

전자석을 사용한 능동 미소진동 절연장치 개발

Development of Active Micro-Vibration Isolator using Electromagnet

이대은* · 박지용* · 한재홍†
Dae-Oen Lee, Gee-Yong Park, Jae-Hung Han

Key Words : Vibration Isolation (진동 절연), Active Vibration Isolation (능동 진동 절연).

ABSTRACT

Observation satellites carrying high precision optical payload require extremely stringent pointing stability that may be violated in the presence of the disturbances coming from reaction wheels, cryocoolers or other actuating components onboard the satellite. The most common method to protect the sensitive payloads from external disturbances is implementation of vibration isolator. In this paper development of a single axis active vibration isolator using electromagnet and its performance in isolating micro-vibration is presented. The main components of the developed isolator are membrane structure providing the isolator with the required stiffness and an electromagnet for active control. The performance test results show that additional damping can be achieved by active control without degrading isolation performance in high frequency region and that the developed isolator can effectively isolate micro-vibration.

고정밀 광학계를 탑재한 관측 위성은 매우 높은 지향 안정성을 요구하지만 인공위성의 자세 제어를 위해 사용하는 반작용 휠, 극저온 냉각을 위한 cooler 등 다양한 진동원에 의해 미소 진동이 발생할 수 있다. 고정밀 탑재체를 외부 진동으로부터 보호하기 위한 가장 일반적인 방법은 진동이 전달되는 경로에 진동 절연장치를 도입하는 것이다. 본 논문에서는 미소진동 절연을 위한 1 축 능동 진동 절연장치의 개발 및 진동 절연 성능 시험을 기술하였다. 개발한 절연장치의 고유진동수를 조절하기 위해 원하는 강성을 갖는 membrane 구조를 설계하였으며 능동 제어를 위해 전자석을 사용하였다. 절연장치의 성능 시험 결과를 통해 능동 제어 시 고주파수 대역에서 절연성능의 저하 없이 공진의 크기를 줄일 수 있는 것과 개발된 진동 절연 장치가 미소 진동을 효과적으로 절연할 수 있는 것을 확인하였다.

1. Introduction

High precision optical payload onboard observation satellites is exposed to vibration disturbances that result from the operation of actuating members including reaction wheel assemblies used for satellite's attitude control, cryocoolers used for temperature control and so on⁽¹⁾. Due to these vibration disturbances, continuous micro-vibration

may be induced on high precision payloads resulting in performance degradation. In case of optical payloads, 10 μ -radian angular vibration corresponds to 5m in the field of view at a distance of 500km. Fig. 1. illustrates how micro-vibration causes changes in the light of sight resulting in performance degradation of optical payloads. Although the quality of the acquired image can be somewhat improved by post image processing, protecting the payload from the induced vibration is always a better and preferred solution⁽²⁾.

The most common method to protect sensitive payload from performance degrading vibration is implementation of vibration isolator in the vibration path. Vibration isolation can be implemented using purely passive components.

† 교신저자; KAIST 항공우주공학전공
E-mail : jaehunghan@kaist.edu
Tel : 042-350-3723, Fax : 042-350-3710
* KAIST 항공우주공학전공

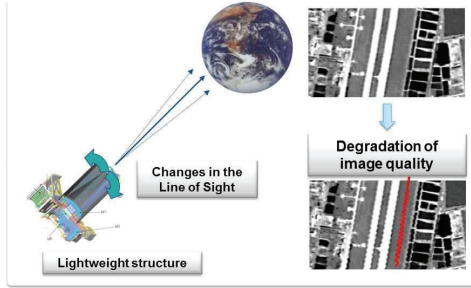


Fig. 1 Effect of micro-vibration on the acquired image

Purely passive isolation systems are reliable and simple and are effective in attenuating high frequency disturbances but tend to aggravate vibration in low frequency region. For applications with stringent requirements that cannot be satisfied by passive systems alone, higher isolation performance can be attained with active systems which are more complex and costly. In this paper, development of a single axis active vibration isolator for micro-vibration isolation is discussed. In section 2, advantages of active vibration isolators over their passive counterparts are introduced. The design and performance test results of the active isolator are discussed in section 3 and 4 respectively.

