chatter in machining is crucial and especially for processing materials for aerospace and military applications. In this research, novel tool holders with internal cavities with innate damping properties will be designed.

Additive Manufacturing (AM) is a layer based manufacturing method also known as 3D printing. Fabricating structures with complex internal cavities is achievable by AM, the intended fabrication method for these tool holders.

The main objective of this research is to develop novel tool holders with intrinsic damping properties and improve machining of metals and alloys using AM. These novel tool holders that will be developed will contain internal cavities filled with metallic powders inside the structure. In the particle damping mechanism, the metal powders
inside the cavity during the vibration cause energy dissipation and thus reduce chatter. This particle damping will be studied in the following section.

Biermann et al. proposed the tool holders shown in figure 2-1 [28]. The first tool holder is an ordinary solid tool holder without the cavity and the second tool holder has internal capsules filled with metal powder. Both of these tool holders were fabricated using selective laser melting (SLM), an additive manufacturing method.

![CAD models of tool holders produced by SLM:](image)

**Figure 2-1: CAD models of tool holders produced by SLM:**

(a) solid holder (b) holder with internal powder capsules [28]

Biermann et al. carried out turning test in order to compare the performance of the solid tool holder and the tool holder with powder capsules. The workpiece material was the alloy, Ti6Al2Sn4Zr6Mo. Figure 2-2 displays the amplitudes of the two tool holders in the frequency domain. The tool holder with internal powder capsules exhibit damping in amplitude compared to the solid tool holder. This proves the damping effect of the tool holder with internal cavity to mitigate chatter vibration. Figure 2-3
demonstrates that the tool holder with powder capsules has improved performance regarding surface quality of the workpiece and tool wear. Compared to the solid tool holder, the tool holder with powder capsules result in a workpiece surface with significantly less roughness fluctuations which means smoother surface finish. Also the tool holder with powder capsules induce less damage on the tool compared to the solid tool holder.

These results definitely show the efficacy of tool holder with internal cavities to abate chatter and the research of this paper aims to extend this proposed design.

Figure 2-2: Influence of different tool holders on static vibration behavior during turning of Ti6Al2Sn4Zr6Mo [28]
Figure 2-3: Confocal laser scanning micrographs of Ti6Al2Sn4Zr6Mo workpiece surfaces processed by using the (a) solid tool holder (b) tool holder with powder capsules. Influence of different tool holders on tool wear at cutting insert after machining Ti6Al2Sn4Zr6Mo with the (c) solid tool holder (d) tool holder with powder capsules [28]
2.1 Particle damping

Wakasawa devised a damper called the impact damper which attenuates vibration via collision of balls inside the vibrating structure. He investigated damping capacity of a square pipe structure filled with glass balls for cases which was close packed [19] and partially packed [20]. In the close packed case, the balls are arranged in a way where the cavity is filled with as many balls as possible. The packing ratio $\beta$ is defined as

$$\beta = \frac{\text{Total volume of packed glass balls}}{\text{Total volume of the cavity structure}}$$

(1.2)

The packing ratio in this close packing is maximum and thus called the maximum packing ratio. The dimension of the square pipe is 25mm×25mm. The arrangement of the balls and maximum packing ratio value of close packed glass balls for various diameter values of glass balls is shown in figure 2-4 and figure 2-5, respectively. In the graph of figure 2-6, when the structure is filled with small balls of diameter less than 5mm, maximum packing ratio is approximately 60%. In other words, 60% of the cavity volume is occupied with glass balls and the other 40% of the cavity volume is unoccupied. The same research also studied the effect of ball size on the damping ratio. An optimal ball diameter exists which in this instance is 12mm. For small sized balls,
friction between the balls and the inner walls of the cavity is the main factor of damping. For large sized balls, there is more room for the balls to move during vibration and subsequently dissipate energy by collision, which is the main factor for damping. Figure 2-6 depicts the aforementioned damping mechanism.

Figure 2-4: Packing arrangement of glass balls [19].

Figure 2-5: Effect of ball size on packing ratio in close-packed structure [19].
Both close packing and partial packing are examined in Wakasawa’s research [20]. The effects of various packing ratios on damping ratio were investigated. In the study of partial packed case, damping performance was characterized by the vibration dissipation time $T$ instead of the damping ratio. In figure 2-7, the maximum amplitude immediately after excitation is defined as the initial amplitude $a_0$. Starting from initial amplitude, the vibration amplitude decreases over time until the residual amplitude becomes infinitesimally small. The time period it takes for this to occur is called the vibration dissipation time.
The vibration dissipation time for various values of packing ratio was examined for glass balls and steel balls as seen in figure 2-8. The diameter of the balls were 3mm. At first, the vibration dissipation time was reduced as the packing ratio increases but it rapidly increases as the packing ratio approaches about 60% the maximum packing ratio during close packing. The optimum packing ratio where vibration dissipation time is minimum which in turn means maximum damping capacity is achieved at the packing ratio slightly smaller than the maximum packing ratio around 50%.

Figure 2-8: Effect of packing ratio on vibration dissipation time [20].
M. Saeki [21] investigated multi-unit particle damping which is similar to the particle damper investigated by Wakasawa [19] [20]. The particles in the cavities induce energy dissipation and attenuate the intense vibration. The novel tool holders with internal cavities filled with metal powders that will be developed using Additive Manufacturing will be based on this multi-unit particle damper. The potential of this multi-unit particle damper will be fully explored in detail in Chapter 5.

