Improvement of Milling Stable Processing Condition for a Cutter With Large Ratio of Length to Diameter

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ABSTRACT In order to machine complex box and shell structures, the cutter with large ratio of length to diameter is usually used, whereas the low rigidity of the cutter hinders the increase of the stable machining parameters. This article develops one kind of damping cutter to improve the stable processing condition in this milling operation. The influences of modal characteristic of the cutter on chatter stability are investigated, where it is proven that increasing the dynamic stiffness can efficiently extend the stable zone. On this basis, two damping cutters are designed by inlaying the strips and damping core into the toolholder body. The influences of geometrical parameters and material property of the damping cutters on modal characteristics are theoretically analyzed, and the optimal geometrical dimensions are determined using the finite element simulation. Then, the two damping cutters are fabricated. After modal tests and milling machining experiments, the measured results validate the effectiveness of the developed damping cutters in terms of improvement of the stability frontier.

INDEX TERMS Milling cutter, ratio of length to diameter, variable cross sections, chatter stability, loss factor.

NOMENCLATURE

- $a$: Axial cutting depth
- $a_{\text{lim}}$: Limit axial cutting depth
- $c_x, k_x$: Damping and stiffness along $x$ direction
- $c_y, k_y$: Damping and stiffness along $y$ direction
- $d$: Dynamic stiffness
- $e$: Elastic modulus ratio
- $D_1, E_1$: Diameter and elastic modulus of the toolholder body
- $D_2, E_2$: Diameter and elastic modulus of the damping core
- $(EI)_w$: Equivalent bending stiffness
- $(EI)_1$: Bending stiffness of the toolholder body and the strips
- $(EI)_2$: Bending stiffness of the damping core
- $f(t)$: Dynamic milling force
- $f_x, f_y$: Dynamic milling forces along $x$ and $y$ directions respectively
- $F(t), F_0$: Milling forces, and its amplitude
- $h$: Thickness ratio
- $H_1, H_2$: Thickness of the toolholder body and damping core
- $K_t$: Cutting force coefficient
- $L$: Overhang length of the cutter
- $m, c, k$: Mass, damping, and stiffness of milling cutter
- $N$: Teeth number
- $w(t)$: System response
- $W$: Total deformation energy of the cutter
- $\Delta W$: Dissipation energy by damping core
\( \beta \) Material loss factor  
\( \eta \) Loss factor  
\( \lambda \) Eigenvalue  
\( \nu \) Frequency ratio  
\( \omega \) Rotational frequency  
\( \omega_n \) Natural frequency  
\( (t) \) Vibration displacement  
\( \Phi(t) \) Transfer function  
\( \psi \) Phase angle  
\( \Omega \) Spindle speed  
\( [\mathcal{A}_0] \) Average directional coefficient matrix

I. INTRODUCTION

Chatter is a harmful self-excited vibration, which directly affects the surface quality of the machined workpiece, and even damages the machine tool and cutter [1]. So far, there have been a considerable amount of literature on chatter control. Considering whether it is real-time, the existing methods can be classified into two kinds, i.e. in-process and out-of-process strategies [2].

The essence of the in-process strategy is to detect and identify the occurrence of chatter and then to adjust the cutting parameters so as to achieve the chatter-free machining process [3]. This strategy generally contains online chatter detection and active control method. Due to the advantages of high efficiency and high accuracy, online chatter detection has been widely investigated [4]–[10]. Various signals are used to identify chatter, such as force signal [4], [5], acceleration signal [6], [7], and sound signal [8]. After signal acquisition, the feature extraction is vital to precisely detect chatter, where many relative signal processing methods, mainly including variational mode decomposition [8], continuous wavelet transform [9], and convolutional neural network [10], are proposed. However, multiple sensors, signal acquisition and processing system, and other auxiliary devices are required, which hinder the wide application of chatter detection. On the other hand, the active control method becomes increasingly important in the field of chatter control thanks to its efficiency and flexibility [11]–[14]. Accordingly, many active devices are designed, such as electromagnetic actuator [12], piezoelectric actuator [13], and magnetorheological fixture [14]. However, these devices usually need the relative elements of monitoring, identification and execution, which make this method complex and costly.