2. Vibration Isolation Characteristics of Passive and Active Isolators

2.1 Passive Vibration Isolator

Typical passive vibration isolator is modeled using a spring element (k) in parallel with a viscous damper (c) as shown in Fig. 2. The isolator is inserted between the vibrating mass and the foundation to reduce the transmission of forces acting on the mass to the foundation. Using the equation of motion given in equation 1, isolation characteristics can be analyzed by taking the ratio of transmitted force (f_{tr}) to input force (f_d) as given in equation 2 where $\omega_n = \sqrt{k/m}$ is the natural frequency of the system, $\zeta = c/2\sqrt{mk}$ is the damping ratio and s is the Laplace variable.

$$m\ddot{x} = -kx - c\dot{x} + f_d \quad (1.a)$$

$$f_{tr} = kx + c\dot{x} \quad (1.b)$$

$$T(s) = \frac{F_{tr}(s)}{F_d(s)} = \frac{2\zeta\omega_n s + \omega_n^2}{s^2 + 2\zeta\omega_n s + \omega_n^2} \quad (2)$$

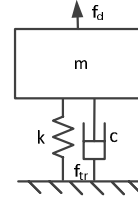


Fig. 2 Typical passive vibration isolator

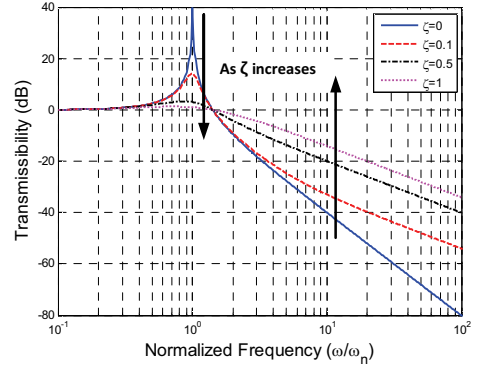


Fig. 3 Transmissibility for various damping values

The magnitude of transmissibility $|T(j\omega)|$ for various damping values is shown in Fig. 3. As shown in the figure, increasing damping lowers amplification at the resonance, but the isolation roll-off rate at high frequency region is degraded. Thus, a compromise between low resonant peak and isolation performance at high frequency must be made for the conventional two parameter passive isolator⁽³⁾.

2.2 Active Vibration Isolator

One method to achieve both the low resonant peak and high roll-off rate at high frequency region is implementation of an active isolator⁽⁴⁾. Consider an active isolation system shown in Fig. 4 where an actuator is placed in parallel with the spring and the damper. A force sensor measures the transmitted force to the foundation which is used by the controller to generate control signal for the actuator. The equation of motion for the active vibration isolator is given in equation 3 where f_a is the control force inserted by the actuator and H is the controller. Using an integral controller ($H = G_a/s$), the transmissibility can be derived as shown in equation 4.

$$m\ddot{x} = -kx - c\dot{x} + f_d + f_a \quad (3.a)$$

$$f_{tr} = kx + c\dot{x} - f_a \quad (3.b)$$

$$F_a = HF_{tr} \quad (3.c)$$

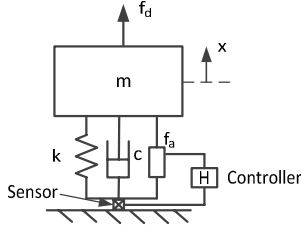


Fig. 4 Active vibration isolator

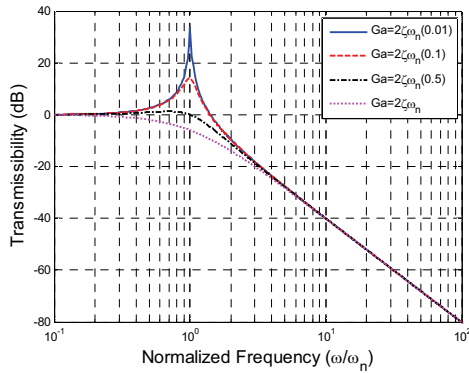


Fig. 5 Transmissibility of active vibration isolator

$$\frac{F_{tr}}{F_d} = \frac{2\zeta\omega_n s + \omega_n^2}{s^2 + (2\zeta\omega_n + G_a)s + \omega_n^2} \quad (4)$$

Fig. 5 shows the magnitude of transmissibility for various gains of the active isolation system with no damping ($\zeta=0$). Unlike the transmissibility of passive isolator shown in Fig. 3, lowering resonant peak does not affect the roll-off rate in high frequency region for the active isolator.

