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DAMPING AND VIBRATION CONSIDERATIONS FOR THE DESIGN OF OPTICAL SYSTEMS IN A LAUNCH/SPACE ENVIRONMENT

by

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1. ABSTRACT

A review of engineering philosophies used for the design of optical systems launched into space and operating a vacuum or cryovacuum environment is presented herein Sources of energy dissipation which are usually lumped under a single modal parameter denoted as the equivalent viscous damping coefficient are reviewed Optical systems operating in a cryovacuum environment are especially difficult to design for the launch inertia and acoustic loadings unless the components of the system are either caged or clamped since general viscoelastic effects, interface and/or joint relative motion are designed to be minimal Moreover, stress levels and stress gradients are also designed to be low in precision instruments

2. DAMPING IN STRUCTURAL DESIGN

Over the past three decades, a large number of high-resolution optical systems have been launched into space, through use of the space shuttle or with expendable rockets These optical systems, designed to operate in a vacuum or in a cryovacuum, are subjected to severe noise and vibration during launch to orbit Design Power Spectral Density (PSD) envelopes for the shuttle cargo bay or the rocket cargo pods, generally range from 0 01 g^2/Hz to 0 2g^2/Hz for frequencies of 20 to 300 Hz, depending on the location within the launch vehicle

The preliminary structural design for the components is, in general, described probabilistically and accomplished through the following scenario using the quasi-static loads and random loads

1) Determine the static and quasi static inertia forces and use these to generate a preliminary finite element model for design

2) Determine the root mean square (rms) forces and stresses from the random input, and multiply these values by a factor (usually ranging from two to three) to obtain an estimated peak value If the factor three is used, for example, a 99 87 percentile level of distribution results

3) Add the forces and stresses obtained from steps 1 and 2 for the finite element model in the design iteration

In step 2, the mode participation factor and the power spectral density may be used to obtain the response of each of the vibration modes considered in the design of the structure These modal responses then are combined in an appropriate manner, usually by the square-root-sum of the squares (srss) if the modal frequencies are distinct, to obtain the combined maximum response

The root mean square acceleration for the single degree of freedom model representing the n[th] mode is

$$a_{rms} = \left[\frac{\pi}{2} \, x \, f_n \, x \, PSD \, x \, Q\right]^{1/2}$$

where,

f_n	=	frequency (Hz) of the n^{th} mode,
PSD	=	power spectral density value of the n^{th} mode frequency (g^2/Hz)
Q	=	quality factor which is equal to $1/2\zeta$ where ζ is the equivalent viscous damping ratio for the n^{th} mode

Alternatively, the modal root mean square displacements are

$$y_{rms} = \left[\frac{PSD}{(4\pi f_n)^3 \zeta}\right]^{1/2}$$

This latter equation may be used to generate a displacement response spectrum from the power spectral design envelope for structural systems with low damping

Modal forces and stresses are determined from these modal accelerations or displacements and the modal participation factors Once the preliminary structural design is accomplished, a complete analysis, using the finite element method, may be performed by either a frequency analysis using the design PSD envelope, or a response spectrum analysis

From the design scenario, it is apparent that both the PSD and the damping must be known to predict, reliably, the dynamic loadings Design PSDs have been determined through tests made in the cargo bay of the space shuttle and in cargo pods of the expendable missiles used to launch satellites Damping, however, is a complex phenomenon, and difficult to determine quantitatively as shown in the papers of numerous professional journals and conference proceedings Damping depends on the existence of joints and interfaces, on the materials used, and on the environment, and cannot be determined theoretically Rather, damping must be determined from experimental data Moreover, practically all results of damping experiments reported in the literature are for single components (beams, plates, etc) and do not include the effects of assemblies, of joints and/or of the environment[(1)]

Damping measurements (refer to Appendix B for general damping terminology) found in the literature are usually based on the following methods

1) A frequency response spectrum as shown in Figure 1, and using either the half power point method or the phase change method