As opposed to in-process strategy, the out-of-process strategy is easy to implement. Stability prediction is a commonly effective way to avoid chatter, in which the predicted stability chart can guide the selection of the suitable machining parameters [15]. Many solutions are proposed to calculate the stability chart, such as the discretization method [16]–[18] and the frequency method [19]. As the input parameters, the precise dynamic parameters and cutting force coefficient of the machining system are crucial to predict the stability chart. However, these parameters are not constant due to the time-varying characteristic of the machining process, which has negative effect on the stability prediction [20]. With the aim to suppress chatter better, the passive control method is proposed, whose essence is to improve the machining stability through introducing an additional device that can consume vibration energy or break the regeneration effect [3]. The designed passive equipment is mainly composed of the friction damper, vibration absorber, tuned mass damper, constrained layer damper, coating damping, and nonstandard cutter. For instance, Madoliat et al. [21] placed the core and multi-fingered cylinder into the axial hold of a slender end milling tool to dissipate energy by the way of the friction. The dynamic vibration absorber with variable stiffness was designed and was filled into a boring bar, where the vibration reduction performance was verified by experiments [22]. A passive damper was developed and was embedded into a milling cutter to achieve the chatter control [23]. According to the constrained layer damping beam theory, the different damping toolholders were separately proposed, and were used in boring, milling and turning operations [24]–[26]. Fu et al. [27] applied the multi-layered nanostructures of carbon on the surface of milling tools to improve the vibration damping, thus increasing the stability limit. Mei et al. [28] designed the milling cutters with alternating variable pitches to improve the machining stability. In addition, Wan et al. [29] proposed a stability improvement method through applying the tensile prestress to the thin-walled piece.

From the above references reviewed, even though the chatter control has been explored broadly, the milling dynamics has received very little attention with regard to the cutter with large ratio of length to diameter and variable cross sections. Therefore, the motivation of this article is to study the dynamics of this kind of milling cutter, and then to design the novel damping cutter, thereby improving the machining stability.

II. DYNAMIC MODEL OF MILLING CUTTER

As shown in Fig. 1, the milling cutter includes a cutter head and a toolholder with the long overhang and variable cross sections. Supposing the positioning connection between the spindle and the toolholder is rigid, the milling cutter can be regarded as the cantilever structure with one end fixed and another end free.

During milling operation, the end free is subjected to the milling force. Note that the axial force along \( z \) direction is neglected, since it has little effect on the radial displacement. The cutter can be simplified as a cutter-spring-damper system. The milling forces attached on cutter head and the
corresponding displacement can meet the requirement of the following equation:

\[ m\ddot{w}(t) + c\dot{w}(t) + kw(t) = F(t) \]  

(1)

where \( m, c \) and \( k \) are mass, damping and stiffness of the cutter separately, \( F(t) \) is the exciting force produced from the cutting process, and \( w(t) \) refers to the response along \( x \) or \( y \) direction. The damping \( c \) can be described as follows:

\[ c = \eta \sqrt{km} \]  

(2)

where \( \eta \) represents the loss factor.

Then, the system response can be obtained from Eq. (1), which is expressed as:

\[ w(t) = \frac{F_0}{k} \cdot \frac{1}{\sqrt{(1 - \nu^2)^2 + (\eta \omega)^2}} e^{i(\omega t - \psi)} \]  

(3)

where \( F_0 \) represents the amplitude of \( F(t) \), \( \nu = \omega/\omega_n \) is the frequency ratio, \( \omega_n \) stands for the natural frequency, \( \psi \) is the phase angle. According to Eq. (3), it can be qualitatively obtained that the vibration displacement can be decreased with the increase of \( k \) and \( \eta \). In order to further analyze the relation between the vibration response of the machining system and its modal characteristic in detail, the dynamic stiffness \( d \) [25] is defined and is expressed as:

\[ d = \eta \cdot k \]  

(4)

According to the dynamics theory of the milling operation [19], the cutter in Fig. 1 and workpiece can be expressed as a vibration system during milling process, as seen in Fig. 2.