3. Design of Active Vibration Isolator

The designed active vibration isolator is shown in Fig. 6. Similar active vibration isolator has been developed by Hanieh⁽⁵⁾. The main components of the isolator include a membrane structure, an electromagnet, outer covers and a mechanical stop to protect isolator from excessive vibration. The permanent magnet is fixed to the cover while the coil is attached to the membrane which is also connected to the payload to be isolated. The targeted natural frequency of the vibration isolator is about 10 Hz for the payload mass of 5 kg.

Because the application of the isolator is in space environment where temperature change is significant and

vacuum environment may cause outgassing, metal material is opted over viscoelastic materials whose properties are known to vary greatly according to temperature⁽⁶⁾.

A flexible aluminum membrane is designed to provide the isolator with the required stiffness. The axial stiffness of the membrane determines the natural frequency of the isolator, thus isolation frequency region. The diameter and thickness of the membrane were constrained at 50mm and 1mm respectively, and various patterns were designed to lower the stiffness of the membrane using ANSYS Workbench. The stiffness of the membrane was estimated by predicting deformation to a known force and using the equation $F=kx$. The finite element analysis result of the simplified version of the isolator predicts natural frequency of the isolator to be about 9.38 Hz for a payload mass of 5 kg. Fig. 7 shows the axial mode shape of the isolator, and Fig. 8 shows the fabricated vibration isolator.

A voice coil motor (VCM) is chosen for active control. VCM is known as an accurate actuator with high bandwidth with very simple structure. The VCM used in the study is from Moticon (HVCM-051-025-013-01) that can provide continuous force of 23.7 N and has a stroke of 12.7mm which covers the requirement. Current amplifying circuit using LM1876 speaker amplifier modulator is used for the actuation of voice coil motor.