2.2 Detection of chatter in machining

It is not possible to study chatter in machining unless the chatter effect is captured. This section will delve into the experimental method of detecting the occurrence of chatter vibration in facing and turning process.

The chatter phenomenon during machining can be detected by measuring three kinds of signals-force, acceleration and acoustic measurements. M. Siddhpura and R. Paurobally performed chatter experiments with a 7.5 hp Macson lathe [22]. Fresh coated carbide tool inserts (ISCAR IC8150) were used for facing of steel (AISI 1045) and aluminum (6061) work pieces. The dimensions of both steel and aluminum work pieces were 60 mm in diameter and 250 mm in length. The steel work piece is of which diameter 70 mm and length of 250 mm. The schematic of experimental setup for facing and turning are shown in figure 2-9 and figure 2-10 respectively.
In the experimental setup, the cutting forces were sensed with a Kistler 9257A three-component piezoelectric dynamometer, the vibration amplitudes were sensed with two PCB 333B piezoelectric accelerometers and the acoustic signal was sensed with a Realistic® electret microphone with an in-house built amplifier.

Figure 2-9: Schematic of experimental setup for facing [22].

Figure 2-10: Schematic of experimental setup for turning [22].
In facing and turning experiments, the feed rate was kept constant and the width of cuts were gradually increased. Tests were carried out for different combinations of depth of cuts and cutting speeds. Waterfall plots were plotted from the measured data. In the schematic of turning and facing experiments in figure 2-9 and figure 2-10, the XYZ coordinate directions are defined as the following: X-direction is the axial direction, Y-direction is the radial direction and Z-direction is the circumferential direction.

For facing, the feed rate was constant at 0.15 mm/rev and the cutting speeds were 470 rpm and 870 rpm. The width of cut for facing was increased from 0.8 mm to 2.8 mm. Then force, acceleration and acoustic signals were collected. Similar procedure was carried out for turning. For turning, the feed rate was constant at 0.3 mm/rev and the cutting speeds were 470 rpm and 870 rpm. The width of cut for turning was also increased from 0.8 mm to 2.8 mm and force, acceleration and acoustic signals were recorded.

During orthogonal turning, also known as facing, width of cut which increased beginning from 0.8 mm was stable and when it reached 2.8 mm, it became unstable and chatter occurred. When the chatter phenomenon was observed, either two things were observed. Either the tool was broken instantly or the tool insert worn very quickly after which the wear rate subsided. Figure 2-11 display the waterfall plots of the cutting forces signals in the frequency domain corresponding to different values of width of cut.
(a) Waterfall plot of Force-X

(b) Waterfall plot of Force-Y
Figure 2-11: Waterfall plots of force in (a) X-direction (b) Y-direction and (c) Z-direction displaying the onset of chatter at 2.8 mm width of cut in facing [22].

The waterfall plots change drastically at 2.8 mm width of cut. The rapid increases in force spectrum at frequency of 234 Hz at 2.8 mm width of cut is observed for all three force directions. The frequency of 234 Hz is the natural frequency. This feature in the waterfall plots indicate that at the 2.8 mm width of cut the vibration becomes unstable and chatter occurs. The width of cut where the spectrum drastically changes and increases is called the stability limit. When the width of cut passes the stability limit, the vibration changes from being stable to unstable and that is when chatter takes place.

Figure 2-12 display the waterfall plots of the acceleration signals in Y-direction and Z-direction.
Figure 2-12: Waterfall plots of acceleration in (a) Y-direction and (b) Z-direction displaying the onset of chatter at 2.8 mm width of cut in facing [22].
In figure 2-12, at 2.8 mm width of cut, when chatter took place, there were large increases in peaks at 234 Hz which is the natural frequency and 29 Hz which is the second harmonic of the speed 870 rpm (14.5 Hz). The acoustic signal of figure 2-13 shows large increase in amplitude at 2.8 mm width of cut when chatter occurred at frequencies of 234 Hz and 7793 Hz. The frequency of 234 Hz is the natural frequency and 7793 Hz is likely due to acoustic resonance in the room.

Figure 2-13: Waterfall plots of amplitude of acoustic signal from microphone displaying the onset of chatter at 2.8 mm width of cut in facing [22].
The force, acceleration and acoustic signals all exhibited similar behavior when the machining system produced chatter. When the width of cut reached the stability limit and lead to chatter, there was a drastic increase in the signals. Obviously, turning similarly exhibited similar characteristics in the acquired signals when chatter happened. Figure 2-14 and figure 2-15 illustrate the plots of force and acceleration signals. In turning, the only difference from facing were these values—the stability limit was 2.4 mm, the natural frequency was 117.2 Hz.

(a) Waterfall plot of Force-X
Figure 2-14: Waterfall plots of force in (a) X-direction (b) Y-direction and (c) Z-direction displaying the onset of chatter at 2.4 mm width of cut in turning [22].
In conclusion, this experimental example demonstrates that all of the force, acceleration and acoustic signals are able to successfully detect chatter.
2.3 Additive Manufacturing

Additive Manufacturing (AM) is a layer-based manufacturing method which eventually produces 3D solid parts. This technology emerged in the mid 1980’s [23] with the advent of rapid prototyping.