2) The starting and decay transient method (logarithmic decrement) as shown in Figure 2

3) The energy dissipation method

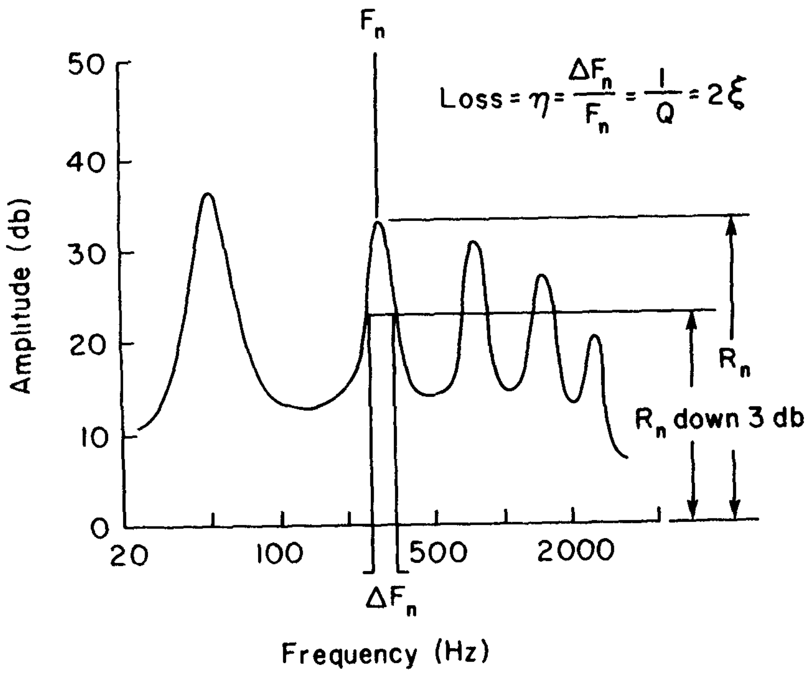
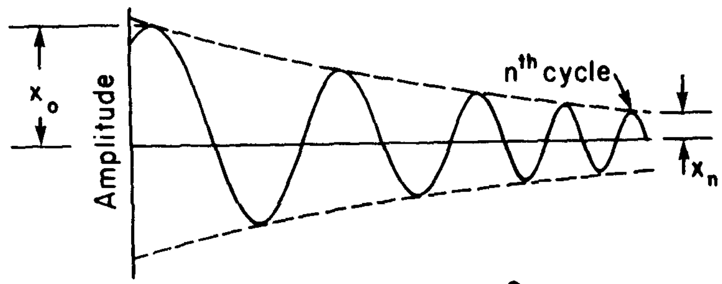


Figure 1 Frequency Response Method



Logarithmic Decrement: $\delta = \ell n(x_o/x_n)/n$

Damping Factor $\xi = \delta/(\delta^2 + 4\pi^2)^{1/2}$

Figure 2 Logarithmic Decrement Method

3. SOURCES OF DAMPING

Damping in spacecraft elements and spacecraft structural systems is quantified through physical testing of materials, structural elements, and structural systems At the material level damping is a result of

a) flow phenomena

b) interactions among molecular forces

c) movements at the microscopic level

d) general viscoelastic and microplasticity effects

At the structural element level (e g , bars, beams, plates, etc), damping is generally quantified using the following parameters

a) temperature

b) stress levels and stress gradients

c) amplitudes and frequencies of vibration

At the structural system level, where assemblies of elements are considered, damping is a primary result of mechanical effects such as

a) interface and/or joint relative motion involving

    (1) dry friction

    (2) fretting

b) plasticity and/or viscoelastic behavior

c) air damping of large flexible structures (e g , solar panels)

Of these three broad categories used to quantify damping, structural system level generally dominates Practically all engineered damping devices involve the three listed methods in which energy is dissipated in structural systems Sucessful use has been made of viscoelastic materials on aerodynamic surfaces, engine components and instrument panels Shim and wire rope devices have been designed to activate dry friction energy dissipation