Since the dynamic milling force \( f(t) \) of \( F(t) \) only affects the chatter stability, \( f(t) \) is thus described as

\[ \{f(t)\} = \frac{1}{2}aK_t \{A_0\} \{\Delta(t)\} \]  

(5)

with

\[ \{f(t)\} = \{f_x \ f_y\}^T \quad \{\Delta(t)\} = \{\Delta x(t) \ \Delta y(t)\}^T \]  

(6)

where \( a \) refers to the cutting depth, \( [A_0] \) is the average directional coefficient matrix, \( K_t \) is the cutting force coefficients, and \( \Delta(t) \) represents the variation of vibration displacements.

Transforming Eq. (5) from time domain to frequency domain leads to:

\[ \{f(\omega)\} = \frac{1}{2}aK_t \{A_0\} (1 - e^{-i\omega T}) \{\Phi(i\omega)\} \{f(\omega)\} \]  

(7)

with

\[ \{\Phi(i\omega)\} = \begin{bmatrix} \Phi_{xx}(i\omega) & 0 \\ 0 & \Phi_{yy}(i\omega) \end{bmatrix} \]  

(8)

where \( \Phi(i\omega) \) stands for the transfer function, and it can be described as:

\[ \Phi_{xx}(i\omega) = \frac{1}{k_x(1 - \nu^2 + i\eta \omega)} \]

\[ \Phi_{yy}(i\omega) = \frac{1}{k_y(1 - \nu^2 + i\eta \omega)} \]  

(9)

When around the chatter frequency, the cutter will be in the limit steady state, and the corresponding characteristic equation is derived as follows:

\[ \det \left\{ \{I\} - \frac{1}{2}a_{\text{lim}}K_t \{A_0\} (1 - e^{-i\omega T}) \{\Phi(i\omega)\} \{I\} \right\} = 0 \]  

(10)

After solving Eq. (10), the limit axial cutting depth \( a_{\text{lim}} \) and the corresponding spindle speed \( \Omega \) can be given by:

\[ a_{\text{lim}} = \frac{-2\pi}{NK_t} \left( \frac{\lambda_{\text{Re}}(1 - \cos(\omega_c T)) + \lambda_{\text{Im}} \sin(\omega_c T)}{(1 - \cos(\omega_c T)) + \lambda_{\text{Re}} \sin(\omega_c T)} \right) \]  

\[ \Omega = \frac{60\omega_c}{N(\pi - 2 \arctan(\lambda_{\text{Im}}/\lambda_{\text{Re}}) + 2n\pi)} \quad n = 0, 1, 2 \cdots \]  

(11)

where \( \lambda \) is the eigenvalue, and \( N \) is the teeth number of the cutter.

Ignoring the imaginary part of Eq. (11), and then combing Eq. (4), the relation among the limit axial cutting depth \( a_{\text{lim}} \), dynamic stiffness \( d \) and spindle speed \( \Omega \) can be obtained and is shown in Fig. 3. It clearly demonstrates that the limit axial cutting depth \( a_{\text{lim}} \) is significantly increased with the gradual increase of \( d \), which is consistent with the results in [25], [29].
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In other words, increasing the dynamics stiffness of the cutter system can improve the chatter stability.

III. DEVELOPMENT OF DAMPING CUTTER

A. DESIGN OF DAMPING CUTTER

As shown in Fig. 4a, the conventional cutter, named CC, is mainly composed of a conventional toolholder, cutter head, inserts, pull nail and axial screw. The cutter can be fixed with the spindle via the pull nail, while the cutter head with four inserts is closely connected with the toolholder by the axial screw.