One of the major advantages of AM is to create parts with complex geometries such as hollow parts and curved holes which is not feasible by other traditional manufacturing methods. Another advantage of AM is a near-net-shape technique where the initial prototype is very close to the final desired product, thereby leading to substantial reduction of production time. In addition, as a layer-based manufacturing method, AM is oriented toward adding material rather than removing material to yield final product and thus minimizes waste of material [24]. Lastly, AM is an excellent manufacturing method for processing Nitinol, a shape memory alloy. The processing and machining of Nitinol is not easy since the thermo-mechanical properties of Nitinol are sensitive to heat generation and could easily be altered in an undesirable way [25]. AM could definitely pioneer a new door for fabricating nitinol. AM could overcome existing limitations of nitinol and create complex shapes beyond simple shapes.
Figure 2-16 illustrates the economic comparison of different manufacturing methods. According to figure 2-16, AM is economic in making highly intricate parts in small quantities.

![Economic comparison of different manufacturing techniques](image)

Figure 2-16: Economic comparison of different manufacturing techniques [25].

A concentrated source of heat energy is necessary to induce consolidation the powder particles. The advancement of fiber laser technology which can consolidate metallic powder bed into solid material facilitated the development of AM. This laser technology is called selective later melting (SLM). Phenix Systems is one of the manufacturers that was established in France in year 2000. The PXM machine made by Phenix Systems is a powder based SLM machine shown in figure 2-17.

Following is the general procedure for AM method. First, a 3D CAD model of the desired product is drawn and imported into the software embedded in the AM process and the CAD model is divided into multiple layers. Layer thickness on average is from
30-150\(\mu m\). During the AM process, a powder delivery piston feeds the powder material vertically and then the roller spreads the powder evenly and smoothly into a layer. A 300W yttrium fiber laser follows the trajectory according to the CAD model and melts and fuses the powder particles together. The particles upon contact with the laser are fused into a dense solid component. The fabrication piston accumulates the subsequent layer on the previously generated layer. Also, the procedure is performed in an atmosphere of inert gas to prevent oxidation and picking up of impurities. This cycle of continuous fabrication of layers is repeated until the entire product is complete as shown in figure 2-18 and figure 2-19.

Figure 2-17: Phenix Systems PXM Selective Laser Melting Machine [23].
Figure 2-18: Sequence of operations of the SLM process conducted on a Phenix Systems PXM [23].

Figure 2-19: Scheme of the AM processing principle: (a) CAD model, (b) sliced CAD model prepared for AM (c) cyclic AM procedure- melting →platform lowering → powder deposition (d) complex Nitinol structure produced by AM [25].

Since AM is a layer-based technique, this manufacturing method will exhibit the staircase effect which is depicted in figure 2-20. A thinner layer thickness of each layer constituting the product will yield finer resolution and better quality. However, the
downside is that thinner layers will result in longer production time. In AM, the main part is fabricated on the substrate with support structures serving as intermediate structure between the main part and the substrate. The support structure must be strong enough to withstand thermo-mechanical stresses and at the same time, weak enough for the final product to be readily removed and minimize post-processing.

![Diagram of support structures](image.png)

Figure 2-20: Visual representation of the Staircase Effect and the principle of support structures [23].

The process parameters for Phenix PXM are laser power, beam radius, hatch spacing, layer thickness and scan strategy. The laser power on the Phenix PXM can reach up to 300W in maximum. The beam radius is measured to be 40 μm [24]. Hatch spacing is the distance between the centerline of the two neighboring laser trajectories. If hatch spacing is less than the trajectory width, some amount of overlap will exist. Sometimes overlap is necessary to create a fully dense part. Thinner powder layers distribute the
laser energy more evenly in the thickness direction while thicker powder layers tend to spread more evenly and saves manufacturing time. Scan strategies is also a parameter that affects the manufacturing process. In a basic scanning strategy, the laser trajectories are all in the same parallel direction and goes from top-to-bottom.

![Basic scan strategy showing laser trajectories all in the same direction](image)

Figure 2-21: Basic scan strategy showing laser trajectories all in the same direction [23].

There are also other more complex scan strategies. For example, alternating x/y scan strategy in figure 2-22 can help minimize the effect of local heat concentrations. This is somewhat similar to the basic scanning strategy in figure 2-21, except that the laser path alternates between two opposite directions as it goes from top-to-bottom. Another strategy is to layer-rotation shown in figure 2-23. This strategy develops from the alternating x/y scanning strategy previously shown in figure 2-22. First, it similarly follows the alternating x/y scanning strategy. But after finishing the first layer, the scan strategy is rotated 90 degrees counterclockwise and now the laser path alternates between
two opposite directions as it goes from left-to-right. It continues to rotate the former scan strategy 90 degrees counterclockwise with each consecutive layers. This strategy reduces the local heat concentrations furthermore.

Figure 2-22: Alternating x/y scan strategy [23].

Figure 2-23: Alternating x/y scan strategy with 90 degree rotation per layer [23]
This AM technology is useful for manufacturing complex parts. It could save tremendous time and cost in applications such as aerospace, biomedical and military. AM is relatively a new but attractive manufacturing method that possesses great potentials.
Chapter 3

Experimental Study

The tool holder undergoes large vibration during the machining operation. Therefore, it is imperative to conduct vibration test on the tool holder and identify the vibration characteristics of the tool holder.

LMS Test.Lab is an engineering solution software for noise and vibration engineering made by Siemens PLM software. LMS Test.Lab is a comprehensive integrated software that is applicable to a wide variety of application such as rotating machinery, structural dynamics, acoustics and vibration control. It is a tool which makes testing more efficient and enhance acquisition of reliable data provided by multichannel data acquisition and built-in analysis and reporting functions.

