JOURNAL OF SPACECRAFT AND ROCKETS
Vol. 32, No. 2, March–April 1995

Engineering Notes

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Evaluation of Passive and Active Vibration Control Mechanisms in a Microgravity Environment

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Introduction

PASSIVE and active vibration control strategies for flexible space structures are fairly well studied; however, their effectiveness in protecting the onboard equipment and nonstructural components is not well understood. In order to meet future goals and designs of large space structures, it will be necessary to use a combination of passive and active control mechanisms. This study is concerned with developing several techniques for vibration control in a microgravity environment. These include the use of passive spring–damper and frictional systems, as well as an active friction control mechanism. Studies on passive and active vibration control devices show that vibration amplitudes and deflections generated in large space structures are significantly reduced by the use of these systems. Grodsinsky[1] describes both an active magnetic suspension system and a passive inertial damping system to achieve microgravity vibration control. Because of the simplicity and the success of frictional dampers for reducing vibration, it is of interest to study the feasibility of applying these techniques to large space structures in a microgravity environment.[2]

In this study, a design for an active vibration control (AFC) mechanism for the microgravity environment is developed. The device is a resilient friction system with a soft spring. The normal contact force is controlled with an electromagnet to minimize the vibration transmitted to the equipment. Equipment responses to STS-40 orbital vibrations are evaluated for various passive and active control mechanisms. The presented results show that the passive vibration control devices could reduce the vibration energy transmitted to the onboard equipment to a certain extent. However, to reduce the peak acceleration to the order of $10^{-6}g$, an active control mechanism is needed.

System and Vibration Control Models

The equipment aboard the orbiter is modeled as an elastic, nonuniform shear beam with a subsystem mounted at the top of the beam and represented as a single-degree-of-freedom oscillator. The governing equations of motion of the shear beam subject to base excitation are detailed by Ellison et al.[3] This model was used earlier by Su et al.[4,5] and Lee-Glauser and Ahmadi.[6]

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In the subsequent discussion, the fixed base (FB) condition represents the baseline condition where the equipment is rigidly attached to the orbiter and no vibration control system is present. Several vibration isolation devices for control of equipment vibrations are considered. They include the spring–damper (SD) system, which consists of a 0.25-Hz spring element and a damper with a damping coefficient of 0.02. We also study a resilient friction (RF) system with the stiffness and damping the same as in the SD system and an additional friction element with a normal force per unit mass of 0.8 g and a coefficient of friction of 0.04. A pure friction control, which is analogous to the current method employed on board the orbiter–payload interface, is also analyzed. A coefficient of friction of 0.2 and a normal force per unit mass of 1.0 g were used. Numerical results show that this allows slipping approximately 50% of the time under microgravity loading.

An active friction control (AFC) system that is an RF system with an active friction control is also proposed. Here, the friction force at the interface of the equipment and orbiter is varied by changing the normal force induced by an active electromagnet. The control of the friction force is achieved by varying the force of the electromagnet in proportion to the base relative velocity. Thus, the friction force decreases as the base velocity decreases, and the friction force becomes zero whenever the equipment reverses its sliding direction. This hybrid (passive–active) system has the advantages of the passive RF systems, e.g., the inherent damping of the viscous damper and restoring force of the spring for keeping the system centered in the absence of excitation. In addition, active control of the friction force significantly improves the effectiveness of the RF system by eliminating slip–stick shock loadings.

For the AFC mechanism, the control system is set to vary the interface normal force in proportion to the base relative velocity. That is, the normal force is given by

$$N(t) = \begin{cases} \alpha|\dot{u}_b(t)|, & \alpha|\dot{u}_b(t)| < N_{\max} \\ N_{\max}, & \alpha|\dot{u}_b(t)| \geq N_{\max} \end{cases} \tag{1}$$

where $\alpha = 5\ \mathrm{s}^{-1}$, $N_{\max} = 0.8\ g$, and $\dot{u}_b(t)$ is the velocity at the base of the equipment. To maximize the frictional damping without incurring the stick condition, the normal force is increased according to Eq. (1) as long as the base velocity is increasing. A maximum cutoff value of $N_{\max} = 0.8\ g$ for the normal force was also imposed. A schematic diagram of equipment with the AFC system is shown in Fig. 1a.

Response Analysis

The base excitation used in this study is the actual flight data from the STS-40 mission. The data segment used is for a quiet period and was measured by a triaxial accelerometer mounted on the floor of the space lab. Data for the x-axis direction, which contains the highest energy, was used to evaluate the performance of the passive and hybrid vibration control systems. One significant feature of the orbital excitation is its low-frequency energy content, which extends to far below 1 Hz. A time history for the STS-40 mission acceleration is shown in Fig. 1b.

