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Damping of transient vibrations of head on top of flexible column of milling machine using electromagnetic hybrid dynamic vibration absorber

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Abstract
The purpose of this study was to develop a novel hybrid dynamic absorber that uses an electromagnetic force to damp the transient vibrations of a spindle head on the column of a milling machine caused by the deceleration of the column on the bed. The hybrid dynamic absorber was fixed to the weight of the primary vibration system placed on a table, which was used to model the spindle head on the column. To mimic the deceleration of the column in the present experiments, the table was quickly stopped. When the proposed hybrid dynamic absorber tuned by an electromagnetic force by optimized proportional–derivative control was used, the time required for the convergence of the residual transient vibrations was reduced by 80%, and the amplitude of the first wave of the transient vibrations was reduced by 65% in comparison with that without the absorber.

Damping, Transient Vibration, Spindle Head, Hybrid Dynamic Vibration Absorber, Electromagnetic Force

1. Introduction

During the process of milling molds using large machining centers, vibration marks may be generated on the finished surface by transient vibrations of the spindle head on top of the long column caused by high deceleration of the column on the bed. The generated vibration marks diminish the quality of the finished surface of the mold and must be removed during final processing. A conventional passive dynamic absorber may damp most transient vibrations but cannot damp the first wave of the transient vibrations, which has the largest amplitude [1]–[3]. The purpose of this study was to develop a novel dynamic vibration absorber to damp the transient vibrations of the spindle head on the column of a milling machine caused by the deceleration of the column on the bed. The natural frequency and damping ratio of the developed hybrid dynamic absorber tuned using an electromagnetic force were investigated. The effect of the magnetic force generated from the acceleration signal of the table on the reduction of the first wave of the transient vibrations of the primary system was also investigated.

2. Electromagnetic hybrid dynamic vibration absorber

A dynamic absorber that can damp the transient vibrations of the spindle head must meet the following criteria: (1) It can damp the first wave of the transient vibrations. (2) It is compact in size. (3) Its free vibration characteristics can be controlled by an electromagnetic force. Figure 1 shows a schematic of the proposed electromagnetic hybrid vibration absorber. The bottom of the dynamic vibration absorber is a ready-made voice coil motor. The weight on the voice coil motor was fixed to the moving coil of the voice coil motor and supported by parallel springs to allow it to move horizontally. Consequently, when a current passes through the moving coil, a magnetic force is generated and moves the weight horizontally. The displacement of the weight in the horizontal direction can be measured using a strain gauge adhered to the surface of the flat spring.

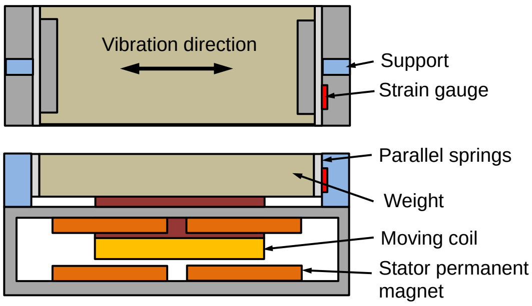

Figure 1. Structure of electromagnetic hybrid vibration absorber

3. Experimental method

Figure 2 shows a schematic of the experimental apparatus. The primary vibration system consists of a weight, parallel springs, and a base plate, and its free vibration characteristics are given in Table 1, together with those for a hybrid dynamic vibration absorber without an electromagnetic force. The primary vibration system with the hybrid dynamic vibration absorber is fixed on a table and is driven by the servomotor on the bed. The vibrational displacement of the weight of the hybrid vibration absorber is measured as a voltage using a strain gauge and a strain amplifier. The measured displacement signal and the velocity signal of the weight, which is generated using a differential circuit, are amplified and subsequently combined using a proportional–derivative (PD) circuit, and the combined signal is input into a current amplifier. Additionally, the acceleration signal is measured by a sensor on the table. The measured signal is amplified and is also input into the current amplifier.

Table 1 Free vibration characteristics (modal parameters)

	Frequency [Hz]	Damping ratio [%]	Mass [kg]	Stiffness [kN/m]
Primary system	43.36	0.64	4.31	320
DVA	15.91	11.23	0.46	4.6

DVA : Dynamic Vibration Absorber

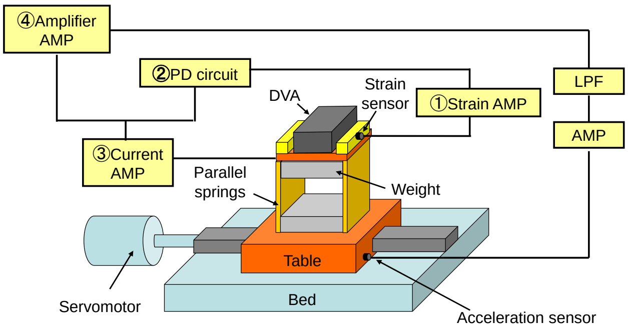

Figure 2. Schematic of experimental apparatus

4. Optimization of parameters of PD electric control circuit

Figure 3 shows a physical model of the table-mounted primary vibration system with the electromagnetic hybrid dynamic vibration absorber shown in Fig. 2. The equations of motion for the vibration system are

$$\left.\begin{aligned} M\ddot{x}_1 &= -k_1x_1 - c_1\dot{x}_1 + f_a + f_v \\ m\ddot{x}_2 &= -f_a \\ f_a &= k_2(x_2 - x_1) + c_2(\dot{x}_2 - \dot{x}_1) + f_c \\ f_c &= K_v(\dot{x}_2 - \dot{x}_1) + K_p(x_2 - x_1) \end{aligned}\right\}, \qquad (1)$$

where K_v is the velocity coefficient and K_p is a proportionality coefficient. The frequency response function $X_1(\omega)/F_v(\omega)$, where ω is the angular frequency, is derived by taking the Fourier transform of Eq. (1), and it is plotted in Fig. 4. The optimized coefficients K_p and K_v obtained by trial and error are 23000 and 45, respectively.

