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A study on parameter optimization for passive vibration damping system by giant magnetostrictive material

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Abstract. In order to achieve vibration damping system with low energy consumption and with high reliability, passive damping is preferred in various system. Due to improvement of manufacturing method of Giant magnetostrictive material (GMM) and some advantages over piezoelectric element that has been used for passive vibration damping system (PVDS), a PVDS using GMM is being studied intensively in recent years, which utilize the Villari effect of GMM. However, study on such a system is not enough compared to other PVDS and parameter design method is not established enough. In our study, fundamental 1-DOF model is introduced and fixed point method is applied to the model in order to study the parameter optimization for PVDS using GMM. According to the optimization condition derived from fixed point method, parameter study is carried out and an appropriate structure of the system for effective vibration damping is proposed. Furthermore, some numerical examples for the proposed method are analyzed and the analyses results show the validity of the proposed method.

1. Introduction

In recent years, many of the satellite have become larger in size due to sophistication of mission. On the other hand, there are restrictions on loads that can be mounted on rockets, and weight reduction is also required to satisfy this. In order to reduce the weight of the system, a flexible structure is mainly used, such as a beam supporting a solar cell panel. Such a flexible structure tends to be excited by vibrations due to movements such as deployment and various external disturbances in outer space. Therefore, damping of vibration is quite important for various missions and especially passive damping is desirable for spacecraft due to its limited energy.

Piezoelectric elements have been studied so far as to enable passive damping [1]. Voltage is generated when deformation is applied to piezoelectric element, and by using this feature, it becomes possible to convert mechanical energy to electrical energy. Furthermore, electrical energy also can be converted to thermal energy by using dissipative element, e.g. electric resistance. By use of these energy conversion, passive damping can be achieved. However, it is not easy to achieve completer passive damping system by piezoelectric element which shows good control performance without external power source, and in order to overcome such a drawback, semipassive method has been also studied[2].

On the other hand, a magnetostrictive material can be cited as a material having properties similar to those of a piezoelectric element. Magnetostrictive material is a material with magnetostrictive effect, which consists of two effects called villari effect and Joule effect[3]. The Villari effect is a phenomenon in which the magnetic flux around the magnetostrictive material changes when strain is applied to the magnetostrictive material. The Joule effect is a phenomenon in which the magnetostrictive material generates deformation by changing the magnetic flux around the magnetostrictive material. In particular, those having a large magnetostriction effect are called giant magnetostrictive materials. By using the magnetostrictive effect, vibration, which is mechanical energy, is transmitted to the giant magnetostrictive material, the magnetic flux generated by the villary effect is converted to electric energy by the coils around giant magnetostrictive material which enable electromagnetic induction, and furthermore the electric energy is converted to thermal energy by resistance[4]. Due to above energy conversion, passive damping by giant magnetostrictive material is achieved.

Studies by Shimazaki et al.[5] grasps the characteristics for actuator use and studies improvement of control performance, but the application range of the proposed method has limitation and was insufficient. Hatakeyama et al.[6] conducted a study for development of a passive damping system for distributed parameter system using giant magnetostrictive material. In the study, possibility of damping was validated, but study on the parameter determination and optimization was insufficient. Therefore, our study focuses on the passive damping of distributed parameter system by giant magnetostrictive material and attempts to improve the control performance by optimizing the parameters from the viewpoint similar to the dynamic vibration absorber using the fixed point theory[7] which is also applied to optimization of passive damping system by piezoelectric element[8].

The structure of this paper is as follows. Section 2 outlines a passive damping system using giant magnetostrictive material. In chapter 3, equation of motion for the system is formulated and parameter optimization is performed. In chapter 4, physical interpretation is discussed and the possibility and condition of optimum tuning and optimum damping are shown. Section 5 presents a summary.

2. Basic principle and analysis target

2.1. Basic principle of giant magnetostrictive material

In this research, Terfenol-D is employed as giant magnetostrictive material. The amount of deformation of the giant magnetostrictive material with respect to the magnetic field is divided into a linear portion and a nonlinear portion. In this study, it is assumed that the giant magnetostrictive material is used in the linear portion, then the following linear magnetomechanical coupling equation is introduced[9].

$$S = S^H \sigma + dH \tag{1}$$

where, S^H, σ, d, H are the elastic compliance of the giant magnetostrictive material, the stress applied to the giant magnetostrictive material, the magnetostriction constant associated with the Joule effect (figure 1), and the strength of the magnetic field around the giant magnetostrictive material respectively, and S^H satisfies $S^H = 1/E$ where E is the Young's modulus of the giant magnetostrictive material. Furthermore, the equation of the magnetic flux density by the giant magnetostrictive material accompanying the Villari effect (figure 2)

$$B = d^* \sigma + \mu H \tag{2}$$

where d^*, and μ are the magnetostriction constant associated with the Villari effect and the permeability of the giant magnetostrictive material, respectively.

