THE INVESTIGATION OF LARGE SPACE STRUCTURE PASSIVE ELECTRODYNAMIC DAMPERS

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ABSTRACT

The focus of this study was on computationally verifying that passive electrodynamic damping was competitive or superior to current damping technologies recommended for Large Space Structures (LSS). Electrodynamic damping is linear and is characterized by a dash pot dissipative force which is proportional the relative velocity of the damper components. The constant of proportionality is c. The study investigated the maximum ratio of c to the mass of the damping system as well as the frequency dependence of c. Both analytic and ADINA models of an LSS-like structure, the Air Force Wright Aeronautical Laboratory 12 Meter Truss (TMT) were used, together with TMT data, to understand and verify Passive Electrodynamic Damper (PED) performance.

The study results indicate that the Auxiliary Mass PED (AM-PED) is competitive or superior to active dampers, in damping TMT bending modes, when the AM-PED weight is comparable to that of active damping actuators. This is important because of the enhanced reliability and cost savings of a passive damping system. An AM-PED does not require sensors, a power source or a computer control system. Although a detailed comparison was not made, it appears that equivalent weight strut PED systems may also be superior to viscoelastic-material strut dampers. This is important because PED systems do not outgas and are stable with respect to environmental temperature variations. In addition PED system performance is easily calculable, c is independent of frequency and of amplitude for the low modal frequencies characteristic of LSS.

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1.0 INTRODUCTION

This paper describes research that was supported by the Strategic Defense Initiative Organization (SDIO). The object of the study was to investigate the feasibility of specially-designed, spacecraft vibration-damping-devices, known as a Passive Electrodynamic Dampers (PEDs). PEDs absorb mechanical energy, by means ohmic heating, when there is motion relative to the field of a permanent-magnet. Future military and non-military spacecraft are expected to be large and flexible, with many low frequency modes. Damping these modes is critical to the operation of some on-board sensors and equipment. A successful PED would therefore be applicable to military non-military space programs.

The goal of the study was to verify PED damping effectiveness by mathematically modeling and evaluating PED electrodynamic characteristics, as they relate to Large Space Structures (LSS), and computationally simulating the mechanical effect of PED configurations on LSS. The approach taken was to focus on one LSS test bed, called a model LSS (MLSS), since there was time to consider only one LSS simulation in the project. The electrodynamic modeling was more general, showing how the key PED design parameters - weight for example - varied as a function of LSS and space environment characteristics - frequency and temperature for example. The idea was to design a PED for the MLSS using the general PED design equations that evolved from the research. The MLSS modal damping was first approximated analytically so that the key parameters and their relationship to the damping could be identified. PED caused, LSS modal-damping was then compared to MLSS data obtained using other damping systems. Because the analytically calculated damping was satisfactory the PED was incorporated into an ADINA (Reference 1) code model of the MLSS. The ADINA model gave the most accurate PED modal damping effectiveness calculation and allowed the most accurate comparison with other damping methods.

After much consideration and discussion with the appropriate government agencies the MLSS chosen for the study was the AFWAL twelve meter truss (TMT, References 2 and 3). The reason for this choice was that the TMT was the simplest, technically acceptable structure for which adequate data was available.

Much of the research presented in this paper is an evaluation of the damping characteristics of a particular PED configuration called the Auxiliary Mass Passive Electrodynamic Damper (AM-PED). Both analytic and computer simulation results show the AM-PED is competitive with active damping systems anticipated for spacecraft use. Substitution of the AM-PED for active damping could mean large increases in space platform damper reliability, weight reduction and a lower cost damping system. The AM-PED is expected to be a very important addition to the technologies used for LSS damping. Other PED configurations are expected to also be important but they have not been studied in as much detail. The Strut Passive Electrodynamic Damper (S-PED) appears to be a particularly promising substitute for viscoelastic materials (VEM) in damping structural truss modes. PED damping concepts, electrodynamics and space environment characteristics are discussed in Section 2. Sections 2 - 4 discuss the results and analysis of this study. Section 5 presents the conclusions and recommendations. In the next subsection the rationale for studying low frequency dampers, in particular PEDs, is discussed

1.1 LARGE SPACE STRUCTURES (LSS) AND DAMPING

Increasingly greater roles are anticipated for satellites in the civilian economy, in government research and in military planning. Planned space structures are therefore becoming larger with more complex missions and increasing power requirements (References 4 and 5). The fiscal and complex-mission, space-structure requirements, for these planned systems, result in lightweight, flexible, loaded, LSS design concepts with very low modal frequencies. In combination with the lack of gravity these requirements also mean that there will be small frictional energy dissipation and modes will be poorly damped.

One proposed solution to the LSS structural requirements has been the use of trusses as the basic support structure. Trusses are both lightweight and rigid and have been designed in beam configurations. The plan is to mount sensors, equipment and solar panels on these lightweight frames. The resulting LSS are truss-type structures connecting a variety of flexible components. Predictions indicate that these flexible components will likely have natural frequencies in the same range as the dominant truss modes (Reference 5).

