ACTIVE VIBRATION ISOLATION USING A POSTBUCKLED SPRING

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ABSTRACT

We study precision positioning and vibration isolation of a platform supported on a post-buckled beam in series with a piezoelectric actuator or in parallel with a voice-coil actuator. The post-buckled beam supports a large weight with small static deformation, maintains a very low resonant frequency for passive vibration isolation, and provides for coarse positioning over a relatively large displacement. Isolation strategies based on feedback control, adaptive feedforward control, and their combination are implemented and compared.

1. INTRODUCTION

This work was motivated by the advanced LIGO (Laser Interferometric Gravity-Wave Observatory) system[1] which requires precision position alignment and seismic isolation of a payload. Many other applications—such as semiconductor manufacturing, interferometry, and microscopy—also call for micron to nanometer precision positioning in addition to vibration isolation.

There are typically three requirements on positioning and isolation systems for instrumentation: support of the payload weight, precision positioning of the payload, and isolation from ground vibration. There are inevitably trade-offs when attempting to meet all of these requirements. For example, a soft mount based on a linear spring designed to have a resonant frequency f will exhibit a static sag Δ_s given by

$$\Delta_s = g/(4\pi^2 f^2) \tag{1}$$

where g is the acceleration of gravity. From this expression, we see that to obtain a resonant frequency of 1 Hz (and therefore 40 dB vibration attenuation above 10 Hz), the static deformation must be 0.248 m. Air legs are frequently used to support large weights and maintain a soft mounting in many precision instruments, but they require a large volume and an air supply. Moreover, it is difficult to obtain static positioning accuracy unless the air leg is combined in parallel with an active element.

Many configurations have been suggested for precision positioning and isolation systems, but most use the parallel combination of (mechanical) springs to support the weight of the payload and stiff actuators (such as hydraulic or piezoelectric actuators) to attenuate vibration and control position (see e.g.,[1,2]). Although controllers based on stiff actuators are able to reduce seismic vibration at low frequencies, there is no vibration attenuation at all above the control bandwidth (which is generally very low, on the order of 10 Hz).

In this paper, we study the use of a post-buckled column in series or parallel combination with an actuator to provide broad-band vibration isolation and precision static positioning of a payload. Nearly buckled and post buckled structures are known to be capable of supporting a large static weight with low resonant frequency.[3,4] Recent studies by Winterflood et al.[5] and Virgin and Davis[6] have studied the use of post-buckled columns for passive vibration isolation. In the present work, we design and test an active positioning and isolation system comprising a post-buckled column and soft spring for coarse position alignment mounted in series with a stiff actuator or in parallel with a soft actuator.

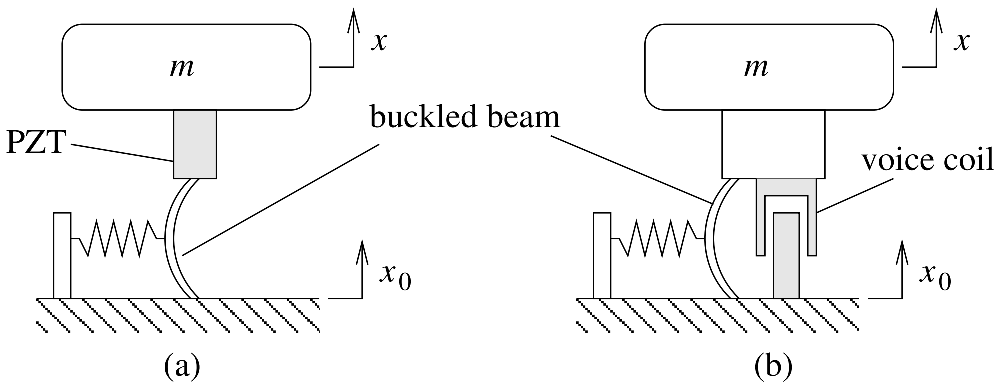

Figure 1. Diagram of a platform for precision position and vibration control based employing (a) a piezoelectric actuator and (b) a voice-coil actuator