2.2. Outline of passive damping using giant magnetostrictive material

In order to achieve passive damping system using giant magnetostrictive material, the system is constructed as shown in figure 3. Since the magnetization of the giant magnetostrictive material changes when an external force is applied, an alternating current flows through the circuit by electromagnetic induction by winding the coil around the material. By connecting a resistance to the circuit, the current flows through the coil and electric energy is converted as thermal energy and passive damping is achieved.

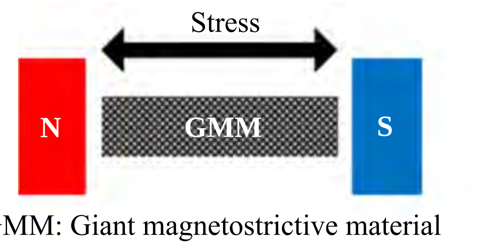

Figure 1. Joule effect

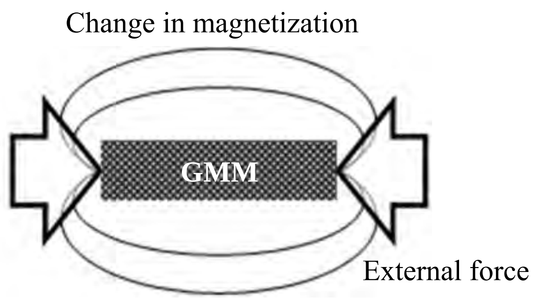

Figure 2. Villari effect

Considering Ohm's law, dissipated energy by electric resistance is given by

$$P = \frac{1}{2} R \left(\frac{V}{Z} \right)^2 = \frac{1}{2} R \frac{V^2}{R^2 + (\omega L)^2} \tag{3}$$

where ω is the angular frequency of the current flowing through the circuit, L is the inductance of the coil and V is electromotive force. Considering the relation on the electromagnetic induction and equation (2), the electromotive force V is expressed as

$$V = -NA \frac{d}{dt} \left(d \frac{\frac{x}{l} - dH}{S^H} + \mu H \right) \tag{4}$$

where N is the number of turns of the coil, A is the cross-sectional area of the giant magnetostrictive material, x is the displacement in the axial direction of the giant magnetostrictive material, l is the length of the giant magnetostrictive material. Note that l and x satisfies $S_m = x/l$ as figure 4 shows. Then, applying the differentiation, the electromotive force is expressed as the following equation.

$$V = -N \frac{NAd}{lS^H} \dot{x} \tag{5}$$

Substitution of equation (5) into equation (3) yields

$$P = \frac{1}{2} R \frac{1}{R^2 + (\omega L)^2} \frac{N^2 A^2 d^2}{l^2 S^{H2}} \dot{x}^2 \tag{6}$$

The above equation represents the energy dissipated by passive damping system by the giant magnetostrictive material. Consequently, considering Lagrange equation, damping coefficient of the system is obtained as

$$c = \frac{RN^2 A^2 d^2}{l^2 S^{H2} \left\{R^2 + (\omega L)^2\right\}} \tag{7}$$

Note that the derived damping coefficient c has the frequency ω of external force in the denominator. That is, the value of the damping coefficient continuously changes according to the frequency of the external force. Because of limitation of the paper, other terms except for damping in the equation of motion are omitted, e.g. inertia term, stiffness term and external force term.