For high precision structural systems operating in a cryovacuum environment, however, only material and structural element level damping generally exist Moreover, if the structure is a high resolution optical system with stress levels specified below the precision elastic limit of the materials, very little damping exists These systems then generally require isolation or clamping or caging designs to limit stresses, strains, and displacements to specified amplitudes

4. QUANTIFYING DAMPING

When structural systems are subjected to cyclic loading, a hysterectic loop in the stress-strain or force-displacement curve results The area within this loop is proportional to the energy absorbed, and is usually used to establish an equivalent viscous damping ratio This damping arises from the sliding of molecules, from intermolecular forces, from plastic flow, and from general viscoelastic effects, and in joined elements, the major source of energy dissipation occurs at interfaces, as a result of friction and local distortions (fretting) arising from relative motion in the interface plane

Published[2] equivalent viscous damping ratios are listed below

Material/Structures	*Range of Equivalent Viscous Damping Ratios*		
Metals	0 0001	-	0 001
Alloys	0 001	-	0 020
Assembled Structures	0 002	-	0 030
Viscoelastic Materials		>2	

From the wide variation shown here, it is apparent that a significant amount of engineering judgement must be used in selecting and using damping coefficients This point is emphasized by the efforts the professional designers have made to arrive at acceptable damping ratios for design For example, there have been numerous conferences and workshops sponsored by various agencies on damping Listed are a number of these

CONFERENCE/WORKSHOP TITLE	SPONSOR	REPORT & DATE
Aerospace Polymeric Viscoelastic Damping Technologies for the 1980s	WPAFB	AFFDL-TM-78078-FBA July 1978
An Introduction to the Problem of Dynamic Structural Damping	NATO	AGARD Rpt No 663 January 1978
Damping Effects in Aerospace Structures	NATO	AGARD Conference Proceedings No 277 April 1979
Vibration Damping 1984 Workshop Proceedings	WPAFB	AFWAL-TR-84-3064 November 1984
Vibration Damping 1986 Workshop Proceedings	WPAFB	AFWAL-TR-86-3059
Response of Symmetric Rectangular Composite Laminates with Nonlinear Damping	AIAA	AIAA 10th Aeroacoustic Conference 1986
Symposium on the Role of Damping in Vibration and Shock Control	ASME	Proceedings of the Winter Meeting Sept 1987, Boston, MA
Vibration Damping 1989 Workshop Proceedings	WPAFB	AFWAL-TR-89-3116 June, 1990

5. DESIGN PRACTICES

Dynamic testing of spacecraft reported in the literature was usually made starting with a modal survey of the test specimen, Modal damping values are then determined using random and/or sinusoidal excitation This excitation is usually done at two different levels at either single or multiple points on the structure These tests have shown a variation in the damping coefficients with amplitude, frequency and methods of excitation and no clear relationship has been determined

In the 1979 AGARD Conference Proceedings, Wada and DesForges[3] presented a "state-of-the-art" paper on damping in spacecraft structural design In addition to these authors' own extensive experience, the work of their colleagues at Martin Marietta, Aerospace Company, Hughes Aircraft, Boeing Aerospace, Johnson Space Center, Marshall Space Flight Center, Goddard Space Center, General Dynamics, and General Electric was included

