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Viscous Damped Space Structure for Reduced Jitter

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ABSTRACT

A technique to provide modal vibration damping in high performance space structures has been developed which uses less than 1 ounce of incompressible fluid. Up to 50 percent damping can be achieved which can reduce settling times of the lowest structural mode by as much as 50 to 1. This concept allows designers to reduce the weight of the structure while improving its dynamic performance.

Damping provided by this technique is purely viscous and has been shown by test to be linear over 5 orders of input magnitude. Amplitudes as low as 0.2 microinch have been demonstrated. Damping in the system is independent of stiffness and relatively insensitive to temperature.

This high resolution damping technique also complements active structural control systems by reducing the structure's amplification ratios (Q) so that active compensation becomes practical.

INTRODUCTION

In the past, engineers have required the structural designer to set the fundamental vibration modes of the spacecraft above the control frequencies. As space structures become larger, this is impractical because designers are having to deal with greater structural flexibility and high Q vibrations; consequently active and passive damping techniques are being explored. The active systems suffer from high cost, lower reliability, less coupling between modes and poor low-level or threshold performance. Passive systems, which to date are primarily viscoelastic, suffer from low damping ratios, sensitivity to temperature, predictability problems and flight qualification difficulties.

A new concept using an incompressible viscous fluid provides some immediate solutions, -- very high damping, linear predictable performance, acceptable temperature sensitivity, very easy qualification for long life space application and relatively low cost.

The concept involves integrating a purely viscous damper using incompressible fluid into the basic strut element of a truss structure. Viscous forces of very high value, roughly equal to the spring or structural compliance force in the strut or tube can be developed. One specific implementation uses a tube within a tube arrangement (see Figure 1). The outer tube provides the basic high stiffness-to-weight characteristic provided for any normal truss structure element. The inner tube is in series with an incompressible fluid which is squeezed through a long orifice that provides a pure velocity sensitive force. The force depends on the viscosity of the fluid and the geometry of the orifice, and can be changed over a wide range. The velocity-sensitive force can be made significant compared to the spring force, and thus provide relatively high damping ratios. It might appear that the dual tube would weigh twice that of the conventional single-tube arrangement, but a structure designed for dynamic performance using highly damped dual-tube struts (D-Strut) could be lighter, since overall static stiffness would be reduced.

