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A STUDY OF THE DESIGN OF A CANTILEVER TYPE MULTI-D.O.F. DYNAMIC VIBRATION ABSORBER FOR MICRO MACHINE TOOLS

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Abstract

This paper presents a study of the design of a cantilever type multi-d.o.f. damper, which can be used on micro machine tools, to reduce resonant vibrations. The proposed micro dynamic damper consists of three cantilever beams and a rigid body suspended from the cantilever beam. For dynamic analysis and design synthesis, a vibrating system with the dynamic damper is mathematically modeled as a lumped parameter model, in which cantilever beams and the rigid body are idealized as springs and a mass respectively. The natural frequencies and harmonic response of the mathematical model are obtained by using modal analysis theory. From the results, the spring constants and mass of the dynamic damper which reduce the resonant vibrations of the main system can be determined. A case study shows the proposed cantilever typed micro dynamic damper can successfully reduce resonant vibrations of the main machine, in the three different directions, at specified resonant frequencies.

1. INTRODUCTION

A simple dynamic damper is made by attaching mass-spring to the main vibrating system to reduce resonant vibration of it. There have been made many researches on dynamic damper since it is one of useful means to control vibrations in structural systems. Choi[2] has done a study on geometrical design theory of a 6-d.o.f. dynamic damper. Nayfeh[3] and Lee[4] have researched the optimization of multi-d.o.f. tuned mass dampers.

In this study a cantilever type dynamic damper is proposed in order to reduce the resonant vibrations of micro-machine tool during machining operations.

The micro-machine tools or micro-factory is increasingly concerned with rapid development of miniaturizing or down-sizing in machine elements. Since micro machine tools are relatively small in size, there is not enough room for extra components.

Therefore small size or miniaturized dynamic damper is more desirable for micro machine tools in order to reduce excessive vibrations during machining operations.

2. VIBRATION ANALYSIS OF MULTI-D.O.F. DYNAMIC DAMPER SYSTEM

2.1 Mathematical Modeling

A schematic of a structural vibrating system with three DOF (Degree-Of-Freedom) is shown in Fig. 1. The main structural vibrating system, as shown in Fig. 1, consists of a mass and three linear springs in the independent direction each other.
It is assumed to experience excessive vibrations in its resonant conditions, which are required to be reduced. For this purpose, a multi-DOF dynamic damper of cantilever type with end mass, whose schematic is shown in Fig. 2, are introduced into the main vibrating system. Consequently, the combined system becomes 6-DOF.

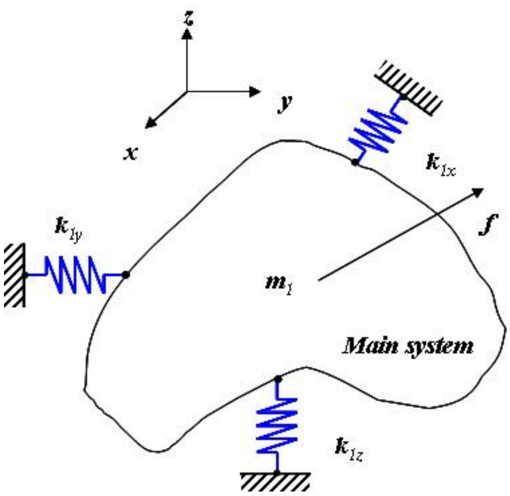


Fig. 1 The main vibrating system with 3-d.o.f.

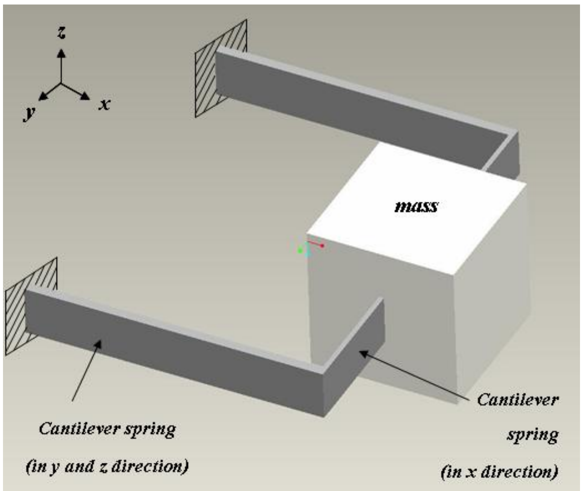


Fig. 2 Schematic of the dynamic damper

And the schematic of the combined dynamic damper system is shown in Fig. 3. Cantilevers of the dynamic damper can be appropriately designed to have required stiffness in each direction. And it is assumed that there are no coupling effects between x-, y- and z-directional springs.

