AN ADVANCED D-STRUT™

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ABSTRACT

A viscous-damping technique offering high damping for spacecraft truss structures has been under development since 1986. The technique, known as the D-Strut™, uses a small mechanical viscous damper configured in an inner-outer tube-strut configuration, and replaces the nominal-type strut. The viscous-damped D-Strut has been employed in more compliant isolation systems for space applications, including the Hubble Space Telescope.

The United States Air Force and Jet Propulsion Laboratory have investigated D-Struts for use in high specific-stiffness truss structures. This technique is an attractive means of attaining significant damping levels in space structures.

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INTRODUCTION

A viscous-damping technique that offers high damping for spacecraft truss structures has been under development now for several years (References [1-3]). The technique, known as the D-Strut™, employs a small, mechanical viscous damper configured in an inner-outer tube-strut configuration. The D-Strut serves as a basic element in a truss structure, replacing the nominal-type strut. The viscous-damping concept, employed in more compliant isolation systems, has been qualified for at least three space applications and is currently flying in the Hubble Space Telescope, where the function is to isolate disturbances emanating from the attitude control reaction wheel assembly (References [4-5]).

The United States Air Force and the Jet Propulsion Laboratory (JPL) have investigated the use of D-Struts in high specific-stiffness truss structures. With considerable development heritage, the viscous D-Strut now provides an attractive means for attaining significant damping levels in space structures. The D-Strut is simple in design and construction, is easy to model, and is readily incorporated into the overall structure design and analysis process.

The advantages of the D-Strut are:

- Very large dynamic range (no rubbing friction or hysteresis)
- Damping independent of stiffness
- High damping
- Low temperature sensitivity compared to viscoelastic materials
- Adjustable performance
- Linear and predictable performance
- Qualified for space application
- Hermetically sealed fluid (fluid exposed satisfies outgassing and mass transfer requirement of NASA)

Design alternatives within the basic concept provide a variety of performance options. Design improvements continue to provide better performance, nearing that of an *ideal damper*. The reference to *ideal* refers to a damper which can be modeled simply as a spring and dashpot in parallel. The following several paragraphs expand on this consideration and develop the necessary mathematics for a more complete understanding. Following that, an improved ***arched*** flexure design with test results is presented. Finally, a glimpse of future plans is provided.

D-STRUT CONFIGURATION

The first D-Struts built, shown in Figure 1(a), employ three basic elements: a small viscous damper, an inner tube, and an outer tube.

The damper is placed in series with the inner tube and the damper/inner tube is placed in parallel with the outer tube. An axial displacement across the strut produces a

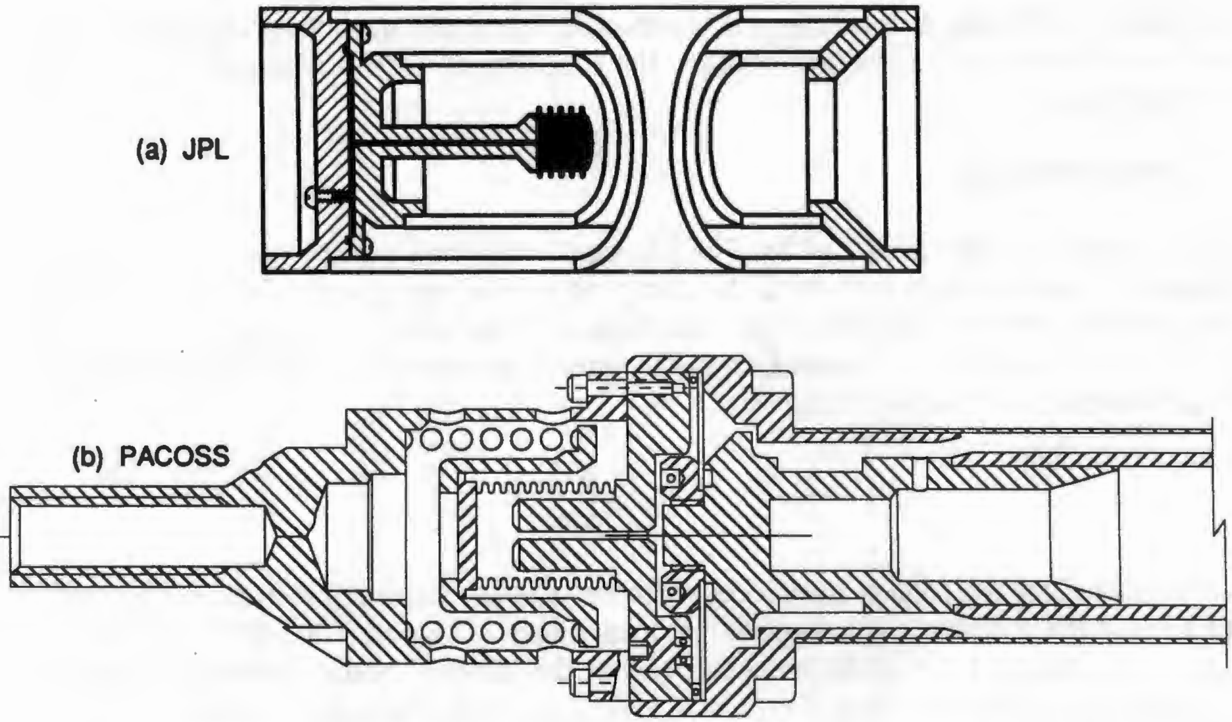

Figure 1. Diaphragm Flexure D-Struts for JPL and PACOSS

displacement across the damper. Under an axial displacement, the damper forces fluid through a small-diameter orifice, thereby causing a shear in the fluid. The fluid shear is proportional to the displacement rate across the damper; thus, a true viscous-damping force is obtained (i.e., a force proportional to velocity).