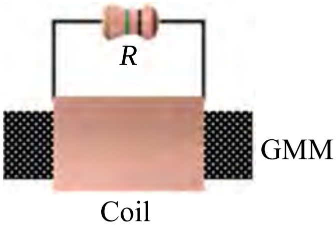

Figure 3. Coil and circuit around GMM

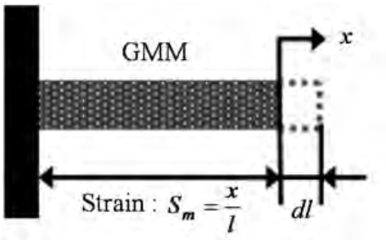

Figure 4. Deformation of GMM

2.3. Model to be controlled

As shown in the left figure of figure 5, a distributed parameter system such as a beam is introduced as controlled object. In our study, the first mode is assumed to be dominant and the first mode is mode to be controlled. Therefore, it can be approximated to a spring-mass system as shown in the figure 5. In addition, the passive damping model proposed in this paper is shown in figure 6. The giant magnetostrictive material is represented by m, k and c and displacement of the left side of the material is given by x_2. Note that c is given by the expression (7) derived in chapter 2. M and K are the mass and stiffness of the approximated system to be controlled and x_1 is the displacement of the mass M. In general, vibration damping is attempted by attaching a spring-mass-damper system to the outside of the vibration control target. On the other hand, the giant magnetostrictive material used in this study produces damping effect by receiving external force, therefore it is necessary to change the vibration due to external force to deformation of giant magnetostrictive material. Considering the above features, a spring-mass-damper system i.e. the giant magnetostrictive material, is installed between the vibration damping object M and the wall as shown in figure 5. In order to conduct fundamental study for passive damping, the applied external force is assumed to be harmonic force, i.e. $Fe^{j\omega}$.

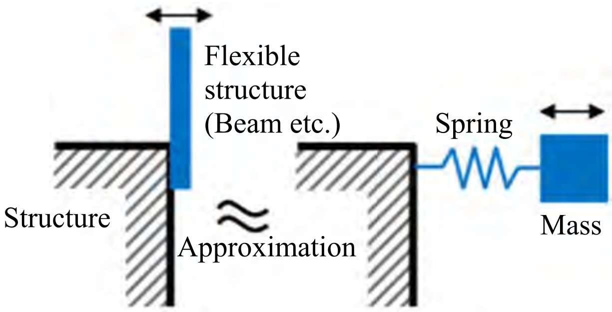

Figure 5. Approximation of flexible structure

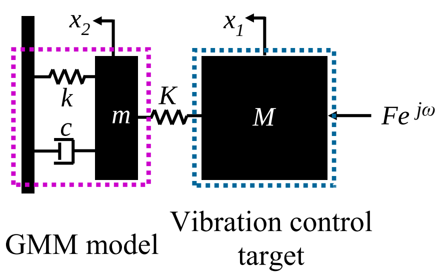

Figure 6. Analysis object model

3. Parameter optimization

3.1. Equation of motion for the controlled object and passive damping system

We formulated an equation of motion for the controlled object and passive vibration damping system by giant magnetostrictive material to obtain the vibration magnification necessary for evaluation of vibration. The equation of motion of the model shown in figure 5 is as follows.

$$\begin{cases} M\ddot{x}_1 = -K(x_1 - x_2) + Fe^{j\omega t} \\ m\ddot{x}_2 = -c\dot{x}_2 - kx_2 + K(x_1 - x_2) \end{cases} \tag{8}$$

Consequently, frequency response for x_1 and x_2 are given as follows:

$$\begin{bmatrix} X_1 \\ X_2 \end{bmatrix} = \frac{1}{\varDelta}\begin{bmatrix} -\omega^2 m + jc\omega + (k+K) & K \\ K & -\omega^2 M + K \end{bmatrix}\begin{bmatrix} F \\ 0 \end{bmatrix} \tag{9}$$

where $\varDelta = \{-\omega^2 m + jc\,\omega + (k+K)\}(-M\omega^2 + K) - K^2$. Let $\Omega_n = \sqrt{K/M}$ be the natural angular frequency of the mass M to be damped, $\omega_n = \sqrt{k/m}$ be the natural angular frequency of the mass m and k, and $\zeta = c/2m\omega_n$ be the damping ratio. Furthermore, we introduce forced frequency ratio $\lambda = \omega/\Omega_n$, mass ratio $\mu = m/M$, frequency ratio $\gamma = \omega_n/\Omega_n$. Then, the vibration magnification $|X_1/X_{st}|$ can be obtained as

$$\left|\frac{X_1}{X_{st}}\right| = \sqrt{\frac{\left(1 - \dfrac{\mu\lambda^2}{\mu\gamma^2+1}\right)^2 + \left(\dfrac{2\mu\gamma\lambda}{\mu\gamma^2+1}\right)^2\zeta^2}{\left\{\dfrac{\lambda^4}{\gamma^2} - \left(\dfrac{1}{\gamma^2} + 1 + \dfrac{1}{\mu\gamma^2}\right)\lambda^2 + 1\right\}^2 + \left(\dfrac{2}{\gamma}\lambda^3 - \dfrac{2}{\gamma}\lambda\right)^2\zeta^2}} \tag{10}$$

where X_{st} is the static displacement when static force F is applied to the system. This equation expresses the vibration magnification of the vibration damping object with respect to the static displacement of the controlled object. It should be noted that the denominator and numerator inside of the radical have the terms with and without damping parameter ζ^2, respectively.