To analyze the performance of the vibration control mechanisms under orbital excitations, peak responses of the equipment and its subsystem are found for a 20-s period. Design requirements for Shuttle space experiments require that equipment frequencies be greater than 30 Hz. Therefore the simulation results are computed for a range of frequencies up to 40 Hz. For the frictional vibration

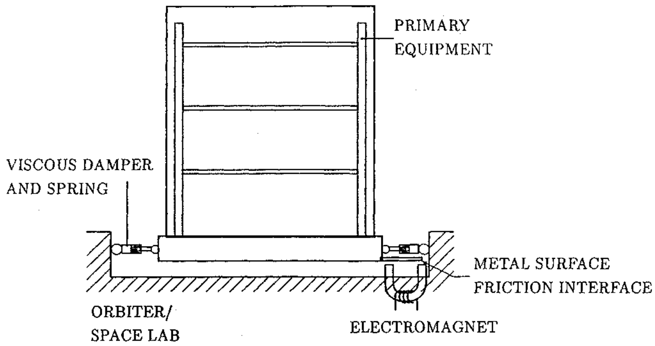

Fig. 1a Schematic diagram of equipment with the AFC system.

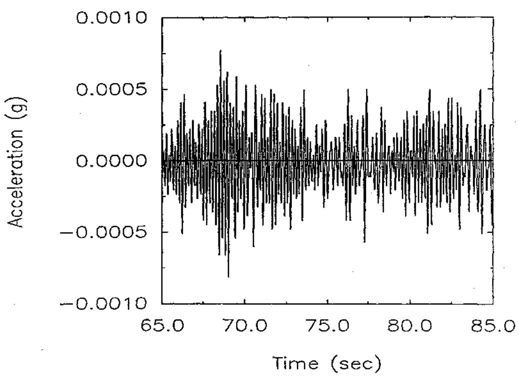

Fig. 1b Time history of the STS-40 x-axis orbital acceleration.

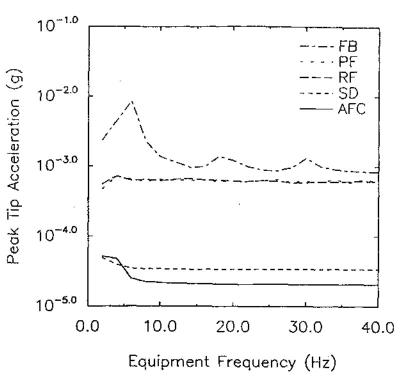

Fig. 2 Equipment peak response for the fixed-base and vibration control mechanisms due to orbital excitation.

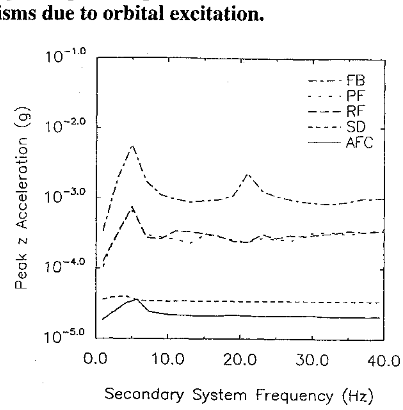

Fig. 3 Secondary system peak response for the fixed-base and vibration control mechanisms due to orbital excitation.

control systems, the transitions from the sliding to the nonsliding phase generate shock loadings and lead to the excitation of the higher harmonics. Therefore, five vibration modes of the equipment are used in the analysis in order to best model the response during this transition.

Figure 2 shows the peak acceleration of the tip of the equipment as a function of the equipment frequency. The fixed-base response is used as a baseline for comparison. The responses of the passive friction-based system are of the same order of magnitude as those of the fixed-base system, because of the slip–stick shock loadings. The SD system, which is the most effective among the passive systems considered, provides an order-of-magnitude decrease in peak responses with respect to the baseline. Figure 2 shows that the AFC system leads to the lowest peak responses.

Peak subsystem responses to orbital excitation are shown in Fig. 3. The response spectra in this figure are plotted versus subsystem frequency. A value of 35 Hz was used for the equipment frequency. The graphs show that the passive friction control systems lead to subsystem responses that are of the same order as the baseline. Other passive control mechanisms provide some isolation from the orbital excitations. The AFC mechanism appears to be the most effective system in reducing the peak responses experienced by the subsystem.

Conclusions

A numerical simulation was performed using a shear beam to represent equipment in a space laboratory. Several vibration control mechanisms, including three passive and one active system, were evaluated. Based on the presented results, the following conclusions are drawn. Use of passive and active vibration control mechanisms reduces the peak accelerations experienced by the equipment and its subsystem in the orbital environment. The active friction control mechanism considered provides the best performance in reducing the peak acceleration transmitted to the equipment and its subsystem. With the use of the AFC mechanism, the peak orbital acceleration experienced by the equipment was reduced to about $20 \times 10^{-6} g$.

Acknowledgments

This work is supported by the NASA Lewis Research Center, Cleveland, Ohio, under Grant No. NGT-50825.

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Reusable Sounding-Rocket Design

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Introduction

IN an era of reduced budgets and project cost cutting, the ideas of reusability and cost-effectiveness in launch vehicles have become highly important. Although these ideas have had some success with

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