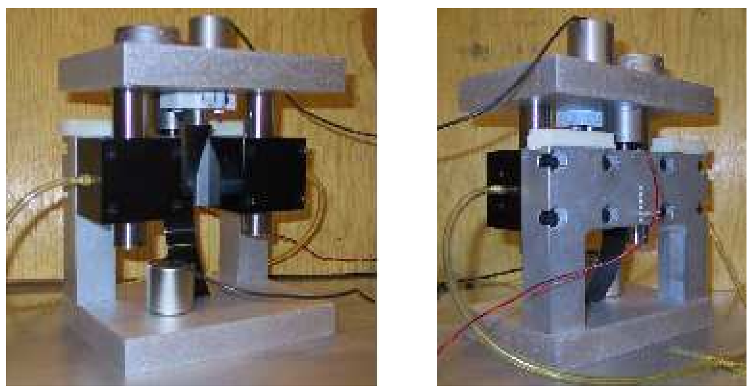

Figure 2. Photographs of the isolation platform used in the experiments from the front (a) and back (b): The platform is guided by a pair of aerostatic bushings and its weight is supported on a buckled spring-steel beam. A voice-coil actuator acts between the base and the platform and a geophone is mounted atop the platform.

2. SYSTEM CONFIGURATION

Figure 1 shows the layout of the proposed system for an SDOF isolation strut. (A combination of such struts could be used to achieve six-DOF isolation.) A post-buckled column supports the weight of the platform and a lateral force generated by a soft spring is used for coarse alignment of position. Active control based on the geophone and displacement sensor signals is carried out using either a stiff piezoelectric actuator as in Fig. 1(a) or a voice coil actuator as in Fig. 1(b). Figure 2 shows the platform used in the experiments (equipped with a voice coil and guided by a pair of air bearings). The platform mass is approximately 3 kg.

One of the key limitations on the performance of feedback or feedforward control arises from the geophone dynamics. The geophones employed have resonances specified to be 4.5±0.75 Hz and a damping ratio of 0.35 (open circuit). The actual measured resonance ω_g is approximately 5.1 Hz. We select the parameters R, R_1–R_3, C, and C_1–C_3 of the circuit of Figure 3 to correct the velocity measurement to 0.5 Hz at low frequency, and set R_4 to obtain a low-pass filter to attenuate high-frequency noise. We use software to further correct the measurement to 0.1 Hz.

3. POST-BUCKLED COLUMN

Figure 4 shows a pinned-pinned column with length l and flexural stiffness EI subject to an axial force P and a lateral force F acting at its midpoint. For the case of $F = 0$, an exact but cumbersome solution for the deflection of the post-buckled column has been obtained using elliptic integrals.[7] A group of governing equations can

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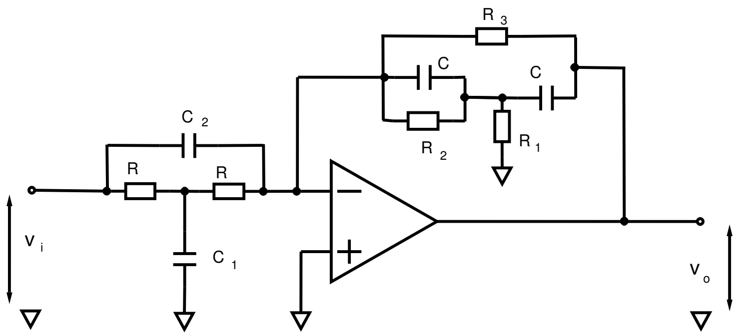

Figure 3. Correction of geophone low-frequency dynamics using analog circuit

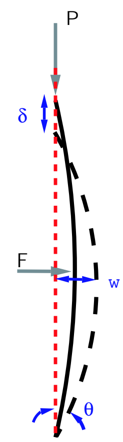

Figure 4. Post-buckled pinned-pinned column

be written, but no exact solution exists for the case where $F \neq 0$. Since we are designing the system to avoid yielding, we can use the Rayleigh–Ritz method to obtain an approximate solution.