A number of groups have developed experimental LSS structural models to verify their structural dynamic computational tool predictions, as well as verify proposed damping concepts. The PACOSS (Passive and Active Control of Space Structures, Reference 6) dynamic test object and Twelve Meter Truss (TMT, References 2 and 3) supported by the Air Force and the Dynamic Scale Model Technology (DSMT, Reference 5) program supported by NASA are examples. The PACOSS program is particularly advanced and experimental results appear to support the current LSS damper design philosophy (Reference 7):

(1) Damp as many modes as possible passively, using VEM.

(2) Damp all remaining modes (assumed to be only a few) by means of active damping.

The TMT approximates a twelve meter beam and experiments have been performed in both a cantilevered and a free-free configuration. The cantilevered configuration is not "realistic" for a complete LSS[1] and is a compromise so that experiments can be performed with low-frequency structural modes (2.25 Hz). TMT cantilever experiments have been performed and analyzed with and without VEM struts. Free-free TMT experiments have been performed with and without VEM struts but the results have not been analyzed in detail. Active-damper, cantilevered-TMT NASTRAN experimental pretest predictions are also available.

Not only damper development but LSS designs and structural dynamic testing are still in the research and development stage. At the present time VEM is the passive damper of choice and for the most part, in practice, it has been used in strut configurations (References 3 and 6). However, there are some shortcomings to the use of this material. The full temperature variation for an exterior spacecraft component could be 150°C, from about -50°C to about 100°C. Some VEM materials have a useful range of only 20 - 30 °C. It is recognized that one material will not suffice for every application and that active heating elements will have to be used in conjunction with VEM to maintain constant damping (Reference 8), for some applications. In addition VEM is nonlinear and its damping characteristics are not easily predicted.

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[1] There are structures that are expected to be cantilevered off the truss. The solar paddles in Reference 5 are an example.

These shortcomings imply uncertainties and expense in damper design as well as possible reliability problems in actual practice. In addition there is the question of the VEM damper effectiveness with respect to its weight. In the TMT experiments the final VEM passive damper configuration weighed about 50% more than the undamped truss. Not all of this damping material was effective in damping the modes, however, and future TMT studies may investigate the elimination of the least effective struts.

At the present time a common active damping system uses a coil and permanent-magnet actuator system. A current is generated in the coil and exerts a force on a moving magnet corresponding to a predetermined algorithm. One such algorithm is to make the force proportional to the velocity of the attachment point for example. The system is very convenient in its application: the actuator is attached at a position of maximum modal amplitude, consistent with dynamic stability requirements. In addition, because it is made of metals, its performance is very stable with respect to expected environmental temperature variations. There are some shortcomings, however. One of these appears to be that the force exerted is limited by the maximum current that can flow through the coil. Too high a current will melt the coil. Most of the power dissipated in the coil appears to result in a restoring force which changes direction as a function of time. Only a small portion of the force actually damps the motion of the LSS modes. Additionally the actuator system requires motion sensors, a computer control system and a power supply. All these system components add weight and contribute to system reliability issues.

If the objective of the active damping system is to damp only a few modes, replacing the electrical-power generated restoring force with a spring and a permanent-magnet system may be the most efficient and cost effective design. The AM-PED herein is a passive damping device which does just that. It has all the advantages of this active damping system but apparently none of its disadvantages. In addition it may be more effective in damping LSS modes than an active damping system.

2.0 THE PED

In this section the PED concept is first discussed from a general point of view. In Section 2.1 the AM-PED is discussed and then in Section 2.2 the design constraints imposed by the electromagnetics is examined. Finally in Section 2.3 the effect of the space environment on the PED is discussed.

The basic PED concept is to mechanically couple LSS vibrations to relative motion between an armature and a magnet. The relative motion gives rise to a dissipative force F which is proportional to the relative velocity of the two PED components. That is

$$F = cv, \tag{1}$$

where v is the armature/magnet relative velocity and c is the constant of proportionality. Figure 1 illustrates the principle. The relative motion generates a current in the armature and vibrational energy is absorbed via ohmic heating. This kind of damping - electromagnetic damping - has been considered in the past for other kinds of systems (Reference 9) and so the concept is not new. What is new is the application of the concept to LSS and the particular LSS PED structural and magnetic configurations. Because of the low LSS frequencies electromagnetic damping, as manifested in the PED design, is a very weight efficient LSS vibration damper. This will be demonstrated in Sections 3 and 4. Because of superior PED, spacecraft-environment, material-properties and its simplicity it is a very desirable damper system.