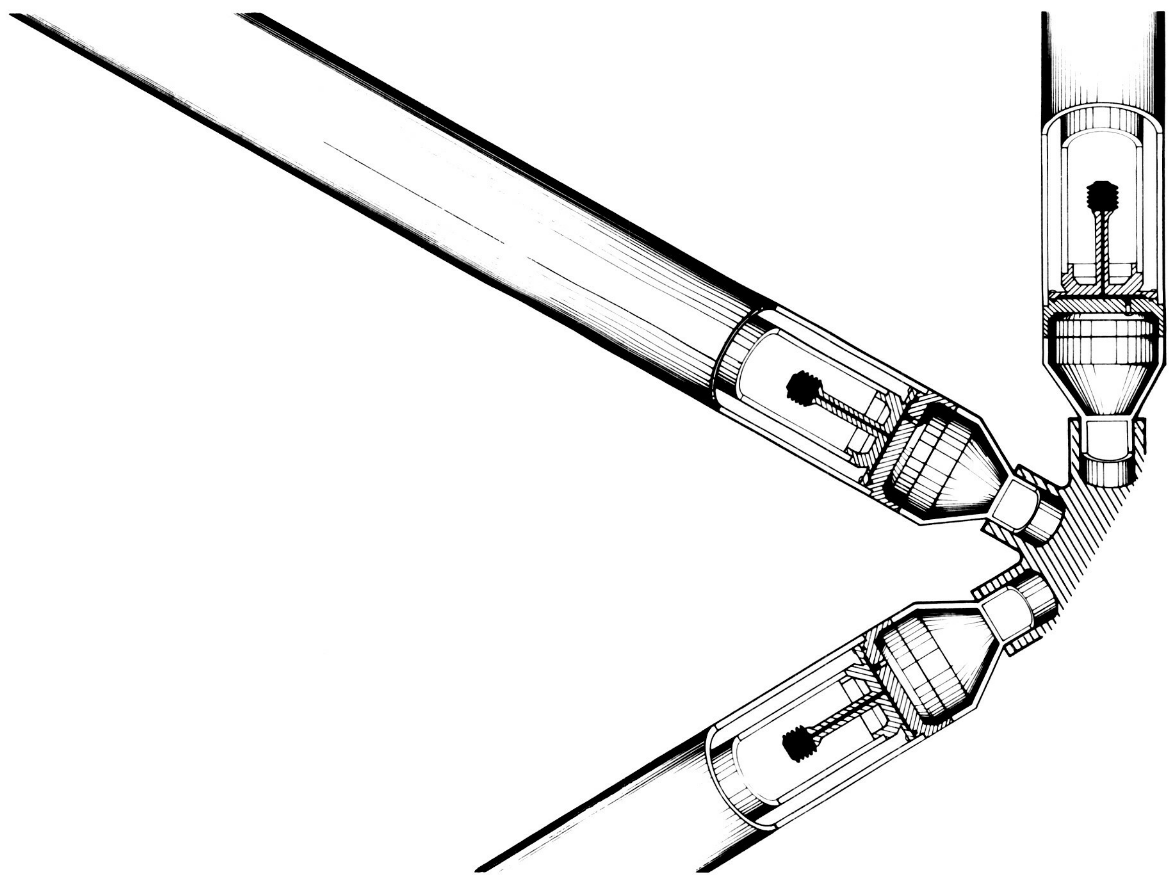


Fig. 1 D-Strut Shown in Truss Arrangement

A means is presented of implementing viscous fluid damping elements for optimal control of structure settling time. A heritage space-qualified damper design has been demonstrated to provide constant damping rate over measurable ranges of amplitude and frequency input. Its linear behavior and deterministic design characteristics allow the structural designer to truly optimize the spacecraft as a dynamically stable platform.

HERITAGE DEVELOPMENT

High structural damping concepts are an evolution of an existing space-qualified vibration-isolation design. This device was used to improve pointing performance of the Hubble Space Telescope (HST) by isolating the Reaction Wheel Assemblies (RWAs) from the space telescope structure. The satellite Attitude Control System employs four RWAs with approximately 220N (50 lb) rotors operating at variable speeds up to ±3000 rpm. Low level forces (millinewton) are produced at many harmonics of wheel speed so that all sensitive frequencies are swept by disturbances during RWA operation. During target acquisition the telescope must maintain precise alignment (<.007 arc sec rms) for periods up to 24 hours. Broadband isolation offered a solution but the application is unique in that performance is required at very low disturbance levels. This was achieved with the device shown in Figure 2. Metal springs act in parallel with a viscous fluid damping element. The spring and damper are physically independent and individually deterministic, permitting precise design for any desired dynamic parameters. Stiffness is provided by coil springs operating in parallel with metal bellows. The springs are positively preloaded to preclude deadband or nonlinearity around the null position. Damping is provided by viscous flow or silicon damping fluid through the annular damping chamber during payload motion. The damping rate is determined by the viscosity of the fluid and the dimensions of the damping chamber. Damping rate has been experimentally verified to be constant over at least five orders of input displacement magnitude. Damping rate is insensitive to input frequency in the region of interest and varies by approximately 2:1 over the qualification temperature range of -20 to +120 °F. The insensitivity to amplitude, frequency, and temperature is in marked contrast to more conventional means of passive damping and facilitates accurate dynamic modeling. Implementation of the isolation system on the HST resulted in approximately 130:1 reduction in peak RWA-induced disturbance in the 0-120 Hz region of interest. This is shown in Figures 3 and 4 where speed maps of RWA disturbances during wheel rundowns are plotted, with and without the isolation system.