In this research, LMS Test.Lab was used for vibration testing of the tool holders. As a precursor trial test to the tool holder test, the LMS Test.Lab was first use to conduct a vibration experiment, specifically an impact test to be exact on a beam that can be modelled as a cantilever beam. The picture of the cantilever beam used in the experiment is shown in figure 3-1. This section will explain how tests were conducted using the LMS Test. Lab.
Figure 3-1: Cantilever beam used in the experiment

The cantilever beam was tested in an impact test. Impact test is a vibration test, where the structure undergoing analysis is excited when it is hit with the impact hammer. The impact hammer has a force transducer embedded in it. The impact force of the hammer will vibrate the structure. Accelerometers are attached on the excited structure. The accelerometers measure and store the resulting vibration response from the impact of the hammer. Piezoelectricity is the underlying mechanism which store the force and vibration data with force transducer inside the hammer and the accelerometer, respectively. The collected vibration response signal is usually analyzed in the frequency domain by using FFT (Fast Fourier Transform). Through the impact test, measurements and experimental data can be acquired which are used to determine the vibration characteristics [26].
In the LMS Test. Lab software, the impact test is comprised of 3 steps: Geometry, Impact Testing and Modal Analysis. Each step will be explored so that better understanding of the experimental process can be grasped.

The first step of the impact test is the Geometry step. The Geometry step will only be briefly explored compared to the other steps. Building the 3D geometry of the actual desired structure is the first common foundational step for most of the CAD software. The 3D geometry drawing of the beam is shown in figure 3-2. However, the 3D geometry model is unique from geometric models of other CAD software in that nodes exist. The nodes in the model are not simply geometry of vertices. The nodes correspond to points which are either excited by the hammer or where the responses are measured.

![Figure 3-2: Geometrical model of the beam used in the experiment](image-url)
The second step of the impact test is the Impact Testing. This is where the main experimental data acquisition and measurements take place. The steps within this main step will be addressed as ‘sections’ and the steps within the sections will be addressed as ‘sub-steps’. The first section is channel setup. Each node is assigned to each channel. The channels where the impact hammer and the accelerometers are connected to must be checked so that they remain active during the vibration test. The minimum number of channels that must remain active are two since one is for the impact hammer which is called the reference and the other channel (or channels) is for the accelerometer. In this test, ‘input 1’ is the reference used for excitation by impact hammer and ‘input 2’ is used for the accelerometer measuring response according to figure 3-3.

<table>
<thead>
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<th>Event</th>
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<th>Channel</th>
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Figure 3-3: Channel setup
It is critical to make sure both the accelerometer and the impact hammer function properly. In the Impact Scope section, the user can briefly observe the force and vibration. In figure 3-4, the top-left screen for force displays a small peak for a short time which is approximately an impulse. In figure 3-4, the top-right screen and the middle screen display the vibration which resembles a typical damped vibration. In the AC Calibration section, accelerometer is calibrated so it can properly make measurements. During the calibration process, the accelerometer is vibrated at a certain known frequency and amplitude, so that the correct sensitivity for the accelerometer is identified. In this case, the accelerometer was vibrated at frequency of 159.2 Hz and amplitude of 1g RMS and for proper calibration. The following screen in figure 3-5 will be displayed if the calibration of the accelerometer was successful.

Figure 3-4: Impact Scope
After calibrating the accelerometer, the next task is to properly calibrate the impact hammer. This task along with other tasks will be achieved in the Impact Setup section. The Impact Setup section is comprised of 4 sub-steps: Trigger, Bandwidth, Windowing and Driving Points.

First, in the Trigger sub-step, the trigger for the hammer is determined. The trigger level is the value of voltage at which it accepts the excitation signal from the impact while filtering the noise signals. After hammering the structure several times, it suggests a new acceptable trigger value in figure 3-6 which is 3.16 V. In figure 3-6, Also, the functions displayed in figure 3-6 are sharp peaks for short time duration which represent the impulse function well.
Second, in the Bandwidth sub-step, it determines the bandwidth which is the frequency range of interest. The Bandwidth sub-step is shown in figure 3-7. Here, the graphs are shown in frequency domain instead of time domain. The left side shows the impact force and the right side shows the FRF (Frequency Response Function). The criterion for selecting the bandwidth is choosing the frequency where the initial maximum amplitude of the force decreases by 2 to 3 orders of magnitude. In figure 3-7, the selected bandwidth is 1600 Hz.
Third, in the Windowing sub-step, the process called windowing is carried out to avoid leakage. In the frequency domain, it should ideally show non-zero value only at a single frequency. However, in reality, error exists and it would exhibit non-zero values at other frequency values. This error phenomenon of collecting spectrum at non-existing frequencies is called leakage. The purpose of the windowing function is to prevent leakage and provide more accurate results. The LMS Test. Lab software will automatically set values for the windowing function to preclude leakage as shown in figure 3-8. Also, the
user can control the number of spectral lines which is the frequency resolution to reduce. A good combination of number of spectral lines and windowing should be used to minimize error.