3.1. Axial Stiffness

We denote the axial displacement at the end of the column by δ, and the slope at each end by θ as shown in Figure 4, and the column is taken to have an initial (unloaded) curvature characterized by θ_0. We obtain the $P - \theta$ relation for the post-buckled column in the form

$$P = P_{cr} \frac{\theta - \theta_0 - \theta_F + \frac{1}{3}\theta_F \theta^2}{\theta(1 - \frac{1}{8}\theta^2)} \tag{2}$$

where P_{cr} is the critical axial force of an initially straight column $P_{cr} = \frac{\pi^2 EI}{l^2}$, and θ_F is the approximate endpoint slope under the lateral force F only

$$\theta_F = \frac{2Fl^2}{\pi^3 EI} = \frac{2F}{\pi P_{cr}} \tag{3}$$

The axial deformation δ can be obtained as

$$\delta = \frac{l}{4}(\theta^2 - \theta_0^2) - \frac{l}{64}(\theta^4 - \theta_0^4) + \frac{Pl}{EA} \tag{4}$$

where A is the the cross-sectional area of the column.

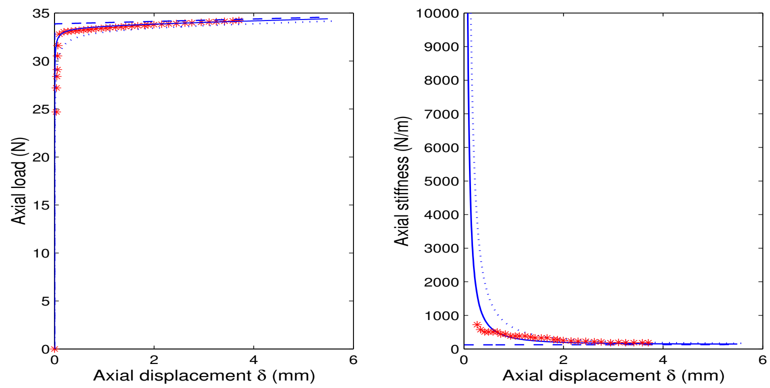

Figure 5. Characteristics of axial load and stiffness versus axial displacement with $\theta_F = 0$: measured (stars), $\theta_0 = 0$ (dashed), $\theta_0 = 2e-3$ (solid), $\theta_0 = 5e-3$ (dots); where $E = 210$ GPa, $l = 137.2$ mm ($f \simeq 1$ Hz from Equation (6)), column width 25.4 mm and thickness 0.526 mm

From the preceding expressions, we find that an initially straight column with no lateral force exerted at the midpoint has an axial stiffness

$$k_p \simeq P_{cr}/2l \tag{5}$$

In the isolation system, we require that $P_{cr} \simeq mg$, where m is the mass of the payload. We find therefore that the natural frequency of the post-buckled system depends only on the column length

$$f = \sqrt{\frac{k_p}{m}} \simeq \sqrt{\frac{P_{cr}}{2ml}} \simeq \sqrt{\frac{g}{2l}} \tag{6}$$

An experiment in which the platform was supported on a spring-steel column was carried out. Figure 5 compares the measured and predicted axial load and stiffness for various axial displacements as the mass of the platform is increased. We see that a 1 Hz resonant frequency can be achieved with only a few millimeters of axial deformation, rather than the 24.8 cm required using linear springs.

3.2. Coarse Position Adjustment

Equations (2) and (4) suggest that we can adjust the position by exerting a force F at the midpoint, as can practically be achieved using a very soft spring on a moveable mount. Figure 6 shows the relationship of F and the static position for a constant axial load $P = 1.01P_{cr}$ with an initial imperfection $\theta_0 = 2e-3$. From the figure, we see that it is possible to adjust the position over a range of approximately a millimeter without changing the stiffness significantly.

4. CONTROL PERFORMANCE

Here, we outline the results of vibration control using the voice coil actuator in parallel with the buckled beam. The voice coil is driven by a voltage amplifier and hence the damping of the system is increased from around 2% to 20%. Three types of control strategies are employed separately: velocity feeback with a constant gain, FIR adaptive feedforward control[8] based on base vibration, and the combination of the feedback and adaptive feedforward controllers. The contollers are implemented using a dSpace 1103 DSP system.

For the results given here, the FIR filter length is 256 and the sampling frequency is 500 Hz. The adaptive FIR filter coefficients will converge to different values dependent on the base excitation. For example, if the base excitation is predominantly sinusoidal at 1 Hz or 5 Hz, the adaptive feedforward controller achieves more than 60 dB more attenuation than the passive isolator.