4. Fixed point theory

4.1. Physical consideration

In order to optimize the parameters, the fixed point theory is employed. From the equation (10), the denominator and the numerator inside of the radical are expressed in terms of A and C which are not related to damping and in terms of B and D which are related to damping, and A, B, C and D are given by

$$A = \left(1 - \frac{\mu\lambda^2}{\mu\gamma^2+1}\right)^2 \tag{11}$$

$$B = \left(\frac{2\mu\gamma\lambda}{\mu\gamma^2+1}\right)^2\xi^2 \tag{12}$$

$$C=\left\{\frac{\lambda^4}{\gamma^2}-\left(\frac{1}{\gamma^2}+1+\frac{1}{\mu\gamma^2}\right)\lambda^2+1\right\}^2 \tag{13}$$

$$D=\left(\frac{2}{\gamma}\lambda^3-\frac{2}{\gamma}\lambda\right)^2\zeta^2 \tag{14}$$

The expression for vibration magnification in terms of above parameter has same structure with that of dynamic vibration absorber[10-12]. Therefore, it is expected that fixed point theory can be applied to the system in our study. In order to investigate the possibility, preliminary numerical analysis is performed and the results are shown in figure 7 and the parameters for the analysis is given by Table 1.

As figure 7 shows, there are clearly two intersection points in the two curves and it can be guessed that there are two fixed points from these results. In the following, fixed points are derived analytically and the points is used for parameter optimizations.

4.2. Optimum tuning condition

Optimum tuning is applied by equalizing the heights of the two fixed points. First, the value of the forced angular frequency ratio λ at which two fixed points occur is obtained. The fixed point is the intersection point of the graph of the maximum and the minimum damping as shown in figure 7, and it is found by solving the following expression.

$$\frac{A}{C}=\frac{B}{D} \tag{15}$$

Solving equation (15) gives the following equation.

$$\lambda^4-\left(\gamma^2+\frac{1}{\mu}+1\right)\lambda^2+\gamma^2+\frac{1}{2\mu}=0 \tag{16}$$

Assuming that the two solutions are λ_P and λ_Q in ascending order of the angular frequency, λ_P and λ_Q are obtained from equation (16). Furthermore, substituting λ_P and λ_Q into the vibration magnification and letting both values to be equal, following relation is obtained:

$$\frac{B(\lambda_P)}{D(\lambda_P)}=\frac{B(\lambda_Q)}{D(\lambda_Q)} \tag{17}$$

Finally solving equation (17) yields the optimum tuning condition as follows.

$$\gamma=\sqrt{-\frac{1}{\mu}+1} \tag{18}$$

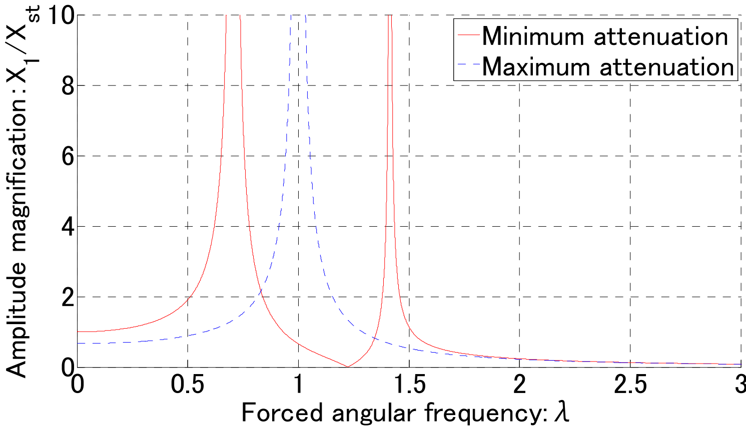

Figure 7. Two fixed points.