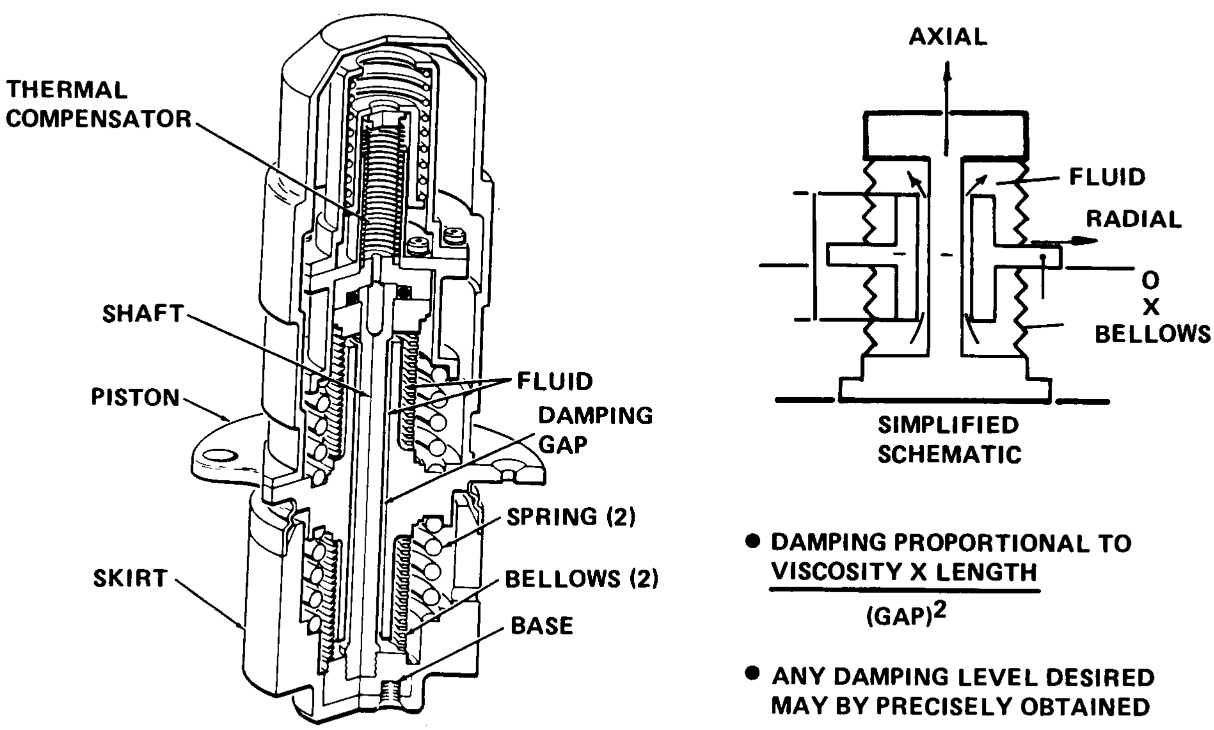


Fig. 2 Heritage Isolator Design

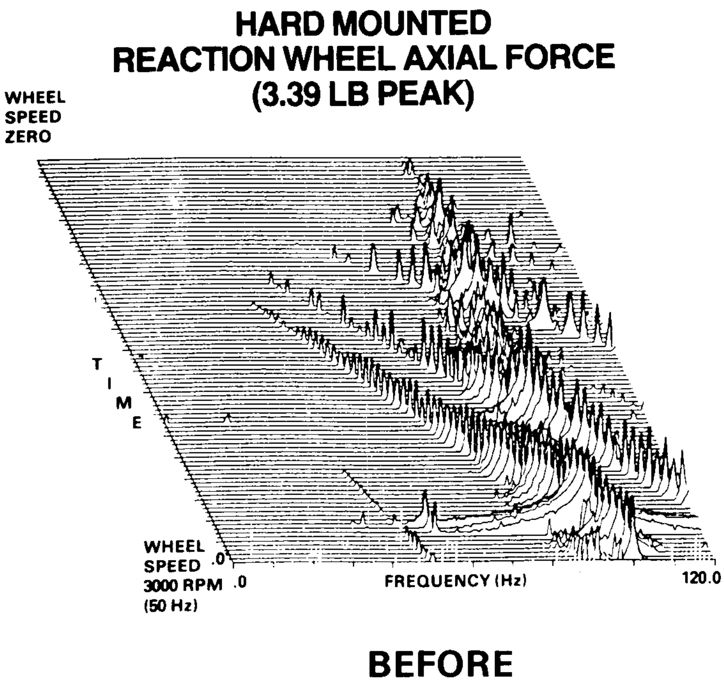


Fig. 3 RWA Disturbances Without Isolation

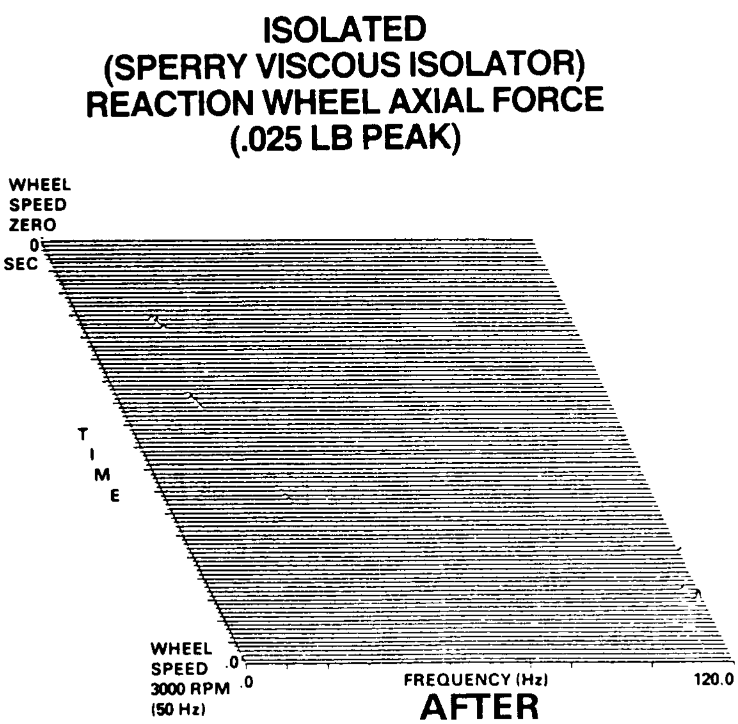


Fig. 4 RWA Disturbances With Isolation

VERY HIGH DAMPING IMPLEMENTATION

The deterministic performance of the HST damping element suggested application to the large space structures area, where very high damping rates at low excitation levels are desirable. An implementation for a standard truss element, the D-Strut, is shown in Figure 5. In this concept concentric truss tubes are connected by a fluid damping element. Design considerations for high damping rate include minimizing the fluid volume to limit compressibility effects and maximizing the ratio of plunger area to damping chamber area. Dampers of this design have demonstrated rates greater than 1000 lb-sec/in. in laboratory testing. A lumped-parameter model of the D-Strut is also shown in Figure 5. The springs k1 and k2 represent the outer and inner truss tubes and c represents the cylindrical damping chamber. The spring k3 represents the axial stiffness due to bending of the thin annular diaphragm. The spring k4 results from a combination of volumetric compression of the fluid and volumetric expansion of the fluid cavity. Due to the series/parallel arrangement of k3 and k4 in the damper, it is desirable to make k3 as small as possible and k4 as large as possible. Optimum sizing of the damper and truss tubes is determined by the structure dynamic requirements and is discussed in this paper. Although only k1 resists static loading, the complete system of dynamic elements resist dynamic loading. This leads to an optimum design since for a satellite on orbit the static loading arises from solar wind, gravity gradient, and other very low amplitude sources. The requirement of a satellite structure is to resist dynamic loading, providing a stable platform with maximum bandwidth and disturbance insensitivity. The optimum solution of this problem requires consideration of both stiffness and damping in the structural design.