The compliances of the damper, the inner tube, and the outer tube are key to the damping performance of the D-Strut. The damper is the most-compliant element and the inner tube is the least-compliant element. The outer tube provides the basic static stiffness of the strut and is pertinent to applications where the strut is a critical load-bearing element in the structure. Otherwise, the outer tube is not necessary and can be eliminated with a resulting improvement in damping performance.

The damper element consists of two compliant metal cavities connected by a small-diameter orifice of a certain length. The damper cavities are hermetically sealed to avoid outgassing and fluid loss. The damper is mechanically simple, has no moving parts or wear mechanisms, and is completely tolerant of space vacuum and radiation.

A diaphragm flexure D-Strut tested by JPL is shown in Figure 1(a). A second diaphragm flexure D-Strut was developed for PACOSS program (Reference [3]) and is illustrated in Figure 1(b). Both systems were tested as single elements and as an integral part of a truss structure. Twelve D-Struts were used in the PACOSS structure.

The flexing of a metal diaphragm is the mechanism that forces fluid through the small orifice. The advanced designs replace the diaphragm with a convoluted cylinder or *arched* flexure.

D-STRUT MODEL

A D-Strut is readily modeled by five physically lumped parameters, as indicated by Figure 2. Considerable insight to the damping performance is gained by regarding the D-Strut as a *mechanical impedance.* Mechanical impedance is somewhat analogous to electrical impedance and related, in the frequency domain, the axial force f to the axial displacement x across the strut:

$$z(s) \equiv \frac{f(s)}{x(s)} \tag{1}$$

with s denoting the *Laplace* transform variable. For no damping, the impedance reduces to a standard stiffness. The mechanical impedance is a good characterization of the D-Strut behavior as long as the mass lumped at the internal nodes, labeled N_3 and N_4 in Figure 2, is negligible. This is typically a very good approximation over the frequency range of interest.

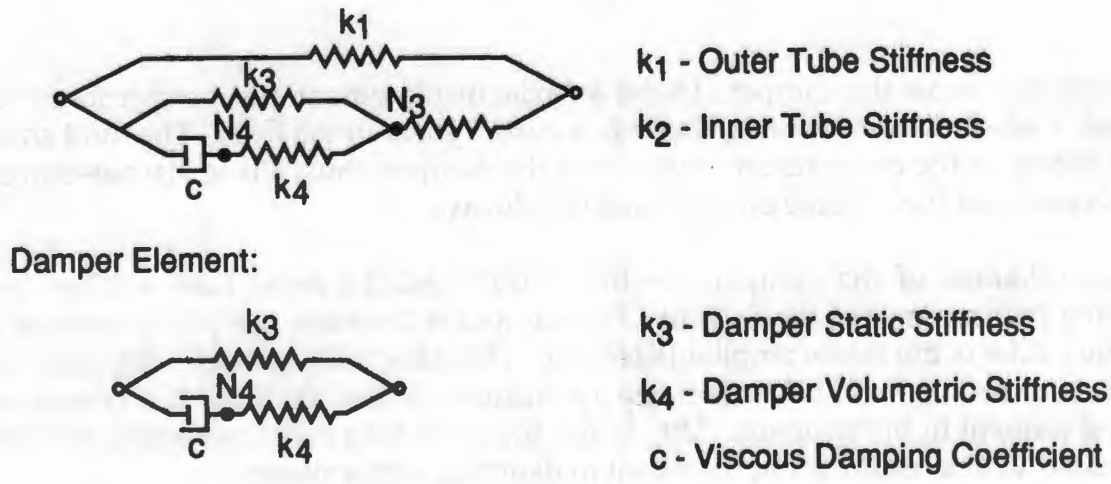

Figure 2. Inner-Outer Tube D-Strut Equivalent Model

The impedance of a D-Strut is a function of three parameters and has a classic *lead-lag* network characteristic:

$$z(s) = k_s \frac{\omega_p}{\omega_z} \frac{s + \omega_z}{s + \omega_p} \tag{2}$$

with:

$$k_s = \frac{k_1k_2 + k_1k_3 + k_2k_3}{k_2 + k_3} \tag{3}$$

$$\omega_z = \frac{k_4}{c} \frac{k_1k_2 + k_1k_3 + k_2k_3}{k_1k_2 + k_1k_3 + k_1k_4 + k_2k_3 + k_2k_4} \tag{4}$$

$$\omega_p = \frac{k_4}{c} \frac{k_2 + k_3}{k_2 + k_3 + k_4} \tag{5}$$

Because the impedance depends only on three parameters, an equivalent three-parameter physical model of the D-Strut can be obtained, as indicated in Figure 3.

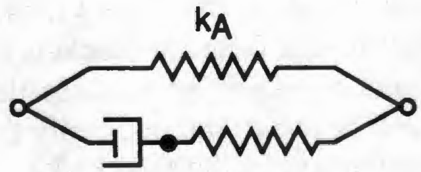

Figure 3. Equivalent D-Strut Model

The equivalent model impedance is:

$$z(s) = k_A \frac{\omega_B}{\omega_A} \frac{s + \omega_A}{s + \omega_B} \tag{6}$$

with:

$$\omega_A = \frac{1}{c_A} \frac{k_A k_B}{k_A + k_B} \tag{7}$$

$$\omega_B = \frac{k_B}{c_A} \tag{8}$$

The relation between the parameters $\{k_A, k_B, c_A\}$and $\{k_1, \ldots, k_4, c\}$ is given by:

$$k_A = k_1 \left[1 + \frac{\frac{k_3}{k_1}}{1 + \frac{k_3}{k_2}} \right] \tag{9}$$

$$k_B = k_2\left[\frac{1}{1+\frac{k_3}{k_2}}\right]\left[\frac{1}{1+\frac{k_2}{k_4}+\frac{k_3}{k_4}}\right] \tag{10}$$

$$c_A = c\left[\frac{1}{1+\frac{k_3}{k_2}}\right]^2 \tag{11}$$

The equivalent model indicates that only three parameters – two equivalent stiffness values and an equivalent dashpot coefficient – are needed to model the D-Strut in overall structure design and analysis. D-Strut testing verifies that this is indeed the case.