Table 1. Analysis conditions

parameters	value	Unit
K	1.7×10^7	[N/m]
M	0.50	[kg]
k	3.4×10^7	[N/m]
m	1.0	[kg]

4.3. Optimum damping condition

The optimum damping is performed by determining the damping condition such that the two fixed points become local vertices of the vibration magnification. Applying second order differential of equation (10) with respect to λ, the condition that the inclination of the graph becomes zero at angular frequencies λ_P, λ_Q is found. That is, solving the following equation yields the optimum condition.

$$\frac{\partial}{\partial\lambda^2}\left|\frac{X_1}{X_{st}}\right| = \frac{\partial}{\partial\lambda^2}\sqrt{\frac{\left(1-\frac{\mu\lambda^2}{\mu\gamma^2+1}\right)^2+\left(\frac{2\mu\gamma\lambda}{\mu\gamma^2+1}\right)^2\zeta^2}{\left\{\frac{\lambda^4}{\gamma^2}-\left(\frac{1}{\gamma^2}+1+\frac{1}{\mu\gamma^2}\right)\lambda^2+1\right\}^2+\left(\frac{2}{\gamma}\lambda^3-\frac{2}{\gamma}\lambda\right)^2\zeta^2}} = 0 \tag{19}$$

Because it is difficult to solve equation (19) analytically, numerical method is used to solve the equation in this study.

4.4. Parameter feasibility for the obtained result

In the previous section, it was confirmed that it is possible to perform optimum tuning and optimum damping, but care should be taken in handling equation (18) which is a parameter condition for realizing optimal tuning. Expression (18) is rewritten as follows.

$$K = \frac{k}{\frac{m}{M}-1} \tag{20}$$

In order to satisfy this condition, the mass m of the vibration damping system must be larger than the mass M to be damped or the spring of the vibration damping system must exhibit negative rigidity. In order to realize passive damping in the proposed system, it is more feasible to change the structure so that $M<m$. However, it is generally not desirable to increase the mass of the damping system. In general, flexible structure which vibration is to be controlled is attached to a rigid structure. Incorporating a damping system using a giant magnetostrictive material in the rigid structure, it becomes possible for the system to satisfy the condition (20).

4.5. A Study on the Structure of Passive Damping System

As an installation example in the case where the mass *m* of the vibration damping system is made larger than the mass *M* to be damped, a system shown in figure 8 is introduced. A giant magnetostrictive material is placed between a rigid structure indicated by mass *M* and a part of the structure to be a mount. Because of such a placement of giant magnetostrictive material, it becomes possible for the system to satisfy $M<m$.

Figure 9 shows the response when an external force is applied to the system of figure 8 at a frequency near the resonance point. Analysis conditions are shown in Table 2. In figure 9, the red line indicate the response of the conventional design method, and the blue line indicates the response of the system to which the proposed method is applied. In contrast to response of red line, the amplitude of the response is decreased and vibration is suppressed in the case of blue line.

5. Summary

In this research, we proposed the passive damping system which utilizes the characteristic of giant magnetostrictive material. Furthermore, fixed point method is applied to the proposed system in order to optimize the design parameter, and optimum tuning condition and optimum damping condition are derived. Furthermore, feasibility of the optimum tuning condition is discussed and an install method of the proposed damping system is proposed to satisfy the optimum tuning condition. Finally, the proposed method is validated by numerical analysis.

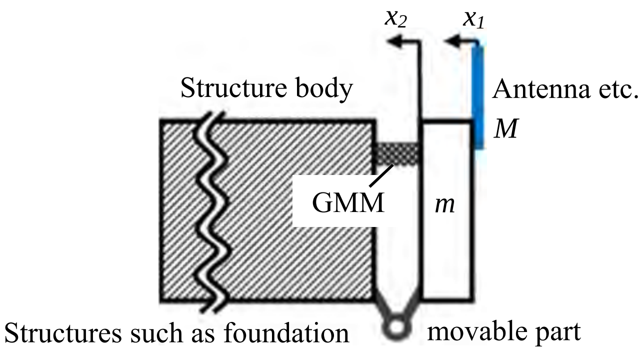

Figure 8. Example of a structure which satisfies $M<m$

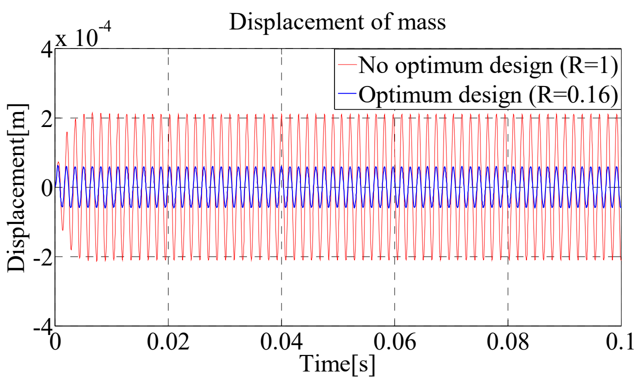

Figure 9. Time history of response.

Table 2. Analysis conditions

parameters	value	unit
K	3.4×10^7	[N/m]
M	0.50	[kg]
k	3.4×10^7	[N/m]
m	1.0	[kg]

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