Regenerative chatter control in turning process using constrained viscoelastic vibration absorber

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Abstract. This paper focuses on utilization of viscoelastic material to control regenerative chatter in turning process. Self-induced vibrations can lead to the condition called regenerative chatter which produces violent relative vibrations between cutting tool and workpiece and reduces tool life and productivity. The regenerative chatter in turning processes can be controlled by maximizing negative real part of the frequency response function of cutting tool structure. In order to achieve this, a constrained viscoelastic vibration absorber (CVVA) is used. A CVVA consists of a viscoelastic material, such as natural rubber, which is sandwiched between two similar or dissimilar metallic layers. The CVVA is used as a cantilever beam whose fundamental natural frequency is tuned to the natural frequency of the dominant mode of cutting tool/tool holder. The optimum stiffness and damping coefficient of the CVVA are found using a numerical optimization technique and these optimal values are used to find the dimensions of CVVA. The resulting natural frequency of CVVA is verified using finite element simulation software ANSYS. The effectiveness of CVVA in controlling regenerative chatter in a compact CNC lathe is also analysed by constructing stability lobes which are plots of depth of cut vs spindle speed.

1. Introduction

Vibrations caused during the machining process, can be responsible for poor accuracy and surface finish of machined components. It also reduces tool life and increases noise level. As the unstable vibration in the machining process tends to cause fluctuating load, the premature cutting tool fracture may occur. Yusuf Altintas (2000) has carried out linear stability analysis of orthogonal machining process in Laplace domain. It was also proved that the critical width of cut in turning process is inversely proportional to the negative real part of frequency response function (FRF). In order to control chatter, passive vibration control techniques have been used by several researchers. Den hartog (1985) introduced the idea of using an auxiliary spring mass system as a tuned mass damper (TMD). The author used equal peaks of magnitude of FRF as a major criterion to find the optimum parameters of TMD. Rivin et al (1989) found optimum tuning parameters of TMD from stability analysis of two degree of freedom system, which consists of a tool and an absorber. The iterative approach was used to apply
various conditions of stability of the system. Tewani (1995) et al compared the effectiveness of passive and active TMD, by calculating the stability boundaries. The piezoelectric material was used as an actuator, which was controlled using a feedback control algorithm. They also discussed different types of control algorithm for the effective use of active TMD. Tarng et al [2000] described that TMD can reduce the magnitude of the turning tool and showed chatter can be suppressed effectively when the TMD is attached to the vibrating main structure. The effective way to control the induced vibration across the system is to provide the optimally Tuned TMD. Sims (2007) presented a new closed form solution to find optimum tuning parameters of TMD for machining process. Numerical optimization technique was adopted to optimize TMD for damped main system. To prove the effectiveness of TMD, a time domain simulation of milling process was performed. Amir Rashid (2008) focused on formulating the design parameters for the TMD directly applying it on the primary system. Yang et al (2010) proposed multiple TMDs to control chatter in turning process. The optimum parameters of TMD were found using minimax numerical optimization technique. Da Silva (2013) showed that the vibration due to the regenerative chatter can be suppressed by using the TMD designed from the tuning parameters, obtained from the minimization of normalized amplitude. Rubio, L., et al (2013) determined the optimum values for TMD for boring bar, by considering boring bar and TMD as continuous and discrete system respectively. Saravanamurugan et al (2013) found optimum parameters of sky hook TMD to control chatter in machining processes. The optimum parameters are found using numerical optimization technique. Saravanamurugan et al (2018) used machine learning algorithm to predict chatter in boring process. The support vector machine algorithm was used to classify the feature. Kishore et al (2018) designed an online condition monitoring system to control chatter in turning process. The dynamics of the cutting tool were controlled by using transmissibility approach and half power band width method. Bansal et al (2018) used receptance coupling method to find the tuning parameters and optimal placement of TMD for a boring tool. The boring bar-TMD system was treated as Euler Bernoulli beam carrying a discrete spring-mass-damper system. Though many theoretical approaches have been suggested for optimisation of TMD, the effective practical implementation of vibration absorbers remains a challenging task. In order to solve this problem, this paper studies the utilization of constrained viscoelastic vibration absorber as a TMD. The CVVA which is simple in construction and can be effectively attached with the metal cutting system for control of regenerative chatter.