![Figure 3-8: Impact Setup – Windowing](image)

Fourth, the Driving Points sub-step is the last sub-step. In the impact test, the nodes are either excited by the impact hammer or mounted with the accelerometer which measures the vibration response. In the impact test, there are two possible options: the roving hammer test or the roving accelerometer test. In the roving hammer test, only the
node for the accelerometer is fixed. In the roving accelerometer test, only the node for the hammer excitation is fixed. In either case, the fixed point is referred to as the driving point. The purpose of the Driving Points sub-step is to decide the best driving point. The driving point is chosen by hammering the same point where the accelerometer is placed on. It is not possible to hit the accelerometer with the hammer and get measurements. In reality, it is not possible to hammer the exact same position where the accelerometer is placed on, the user should hit position as close to the accelerometer as possible. The point which yields FRF with most resonant peaks will be the driving point. In this case, the roving hammer test option was used.

After finishing the Impact Setup section and all its four sub-steps, then the actual measurement begins in the Measure section presented next.

The Measure section is shown in figure 3-9. In the Measure section, each node is struck 5 times and the corresponding 5 measured responses are averaged for reliable results. Then the same procedure is repeated for all the other points until the measurement is complete.

Now we have acquired all the data to carry out the Modal Analysis, the third and final step. During the Modal Analysis step, vibration characteristics such as the natural frequency, damping ratio and the mode shape are acquired from the collected data to fully describe the system.
During the modal analysis process, it is crucial to know if the number of nodes in the system model appropriately represents the system. The MAC (Modal Assurance Criterion) is used to validate the measurements from the experimental model. The mathematical definition of MAC can be expressed as [27]

$$MA C\left( \mathbf{x}^{(1)}, \mathbf{x}^{(2)} \right) = \left( \frac{\mathbf{x}^{(1)T} \mathbf{x}^{(2)}}{\| \mathbf{x}^{(1)} \| \| \mathbf{x}^{(2)} \|} \right)^2 (3.1)$$

The MAC is a matrix which indicates the similarity between the two families of mode shapes $\mathbf{x}^{(1)}$ and $\mathbf{x}^{(2)}$. In this equation (3.1), the MAC is calculated between mode $i$
of the first family and mode \( j \) of the second family. The range of MAC values is from 0 to 1 which shows correlation. Higher value means more similarity between two mode shapes and value of 1 means a perfect match.

The LMS Test. Lab uses its own version of MAC function called the AutoMAC. The AutoMAC compares and correlates the mode shape calculated from the measured data with itself. Each natural frequency in the system has a unique corresponding mode shape. In the MAC matrix, the main diagonal elements are the correlation between two identical mode shapes so it will yield 1. The off-diagonal elements are the correlation between different mode shapes. Each mode shapes at different unique natural frequencies should not be same or very similar to each other. So the off-diagonal entries should be values close to 0. If the off-diagonal values possess high value, then there is an error. An ideal representation of MAC will have entries of 1 on the main diagonal and small values and close to 0 on entries elsewhere. In other words, an ideal MAC should resemble an identity matrix. Otherwise, more number of points need to be measured. The MAC indicates whether the proposed experimental model, specifically the number of nodes are sufficient to fully describe the model.

The MAC for this beam experiment is shown in figure 3-10. The MAC was not satisfactory for this beam experiment because it shows considerably high off-diagonal values. This implies that the initial geometrical model for the beam (figure 3-2) is not satisfactory. It is important to find the appropriate modeling for the given structure undergoing vibration. The realization was that the 3D modelling of the beam is redundant. First, only the bending deflection in a single direction is analyzed. Bending in other direction or torsion is not the motion of interest. Second, the effect of dimensions in
transverse directions compared to the longitudinal direction is negligible. The exact and elaborate 3D geometrical model would be necessary if other motions are considered or the geometry in other directions are not marginal.

Figure 3-10: MAC of the cantilever beam

The conclusion was that 3D model of the beam is unnecessary and simple one-dimensional model of the cantilever beam which deflects in a single direction is sufficient for modal analysis. Therefore, the new following geometric models for the cantilever beam were proposed in figures 3-11 & 3-12. The beam model in figure 3-11 is composed of 6 nodes and the beam model in figure 3-12 is composed of 10 nodes. Impact test was conducted on both new beam models. For both models, the first five natural frequencies (Table 3.1) and MAC were acquired (figure 3-13 & 3-14). Comparing the results, the
natural frequency values of the two models are quite similar. However, the off-diagonal elements of MAC for the 6 node model are lower compared to the 6 node model. The 10 node model yields better MAC data compared to the 6 node model. The 1D model shows definite improvement compared to the previous 3D model evidenced by MAC.

![Figure 3-11: 1D beam model with 6 nodes](image)

![Figure 3-12: 1D beam model with 10 nodes](image)

**Table 3.1: The first 5 natural frequencies**

<table>
<thead>
<tr>
<th>Natural frequencies</th>
<th>6 node beam model</th>
<th>10 node beam model</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\omega_1$</td>
<td>104.207 Hz</td>
<td>101.148 Hz</td>
</tr>
<tr>
<td>$\omega_2$</td>
<td>307.595 Hz</td>
<td>302.664 Hz</td>
</tr>
<tr>
<td>$\omega_3$</td>
<td>595.972 Hz</td>
<td>594.949 Hz</td>
</tr>
<tr>
<td>$\omega_4$</td>
<td>964.715 Hz</td>
<td>959.631 Hz</td>
</tr>
<tr>
<td>$\omega_5$</td>
<td>1427.124 Hz</td>
<td>1428.666 Hz</td>
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</table>
Figure 3-13: MAC of the 6 node beam model

Figure 3-14: MAC of the 10 node beam model
Chapter 4

Results

The example of the cantilever beam is illustrated how to perform impact test using the LMS Test. Lab software. Now the impact test using LMS Test. Lab will be applied to the tool holder for turning so that its vibration characteristics can be known. The photo of the tool holder is shown in figure 4-1.