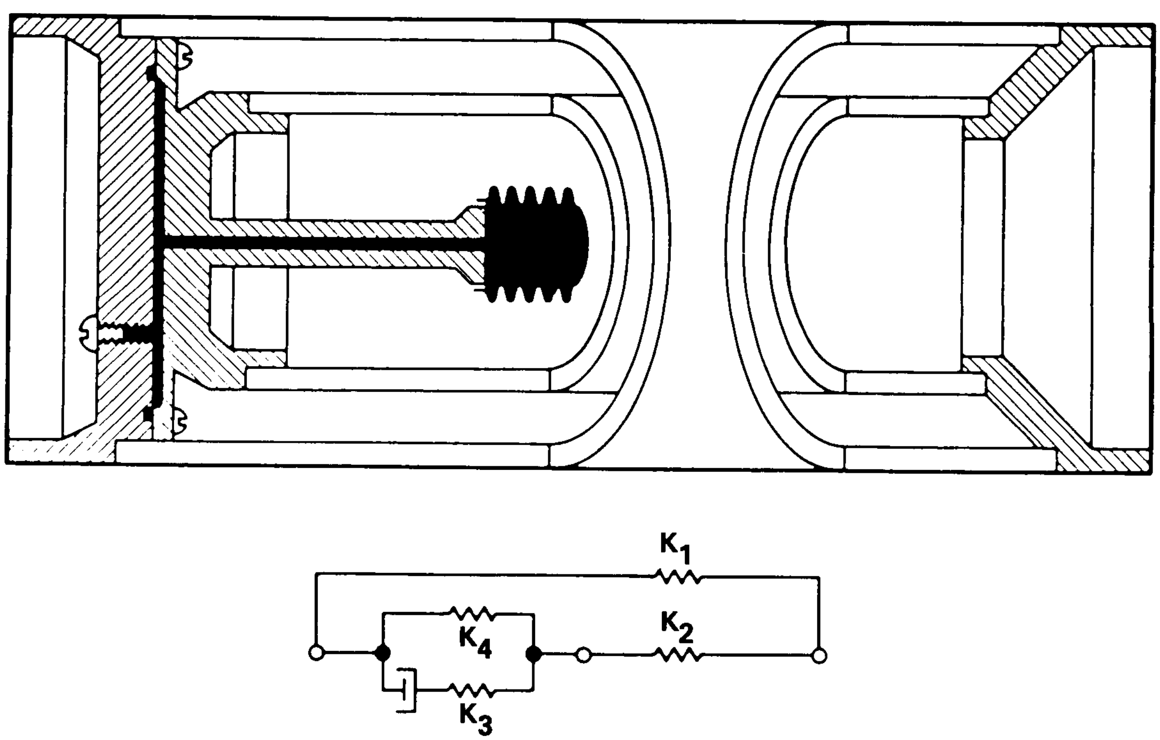


Fig. 5 D-Strut Truss Element

MODELING OF HIGHLY DAMPED STRUCTURES

When damping becomes a significant fraction of critical, its effect on a structure's natural frequencies and mode shapes is not negligible. Since damping on the order of 10 to 20 percent of critical is anticipated, special modeling techniques are required. Computer algorithms for solution of the complex eigenvalue problem

$$Kx + sCx + s^2Mx = 0 \tag{1}$$

are available and the most direct approach would be to include all the dynamic elements. This would require using four nodes per truss element and would lead to fairly costly complex eigensolutions. If the interior nodes in the model in Figure 5 are assumed to be massless, the equations of motion for a damped strut become

$$Zo \begin{bmatrix} k & -k \\ -k & k \end{bmatrix} \begin{Bmatrix} x1 \\ x2 \end{Bmatrix} + s^2 \begin{bmatrix} M & o \\ o & M \end{bmatrix} \begin{Bmatrix} x1 \\ x2 \end{Bmatrix} = \begin{Bmatrix} o \\ o \end{Bmatrix} \tag{2}$$

where Zok is the complex impedance of the strut and Zo is a dimensionless complex number. If a model is assembled from a number of these elements, each with approximately the same value of Zo, this term can be factored out of the equations. Under these assumptions the damping affects the natural frequencies but not the mode shapes of the model. The decoupled frẽquency equations become

$$Zo + S^2/pi = 0 \tag{3}$$

where pi is the natural frequency obtained from the real eigenvalue solution obtained by setting Zo = 1. For the damped strut of Figure 5, if k3 = 0 and k4 = ∞;

$$Zo = \frac{\alpha(1 - \alpha) + \beta\, s/p}{1 - \alpha + \beta\, s/p} \tag{4}$$

where kl = α k, k2 = $(1 - \alpha)$k, and β = cp/k. The frequency equation becomes

$$0 = \alpha(1 - \alpha) + \beta(s/p) + (1 - \alpha)(s/p)^2 + \beta(s/p)^3 \tag{5}$$

Equation (5) always has one real negative root and one complex pair with a negative real component for physically-realizable parameters α and β. The complex pair is associated with normal exponentially-damped oscillation. The real root is associated with nonoscillating motion of the assumed massless internal node. The roots are plotted as a function of β for α = 1/2 in Figure 6. The time constraints associated with the mode are l/rp for nonoscillating motion and l/xp for oscillating motion. Since r is always greater than x, the minimum settling time design is where x is a maximum, β = .5. The optimal damper value is c = .5k/p and the fraction of critical damping is s/p = .175.

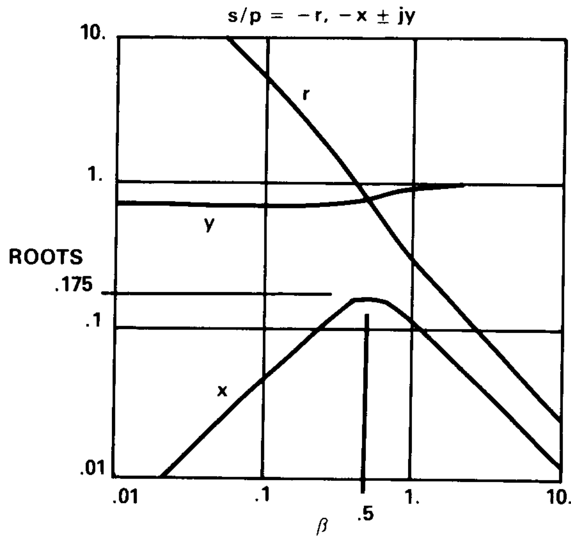
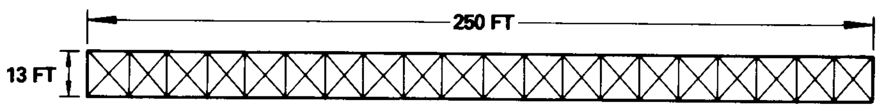