2. Chatter Control Using Dynamic Vibration Absorber

The regenerative chatter occurs due to the interaction between cutting process and the structural dynamics of cutting tool. In order to control the regenerative chatter, an auxiliary vibrator system called TMD is attached to a primary system as shown in figure 1. The primary system is subjected to a dynamic cutting force due to the variation in chip thickness. By applying principle of dynamics, the governing equation of motion of the 2-DOF parameter system is obtained as (Singiresu S Rao (2010)).

![Figure 1. Dynamic model of 2-DOF system](image-url)
where, \( m_t \), \( c_t \) and \( k_t \) are the mass, damping and stiffness of the primary system; \( m_a \), \( c_a \) and \( k_a \) are the mass, damping and stiffness of TMD; \( X_t(s) \) and \( X_a(s) \) are the amplitudes in laplace domain of the primary and TMD systems respectively; \( F(s) \) is the external force in laplace domain acting on the primary system. Using the following non-dimensional parameters provided in table 1, the expressions for real and imaginary part of the FRF of the primary system are expressed in equation (2) and (3) (Singiresu S Rao (2010)).

### Table 1. Non-Dimensional Parameters

| Parameters                   | Equations                      |
|------------------------------|--------------------------------|
| Mass ratio                   | \( \mu = m_a/m_t \)            |
| Forced frequency ratio       | \( g = \omega/\omega_t \)      |
| Main system natural frequency| \( \omega_t = \sqrt{(k_t/m_t)} \) |
| Main system damping ratio    | \( \xi_t = c_t/2\omega_t m_t \) |
| Absorber natural frequency   | \( \omega_a = \sqrt{(k_a/m_a)} \) |
| Absorber damping ratio       | \( \xi_a = c_t/2\omega_a m_a \) |
| Frequency ratio              | \( f = \omega_a/\omega_t \)    |

The real and imaginary part of the FRF are provided as,

\[
\text{Re}(G(g)) = \frac{(t^2 - g^2)((1 - g^2)(t^2 - g^2) - \mu t^2 g^2) - 4\xi_t^2 t g(1 - g^2 + \mu g^2) + 2\xi_a t g(f^2 - g^2)}{(1 - g^2)(t^2 - g^2) - \mu t^2 g^2 - 4\xi_t^2 g(t^2 - g^2)^2 + 4\xi_a t g(1 - g^2 + \mu g^2) + \xi_a g(f^2 - g^2))^2} \tag{2}
\]

\[
\text{Im}(G(g)) = \frac{-(t^2 - g^2)2\xi_t^2 f g(1 - g^2 + \mu g^2) + (2\xi_a^2)(t^2 - g^2)) + 2\xi_a t g(1 - g^2)(t^2 - g^2) - \mu^2 g^2 - 4\alpha \xi_t \xi_a g^2}{((1 - g^2)(t^2 - g^2) - \mu t^2 g^2 - 4\xi_t^2 g(t^2 - g^2)^2 + 4\xi_a t g(1 - g^2 + \mu g^2) + \xi_a g(f^2 - g^2))^2} \tag{3}
\]

The expression for the critical width of cut is determined as, (Yusuf Altintas (2000)) and Schmitz et al (2008).

\[
a_c = -\frac{1}{2K_f \text{Re}(G(\omega))} \tag{4}
\]

where, \( a_c \) is the critical width of cut, \( K_f \) is the cutting constant, \( \text{Re}(G(\omega)) \) is the real part of the transfer function of metal cutting system and \( \omega \) is the chatter frequency.

From equation (4), one can infer that the critical width of cut is inversely proportional to the normalized amplitude of real part of the FRF. As the terms of \( a_c \) consists of the negative sign, it is important to maximize the negative real part of FRF \( \text{Re}(G(g)) \) in order to increase the width of cut.

### 3. Optimisation of TMD using Genetic Algorithm

The optimisation of TMD is carried out using MATLAB with the genetic algorithm (GA) as optimisation tool. The tuning parameters which provides response with two equal troughs for the
negative real part of $G(g)$, is considered as the optimum. The criteria to obtain equal trough of the Re($G(g)$) is obtained through finding the maximum value across the minimum values of Re($G(g)$). This criterion is met through two processes, where the minimum value of Re($G(g)$) is found analytically through an iterative process and these results are fed as input values to GA, thereby it performs maximisation on the available minimums. The frequency response of the primary system, coupled with the optimum TMD is shown in figure 2.