To obtain clean measurements of the base-to-platfrom velocity transmittance, we place the entire isolation system atop a small optical table whose mass is approximately 100 kg and is supported relative to ground with

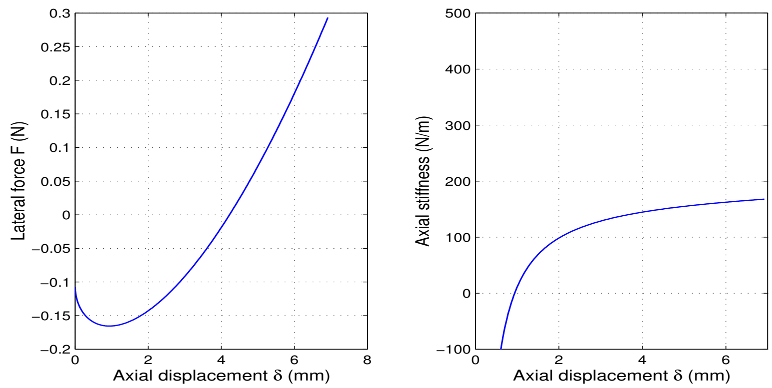

Figure 6. Relation of lateral force F and axial displacement δ and the corresponding stiffness for a constant loading force $P = 1.01P_{cr}$ with geometric imperfection $\theta_0 = 2e-3$. Where $E = 210$ GPa, $l = 137.2$ mm, beam width 25.4 mm, thickness 0.526 mm

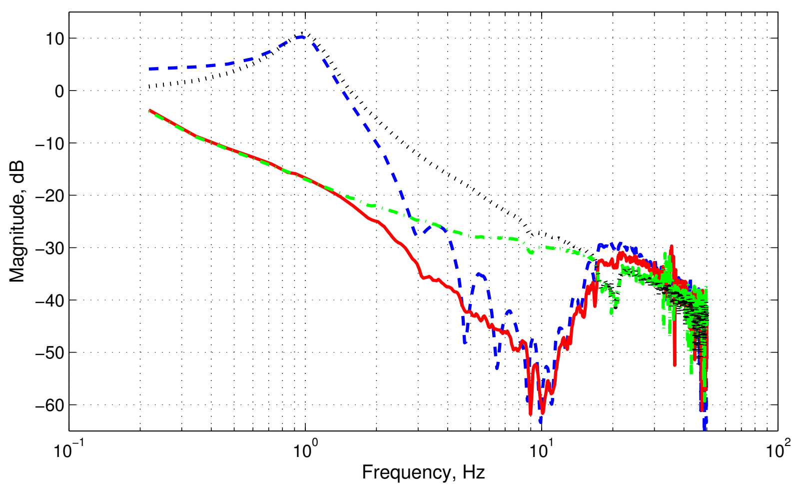

Figure 7. Vibration transmission from base to platform: passive system (dot), feedback control (dash dot), adaptive feedforward control (dash), and combination of feedback and adaptive feedforward control (solid)

a first resonance of approximately 10 Hz. A shaker is attached to the table to provide a predominantly vertical base excitation. A geophone mounted to the table and corrected to 0.1 Hz (in the same manner as the payload geophone) measures the base motion.

Figure 7 compares the vibration transmission from the base to the platform for the passive system and the system with three different controllers. The feedforward controller attenuates vibration most strongly at 10 Hz because the base resonance leads to relatively large disturbances at this frequency and the FIR filter adapts accordingly. The combination of feedback and feedforward control obtains 20-30 dB vibration reduction over the entire range of frequencies from 0.6 to 13 Hz.

A capacitive sensor is employed to measure the relative position of the platform and base for feedback control. The results of position control, as well as a comparison of the performances achieved using the voice-coil and piezoelectric actuators, will be presented at the conference.

5. CONCLUSION

The combination of a buckled column and series or parallel actuators provides a simple and compact method for precision positioning and isolation systems, providing for coarse adjustment over a relatively large displacement without a large variation in stiffness. The combination of adaptive feedforward and feedback control obtains strong attenuation of seismic vibration beyond that of the passive system over a broad frequency range.

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