The tool holder exhibits a single natural frequency. The natural frequency value is very high as shown in the results. So the frequency range of interest that is aimed in this test should be high. The tip of the hammer play an important role in determining the frequency range. The hammer tip material can either be soft rubber tip or hard metal tip. A hard tip with higher stiffness would be more effective in exciting the higher frequencies. Therefore, hard metal tip for the impact hammer was used.
Impact test was performed and the vibration response was measured in the three directions X, Y and Z. The X, Y and Z directions will be defined as the following: the X-direction is along the length of the tool, the Y-direction and the Z-direction is along the transverse directions. The picture of the impact test and measured vibration response in the three directions X, Y and Z are shown in figure 4-2, 4-3 and 4-4, respectively. The values of the natural frequencies and the maximum amplitude found in three directions are shown in Table 4.1 & 4.2.
Figure 4-2: Impact test and measurement in X-direction
Figure 4-3: Impact test and measurement in Y-direction
Figure 4-4: Impact test and measurement in Z-direction
Table 4.1: Natural frequencies in X, Y and Z directions for the tool holder

<table>
<thead>
<tr>
<th>Directions</th>
<th>Natural frequency</th>
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<tbody>
<tr>
<td>X</td>
<td>7081 Hz</td>
</tr>
<tr>
<td>Y</td>
<td>6069 Hz</td>
</tr>
<tr>
<td>Z</td>
<td>2787 Hz</td>
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Table 4.2: Maximum amplitudes in X, Y and Z directions for the tool holder

<table>
<thead>
<tr>
<th>Directions</th>
<th>Maximum amplitudes</th>
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<tbody>
<tr>
<td>X</td>
<td>11.78 g/N</td>
</tr>
<tr>
<td>Y</td>
<td>62.10 g/N</td>
</tr>
<tr>
<td>Z</td>
<td>56.94 g/N</td>
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</table>

From Table 4.1, the natural frequency value is higher in the X-direction than the Y & Z directions. This implies that the tool holder is more flexible in Y & Z directions than the X direction. From Table 4.2, although the value of the maximum amplitude in the X-direction which is the longitudinal direction of the tool holder is small compared to the other directions, the maximum amplitude values of the tool holder in Y & Z-directions which are the transverse directions where it has more flexibility are large. Reaching such high maximum amplitudes is not an attribute of an ideal tool holder. This prompts the necessity to create a tool holder with alternative design which was proposed in this research using Additive Manufacturing and internal cavity which will consequently yield better and improved damping capacity to mitigate chatter.
Chapter 5

Analytical Study

There are two possible approaches for analyzing the tool holder: experimental and analytical. The experimental approach using LMS Test. Lab was explained in the previous section. In this section, the analytical approach will be explained. It is worth noting that the analytical approach has not been implemented for the design of the tool holders. Based on Saeki’s multi-unit particle damping research [21], the purpose of this section is to review the methodology and show the potentials of this approach in developing optimal tool holders.

M. Saeki created the multi-unit particle damper which shares particle damping mechanism similar to the damping mechanism examined by Wakasawa [19] [20]. Wakasawa’s research can be considered a precursor to Saeki’s research and the additively manufactured tool holder with internal cavity filled with powder metal in this project. Figure 5-1 shows a lumped parameter model of the multi-unit particle damper. There are five cavities in the structure. The granular particles in the 5 cavities induce energy dissipation and attenuate the intense vibration. The shape of the cavity is cylinder. The
experimental apparatus is shown in figure 5-2. The cavity cylinder and the granular particles are composed of acrylic resin.

Figure 5-1: Model of a multi-unit particle damping system [21].

Figure 5-2: Schematic of experimental apparatus [21].

Saeki also constructed an analytic model to analyze the damping performance of the multi-unit particle damper. The equations which describe the system are reviewed in this section. The motion of the particles must be evaluated in order to evaluate the behavior of the entire system. Equations of the motions of particle $i$ are expressed as
\[ m \ddot{p}_i = F_i - mg \quad (5.1) \]
\[ I \ddot{\varphi}_i = T_i \quad (5.2) \]

where \( m \) is mass of the particle, \( I \) is moment of inertia of the particle, \( g \) is acceleration vector due to gravity, \( p_i \) is position vector of the particle, \( \varphi_i \) is angular velocity vector of the particle, \( F_i \) is sum of the contact forces acting on the particle, \( T_i \) is sum of the torque caused by the contact forces.

The damper is modelled as a single-degree-of-freedom system. It oscillates in a single direction which is labeled as the \( x \)-direction. The primary vibrating system has mass of \( M \), damping coefficient of \( c \) and spring constant of \( k \). The support point excites the system with the harmonic motion \( u = a \cos \omega t \). The equations of motion for the system and the support point are expressed as

First, the total contact force \( F_i \) for each individual particles is calculated in Eq (5.1). Then taking the aggregate of all of each \( F_i \) for each individual particles, the component of the total force \( F \) exerted by all the particles on the primary system in the direction of the excitation \( x \) is obtained for solving Eq (5.3).

\[ M \ddot{x} + c(\dot{x} - \dot{u}) + k(x - u) = F \quad (5.3) \]
\[ u = a \cos \omega t \quad (5.4) \]