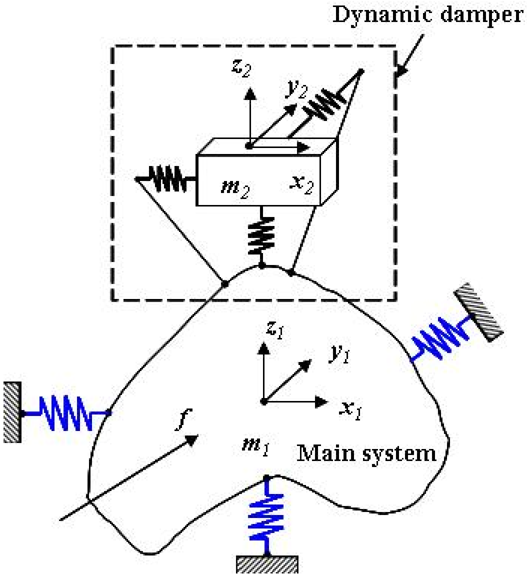


Fig. 3 The schematic of 6-d.o.f. vibrating system with dynamic damper

The main vibrating system combined with the dynamic damper was modeled mathematically as a lumped parameter model with 6 DOF as shown in Fig. 4. In the model, symbols, m, k, and f stand for mass, spring constant and exciting force, respectively. Symbols, x, y, and z represent degrees of freedom or coordinates respectively. And subscripts 1 and 2 mean mass of the main vibrating system mass and dynamic damper respectively.

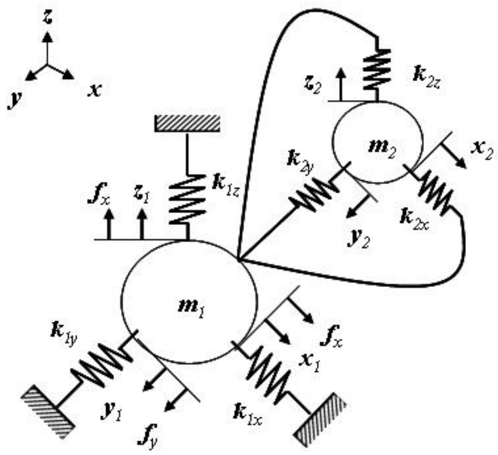


Fig. 4 Mathematical model of 6-d.o.f. vibrating system with dynamic damper

2.2 Equation of Motion

The equations of motion for the 6-DOF dynamic damper system can be derived as follows by using Newton's second law.

$$[M]\{\ddot{u}\}+[K]\{u\}=\{f\} \tag{1}$$

Where, $[M]$ and $[K]$ are mass and stiffness matrices respectively. $\{u\}$ and $\{f\}$ are displacement and exciting force vectors respectively.

$$[u]=[x_1\ y_1\ z_1\ x_2\ y_2\ z_2]^T \tag{2}$$

$$[f] = [f_x \; f_y \; f_z \; 0 \, 0 \, 0]^T \tag{3}$$

$$[M] = \begin{bmatrix} m_1 & 0 & 0 & 0 & 0 & 0 \\ 0 & m_1 & 0 & 0 & 0 & 0 \\ 0 & 0 & m_1 & 0 & 0 & 0 \\ 0 & 0 & 0 & m_2 & 0 & 0 \\ 0 & 0 & 0 & 0 & m_2 & 0 \\ 0 & 0 & 0 & 0 & 0 & m_2 \end{bmatrix} \tag{4}$$

$$[K] = \begin{bmatrix} k_{1x} + k_{2x} & 0 & 0 & -k_{2x} & 0 & 0 \\ 0 & k_{1y} + k_{2y} & 0 & 0 & -k_{2y} & 0 \\ 0 & 0 & k_{1z} + k_{2z} & 0 & 0 & -k_{2z} \\ -k_{2x} & 0 & 0 & k_{2x} & 0 & 0 \\ 0 & -k_{2y} & 0 & 0 & k_{2y} & 0 \\ 0 & 0 & -k_{2z} & 0 & 0 & k_{2z} \end{bmatrix} \tag{5}$$