The expressions above for $\{k_A, k_B, c_A\}$ show that $k_A \sim k_1, k_B \sim k_2$, and $c_A \sim c$. Actually, k_A is larger than k_1, since its multiplying factor in brackets is larger than 1 and both k_B and c_A are smaller than k_2, and c, respectively, since their multiplying factors in brackets are smaller than 1. It will be shown momentarily that the maximum damping performance of the D-Strut is established by the ratio k_B/k_A. This ratio depends only on the stiffness elements in the damper and the stiffness of the inner and outer tubes. To maximize the D-Strut damping performance, the damper element should be made to approach the characteristic of an ideal dashpot. This is accomplished by driving $k_3 \to 0$ and $k_4 \to \infty$. In this situation $k_A \to k_1, k_B \to k_2$, and the maximum damping performance are established by the ratio of the inner-to-outer tube stiffness k_2/k_1. A damper with nonzero stiffness for k_3 and a finite stiffness for k_4 reduces the D-Strut maximum damping performance from the theoretical limit.

D-STRUT PERFORMANCE

The D-Strut damping performance is easily understood under the condition of sinusoidal displacement and forces. If a sinusoidal displacement:

$$x(t) = X \sin \omega t \tag{12}$$

is prescribed across the D-Strut, then the resulting force developed in the strut is also sinusoidal:

$$f(t) = XA(\omega) \sin(\omega t + \phi(\omega)) \tag{13}$$

where $A(w)$ and $\phi(w)$ are the amplitude and phase angle of the impedance at the frequency w:

$$z(j\omega) = z_R(\omega) + jz_I(\omega) = A(w)e^{j\phi(\omega)} \tag{14}$$

Defining the parameters α as:

$$\alpha \equiv \sqrt{\frac{\omega_B}{\omega_A}} = \sqrt{1 + \frac{k_B}{k_A}} \tag{15}$$

the amplitude and phase of the impedance are given by:

$$A(\omega) = k_A\, \alpha^2 \frac{\sqrt{1 + (\omega/\omega_A)^2}}{\sqrt{\alpha^4 + (\omega/\omega_A)^2}} \tag{16}$$

$$\tan \phi(\omega) = \frac{(\alpha^2 - 1)\,(\omega/\omega_A)}{\alpha^2 + (\omega/\omega_A)^2} \tag{17}$$

A typical impedance characteristic is illustrated in Figure 4.

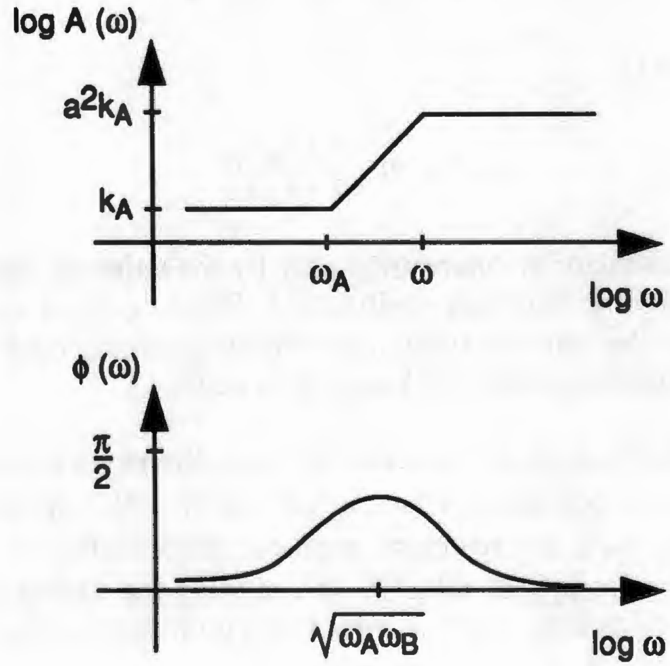

Figure 4. D-Strut Impedance Characteristic

The energy dissipated per cycle due to the damping is determined by $\phi(w)$. In fact, using the classical definition of damping loss factor:

$$\eta(\omega) \equiv \frac{1}{2\pi} \frac{\textit{energy dissipated / cycle}}{\textit{max energy stored / cycle}} \tag{18}$$

then:

$$\eta(\omega) = tan\ \phi(\omega) \tag{19}$$

and the impedance can be expressed as:

$$z(j\omega) = z_R(\omega)\,(1 + j\eta(\omega)) \tag{20}$$

It is easy to demonstrate that the maximum loss factor is given by:

$$\eta^* \equiv \max_{\omega} \eta(\omega) = \frac{1}{2}\,\frac{\alpha^2 - 1}{\alpha} = \frac{1}{2}\,\frac{k_B/k_A}{\sqrt{1 + k_B/k_A}} \tag{21}$$

and that η^* occurs at:

$$\omega^* = \sqrt{\omega_B \omega_A} = \alpha\omega_A = 2\,\frac{k_A}{c_A}\,\eta^* \tag{22}$$

The value of $z_R(w)$ at w^* is:

$$z_R(\omega^*) = 2k_A\,\frac{1 + k_B/k_A}{2 + k_B/k_A} \tag{23}$$

Thus, the maximum loss factor is determined only by the stiffness characteristics of the damper and tubes, not by the damping coefficient *c*. Since k_A is the strut static stiffness, which is determined by the load capability needed, the damping coefficient *c* is used to set the frequency at which the maximum loss factor occurs.