![Figure 2](image-url)

**Figure 2.** Frequency response of main system with and without TMD

4. Design of Constrained viscoelastic vibration absorber

In this section, the design procedure of a CVVA, which can be used as a TMD, is explained. The CVVA is a composite beam that consists of a viscoelastic layer sandwiched between two metallic constrained layers, as shown in figure 3. The two stiff metal layers (Base and Constraining) are used to produce shear deformation in viscoelastic layer in such a way that the energy dissipation occurs in the system effectively. The fundamental natural frequency of the CVVA is tuned to the natural frequency of the primary system. Moreover the dynamics of the CVVA is also characterized by its loss factor, hence the dimensions of the CVVA are found based on the values of natural frequency and loss factor.

The expression for the fundamental natural frequency of CVVA is given as follows, (Mead 1999)

$$\omega_a = \frac{8\beta^4}{m_a l^3}$$  \hspace{1cm} (5)

where $\beta$ is the wave number, $l$ and $m_a$ are length and mass of CVVA respectively and $B$ is the flexural rigidity.

The overall mass ($m_a$) of the composite beam is expressed using the following relation,

$$m_a = (\rho_1 a_1 + \rho_2 a_2 + \rho_3 a_3) l$$  \hspace{1cm} (6)

where $\rho_1$, $a_1$ are the density and cross-sectional area of the base layer, $\rho_2$, $a_2$ are the density and cross-sectional area of the constrained layer respectively; $\rho_3$ and $a_3$ are the density and cross-sectional area of the viscoelastic layer.

The flexural rigidity ($B$) of the composite beam depends on shear ($Y$) and geometric ($X$) parameters. The shear and geometric parameters are the functions of material and geometric properties respectively. The expression for flexural rigidity, shear and geometric parameter are provided in equations (7), (8) and (9) respectively.

$$B = \frac{1 + YX(1 + X(1 + e^2))}{(1 + 2X + X^2(1 + e^2))(E_1 I_1 + E_3 I_3)}$$  \hspace{1cm} (7)

...
\[ Y = \frac{E_1a_1E_2a_2t^2}{(E_1a_1+E_3a_3)(E_1I_1+E_2I_2)} \]  
\[ X = \frac{Gb(E_1a_1+E_3a_3)}{t_2k^2E_1a_1E_3a_3} \]  

where, \( E_1, I_1 \) are the Young’s modulus and moment of inertia of the base layer about its neutral axes; \( E_3, I_3 \) are the Young’s modulus and moment of inertia of constraining layer about its neutral axes respectively; \( e \) is the shear modulus of viscoelastic layer.

The loss factor (\( \eta \)) is used to quantify the damping effect of the composite beam, which depends on shear and geometric parameters.

\[ \eta = \frac{eXY}{1+X(2+Y)+(X^2(1+e^2))(1+Y)} \]

### 4.1. Algorithm for CVVA

The algorithm which is used to find the dimensions of the CVVA is explained as follows,

1. Find the natural frequency (\( \omega_a \)) and the mass (\( m_a \)) of the damper for the required mass ratio (\( \mu \)) and the corresponding optimum tuning parameters (\( f \) and \( \eta \)), using expressions from table 1.
2. The viscoelastic material, with high loss factor (\( \eta \)) across the working temperature and frequency is selected.
3. The flexural rigidity (\( B \)) is calculated using the geometric (\( Y \)) and modified shear parameter (\( X \)) using the equations (7), (8) and (9) respectively.
4. The different combination of the length (\( l \)), breadth (\( b \)) and thickness of the layers (\( t_1, t_2 \) and \( t_3 \)) with the shear modulus and loss factor are iterated till the combination satisfies the required frequency and mass of the CVVA.

#### 4.1.1 Experimental analysis

Using the above algorithm, a CVVA is designed for a compact 2-axis lathe machine and the dynamic characteristics of the lathe machine is found using impact hammer test. The experimental setup to obtain the natural frequency and damping ratio of the primary system (Denford compact turning lathe), is shown in figure 4.
The system was perturbed with a short impulse of force by means of an impact hammer consisting a load cell and the induced response of the system, was picked up by an accelerometer and further the response was analysed using photon data acquisition system and RT Pro software for obtaining the frequency response function (FRF) which is shown in figure 5.