They reported that estimates of the overall system modal damping ratios may be obtained by establishing the kinetic energy contributions of the various component modes to the kinetic energy of a given system mode so that the system modal damping ratio is a weighted sum of the component modal damping ratios For a coupled spacecraft/launch vehicle loads analysis, the modal damping values to be used for the coupled-system modes are derived from the individual spacecraft and launch vehicle modal damping matrices or are based on either flight data or engineering judgment The current procedure for transient loads analysis for space shuttle payloads is to assign a modal damping value of one percent of critical for all payload modes They reported that at the Goddard Space Flight Center, the general procedure is to assign one percent of critical modal damping for all loads analysis If, at the time of the verification loads analysis (which occurs after the spacecraft modal test), load problems with payloads are anticipated, damping values obtained from the modal test are incorporated into the loads analysis For the Delta launch vehicle, a value of 1 5 percent of critical damping is assumed for all coupled system modes Modal response data measured in-flight have provided a sound basis for this value Wada and DeForges[3] state that the use of this constant value for all modes will yield slightly conservative results, since certain local modes may exhibit higher damping values They reported that on Viking flight program that instrumentation was employed and extensive analyses were performed to predict responses A good analytical model verified by test was available, and excellent correlation between flight and analysis for both Viking and Voyager was obtained

Appendix A contains part of the results of the research reported in Reference 3 These data are included in this paper to provide designers with a ready reference for a range of damping ratios that were obtained by test from a wide variety of spacecraft

In 1987, Simonian[4] gave a data base comprising 23 satellites (Table 1) with a total of 290 sample measurements spanning the frequency range 0 15 through 195 Hz The surveyed data were statistically analyzed in various combinations First, the total sample statistics were computed and was followed by taking statistical computations of frequency groupings of a 10Hz band with each, up to 50Hz Above 50Hz the data were lumped and analyzed as a single group For each group of data analyzed, histograms were constructed and probability density functions (pdf) were fitted to the data The parameters of fitted pdf's of the gamma and lognormal distributions were determined Summarized in Table 2 are the computed statistical damping parameters for various combinations of the data grouped according to frequencies

VEHICLES	
INTELSAT IV (IN-ORBIT DATA)	NINBUS
SPACE TELESCOPE METERING	ATS-F
TRUSS	777 REDESIGN
RANGER III STM	777
SEASAT	M35 PHASE II UPGRADE
HUGHES SATELLITE (3)	M35 PHASE I
SCATHA SATELLITE	M35 PHASE II
HERMES (IN-ORBIT DATA)	FLTSATCOM
INTELSAT IV-A	OGO-3 (IN-ORBIT DATA)
SKYLAB	OSO-8 (IN-ORBIT DATA)
APOLLO-SATURN V, 1/10 SCALE	FRUSA (IN-ORBIT DATA)

Table 1. Satellites Surveyed

Frequency Interval (Hz)	ξ Mean (% Critical)	ξ Standard Deviation (% Critical)	COV (%)	Gamma[1]			Lognormal[2]		Number of Samples
				(%)	b	$\Gamma(b)$	c	d	
A. 0.14 - 195	1.20	0.92	76.7	0.705	1.701	0.909	-0.0488	0.680	290
B. 0.14 - 9.99	1.90	1.58	83.2	1.314	1.446	0.886	0.379	0.725	39
C. 10.00 - 19.99	0.94	0.57	60.6	0.346	2.800	1.569	-0.218	0.560	85
D. 20.00 - 29.99	1.18	0.86	72.9	0.627	1.883	0.956	-0.0475	0.653	56
E. 30.00 - 39.99	1.09	0.75	68.8	0.516	2.112	1.053	-0.108	0.623	51
F 40.00 - 49.99	1.27	0.79	62.2	0.491	2.584	1.413	0.0755	0.572	29
G. 50.00 - 195	1.16	0.50	43.1	0.216	5.382	43.369	0.0632	0.413	30

Table 2. Summary of Spacecraft Measured Damping Statistics

6. SUMMARY

The need for accurate quantitative values of energy dissipation for spacecraft operating in a vacuum or cryovacuum environment is apparent However, structural and instrument designers will have to continue to use engineering judgement along with parametric studies to complete a design and then depend upon qualification tests to verify the structural integrity and operational quality of the design Caging and/or damping system components or using viscoelastic materials and/or dry friction devices are alternative methods for stabilizing instruments sensitive to motion

7. ACKNOWLEDGEMENTS

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