The above equations indicate an equivalence between the *physical* parameters $\{k_A, k_B, c_A\}$ and the *performance* parameters $\{\eta^*, \omega^*, zR(\omega^*)\}$. When *designing* damping performance into a structure, the structure engineer often prefers to work in terms of the performance parameters $\{\eta^*, \omega^*, zR(\omega^*)\}$. In *analyzing* the damping performance of the structure, the physical parameters $\{k_A, k_B, c_A\}$ are more appropriate.

From the above equations it is clear that η^* is maximized by maximizing k_B/k_A. This ratio is related to the four stiffness parameters $\{k_1, \ldots, k_4\}$ by:

$$\frac{k_B}{k_A} = \frac{k_2}{k_1}\;\frac{1}{1 + \frac{k_3}{k_2} + \frac{k_3}{k_1}}\;\frac{1}{1 + \frac{k_2}{k_4} + \frac{k_3}{k_4}} \tag{24}$$

It appears from this equation that, to maximize the damping performance, the stiffness ratios k_2/k_1 and k_4/k_3 should be maximized. The first ratio is that of the inner-to-outer tube stiffness and the second ratio is the damper's *static-to-volumetric* stiffness. The damper volumetric stiffness is due to the fluid bulk modulus and the change in cavity volume due to stretching of the metal under fluid pressure.

Thus, from the performance viewpoint, the damper element of a D-Strut should be designed to have as large a ratio of k_4/k_3 as possible. An achievable stiffness ratio for a typical diaphragm flexure, as designed for the PACOSS program, is $k_4/k_3 = 20$. A significant factor preventing a larger ratio for the diaphragm flexure is the difficulty increasing the volumetric stiffness k_4 while not compromising the strut's static deflection capability, which is determined by k_3. This limiting factor of the diaphragm flexure has lead to the development of an improved damper employing an ***arched*** flexure with the capability of achieving considerably greater ratios of k_4/k_3. As discussed in a latter section, preliminary testing of several prototype designs has indicated attainable ratios of 50, more than double that for the diaphragm flexure.

D-STRUT DESIGN

Performance is not the only consideration in D-Strut design. Strut weight and load capability are two more important considerations. There are four basic elements contributing to strut weight: the inner tube, outer tube, damper elements, and strut end fittings that interface the strut to the structure. A typical damper element employing an ***arched*** flexure weighs approximately 0.1 lb. The end fittings also tend to have a rather small, fixed weight. Thus, the inner and outer tubes are the major weight contributors that vary in the design process.

The tube stiffness is AE/L and the tube weight is $AL\rho$, where A, L, E, and ρ are the tube cross sectional area, length, material elastic modules, and material density, respectively. Thus, for a given tube length and a selected material, the tube weight varies in proportion to its stiffness. Therefore, the sum of the inner and outer tube weights, and thus the strut weight varies as:

$$W_i + W_o \sim k_1\left(a + \frac{k_2}{k_1} b\right) \tag{25}$$

where a and b are constants. The outer tube stiffness k_1 is now the major factor determining the strut's static stiffness (recall that the strut's static stiffness is k_A, which is proportional to k_1). The strut-load requirement essentially establishes k_1 and the strut weight then varies as the stiffness ratio k_2/k_1.

The strut load requirement leads to consideration of allowable stresses and strains in the strut elements. The two most important elements in terms of stresses are the outer tube and the damper. Consider a static-load condition. If x denotes the resulting static displacement across the strut (outer tube), y denotes the static displacement across the inner tube, and β denotes the ratio of the displacement across the damper to the displacement across the strut:

$$\beta \equiv \frac{x - y}{x} \tag{26}$$

then the ratio of the axial stress in the damper to the axial stress in the outer tube is proportional to β. An optimal D-Strut design should tend to have the stresses in the damper and outer tube approximately equal.

Therefore, three important considerations in D-Strut design are performance, weight, and allowable stress in the outer tube and damper. At the first level, D-Strut design involves determining values for the five parameters $\{k_1, \ldots, k_4, c\}$ to address performance, weight, and allowable stress. Specification of performance in terms of the three parameters $\{\eta^*, \omega^*, zR(\omega^*)\}$ leads to conditions for determining three of the five parameters $\{k_1, \ldots, k_4, c\}$. Conditions for determining the remaining two parameters are derived from weight and stress considerations.

To be specific, let the two parameters M and N be defined by:

$$M \equiv \frac{k_2}{k_1} \quad \text{and} \quad N \equiv \frac{k_4}{k_3} \tag{27}$$

M is indicative of D-Strut weight and N is the ratio of the damper's volumetric stiffness to the static stiffness. D-Strut design addressing performance, weight, and stress can be accomplished via the equation:

$$\frac{k_B}{k_A} = \frac{M}{1 + (1 + M)\,\delta}\,\frac{N\delta}{1 + (1 + N)\,\delta} \tag{28}$$

$$\delta \equiv \frac{1 - \beta}{\beta} \tag{29}$$

and the previous equation relating η^* to k_B/k_A. As an example, Figure 5 shows M as a function of N for various values of η^* and a value of $\beta = 0.95$. This figure clearly illustrates the benefit of maximizing the damper's stiffness ratio N. For a fixed level of performance (η^*), maximizing N tends to minimize the D-Strut weight (M). Conversely, for a fixed weight (M), maximizing N leads to improved damping performance (η^*). The ***arched*** flexure damper, described next, is able to attain values of N greater than 50, which provides a significant improvement over the diaphragm flexure $N = 20$).