2.3 Solution of Equation of Motion

When harmonic exciting force $\{f\} = \{F\}e^{i\omega t}$ is applied on the system, its harmonic response $\{u\} = \{U\}e^{i\omega t}$ can be obtained as followings.

$$\begin{Bmatrix} X_1 \\ Y_1 \\ Z_1 \\ X_2 \\ Y_2 \\ Z_2 \end{Bmatrix} = \left[[K] - \omega^2 [M] \right]^{-1} \begin{Bmatrix} F_x \\ F_y \\ F_z \\ 0 \\ 0 \\ 0 \end{Bmatrix} \tag{6}$$

3. DESIGN OF MICRO DYNAMIC DAMPER

3.1 Approach of Dynamic Damper Design

In this study, it is required that only resonant vibrations in the three independent translational directions should be reduced. This requirement can be accomplished by letting response of the main system be zero in the Eq. (6) as follows.

$$\left[[K]-\omega^2[M]\right]\begin{Bmatrix} 0 \\ 0 \\ 0 \\ X_2 \\ Y_2 \\ Z_2 \end{Bmatrix} = \begin{Bmatrix} F_x \\ F_y \\ F_z \\ 0 \\ 0 \\ 0 \end{Bmatrix} \tag{7}$$

Solving the Eq. (7), following dynamic damping criteria is obtained.

$$k_{2i} - \omega^2 m_2 = 0, \qquad (i = x,\ y,\ z) \tag{8}$$

In case of reducing three independent vibration modes in x-, y-, and z-directions at resonant frequencies, Ω1, Ω2, and Ω3, Eq. (8) can be rewritten as follows.

$$k_{2x} - \Omega_1^2 m_2 = 0 \tag{9a}$$

$$k_{2y} - \Omega_2^2 m_2 = 0 \tag{9b}$$

$$k_{2z} - \Omega_3^2 m_2 = 0 \tag{9c}$$

From this criterion, the design parameters of multi-DOF dynamic damper can be determined.

3.2 Design Example

Design example is to design a multi-DOF dynamic damper that can reduce the resonant vibrations of a main vibrating system, whose modeling parameters are given in Table 1.

Table 1. Material properties of structure

m_1 (kg)	k_{1x} (N/m)	k_{1y} (N/m)	k_{1z} (N/m)
10	0.5×10^6	3×10^6	10×10^6

Three natural frequencies of 35.6, 87.2, and 159.2 Hz can be determined from modal analysis of the main vibrating system. Vibration modes corresponding to these natural frequencies are x-, y-, and z-directional vibration modes accordingly.

Design parameters of dynamic damper to reduce the resonant vibrations can be determined by substituting those frequencies into Eq. (9) with mass of the dynamic damper of 1 kg. In this example, the mass of dynamic damper was assigned around 10 % of the main system mass.

From the frequency response curve shown in Fig. 5, it is apparent that peak vibrations at three resonant frequencies are successfully diminished.

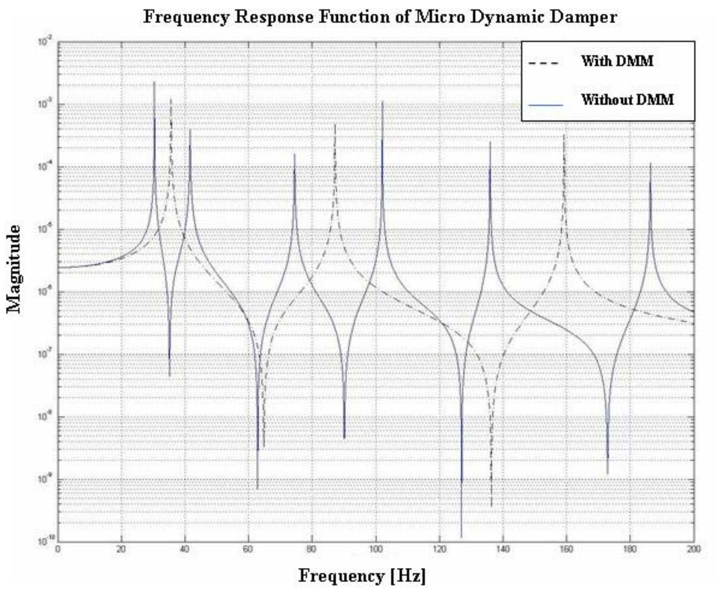
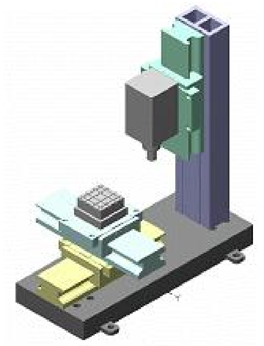


