Chatter suppression in turning operations with a tuned vibration absorber

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Abstract

In this paper, a piezoelectric inertia actuator is mounted on the cutting tool and acted as a tuned vibration absorber for the suppression of chatter in turning operations. It is shown that the tuned vibration absorber can modify the frequency response function of the cutting tool so as to improve cutting stability in turning operations. Chatter can then be effectively suppressed due to the increase of cutting stability. Theoretical studies and experimental results performed in turning operations are presented to illustrate this approach.

Keywords: Chatter; Turning; Vibration absorber

1. Introduction

In turning operations, chatter is a dynamic instability of the cutting process, which results from the interaction of the dynamics of the metal cutting process and the structural dynamics of the machine tool [1–3]. The presence of chatter, if uncontrolled, can easily spoil surface accuracy, damage the cutting tool, and produce irritating unacceptable noise. Therefore, many techniques have been investigated to suppress chatter in turning operations [4–8]. The present study focuses on developing a simple chatter suppression method based on the improvement of the dynamic response of a cutting tool with a tuned vibration absorber. Basically, the use of the tuned vibration absorber for the suppression of chatter can be considered as a passive control method. Although active control methods have become increasingly popular, passive control methods remain an important and effective vibration control tool. Passive control, compared to active control, exhibits the advantages of easy implementation, low cost, and no need for external energy. More importantly, passive control methods never drive the controlled system to instability, while the active control methods might [9].

It is known that the negative real part of the frequency response function of the machine tool structure plays an important role in determining cutting stability [10]. The stability limit of the cut width is inversely proportional to the magnitude of the negative real part of the frequency response function of the machine tool structure. Therefore, the suppression of chatter is feasible if the magnitude of the negative real part of the frequency response function of the machine tool structure is reduced. In this study, a piezoelectric inertia actuator mounted on the cutting tool is used as a tuned vibration absorber for reducing the magnitude of the negative real part of the frequency response function of the cutting tool in turning operations. This is because the tuned vibration absorber introduces a new degree of freedom to the dynamic system of the cutting tool. The new frequency response function of the cutting tool due to the tuned vibration absorber mounted on the cutting tool can then be obtained. The absorber with damping, when tuned to the proper frequency, can greatly reduce the magnitude of the negative real part of the frequency response function of the cutting tool so as to improve cutting stability in turning operations. As a result, chatter can be effectively suppressed due to the increase of cutting stability.

In the rest of the paper, the analysis of chatter in turning operation is first given. Then, a mathematical model for the dynamics of the cutting tool with the vibration absorber is established. Through the analysis of the solution of the mathematical model, cutting stability can be improved by
2. Analysis of chatter in turning operations

Fig. 1 shows the control block diagram of chatter in turning. As shown in Fig. 1, the uncut chip thickness \( h(s) \) is composed of the mean uncut chip thickness \( h_m(s) \), the inner modulated cut surface \( y(s) \), and the outer modulated workpiece surface \( y(s)e^{-Ts} \), where \( T(s) \) is the time delay between the inner and outer modulated surface. The uncut chip thickness \( h(s) \) is fed into the cutting process to produce the cutting force \( F(s) \) acting on the cutting tool. The transfer function of the cutting process \( G_c(s) \) can be expressed as

\[
G_c(s) = k_s\beta
\]

(1)

where \( k_s \) is the specific cutting force and \( \beta \) the cut width.

The cutting force \( F(s) \) then excites the cutting tool to generate the vibration \( y(s) \) between the tool and workpiece. The oriented transfer function of the cutting tool \( G_m(s) \) can be expressed as

\[
G_m(s) = \sum_{i=1}^{n} u_i G_i(s)
\]

(2)

where \( n \) is the number of modes, \( u_i \) the directional factor for mode \( i \), and \( G_i(s) \) the direct transfer function for mode \( i \), which can also be expressed as

\[
G_i(s) = \frac{1}{m_i\beta^2 + c_i\beta + k_i}
\]

(3)

where \( m_i, c_i, \) and \( k_i \) are the mass, damping, and spring parameters, respectively.

Based on the Nyquist stability criterion [11], the control system for turning is at the stability limit when the gain of the open loop transfer function has the critical value of \(-1\). The cut width \( b \) at the stability limit is called the critical cut width \( b_{\text{lim}} \) which can be expressed as

\[
b_{\text{lim}} = \frac{-1}{2k_s\text{Re}(G_m(j\omega))_{\text{neg}}}
\]

(4)

where \( \text{Re}(G_m(j\omega))_{\text{neg}} \) is the negative real part of the frequency response function of the cutting tool and \( \omega \) the chatter frequency in rad/s.