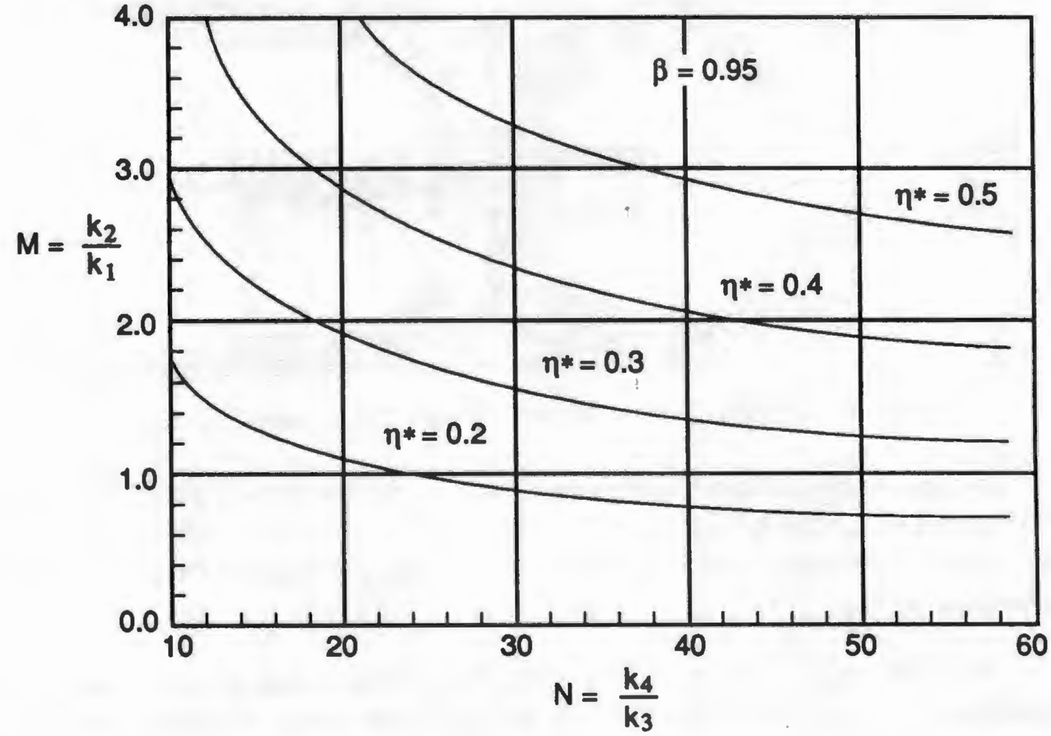

Figure 5. D-Strut Design Curves

ARCHED FLEXURE D-STRUT

The name ***arched*** flexure was chosen because of the similarity with a two-dimensional semicircular arch. The design is more accurately a convoluted or corrugated cylinder. Figure 6 shows a single-convoluted design and Figure 7 shows a multiconvoluted design.

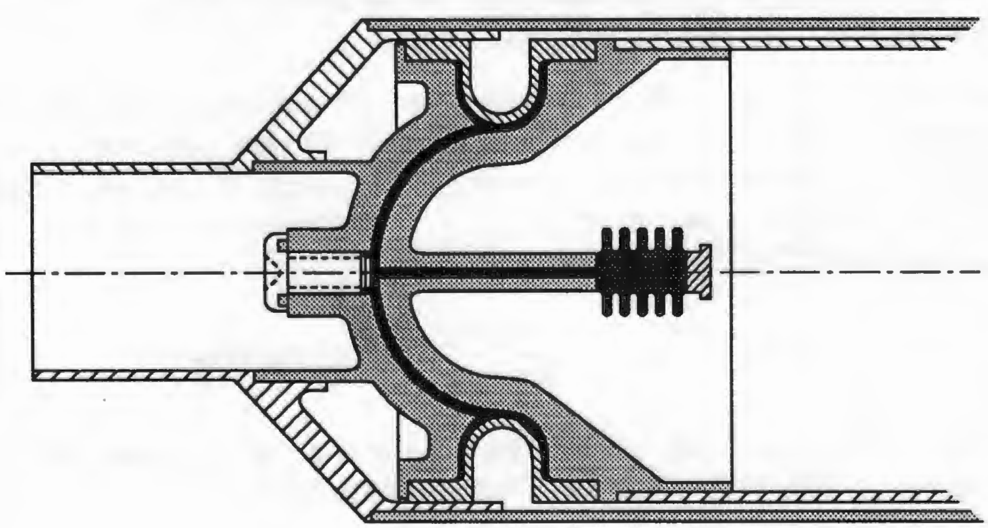

Figure 6. Arched Flexure D-Strut

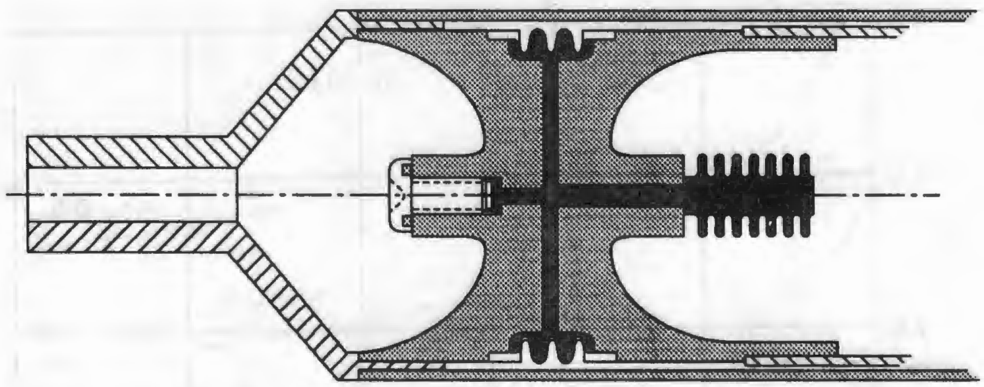

Figure 7. Arched Flexure D-Strut (Multiple Convolutions)

The ***arched*** flexure configuration was selected because its shape will provide the highest possible ratio of k_4/k_3, which in turn will minimize the needed ratio of k_2/k_1. This is equivalent to minimizing k_2,which will also minimize the weight of the system for a given performance.