![Figure 4. Experimental setup](image)

![Figure 5. FRF provided by the Impact Test](image)

The datas obtained from the impact hammer test are depicted in the table 2 and these values are used to design the constrained viscoelastic vibration absorber (CVVA).

| Parameters                      | Values(units)     |
|---------------------------------|-------------------|
| Mass of the Tool post (m_t)    | 1.005 kg          |
| Damping ratio (ξ_t)             | 0.0488            |
| Natural frequency (ω_t)         | 377.208 Hz        |
| Cutting constant (K_f)          | 1.462e9N/m²       |
Having got the parameters of the primary system, the dimension of the CVVA are found for the mass ratio of 1% (μ) and the results are provided in table 3.

| Parameters                        | Values(units) |
|-----------------------------------|---------------|
| Mass of the CVVA (m_a)            | 0.01005 kg    |
| Natural frequency of the CVVA (ω_a)| 390 Hz        |
| Optimum frequency ratio (f)       | 1.0601        |
| Optimum damping ratio for CVVA (ξ_a)| 0.209         |
| Thickness of base plate (t_1)     | 1mm           |
| Thickness of viscoelastic layer (t_2)| 24mm         |
| Thickness of constraining layer (t_3)| 1mm           |
| Width of CVVA (b)                 | 15mm          |
| Length of CVVA (l)                | 70mm          |
| Loss factor for viscoelastic layer (η)| 0.1045       |

4.2. Finite Element Analysis
In order to validate the design procedure of CVVA, a finite element package Ansys is used. The constraining layer of cantilever composite beam was meshed using SOLID 45 element, whereas the viscoelastic layer was meshed using VISCO89 element. The entire finite element model consists of 18,200 number of elements. The material used for the two metal layers is steel (E=200MPa; ρ = 7850 kg/mm^3) and the viscoelastic layer is 3M112 (Amir Rashid 2008). After assigning the material properties and necessary boundary conditions the modal analysis was performed. The result obtained from the simulation is shown in figure 6.

![Fundamental mode of composite beam](image)

The value of the natural frequency of the CVVA is found to be 365Hz, which deviates from the analytical result by 6.4%. This shows CVVA can be accurately tuned with the main system based on the tuning parameters which are obtained using numerical optimisation.
5. Construction of Stability Lobes
The prediction of improvement in machining stability is done by constructing stability lobes which are the graphical representation of width of cut \( (a_c) \) and spindle speed \( (N) \). The stability lobe diagrams are used to predict and control the regenerative chatter produced under machining. The procedure which was suggested by Yusuf Altintas (2000) is adopted to construct the stability lobes. The flowchart shown in figure 7, explains the various processes that are involved in the construction of stability lobes. The flow starts by obtaining the tuning parameters, dynamic parameters and machining constants (Schmitz, T. L., and Smith, K. S. (2008) of the system. The real and imaginary parts of the FRF is calculated by using the equation (2). The critical width of cut and spindle speed are computed using the relations, shown in equations (8), (9) and (10). To plot the stability lobe a MATLAB code is formulated for mass ratios 1% and 5%.

![Flowchart for stability lobe construction](image)

**Figure 7.** Flowchart for stability lobe construction

The resulting stability lobes are shown in figure 8, which shows that the width of cut can be significantly increased with the help of the tuned mass damper (TMD) in machining process. The improvement in the width of cut increases with the increase in mass ratio.
6. Conclusion
The constrained viscoelastic vibration absorber is designed for the purpose of controlling regenerative chatter in external turning process. The design of CVVA is carried out by tuning its fundamental natural frequency to the natural frequency of tool holder system and the optimum tuning parameters of the absorber system are found using numerical optimization and the dimensions of the CVVA are obtained based on these tuning parameters. The design procedure is developed to find the natural frequency of CVVA analytically. The finite element simulation is used to verify the natural frequency of CVVA and the simulation result is found to be in agreement with the analytical result. The stability lobes, which are the plots of spindle speed versus depth of cut, for the 2-DOF system are constructed and it shows improvement in stability margin by 25%, for low mass ratio($\mu=0.01$) by 1.17% and for high mass ratio($\mu=0.05$) by 2.5%.

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