In order to reduce the computation time, the following simplifying assumption is established. The assumption is that the behavior of all the particles in every cavity is the same. This simplifies calculation of the component of the contact force \( F \) as the product of the number of cavities \( N_U \) and the contact force component in each cavity \( F_u \). With this assumption instead of analyzing all the particles in every cavity, only the particles in a
A single cavity is analyzed. Thus, the equation of motion of the primary system can be re-written as

\[ M\ddot{x} + c(\dot{x} - \dot{u}) + k(x - u) = N_u \times F_u \quad (5.5) \]

In this model, contact occurs either between two particles or between a particle and the cavity wall. The two particles will be distinguished as particle \( i \) and particle \( j \) in this text. The contact force is composed of the normal component \( f_n \) and the tangential component \( f_t \). This contact force is derived from the Hertzian contact theory. The equations for the contact force are given as

\[ f_n = k_n \delta_n^{3/2} + c_n \delta_n^{1/4} \dot{\delta}_n \quad (5.6) \]
\[ f_t = -\mu f_n \delta_t/|\delta_t| \quad (5.7) \]

where \( \delta_n \) is the normal displacement and \( \dot{\delta}_n \) is the normal velocity of particle \( i \) relative to particle \( j \). \( \delta_t \) is the tangential velocity, \( c_n \) is the damping coefficient and \( \mu \) is the coefficient of friction between two particles or between a particle and the cavity wall. \( k_n \) is the spring constant.

The normal displacement \( \delta_n \) between two particles \( i, j \) and between a particle and the cavity wall which is cylindrical are respectively expressed as

\[ \delta_n = 2r - |\mathbf{p}_j - \mathbf{p}_i| \quad (5.8) \]
\[ \delta_n = \sqrt{(x_i - x_c)^2 + (y_i - y_c)^2} + r - R \quad (5.9) \]

where \( R \) is the radius and subscript \( C \) is the center of the cylindrical cavity. \( r \) is the radius of the particle. \( x_i \) and \( y_i \) are the coordinates of the particle \( i \) inside the cavity. The contact between two particles and particle and the cavity wall is recognized when the normal displacement meets the condition \( \delta_n > 0 \). \( k_n \) is the spring constant defined according to
the Hertzian contact theory. The particles are modelled as spherical shapes. The spring constant $k_n$ for inter-particle contact is equivalent to contact between two spheres and is expressed as

$$k_n = \frac{\sqrt{2r}}{3} \frac{E_p}{1 - \nu_p^2}$$  \hspace{1cm} (5.10)

The subscript $p$ is for the particle. $E$ is the modulus of elasticity and $\nu$ is the Poisson’s ratio. The contact between the particle and the cavity wall is equivalent to sphere-cylindrical wall contact. The spring constant $k_n$ between particle and the cylindrical cavity wall is given by

$$k_n = \frac{128}{9} \frac{\zeta E_i E_0}{(1 - \nu_i^2)E_0 + (1 - \nu_0^2)E_i}$$  \hspace{1cm} (5.11)

The constant $\zeta$ is a constant dependent on the cavity radius $R$ and the particle radius $r$. For example, when $r = 0.003m$, the constant $\zeta$ can be found from the graph in figure 5-3. By using the graph in figure 5-3, the spring constant $k_n$ between cylindrical wall and particle can be found.

For the contact between sphere and flat wall, the spring constant $k_n$ in this case can be expressed as

$$k_n = \frac{4\sqrt{r}}{3} \frac{E_p E_0}{(1 - \nu_p^2)E_0 + (1 - \nu_0^2)E_p}$$  \hspace{1cm} (5.12)

where the subscript 0 stands for the wall.
Figure 5-3: Relation between the cavity radius $R$ and $\zeta$ ($r = 0.003m$) [21].

The plot in figure 5-4 compares the spring constants for the sphere-cylindrical wall contact and the sphere-flat wall contact. In reality, the cavity wall is a cylindrical shape. However, as seen in figure 5-4, when the cavity radius $R$ is large enough ($R > 0.02m$), the spring constant $k_n$ values become constant and the difference between the two $k_n$ values for the sphere-cylindrical wall contact and the sphere-flat wall contact becomes marginal. So when the cavity radius $R$ is large enough ($R > 0.02m$), the spring constant $k_n$ can be considered constant and the spring constant $k_n$ can be equal to equation (5.12) instead of (5.11).
Figure 5-4: Influence of the cavity radius on the spring constant \((r = 0.003m)\) [21].

Each particle can be in a state of contact with more than one particle and the wall simultaneously. The sum \(F_i\) of the contact force acting on the particle and the sum \(T_i\) of the torque of the contact force are calculated as

\[
F_i = \sum (f_n + f_t) \tag{5.13}
\]

\[
T_i = \sum (\eta n_{ij} \times f_t) \tag{5.14}
\]

where \(n_{ij}\) is the unit vector from particle \(i\) to \(j\).

Saeki also examined the effect of several parameters that comprise the system. The effect of the number of cavities filled, cavity dimensions, and mass ratio on the damping performance are examined. The mass ratio \(\lambda\) is defined as

\[
\lambda = \frac{\text{Total mass of the particles}}{\text{Total mass of the system}} \tag{5.15}
\]

The mass ratio \(\lambda\) is similar to the packing ratio in that it indicates how much of the cavity is filled with particles. In order to investigate the influence of the number of cavities
on the damping capacity, 3 to 5 cavities were filled with particles in the formations shown in figure 5-5.

Figure 5-5: Cavity arrangement [21].