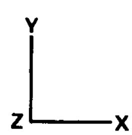
Fig. 6 Roots of Frequency Equation

GENERIC TRUSS EXAMPLE

In Figure 7 a generic truss structure is shown such as might be used for the Space Station or other large space structures. The frequencies of the first 20 modes are tabulated. The assumed goal of the structure designer is to make the longest system time constant as small as possible. The lowest mode is at 1.55 Hz for the lightly damped structure. If the damping associated with this mode is 1 percent (Q = 50), the time constant will be 1/.01x2x π x1.55 = 10 sec. If highly damped struts are used with a = 1/2 (1/2 of existing structure devoted to damping), the frequency of the mode will drop to 1.33 Hz but the time constant will become 1/.175x2x π x1.55 = .58 sec.



GRAPHITE EPOXY TUBING
2 IN OD × .06 WALL



0 ROOT NUMBER	FREQUENCY (CYCLES/UNIT TIME)
1	1.5502
2	2.7361
3	3.6807
4	4.0358
5	5.3738
6	6.7065
7	7.3319
8	8.0105
9	9.2717
10	10.4813
11	10.9306
12	11.6318
13	12.7172
14	13.7304
15	14.6671
16	15.5136
17	16.2750
18	16.7708
19	17.2084
20	17.4099
21	18.0131

Fig. 7 Generic Truss

To achieve the same time constant reduction by stiffening the structure would require raising the first mode to over 27 Hz, an improbable goal. In Figure 8 the time constant reductions for the lowest few modes are shown. Only the lower modes with long time constants are affected. Since the damping is applied in every member the result is insensitive to configuration changes that modify the mode shapes. The damping elements are sized to be most effective at 1.5 Hz and the system effectiveness would be compromised if the frequency of the lowest mode were to change significantly.

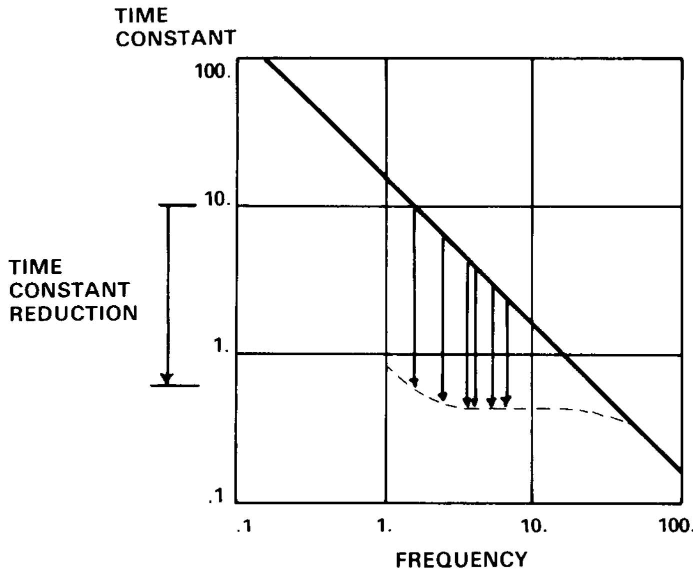


Fig. 8 Time Constant Reduction for Generic Truss

In Figure 9 the effect of high passive damping on structural control considerations is shown. Control bandwidth is often limited by the requirement to provide some level of margin, such as the 6 dB shown at the first structural resonance. Resonances with Qs of 3 and 50 are shown occurring at the same frequency. The very low Q system can implement approximately one order of magnitude higher bandwidth because of the greatly reduced gain at the first resonance.