The volumetric stiffness, k_4, can be characterized as a ballooning effect. It specifically is the axial stiffness of the system that would result if the shear annulus were plugged. Both the flexure and the fluid contribute to k_4. The fluid stiffness is generally not a problem if the depth of the fluid is minimized. The fluid stiffness will range from 1 to 15 million pounds per inch. The D-Strut configuration used in the PACOSS testing, using a diaphragm flexure, typically exhibited a k_3 in the range of 5,000 to 10,000 lb/in. and a k_4 in the range of 100,000 to 250,000 lb/in. The ratio of k_4/k_3 ranged from 20:1 to 25:1.

The ***arched*** flexure has the potential for much higher values. Several single convoluted systems have been fabricated and tested. The result of the first prototypes was a k_4/k_3 ratio of 52 to 1, or a 2-to-1 improvement over the diaphragm designs. Much higher values are expected with a second-generation design.

The k_4/k_3 ratio of 52 to 1 was obtained by dynamic test methods as opposed to direct static-load testing. In the test setup shown in Figure 8, k_3 was measured by removing the fluid, adapting a known mass, and vibrating the system to determine its resonance. One result of such a test is shown in Figure 9. The resonance was 64.2 Hz. The suspended mass was 23 lb. Thus:

$$k_3 \cong (2\,\pi f)^2 \frac{w}{g} = \frac{(2\,\pi \times 64.2\ Hz)^2\ 23\ lb}{386\ rad/s^2} = 9686\ lb/in. \qquad (30)$$

To determine k_4, the fluid cavity was refilled and the annulus plugged. The resulting resonance, shown in Figure 10, is 463 Hz. Thus k_4 is approximated:

$$k_4 \cong (2\,\pi f)^2 \frac{w}{g} = \frac{(2\,\pi \times 463\ Hz)^2\ 23\ lb}{386\ rad/s^2} = 503{,}776\ lb/in. \qquad (31)$$