Figure 5-6 and figure 5-7 both illustrate the effect of different number of cavities and mass ratio $\lambda$ on the RMS (Root Mean Square) values of the vibration amplitude. When cavities exist in the structure, it clearly shows that the vibration amplitude decreases more than the cases with no cavities. It clearly proves that there is a definite improvement in damping when cavities filled with particles exist. Also the effect of mass ratio $\lambda$ on damping performance in figure 5-8. The vibration amplitude reduces as mass ratio increases. The damping capacity increases as the mass ratio $\lambda$ increases.
Figure 5-6: Comparison between experimental and calculated results

(4 cavities, \( \lambda = 0.098 \)) [21].

Figure 5-7: Comparison between experimental and calculated results

(5 cavities, \( \lambda = 0.098 \)) [21].
Figure 5-8: Response of the system amplitude vs. the frequency

(3 cavities, $R = 0.038 \, m$) [21].

Lastly, another parameter was investigated. The influence of size of the cylindrical cavity on damping capacity was investigated. The parameter which represents the size of the cavity is the radius $R$ of the cylindrical cavity. Figure 5-9 shows the relationship between the maximum amplitude value and the cavity radius $R$. In figure 5-9, the effect of the cavity radius $R$ on the maximum vibration amplitude is less significant for 4 cavities and 5 cavities which is greater than 3 cavities. Furthermore, there is an optimum cavity radius $R_{opt}$ value where the maximum amplitude is minimum which indicates maximum damping effect. The optimum cavity radius $R_{opt}$ seems to decrease with increasing number of cavities.
The tool holder that will be additively manufactured containing cavity filled will be very similar to this multi-unit particle damper. The cavities are filled with metallic powder. There is multitude of powder particles inside the cavity and it is crucial to analyze all of the each individual particles in order to properly evaluate the damping performance of the tool holder during vibration in machining. Using the equations from Eq (5.1) to (5.14) from Saeki’s material, the contact force and the equation of motion can be analyzed for each particle. This procedure is repeated for all particles. Particles interact with the cavity wall and in turn, affect the vibration motion of the entire tool holder. The particles exert force on the cavity by collision or friction. By analyzing each of the particles, the total contact force acting on the cavity by the particles can be acquired. The force acting on the tool holder by the particles inside the cavity is important and must be taken into account in Eq (5.3), the equation of motion for the entire system. Finally, this equation of motion for the primary system is solved to observe the damping performance and the capacity of this tool holder to suppress chatter.
Saeki’s multi-unit particle damper is the very design that the novel tool holder design is inspired by and that it will emulate. Therefore, Saeki’s research will be very useful for this research. Based on the study of multi-unit particle damper, it is possible to predict the effect of design parameters on the performance of the new tool holder. This effect of design parameters are exhibited in figures 5-6 to 5-9. Design parameters such as how much the cavity is filled metal powder, number and size of cavities can be addressed in the design of tool holder. It is expected that the presence of cavities in the tool holder will certainly have better outcome compared to when there isn’t. And as the number of cavities increase, the damping capacity will increase. It is expected that the more cavities are filled with powder, the more it will be effective in mitigating chatter. Lastly, an optimal value of cavity size where maximum damping is achieved exists and this optimal value will decrease with increasing number of cavities. The additively tool holder that is somewhat analogous to the multi-unit particle damper is anticipated to exhibit similar damping characteristics, show favorable results and effectively suppress chatter.
Chapter 6

Conclusion

Chatter is the unstable vibration due to regeneration of wavy surface and this is the central challenge that prevents attaining the maximum productivity and undermines the entire machining procedure. Thus, it is critical to mitigate the chatter vibration. The Stability Lobe Diagram is helpful in understanding and controlling chatter. There are many solutions devised by researchers to suppress and control chatter and they can be classified into the following four categories: out-of-process, in-process, passive and active strategies. The particular study of particle damping became the catalyst for the solution for chatter this paper is working toward. The purpose of this research was to develop tool holders with internal cavities filled with metal powders, using the Additive Manufacturing technique which is recently gaining interest. This tool holder shares the same intrinsic damping mechanism of the particle damping.

The impact test provides information about the vibration characteristics of the structure. Impact test was carried out on the tool holder for turning using the state-of-the-art vibration measurement software called the LMS Test.Lab. The impact test results for the tool holder for turning was not satisfactory and showed high value of vibration
amplitudes. The experimental results pointed to a need for a better improved tool holder design which was proposed in this thesis.

Analytic approach was taken by discussing how to apply the analytical and numerical procedure from Saeki’s research on particle damping in this research for the tool holders that this research aims to develop. The design parameters considered here were the mass ratio which indicates how much of the cavities are filled, number of cavities and the size of the cavity. Furthermore, the influence of the design parameters on the performance of the new tool holder was predicted based on the study of Saeki’s particle damping and favorable outcomes were expected.

The future work can focus to expand the methodology for designing optimum cavity for the novel tool holder based on the analytical and numerical work shown in this paper. It is desirable to design various tool holders using the Additive Manufacturing and compare their performance with the conventional tool holders. Lastly, the future work can expand the parametric study based on Saeki’s work and investigate the effect of other design parameters such as cavity shapes and position. The cavity can adopt various shapes such as cubic and spherical shapes, not only cylindrical shape as which it was chosen in this research. The position of cavities within the tool holder can be varied to investigate its influence. Through expanding the given parametric study, it will facilitate the optimization process for tool holder design. The Additive Manufacturing technique is able to achieve this.
References