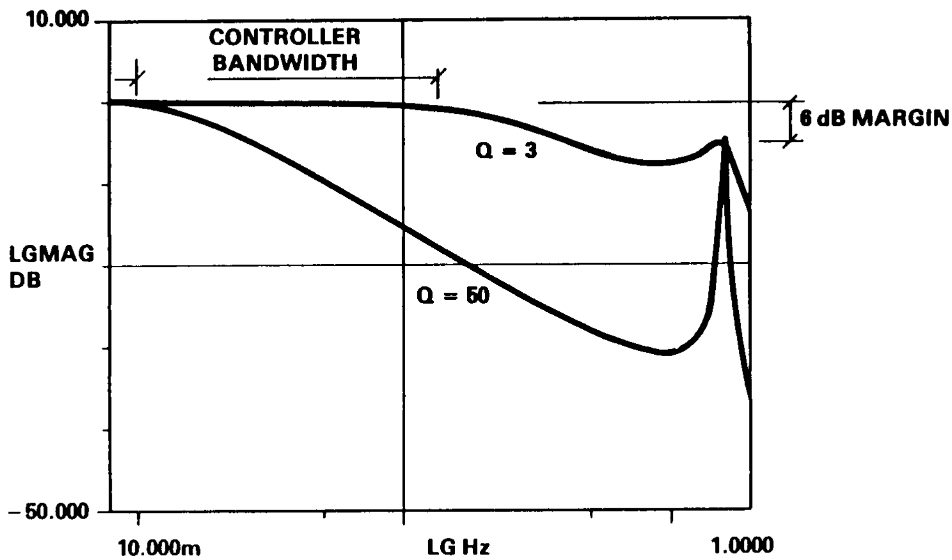


Fig. 9 Bandwidth Increase for Highly Damped Structure

SUPPLEMENTAL DAMPING IMPLEMENTATION

In some applications it may be desirable to merely add supplemental damping rather than build the entire structure from damped elements. One possible implementation of this is shown in Figure 10. Here a damping element is mounted in series with a stiff structural tube to provide damping along the axis of the strut. This strut may be derived from the one in Figure 5 by removing the outer tube. Such a strut might be used to limit response of a localized mode to the launch environment or on-orbit disturbance sources.

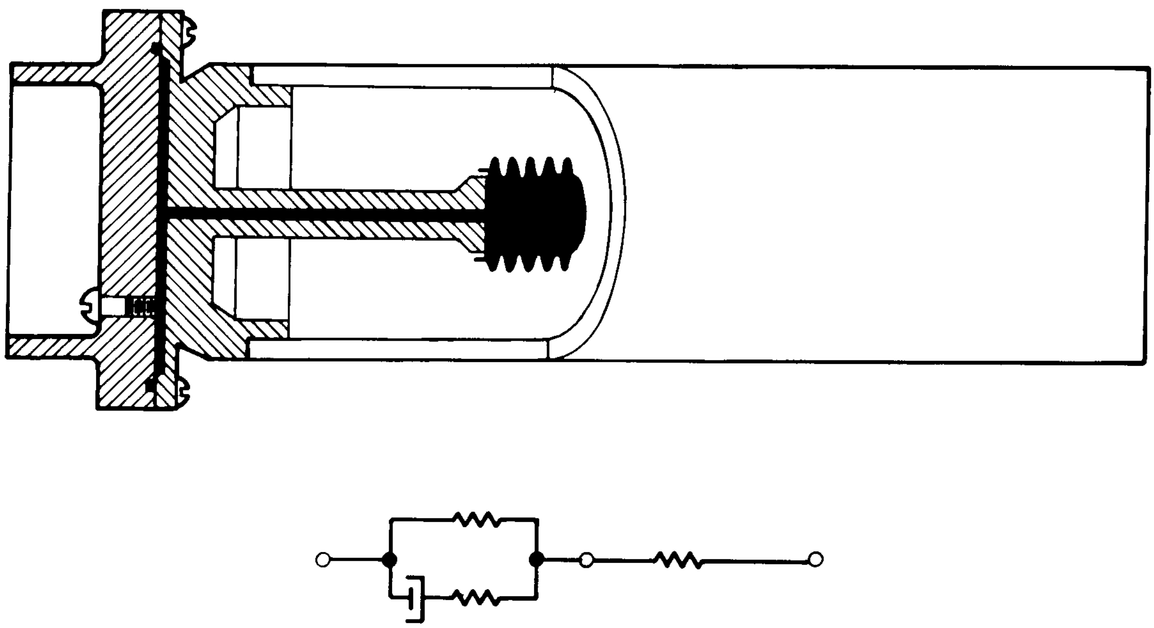


Fig. 10 Supplemental Damper

MODELING OF SUPPLEMENTAL DAMPING

It is assumed that a real eigenvalue solution has been performed for the lightly damped structure and natural frequencies p and mass-normalized mode shapes are available for the modes of interest. The complex impedance of the series spring damper is

$$Z = k \frac{cs/k}{1 + cs/k} \qquad (6)$$

and at the frequency of interest

$$Z = k \frac{jcp/k}{1 + jcp/k} \qquad (7)$$

The impedance may be expressed in real and imaginary components,

$$Z = k \frac{(cp/k)^2 + j(cp/k)}{1 + (cp/k)^2} \qquad (8)$$

These are plotted in Figure 11. The optimal design is that for which the imaginary component is maximum, with $c = k/p$. The value of the imaginary impedance at this point is

$$\text{Im}\,(Z) = .5cpj = .5kj \tag{9}$$

indicating the damper loses half its effectiveness due to the presence of the series spring. If ϕ is the mass-normalized modal displacement across the strut, the resultant modal damping ratio is