For simplicity of presentation, the cutting tool is taken as a single-degree-of-freedom vibration system. The parameters of the vibration system for the cutting tool are selected as mass, \( m=1 \) kg, damping ratio, \( \zeta=0.03 \), natural frequency, \( f_n=100 \) Hz. The specific cutting force is equal to \( k_s=1000 \) N/mm². Fig. 2 shows the real part and magnitude of the frequency response function of the cutting tool. Based on Eq. (4), the chatter frequency and the critical cut width \( b_{\text{lim}} \) can be calculated [10]. Fig. 3 shows the chatter frequency and the critical cut width \( b_{\text{lim}} \) as a function of spindle speed. Chatter will occur when the cut width \( b \) is greater than \( b_{\text{lim}} \). However, the critical cut width \( b_{\text{lim}} \) can be increased if the magnitude of the negative real part of the frequency response function of the cutting tool is reduced (Eq. (4)). In the following section, the use of a tuned vibration absorber to reduce the magnitude of the negative real part of the frequency response function of the cutting tool in turning will be investigated.
3. Modeling of the cutting tool with a tuned vibration absorber

As discussed before, the cutting tool is taken as a single-degree-of-freedom vibration system. Fig. 4 shows the single-degree-of-freedom vibration system with an additional degree of freedom due to a vibration absorber. The equation of motion governing the cutting tool with the vibration absorber can be expressed as

\[ [M] \ddot{x} + [C] \dot{x} + [K] x = \begin{bmatrix} F_0 \sin \omega t \end{bmatrix} \]

where \( F_0 \sin \omega t \) is the external force of the vibration system and the system coefficient matrices \([M]\), \([C]\), and \([K]\) can be expressed as

\[
[M] = \begin{bmatrix} m & 0 \\ 0 & m_a \end{bmatrix}, \quad [C] = \begin{bmatrix} c + c_a & -c_a \\ -c_a & c_a \end{bmatrix}, \quad [K] = \begin{bmatrix} k + k_a & -k_a \\ -k_a & k_a \end{bmatrix}
\]

where \( m, c, \) and \( k \) are the mass, damping, and spring parameters of the cutting tool, and \( m_a, c_a, \) and \( k_a \) are the mass, damping, and spring parameters of the vibration absorber.

Let the steady-state solution of \( x \) and \( x_a \) be of the following form:

\[
x = X \sin \omega t \quad \text{(7)}
\]

\[
x_a = X_a \sin \omega t \quad \text{(8)}
\]

where \( X \) is the magnitude of the steady-state vibration of the cutting tool and \( X_a \) is the magnitude of the steady-state vibration of the absorber.

Substituting Eqs. (7) and (8) into Eq. (5), the magnitude of the steady-state vibration of the cutting tool \( X \) and the magnitude of the steady-state vibration of the absorber \( X_a \) can be calculated. Once the magnitude of the steady-state vibration of the cutting tool \( X \) is obtained, a dimensionless magnification ratio \( R \), commonly used to present the steady-state vibration response of the vibration system, [12,13] can be expressed as

\[
R = \frac{X}{F_0/k}
\]

Fig. 5 shows the magnification ratio \( R \) of the cutting tool with the vibration absorber as a function of the frequency ratio \( \omega/\omega_n \). The ratio of the mass of the vibration absorber to the mass of the cutting tool is assumed as \( \mu = m_a/m = 0.25 \). In addition, the ratio of the natural frequency of the vibration absorber to the natural frequency of the cutting tool is chosen as \( \beta = \omega/\omega_n = 1 \). It is shown that the magnification ratio \( R \) at the natural frequency of the cutting tool is almost reduced to zero. However, two new natural frequencies at the left-hand and right-hand sides of the natural frequency of the cutting tool are generated (Fig. 5). The introduction of damping into the vibration absorber can modify the magnification ratio \( R \) at the two new natural frequencies. The magnification ratio \( R \) at the left-hand and right-hand sides of the natural frequency of the cutting tool is greatly reduced if the damping ratio \( \zeta_a\) of the vibration absorber is increased. Therefore, the reduction of the magnification ratio \( R \) of the cutting tool can be achieved by tuning the natural frequency of the vibration absorber to be equal to the natural frequency of the cutting tool and using a vibration absorber with a larger damping ratio.

Fig. 6 shows the real part and magnitude of the frequency response function of the cutting tool with a vibration absorber. The parameters of the vibration absorber are selected as mass, \( m_a = 0.25 \text{ kg} \) (\( \mu = 0.25 \)), damping ratio, \( \zeta_a = 0.1 \), natural frequency, \( f_a = 100 \text{ Hz} \) (\( \beta = 1 \)). As compared with Figs. 2(a) and 6(a), the magnitude of the negative real part of the frequency response function is greatly reduced because the vibration absorber is attached to the cutting tool. Fig. 7 shows the chatter frequency and the cut width \( b \) at the stability limit versus spindle speed. It can be seen that the critical cut width \( b_{\text{lim}} \) (Fig. 7) becomes larger due to the decrease of the magnitude of the negative real part of the frequency response function (Fig. 6(a)). Therefore, cutting
stability can be greatly improved because a well-tuned vibration absorber is attached to the cutting tool.

