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A novel vibration isolation system for reaction wheel on space telescopes ☆

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ABSTRACT

A reaction wheel (RW) is commonly used as an actuator for attitude control on space telescopes. The RW can also produce tonal disturbances and broadband noises when the wheel spins. In this work, a novel vibration isolation system is proposed to attenuate the disturbances caused by the RW. This novel vibration isolation system includes a multi-strut vibration isolation platform and multiple tuned-mass dampers, and each strut of the vibration isolation platform includes a negative stiffness structure in parallel with a positive stiffness structure. This study aims to validate the feasibility and effectiveness of this new vibration isolation system from a theoretical perspective. First, the integrated satellite dynamic model is constructed, including the RWs and the vibration isolation systems. Next, its frequency domain characteristics are described, and the application of the vibration isolation system for RWs is presented. Finally, the effective attenuation of RW disturbances is illustrated via the new vibration isolation system, and its safety performance is verified with numerical simulations.

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1. Introduction

Space telescopes with high pointing accuracy and stability have gained increasing importance in space missions. The Hubble Space Telescope (HST), which was launched in 1990, required the pointing stability to be less than 0.007 arc-seconds within periods of up to 24 h [1]. The James Webb Space Telescope (JWST) requires that the line-of-sight motion should be 4 milli-arc-seconds [2]. The Terrestrial Planet Finder Coronagraph (TPF-C), which is planned for launch in five years, must also maintain a pointing accuracy of 4 milli-arc-seconds to meet the minimum scientific requirements [3]. To achieve this high performance, it is necessary to understand the characteristics of the disturbance sources, structural dynamics and optical payload of the spacecraft and to develop the appropriate vibration control techniques.

In these space telescopes, the attitude control torques are all provided by reaction wheels (RWs), which produce continuous torques to deliver high-precision pointing control and perform large-angle slewing maneuvers. However, due to the imbalance of the rotor, imperfections in the spin bearings, motor disturbances and motor-driven errors, the reaction wheel assembly (RWA) produces tonal disturbances and broadband noises when the wheel spins, thus making it one of the largest disturbance sources onboard the spacecraft [4]. To mitigate these effects on the spacecraft pointing control, vibration isolation technology is often used in the reaction wheel. The HST used viscous fluid dampers known as D-Struts to attenuate the axial disturbances of the RWAs [5]. The Defense Satellite

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Communications System III Spacecraft applied four damped stainless steel spring isolators to support an RW to provide six degrees of freedom in the wheel isolation [6]. The Chandra X-ray Observatory employed a hexapod isolator at each of its six reaction wheels to achieve multi-dimensional vibration isolation [7]; this passive vibration isolation platform will also be employed in JWST as the first isolation stage [8]. Kamesh et al. [9] designed a flexible platform consisting of four folded continuous beams arranged in three dimensions to act as a mount for each flywheel.

In designing the abovementioned vibration isolation systems for reaction wheels, the researchers ignored the fact that the reaction wheel must constantly adjust its speed to realize attitude stabilization or attitude maneuvers. Furthermore, the RW disturbance force is caused by the static imbalance of the rotor, whose amplitude is proportional to the square of the wheel speed and whose frequency is the same as the wheel speed. Consequently, a resonance condition will occur when the wheel speed and the corner frequencies of the abovementioned vibration isolation systems are similar. This resonance will cause more serious vibration in the satellite and significant influence on the image performance of the optical payload on the satellites. If the resonance amplitude is sufficiently large, the vibration isolation system will lose the ability to transfer effective torques, causing rapid and catastrophic attitude control system failure. A tuned-mass damper is a device mounted in the structures to reduce the amplitude of mechanical vibrations, the application of which can prevent discomfort, damage, or outright structural failure. Therefore, tuned-mass dampers are frequently used in power transmission, automobiles, and buildings [10–12]. However, the tuned-mass damper has not been applied to the most recent vibration isolation systems for RWs.

In addition, the spring–damper system is used in a commonly applied passive vibration isolator, which consists of a linear stiffness spring in parallel with a damper. To provide the proper resonance frequency, the spring stiffness is selected relative to the payload mass. The damping coefficient is selected to provide a trade-off between resonance amplification and high frequency attenuation. In this method, the spring stiffness should be decreased to reduce the resonance frequency. However, a reduction in the spring stiffness leads to a low load-bearing capacity. To overcome this problem, this work uses negative stiffness structures for a vibration isolation system under low-frequency vibration conditions. The recent achievements in negative stiffness structures are worthy of note. Carrella and Brennan [13] studied the use of magnets arranged in an attracting configuration to obtain a negative stiffness element. Lee et al. [14] presented a useful approach for designing a compact spring employed to effectively minimize suspension stiffness. Le and Ahn [15] used the negative stiffness technique to design a vibration isolation system for improved vibration isolation effectiveness for a vehicle seat under low excitation frequencies. The relationship between the configuration parameters and the stiffness of the elastic element (for which the dimensionless nonlinear stiffness of the proposed system ranges from 0 to 1 and the displacement of the isolation equipment [mass] reaches a maximum without exceeding a desired value of stiffness) was obtained by Le and Ahn [16]. Yang et al. [17] studied the dynamics and power flow behavior of a nonlinear vibration isolation system with a negative stiffness mechanism. However, the dynamics of the vibration isolation system, including the negative stiffness structures, were derived in a single-degree-of-freedom context.

Predictions of jitter performance using vibration isolation systems require the construction of a dynamic model of the spacecraft that includes this vibration isolation system. Recently, notable progress has been made in the dynamic analysis and simulation of parallel manipulators. Lebert et al. [18] constructed a dynamic model of a vibration isolation platform with six degrees of freedom using the Lagrange method. Tsai developed a dynamic model of a vibration isolation platform using the principle of virtual work. Liu et al. [19] developed general dynamic equations for an arbitrary parallel manipulator using the Kane method and the substructure method. Dasgupta and Choudhury [20] developed a dynamic model of a vibration isolation platform using the Newton–Euler method. In fact, the performance of vibration isolation platforms used in spacecraft is affected by the spacecraft motion and the coupling between the vibration isolation system and the spacecraft. However, most of the aforementioned studies neglected the effect of the base platform motion and did not analyze the frequency domain characteristics.

This paper presents a new vibration isolation system containing a Stewart platform and multiple tuned-mass dampers that not only satisfies the brace stiffness requirement but also reduces the resonance amplitude. For the Stewart platform, each strut uses the negative stiffness technique to greatly enlarge the frequency band for effective vibration isolation. In addition, for predictions of jitter performance using this vibration isolation system, we construct an integrated satellite dynamic model and discuss the characteristics of the integrated satellite system.

2. Theory of single-axis vibration isolation

2.1. Theory of a single-axis tuned-mass damper

A single-axis isolator is shown in Fig. 1, where M is the mass of the sensitive equipment, and K and c' are the stiffness and the damping of the isolator, respectively.

The transfer function of the passive vibration isolator can be written as

$$G(s)=\frac{c's+K}{Ms^2+c's+K} \tag{1}$$

The undamped natural frequency of the transfer function is ω_n, and the damping ratio is ξ. Substituting $s=j\omega$

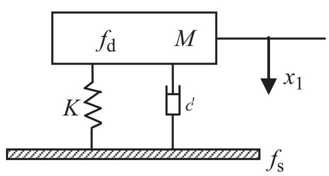

Fig. 1. Schematic of a single-axis isolator.

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