$$\zeta = c\phi^2/4p \tag{10}$$

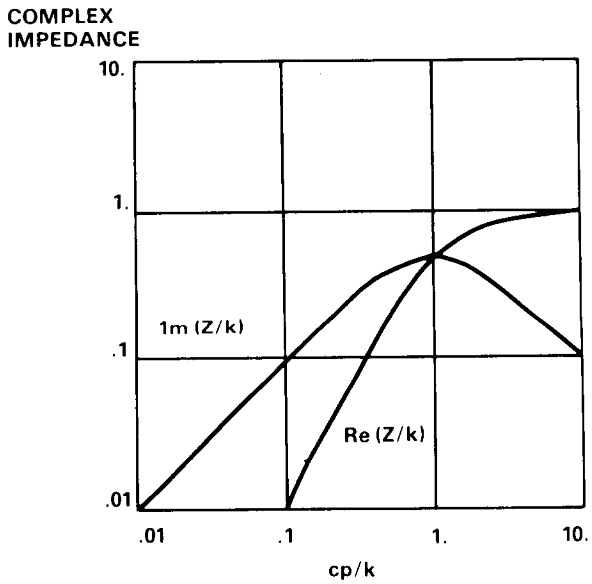


Fig. 11 Complex Impedance

CMG GIMBAL SUPPORT EXAMPLE

A proposed design for a CMG mounting application employed supplemental damping to reduce the CMG support loads during launch. Two stacks of two CMGs each were employed, as in Figure 12. Because of envelope restraints the stacks were cantilevered, although hard points were available near the top of the stacks. The stack consists of two 750N (170 lb) CMGs mounted in a cylindrical aluminum Gimbal Mounting Structure. The cylinders are approximately .6m in diameter and 1.3m high. Their fundamental rocking mode was predicted to be at 31 Hz. Consideration was given to structurally tie the tops of the stacks to the hard points to increase overall strength. This also has the effect of raising the frequency of the rocking mode and possibly increasing the input vibration level. It was decided to add a damping element rather than a stiffener to exploit whatever isolation may be obtained from the 31 Hz resonance while limiting its response. The proposed strut employed an aluminum tube with $k = 3.5 \times 10^6$ N/m (20,000 lb/in.), including joint flexibility. For optimum energy absorption the damper rate is

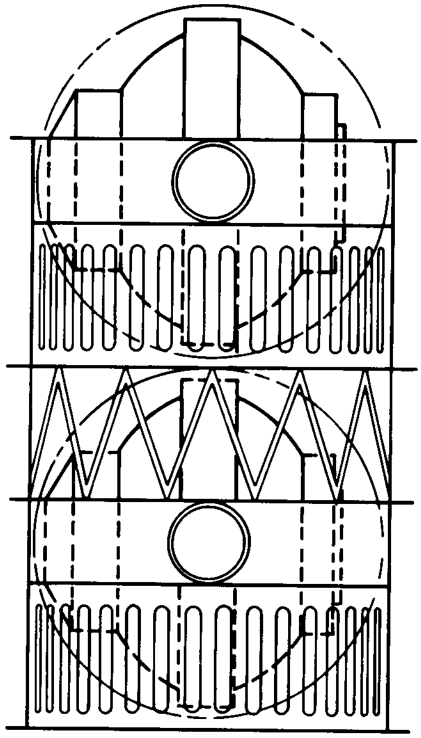


Fig. 12 CMG Mounting Application

$$c = k/p = 18{,}000 \text{ Ns/m } (103 \text{ lb sec/in.}) \tag{11}$$

The modal displacement predicted across the strut is 1.21 and the increase in modal damping ratio is given by (10) as

$$\zeta = c\phi^2/4p = .19 \tag{12}$$

This solution produces a low frequency isolation mode with a Q of approximately 2.5 and appears to offer the optimum launch load reduction for the CMG stacks.

SUMMARY

The current concepts for large space-structure design are the evolution of a space-qualified precision isolation system design. Damper elements with very high damping rates, >170,000 N.S/m (1,000 lb-sec/in.), have been demonstrated in the laboratory. Analytical techniques for optimal implementation of these devices in large space structures are presented. Significant reductions in structure settling time appear feasible with optimal use of damping in the structural design. Means are presented for controlling the gains of all the modes in a given frequency region, facilitating active control techniques. A method for optimum implementation of supplemental damping for localized modes is also presented. These design approaches appear uniquely suited to satellite applications where static loading is virtually nonexistent and the requirement is for a dynamically stable mounting platform.