4. Computer simulations and discussion

In this section, computer simulations performed in turning operations are presented to support the proposed chatter suppression strategy using the vibration absorber. Fig. 8(a) shows the simulated displacement signal of the cutting tool without the vibration absorber during turning. The cutting parameters are selected as: a cut width of 2 mm and a spindle speed of 1000 rpm. As shown in Fig. 3(b), the cut width of 2 mm is still smaller than the critical cut width \(b_{\text{lim}}\). Therefore, the simulated displacement signal (Fig. 8(a)) is decreasing revolution after revolution. However, unstable cutting occurs if the cut width is increased to 5 mm and the spindle speed of 1000 rpm is still unchanged. The growth of the simulated displacement signal of the cutting tool becomes very fast due to the occurrence of strong chatter (Fig. 8(b)). In order to suppress the unstable cutting, a cutting tool mounted with a well-tuned vibration absorber is applied to this cutting test. Fig. 9 shows the simulated displacement signal of the cutting tool with the vibration absorber during turning using a cut width of 5 mm and a spindle speed of 1000 rpm. It is shown that a stable simulated displacement signal of the cutting tool is recorded. This is because the cutting stability is greatly improved owing to the vibration absorber attached to the cutting tool. Therefore, the cut width of 5 mm is still much below the critical cut width \(b_{\text{lim}}\) (Fig. 7(b)).
5. Experimental results and discussion

A number of cutting experiments into the turning of 6061 aluminum bars were carried out on a CNC turret lathe. Fig. 10 shows a schematic diagram of the experimental set-up. An accelerometer (Beetle Nevada 23732) mounted on the cutting tool was used to sense the vibration signal of the cutting tool. The accelerometer signal was then recorded on a PC-486 through a data acquisition board (DT2828). Fig. 11 shows the real part and magnitude of the frequency response function of the cutting tool. It is shown that the natural frequency of the cutting tool is close to 2000 Hz. In this study, a piezoelectric inertia actuator (PCB 712A01) with an initial mass acted as the tuned vibration absorber. The piezoelectric inertia actuator behaves like a simple mass–spring–damper system. The natural frequency of the actuator \( f_n \) can be tuned by adjusting the size of the inertia mass \( m_{in} \), i.e.,

\[
f_n = \frac{1}{2\pi} \sqrt{\frac{k_a}{m_{in} + m_a}} \tag{10}
\]

As shown in Eq. (10), the natural frequency of the actuator \( f_n \) increases with the decrease of the size of the inertia mass \( m_{in} \). On the other hand, the natural frequency of the actuator \( f_n \) decreases with the increase of the size of the inertia mass \( m_{in} \). Therefore, the natural frequency of the actuator \( f_n \) can be close to 2000 Hz by tuning the size of the inertia mass \( m_{in} \). Fig. 12 shows the frequency response function of the piezoelectric inertia actuator between the input voltage of the actuator and the output vibration of the actuator. It can be seen that the natural frequency of the actuator \( f_n \) is almost equal to the natural frequency of the cutting tool after the proper selection of the inertia mass \( m_{in} \). In addition, a vibration absorber with a large damping ratio is also shown in Fig. 12. Once the structure of the vibration absorber is well tuned, the vibration absorber is mounted on the cutting tool. Fig. 13 shows the real part and magnitude of the frequency response function of the cutting tool with the tuned vibration absorber. Comparing Fig. 11(a) with Fig. 13(a), the magnitude of the negative real part of the frequency response function is greatly reduced when the tuned vibration absorber is attached to the cutting tool. Hence, the experimental results shown in Figs. 11(a) and 13(a) are consistent with the simulation results demonstrated in Figs. 2(a) and 6(a).

Fig. 14(a) shows the accelerometer signal of the cutting tool without the vibration absorber during the turning of an aluminum bar. The cutting parameters are selected as: a cut

![Figure 10: Schematic diagram of the experimental set-up.](image)

![Figure 11: Frequency response function of the cutting tool: (a) real part; (b) magnitude.](image)

![Figure 12: Magnitude of the frequency response function of a vibration absorber.](image)

![Figure 13: Frequency response function of the cutting tool with a vibration absorber: (a) real part; (b) magnitude.](image)
width of 0.5 mm, a feed of 0.05 mm/rev and a spindle speed of 1200 rpm. The accelerometer signal becomes unstable within a short period of time due to the occurrence of strong chatter. Next, Fig. 14(b) shows the accelerometer signal of the cutting tool with the well-tuned vibration absorber (Fig. 12) using the same cutting parameters as shown in Fig. 14(a). The accelerometer signal becomes stable due to the well-tuned vibration absorber (Fig. 12) mounted on the cutting tool. Fig. 15 shows the accelerometer signal of the cutting tool with the vibration absorber using a cut width of 3 mm. It is seen that the accelerometer signal is still stable and therefore cutting stability is greatly enhanced six times, through this approach. In the foregoing discussion, the application of the tuned vibration absorber to the suppression of chatter in turning is clearly demonstrated.

6. Conclusions

In this paper, the use of a piezoelectric inertia actuator mounted on a cutting tool and acting as a tuned vibration absorber for the suppression of chatter in turning operations has been explored. The piezoelectric inertia actuator must satisfy the following criteria to suppress chatter in turning effectively. First, the natural frequency of the vibration absorber must be equal to the natural frequency of the cutting tool. Next, the vibration absorber must have a larger damping ratio. Experimental results have shown that chatter in turning can be greatly suppressed and the cutting stability increased six times by using the technique proposed in this study.

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References