Based on the above finding, two damping cutters are initially designed as shown in Figs. 4b and 4c, where they are respectively called DCDS and DCCS for analytical convenience. In Fig. 4b or Fig. 4c, a damping core is filled into the toolholder body to improve the damping performance, and the strips made of material with high modulus are embedded into the toolholder using the welding method to increase the rigidity, which can enlarge the dynamic stiffness. It should be pointed out that every strip embedded into the DCCS matches with the external shape of the toolholder, whereas the strips in the DCDS are not continuous.

The material of 35CrMo steel is used to manufacture the toolholder body. Owing to high damping performance, Foamed aluminium (FA) and Polyurethane rubber (PR) are selected as the materials of damping core in the DCDS and DCCS respectively [24], [30]. And YG6 steel is chosen to make the strip, since it has high modulus. The properties of the used materials are given in Table 1.

B. MODAL CHARACTERISTICS OF DAMPING CUTTER

To analyze the modal characteristics of the damping cutter, some assumptions are firstly provided as follows [25]:

1. The bending strains produced from damping core, strip and toolholder body show the same linear distribution in vertical direction;
2. There is not relative sliding between the contact surfaces among the above three parts;
3. The material of damping core is uniform.

The stability limit of milling cutter is characterized by dynamic stiffness as shown in Fig. 3. According to the dynamic model in Fig. 1, the cutter is regarded as a cantilever structure [29], whose stiffness can be expressed as:

$$k = \frac{3(\frac{EI}{w})}{L^3}$$  \hspace{1cm} (12)

where $L$ represents the overhang length of milling cutter, $(\frac{EI}{w})$ denotes the equivalent bending stiffness which can be described as:

$$\frac{EI}{w} = (\frac{EI}{1}) + (\frac{EI}{2})$$  \hspace{1cm} (13)

where $(\frac{EI}{1})$ is the bending stiffness of both toolholder body and strips, and $(\frac{EI}{2})$ is the bending stiffness for the damping core.

The substitution of Eq. (13) into Eq. (12) gives:

$$k = \frac{3\pi[D_1^3 \cdot E_2 + (D_1^3 + D_2^3) \cdot E_1]}{L^3}$$  \hspace{1cm} (14)

where $E_1$ and $D_1$ are elastic modulus and diameter for the toolholder body, and $E_2$ and $D_2$ are the elastic modulus and diameter for the damping core, respectively.

During milling operation, the cutter is suffered from the milling vibration, where the damping core can dissipate the vibration energy by the resulting bending strain. Based on the bending strain energy method [25], [31], the loss factor of the damping cutter can be represented by:

$$\eta = \frac{\Delta W}{\dot{W}}$$  \hspace{1cm} (15)

where $\Delta W$ stands for the dissipation energy by damping core, and $W$ is the total deformation energy of the cutter.

Substituting the geometrical parameter and material property of the damping cutter into Eq. (15), then the following

TABLE 1. Material properties used in cutters.

| Material | Density ($\text{g/cm}^3$) | Elastic modulus (GPa) | Poisson’s ratio | Material loss factor (%) |
|----------|--------------------------|-----------------------|----------------|-------------------------|
| 35CrMo   | 7.87                     | 213                   | 0.29           | 0.1                     |
| YG6      | 14.5                     | 600                   | 0.21           | 0.1                     |
| FA       | 0.65                     | 12                    | 0.33           | 12                      |
| PR       | 1.11                     | 30                    | 0.3            | 47                      |
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FIGURE 5. Stiffness (a) and loss factor (b) with respect to the geometrical dimension and material property.

The equation can be obtained as:

$$\eta = \frac{\alpha \beta [H_{12}^2 + r_2^2(1 + 2\alpha)]}{\alpha H_{12}^2 + (\alpha r_2^2 + r_1^2)(1 + 2\alpha)}$$  \hspace{1cm} (16)

with

$$r_1^2 = H_1^2/12, \quad r_2^2 = H_2^2/12, \quad H_{12} = (H_1 + H_2)/2$$

$$e = E_2/E_1, \quad h = H_2/H_1, \quad \alpha = e \cdot h$$  \hspace{1cm} (17)

where $H_1$ is the thickness of the toolholder body, $H_2 = D_2/2$ is the thickness of damping core, and $\beta$ is the material loss factor.