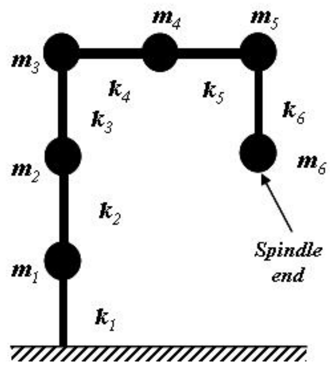
Fig. 5 FRF of 6-d.o.f. model

4. DESIGN OF MICRO DYNAMIC DAMPER

A modular micro milling structure and its simplified model are shown in Fig. 6. Natural frequencies for this simplified model were determined by using finite element analysis as shown in Table 2.



(a) Micro milling machine



(b) Simplified lumped parameter model

Fig. 6 The image of micro milling machine

Table 2. Natural frequency of μ-milling machine without dynamic damper

Mode	Natural frequency (Hz)	Mode	Natural frequency (Hz)
1	85.3	6	831.8
2	94.0	7	1375.3
3	205.3	8	1497.8
4	217.0	9	2144.9
5	535.4	10	2400.6

The three modes corresponding to 217.0, 535.4, and 831.8 Hz are critical modes in z-, x-, and y-directions. Therefore vibrations at these frequencies are required to be diminished. When the mass of dynamic damped was set 10 % of the main vibrating system mass, parameters of dynamic damper were designed as shown in Table 3. The frequency response function shown in Fig. 7 represents that the resonant vibrations at the three critical frequencies were perfectly decreased.

Table 3 Designed parameters of dynamic damper for a micro milling machine

m_1 (kg)	k_{1x} (N/m)	k_{1y} (N/m)	k_{1z} (N/m)
0.413	4673761	11281001	767767

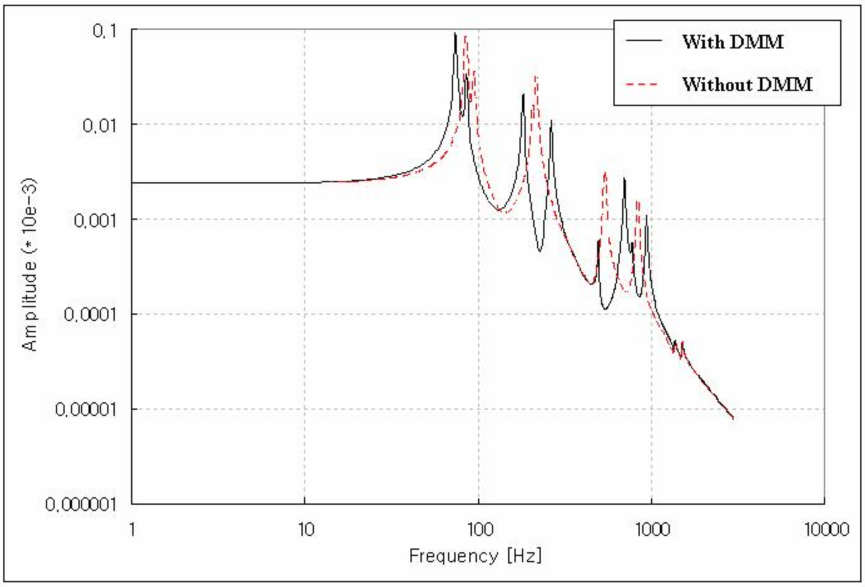


Fig. 7 Comparison of frequency response function

5. CONCLUSION

In this study, a multi-DOF (Degree-Of-Freedom) dynamic damper of cantilever type with end mass was introduced to reducing micro-machine tool vibrations. The micro dynamic damper consists of a single mass having three DOF and flexible cantilevers. Dynamic characteristics of the dynamic damper become different with designs of cantilever beam and end mass. The main vibrating system, a micro-milling machine in this case, has greatly diminished in the resonant vibrations at its several natural frequencies with the aid of the properly designed multi-DOF micro dynamic damper.

Therefore multi-DOF dynamic damper of cantilever type with end mass can be successfully applicable to reducing unwanted resonant vibrations in micro machine tools.

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