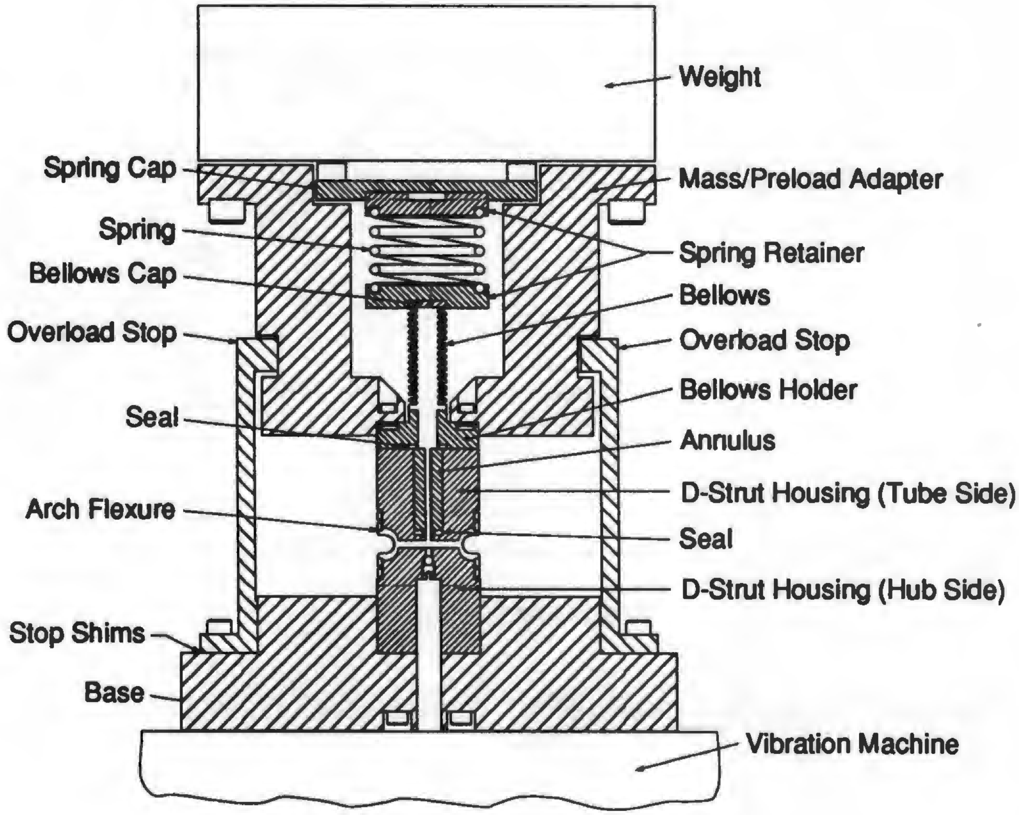

Figure 8. D-Strut Test Setup

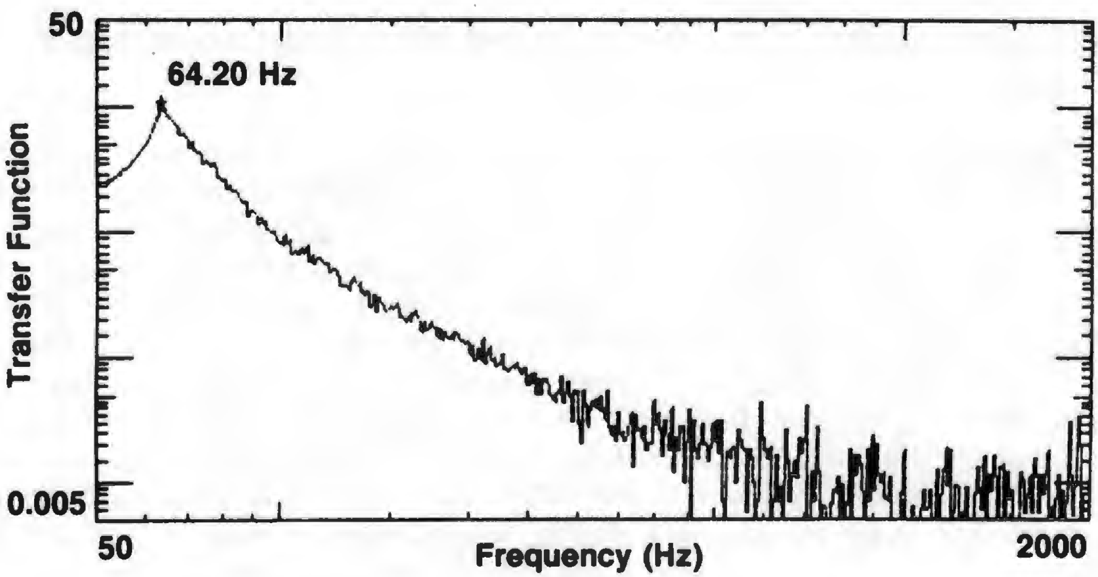

Figure 9. Transfer Function Without Fluid

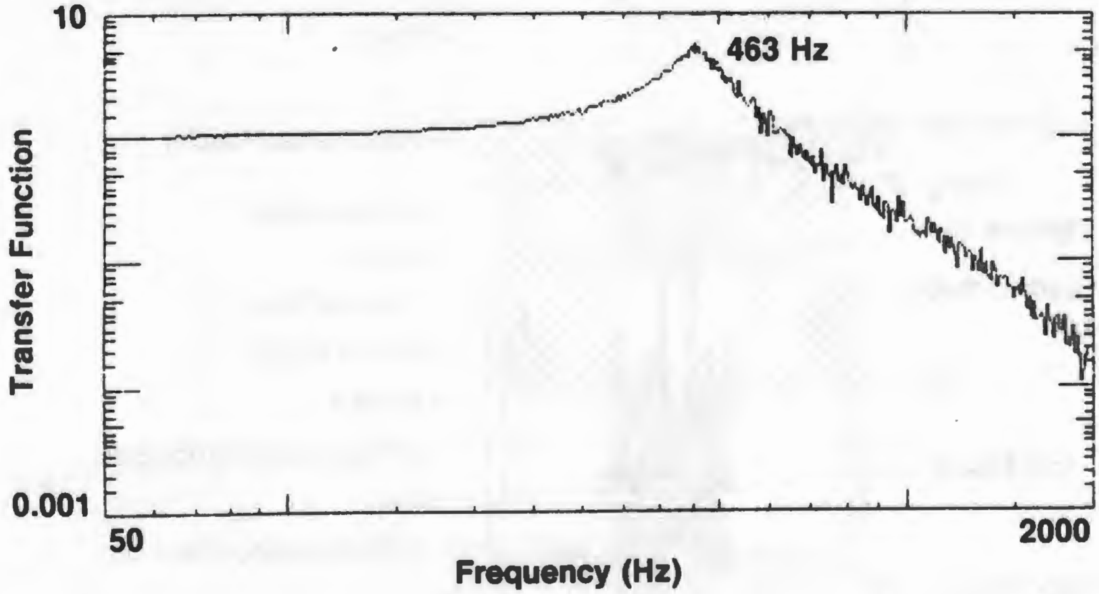

Figure 10. Transfer Function With Annulus Block (With Fluid)

For a ratio:

$$\frac{k_4}{k_3} = \frac{503776}{9686} = 52 \tag{32}$$

Calculations show that the fluid stiffness is 15,000,000 lb/in. Using this value, the metal volumetric stiffness is calculated to be 521,100 lb/in.

Parametric optimization using closed form stiffness equations lead to the conclusion that k_4/k_3 ratios much higher than 52 to 1 can be achieved through parametric optimization. Further, axial strokes can be achieved greater than the deflection capacity of the tubular part of any strut. This means that the addition of a D-Strut element will not reduce the static load capacity of the system. Figure 11 shows a table of ***arched*** flexure designs that point to these conclusions. Note that design No. 4 approximates the results of the single convoluted design just discussed. This design has one convolution, $N = 1$; the radius of the arch is $b = 0.125$ in., the radius of the tube forming the convolute is $a = 0.445$ in.; the OD (outside diameter) of the element is 1.34 in.; the modulus $e = 16$ million lb/in., which corresponds to titanium; the stroke of $s = 0.006$ in. results in a stress of 55,418 psi; $k_3 = 9.68$ klb/in.; $k_4 = 503$ klb/in.; $k_4/k_3 = 52$; the outer tube stiffness $k_1 = 67$ klb/in.; the 3-model equivalent paralleled spring stiffness k_A = 76 klb/in.; the inner tube stiffness $k_2 = 165$ klb/in.; and the 3-model equivalent series spring stiffness $k_B = 156$ klb/in. In this case, k_1 and k_2 were somewhat arbitrarily selected to represent the character of the PACOSS structure. K_1 and k_2 were not part of the testing: only the basic D-Strut element was tested. However, previous correlation between D-Strut element tests and D-Strut tests is evidence that this process is accurate. Notice that the damping loss factor ETA = 0.586 or 58% (in the text ETA is *N*). Had k_4/k_3 been larger than 52, ETA would have been larger. Also the effective damping constant (CA) = 1784.5 lb-s/in. would have been closer to the actual damping constant (c_1) = 2000 lb-s/in. Notice further that the range of frequencies where the damping is effective from OMEGA A = 4.56 Hz (minimum) to OMEGA A* = 7.97 Hz (maximum).