Thus, Eq. (16) can be further transformed into the following expression:

$$\eta = \frac{eh(3 + 6h + 4h^2)}{1 + eh(5 + 6h + 4h^2)}$$  \hspace{1cm} (18)

difficult to obtain the suitable structural parameters for the strips and damping core from the qualitative theory analysis, since the cross sections of the cutter are variable. Therefore, the finite element method (FEM) is employed to determine the optimum dimensions for the damping cutter in the next subsection.

C. PARAMETER OPTIMIZATION AND MANUFACTURE OF DAMPING CUTTER

As seen in Fig. 4, the thicknesses of the inlaid strip and the diameter of the damping core, i.e., $H_{y1}$ and $H_2$, are selected as the design variables. For convenience of the optimization analysis, the thicknesses of the strips embedded into the different cross sections of the toolholder body are proportional to the corresponding diameters of the cross sections. The optimization goal is to make the maximum response amplitude of the cutter head smallest.

According to the optimal design parameters, the two damping cutters are manufactured as shown in Fig. 7, where the dynamic balance tests for the damping cutters are performed to meet the good rotation precision.

IV. PERFORMANCE TESTS AND COMPARISON

In order to verify the chatter stability of the damping cutters developed above, modal testing and milling experiments are conducted on a CNC machining center (SPN50R) in the workshop of KSD. Before milling tests, the modal parameters of the machining system are measured by modal testing to predict the stability chart, and simultaneously is used to verify the above simulated modal characteristic. The used cutter head with the diameter of 80 mm has four inserts. The material of ductile cast iron is used to make workpiece.

A. MODAL TESTING

During modal testing, the hammer (CL-YD-303) with the sensitivity of 4.29 pC/N and accelerometer sensor
FIGURE 6. The simulated dynamic stiffness (a) and response amplitude (b) for the CC, DCDS and DCCS.

FIGURE 7. Photographs of the CC, DCDS and DCCS.

FIGURE 8. Experimental setup for modal testing.

(Dytran 3035B) are employed as shown in Fig. 8. The hammer provides the impact forces at cutter head and synchronously the responses signals are collected by B&K Pulse 3560C data acquisition system.

FIGURE 9. Measured frequency curves of the CC, DCDS and DCCS.

TABLE 3. Measured modal parameters of three cutters.

| Modal parameters          | CC   | DCDS | DCCS |
|---------------------------|------|------|------|
| Natural frequency (Hz)    | 312  | 316  | 340  |
| Damping ratio (%)         | 1.28 | 2.53 | 2.35 |
| Stiffness (MN/m)          | 8.07 | 9.3  | 16.08|
| Dynamic stiffness (MN/m)  | 0.21 | 0.47 | 0.76 |

FIGURE 10. Experimental setup for milling force testing.

The measured frequency curves along y direction for three milling cutters are illustrated in Fig. 9, where the modal characteristic along x direction can be considered to be same with that along y direction because of the symmetrical structure. The corresponding modal parameters are identified and given in Table 3, which experimentally shows the obvious improvement of dynamic stiffness for the damping cutters, especially the DCCS, compared with the conventional cutter. In addition, the measured results verify that the simulated results from the FEM is effective under the range of the errors permitted.

B. FACE MILLING EXPERIMENTS

Fig. 10 presents the cutting experimental setup. The workpiece is mounted on the dynamometer (Kistler 9257B) fixed on the vertical machine table. During milling process, the milling forces can be measured to detect the cutting state. When the conventional cutter is used in workshop, the relative machining parameters include the spindle speed of 796 rpm, axial depth of cut of 1.5 mm, and feed rate of 31 mm/min. In order to compare the machining stability for the CC, DCDS
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FIGURE 11. Stability charts for the CC, DCDS and DCCS.