$$\frac{X_1}{F_v} = \frac{\left\{-m\omega^2 + i(c_2 + K_v)\omega + (k_2 + K_p)\right\}}{B_{(\omega)}} \qquad (2)$$

$$B_{(\omega)} = \left[(k_1 - M\omega^2)\left\{(k_2 + K_p) - m\omega^2\right\} - m(k_2 + K_p)\omega^2 - c_1(c_2 + K_v)\omega^2\right] + i\left[c_1\left\{(k_2 + K_p) - m\omega^2\right\}\omega + (c_2 + K_v)\left\{k_1 - (m + M)\omega^2\right\}\omega\right]$$

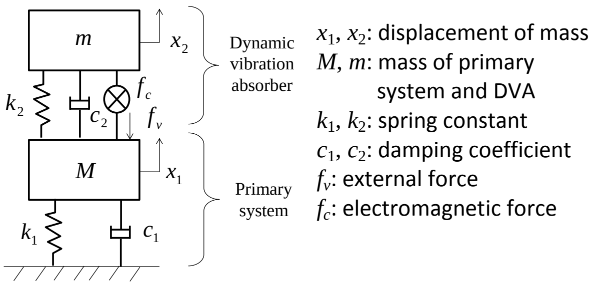

Figure 3. Model of table-mounted vibration system

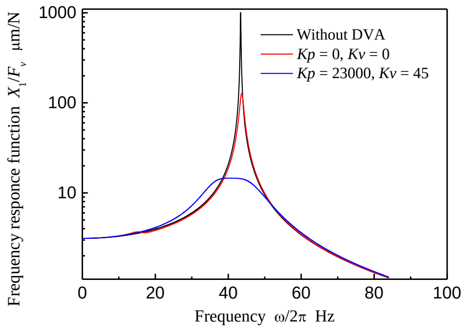

Figure 4. Frequency response function for primary vibration system

5. Experimental results

Figure 5 shows the transient vibration amplitude for the primary vibration system and the acceleration of the table after rapid deceleration of the table without an electromagnetic force. The amplitude of the first wave of the transient vibration is 150 μm, and the amplitude decreases to 5% (7.5 μm) of this value in 2 s. The acceleration has the appearance of a pulse. Figure 6 shows the transient vibration amplitude for the primary vibration system with an applied electromagnetic force. The amplitude of the first wave of the transient vibration with an electromagnetic force for the hybrid dynamic vibration absorber including optimized PD control (Fig. 6(a)) is almost the same as that without the electromagnetic force (Fig. 5), but it rapidly decreases to 7.5 μm in 0.4 s. When an additional electromagnetic force is applied to compensate for the inertial force of the weight of the primary vibration system, the amplitude of the first wave is 50 μm (Fig. 6(b)).

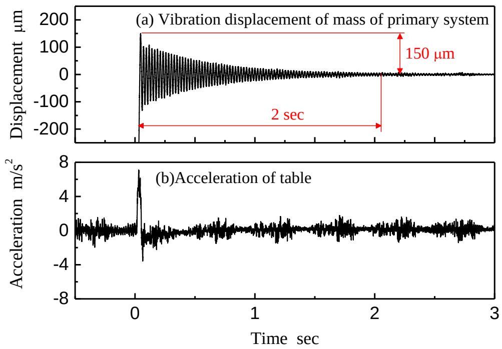

Figure 5. Transient vibration amplitude for primary vibration system and acceleration of table without applied electromagnetic force

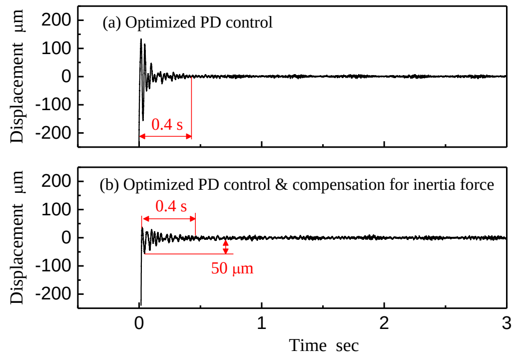

Figure 6. Transient vibration amplitude for primary vibration system with applied electromagnetic force

6. Conclusions

The use of the proposed hybrid dynamic absorber with an electromagnetic force tuned by optimized PD control reduced the time required for the convergence of residual transient vibrations by about 80% in comparison with that without the absorber. Furthermore, the amplitude of the first wave of the transient vibrations was reduced by about 70% by the pulsed electromagnetic force generated by the acceleration of the table.

References

[1] Tanaka N and Kikushima Y 1986 Trans. of the Japan Society of Mechanical Engineers, C (in Japanese) **52** 3175-3183

[2] Yamaguchi H 1987 Trans. of the Japan Society of Mechanical Engineers, C (in Japanese) **53** 561-568

[3] Chatterjee S 2010 J. of Sound and vibration **329** 5397-5414

[4] Inoue T, Ishida Y and Sumi M 2004 Trans. of the Japan Society of Mechanical Engineers, C (in Japanese) **70** 2565-2572