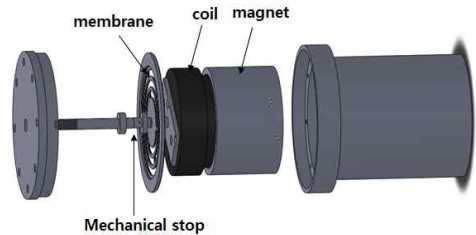


Fig. 6 Components of designed vibration isolator

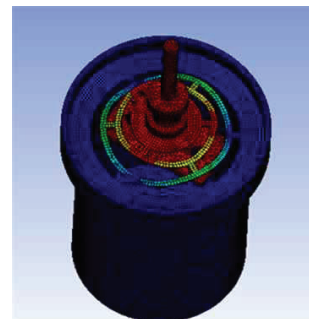


Fig. 7 Axial mode shape of the designed isolator

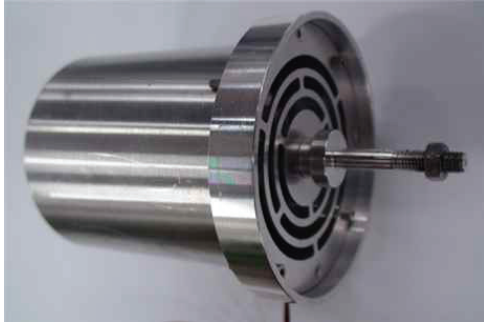


Fig. 8 Fabricated vibration isolator

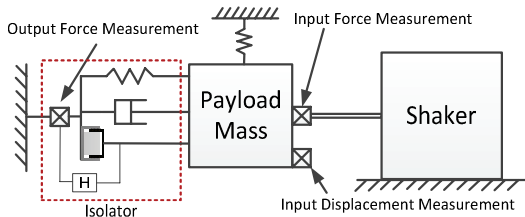


Fig. 9 Schematic of the test setup

4. Isolation Test

4.1 Isolation Test Setup

The designed vibration isolator is manufactured and tested for performance evaluation. For the performance evaluation, the force transmissibility is measured using the test setup shown in Fig. 9. The frequency range of interest is from 1 to 200 Hz, and sine sweep signal from 1~257 Hz with sweeping rate of 16 Hz/s is generated using a shaker from MB Dynamics (Model PM50A) with SS250 power amplifier. The input and output forces are measured using the force sensors from PCB (Model 208C02), and Pulse FFT signal analyzer is used for data analysis. The output force signal is also fed to DS1103 PPC controller board to generate control signal using an integral force feedback controller shown in Fig. 10. Laser displacement sensor (Keyence LK-031) is used to measure the input displacement level. The RMS of the input displacement is in 100 μ m level. The test is repeated 5 times and the average is taken.

4.2 Isolation Test Results

Isolator's performance test is conducted for passive and active cases and the results are shown Fig. 11. The passive

isolation test results show that the passive damping ratio is 0.097 and the damped natural frequency is 8.6875 Hz which is close to the predicted undamped natural frequency of 9.38 Hz.

Active control results show that the resonant peak can indeed be lowered without affecting isolation performance in high frequency region. For the case with gain #2, calculated damping ratio is 0.2084 which is twice the damping ratio of the passive case, while isolation performance in high frequency region is almost identical to the passive case. Fig. 11 also shows that the developed isolator can achieve vibration reduction about 20 dB from 40 Hz onwards. The input and output time signal as well as the control signal for the case with gain #2 are plotted in Fig. 12. The RMS values of the input and output forces are 6.71 N and 1.64 N respectively, which is 75.7% reduction in the transmitted vibration. The test results show that the developed vibration isolator is effective in isolating micro-vibration.

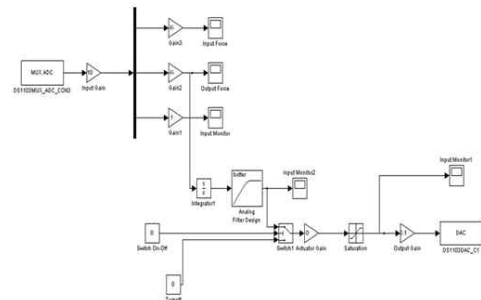


Fig. 10 Integral force feedback controller

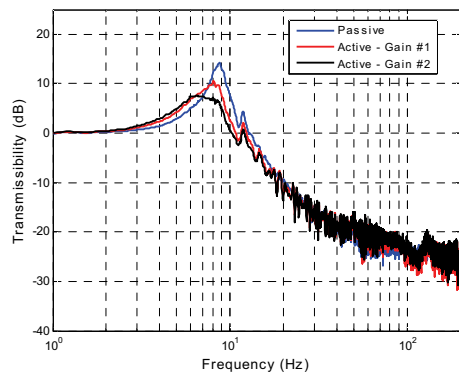


Fig. 11 Measured transmissibility of the developed isolator

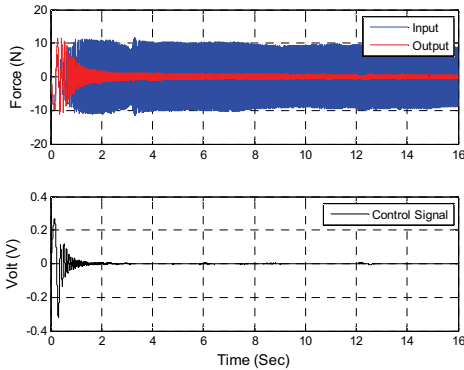


Fig. 12 Time history of input, output and control signal for active isolation with gain #2

3. Conclusion

An active vibration isolator for micro-vibration is developed and its isolation performance is tested. The main components of the developed isolator are flexible membrane designed to set the natural frequency of the isolator to the desired value and a voice coil motor used as an actuator for active control. The test results show that damping at resonance can be increased without degrading roll-off rate at high frequency region using integral force feedback controller, and greater than 20 dB reduction in vibration amplitude is achieved from 40 Hz onwards.

후 기

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