FIGURE 12. Milling force in frequency domain under $a = 1.0 \text{ mm}$ when using the CC (a), DCDS (b) and DCCS (c), respectively.

FIGURE 13. Milling force in frequency domain under $a = 1.5 \text{ mm}$ when using the CC (a), DCDS (b) and DCCS (c), respectively.

and DCCS, three axial cutting depths from $1 \text{ mm}$ to $2 \text{ mm}$ at the $0.5 \text{ mm}$ intervals are chosen as shown in Fig. 11, and other machining parameters are constant. Thus, under the different axial depth of cut, the milling forces from the three cutters are measured during milling process, whose spectra are presented in Figs. 12-14 in detail to reveal the machining stable state.

When the axial cutting depth is equal to $1 \text{ mm}$, i.e. point a in Fig. 11, the rotational frequency ($13.27 \text{ Hz}$) and its multiplication frequencies ($39.8 \text{ Hz}$, $53.1 \text{ Hz}$, $79.6 \text{ Hz},...$) are the dominant frequency in Fig. 12 whether the damping cutters are used or not. This always means that the milling state is stable by using the conventional or damping cutters.

As the axial cutting depth reaches to $1.5 \text{ mm}$ in point b, complex frequencies appear in Figs. 13a and 13b. Specifically, the vibration responses gather around $314.3 \text{ Hz}$ and $367.5 \text{ Hz}$ in Fig. 13a when using the conventional cutter. These frequencies are approximatively expressed by $314 \text{ Hz}$ and $314 \text{ Hz}$ plus 4 times of rotational frequency, where frequency value $314 \text{ Hz}$ is near to the natural frequency along $y$ direction. In other words, the responses near the natural frequency are dominant, which means the occurrence of chatter. Similarly, it can be found in Fig. 13b that the chatter also appears when using the DCDS, whereas the vibration...
amplitude is much less than that in Fig. 13a. When the DCCS is used, the milling process is still stable as shown in Fig. 13c. The axial depth of cut further increases to 2 mm in point c. Compared to Figs. 13a and 13b, both of the chatter phenomenon in Figs. 14a and 14b becomes serious when using the CC and DCDS. Additionally, it shows the slight unstable in Fig. 14c when DCCS is used, where some complex frequencies appear. Under the different axial depths of cut, the machining state shows various forms. The measured results show that the developed damping cutters, especially the DCCS, can improve the stable machining parameters obviously.

In order to further compare the machining performance, the damping cutters are then used to machine the motor shell in workshop as shown in Fig. 15. The cutting parameters of the conventional cutter used in workshop are selected. The corresponding acceleration responses are acquired via the acceleration sensor during milling process. The measured results are shown in Fig. 16. It can be found that the chatter occurs when using the CC and DCDS, whereas the amplitude in Fig. 16b is far below that in Fig. 16a. From Fig. 16c, the machining state is stable when using the DCCS. These imply that the chatter stability performance in the order from high to low is the DCCS, DCDS and CC.
V. CONCLUSION
In this article, the novel damping cutter is proposed to improve the stable zone of the milling cutter with large overhang and variable cross sections. The dynamic model for the cutter system is built to reveal the effect of the stiffness and loss factor on the chatter stability. According to the passive control strategy, two damping cutters are designed by embedding the strips and damping core into the toolholder body, whose modal characteristics analysis shows that the geometrical dimension and damping materials have the direct effects on the dynamic performance. Through the optimization analysis, the optimal geometrical dimensions are determined, and then the damping cutters are manufactured. Modal testing is carried out to compare the dynamic characteristic of the conventional and damping cutters, and the measured dynamic stiffness for the proposed cutter with 0.47 MN/m or 0.76 MN/m is much larger than that of the conventional cutter with 0.21 MN/m. The predicted stability charts for three cutters are verified by the cutting tests. Compared with the conventional cutter, the stability performance is improved significantly when using the damping cutters. It is worth noting that the designed damping cutters have been practically applied in the workshop of KSD.

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