DSGN	(n)	(b)	(a)	(OD)	(t)	(e)	(s)	sigma 1	(k_3)	(k_4)	(k_4/k_3)
1	1	0.15	0.445	1.39	0.006	16.00	0.006	49,073	8.06	255	32
2	1	0.125	0.445	1.34	0.012	16.00	0.007	81,497	38.71	1,006	26
3	1	0.125	0.445	1.34	0.015	16.00	0.006	75,260	60.49	1,258	21
4	1	0.125	0.445	1.3	0.01	16.00	0.006	55,418	9.68	503	52
5	1	0.125	0.6	1.65	0.01	16.00	0.006	53,849	26.88	2,425	90
6	2	0.125	0.6	1.65	0.006	16.00	0.014	52,970	4.84	727	150
7	2	0.125	0.8	2.05	0.005	16.00	0.018	52,892	3.36	1,611	480
8	4	0.125	0.8	2.05	0.006	16.00	0.036	56,213	2.42	967	400
9	6	0.1	0.8	2	0.003	16.00	0.06	57,503	0.50	673	1,336
10	10	0.1	0.8	2	0.003	16.00	0.1	57,503	0.30	404	1,336
	(k_1)	(k_A)	(k_2)	(k_B)	ALPHA	(ETA)	(c_1)	(c_A)	OMEGA A	OMEGA B	OMEGA *
1	67	75	165	157	1.76	0.598	2000	1817.9	4.44	13.78	7.82
2	67	98	165	134	1.54	0.442	2000	1312.1	6.88	16.22	10.56
3	67	111	165	121	1.44	0.376	2000	1070.9	8.61	17.95	12.43
4	67	76	165	156	1.75	0.586	2000	1785	4.56	13.91	7.97
5	67	90	165	142	1.60	0.491	2000	1478.9	5.93	15.28	9.52
6	67	72	165	160	1.80	0.621	2000	1887.7	4.18	13.52	7.52
7	67	70	165	162	1.82	0.633	2000	1921	4.06	13.40	7.38
8	67	69	165	163	1.83	0.641	2000	1942.6	3.99	13.33	7.29
9	67	68	165	164	1.85	0.657	2000	1987.8	3.83	13.18	7.11
10	67	67	165	165	1.86	0.659	2000	1992.7	3.82	13.16	7.09

Poisson's Ration (v)	(b) Arch Radius	(n) Number of Convolutions
Number of Convolutions (n)	(t) Thickness	(OD) Outside Diameter
Stroke (s)	(a) Tube Radius	(k_1) Outer Tube Stiffness
Validation Factor 4 < u < 40	(e) Modulus of Elasticity	(k_2) Inner Tube Stifness
Load to Produce Deflection (p)	(k_3) Axial Stiffness of Arch	(c_1) Two Spring Damping Coef
SIGMA 1 - Stress	(k_4) Volumetric Stiffness	(c_A) Four Spring Damping Coef
SIGMA 2 - Stress		

Figure 11. Damping Spring Design Alternatives (D-Strut)

Two questions arise, as follows:

- How high must the ratio of k_4/k_3 be to obtain practically ideal performance?
- Once this ratio is known, can the D-Strut be optimized to provide that capability?

To answer these questions, compare the sample designs shown (designs 1 through 10 in Figure 11), and consider the values of k_4/k_3. Notice that, as the design parameters of the arched flexure change, substantial improvement in k_4/k_3 is realized, particularly for designs 9 and 10 ($k_4/k_3 = 1{,}336$). Also notice that, for these designs, $c_1 \approx c_A$, $k_1 \approx k_A$, $k_2 \approx k_B$. Therefore, it is clear that this value of k_4/k_3 results in essentially ideal performance and further increase will not produce additional value. Further analyses of the 10 designs suggest that values of k_4/k_3 above 100 are of little additional benefit. It should also be clear from this data that a second-generation ***arched*** flexure D-Strut, with better parametric optimization, could easily reach the $k_4/k_3 > 100$ level. The more optimum design would appear to be a multiconvoluted design not only because of the better k_4/k_3 ratio, but also because of the large stroke capacity.

The data presented here suggest near-perfect correlation between empirical data and analytical calculations for stiffness. Actually this was not the case; some substantial differences existed. Specifically, the coefficient of the equation for k_3 was factored by 1.156 and for k_4 by 2.35. Therefore, only limited value can be placed on the specific numerical results. The factors were applied to facilitate limited design trades and trend considerations.

The reason for these discrepancies is believed to be primarily the differences between the actual thickness of the flexures manufactured and the intended design thickness. Improved controls are planned for future parts. Some error has likely been contributed due to the limitations of the equations used. We are currently conducting fundamental work to improve these.

FUTURE PLANS

Future plans for the D-Strut involve further improvements in the damper element of the strut, based substantially on the factors and optimization trends discussed in the preceding paragraphs. A multiconvoluted design will be a first priority.

Another factor significantly influencing D-Strut performance is the ratio of inner-to-outer tube stiffness. D-Struts fabricated to date have used the same material for inner and outer tubes. The lengths of the two tubes are also approximately equal. Thus, using the same material for both tubes, the only way to increase the stiffness ratio k_2/k_1 is to either decrease the outer tube cross sectional area or increase the inner tube cross sectional area. Decreasing the outer tube area will affect static stiffness requirements, while increasing the inner tube area leads to a considerable weight penalty.

An obvious alternative is to use different materials for the inner and outer tubes. For example, an aluminum outer tube with a metal matrix composite inner tube would give a factor of 2 improvement in the ratio k_2/k_1, due solely to the difference in the modulus of elasticity. The use of different materials for the inner and outer tubes is an important aspect under investigation.

Of course, for a nonload-bearing strut, the outer tube may be eliminated, providing a significant improvement in damping. Other factors then become important. To date, no D-Struts have been fabricated without an outer tube. There are applications where this will be an important consideration.

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