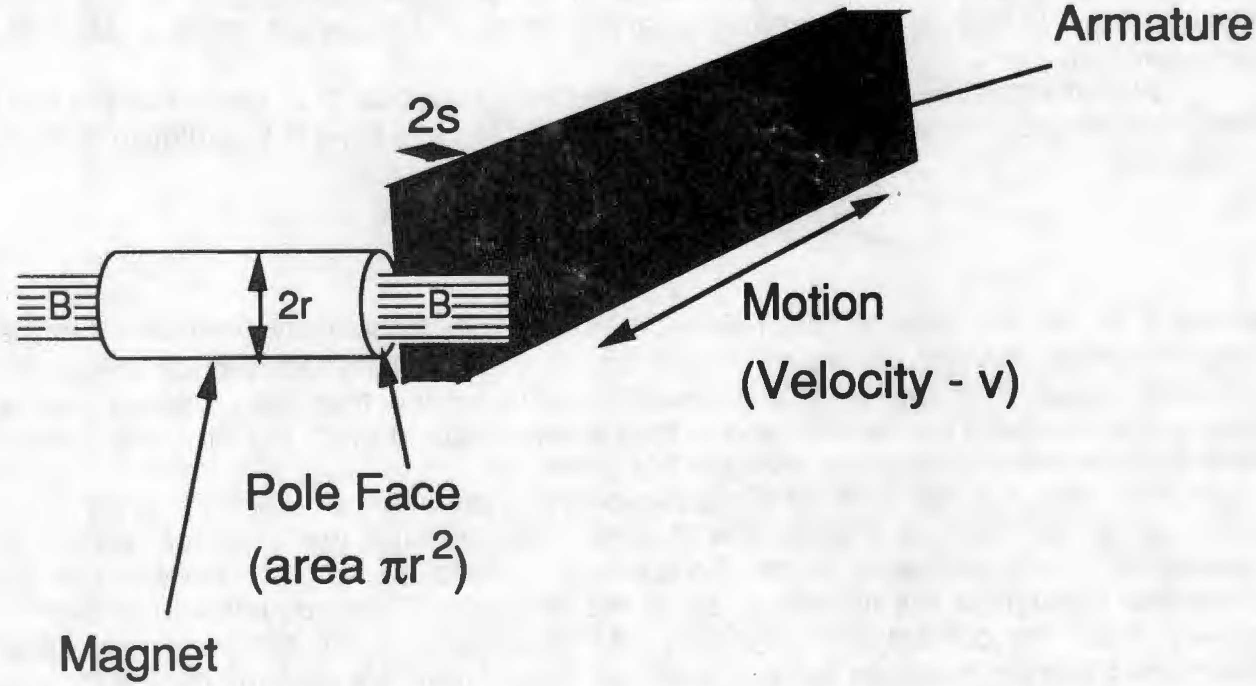

Figure 1. Generic PED Components

The PED is essentially a dash pot, for all LSS vibrational amplitudes of concern, and the mechanical analysis is relatively straight forward. Difficulties lie in efficiently designing the magnetic circuit and in coupling the dissipative force to the complex multi-modal mechanical motion. Two coupling configurations were studied: (1) the Auxiliary Mass Passive Electromagnetic Damper (AM-PED), and the (2) Strut PED (S-PED). The idea behind the AM-PED is to transfer the LSS vibrational energy to a proof mass and then dissipate the the proof mass energy via ohmic heating. The AM-PED can theoretically be attached anywhere on the LSS the vibration amplitude is large. The S-PED is used mainly as a component of the LSS truss support structure to dissipate truss vibrations.

For most coupling the PED is designed to have is own restoring force proportional to displacement and consequently its own resonant frequency ω_0. The PED has, of course, its own mass, m, as well. One design problem is choosing the PED parameters, c, ω_0, and m to optimize the damping over the LSS frequency range of interest. For a given m, we are actually optimizing the damping by appropriately choosing the parameters c/m and ω_0.

One of the advantages of the PED is that it is a simple, linear, mechanical system and its effect can be calculated. However, coupling to a complex LSS means the analysis is more complex than for a one dimensional system. In general the PED damping of a given LSS of mass M and frequency Ω_{0n} ($n = 0, 1, 2, 3.........$) will not be the same as for a one dimensional system of mass M with a damping force proportional to velocity. That is the percent of critical damping, γ_n, of the particular LSS mode will not be simply $c/(2M\Omega_{0n})$. We can expect that γ_n will be related to the <u>effective</u> mass of the LSS, for the particular modal vibration of concern (the total LSS mass is not necessarily effective in the vibrations of a particular mode), and to the PED parameters. These latter comments are particularly relevant to the AM-PED configuration.

We now consider how the PED parameter c (equation 1) is related to the PED design parameters. As expressed in equation 1, the force is cv, v is the relative velocity (m/sec), and

$$c \approx \sigma\pi r^2(2s)B^2 \qquad \text{(kg/sec)}, \tag{3}$$

where B is the flux density field (Weber/m^2), σ is the conductivity (mhos/m) r is the magnet radius (m) and 2s the thickness (m) of the armature. Reference should be made to Figure 1. Equation 3 assumes that none of the magnetic leakage flux is effective in damping the system and is thus a lower bound on c: the armature will be wide enough to cut most of the leakage flux lines.

Equation 3 is not valid for all frequencies and although an arbitrarily large c can be developed simply by making the magnet large enough the important ratio c/m cannot be made arbitrarily large. Equation 3 is valid so long as current can be generated throughout the thickness, 2s, of the armature. If the frequency of oscillation is very large the current will only exist on the surface of the armature and 2s in equation 3 will be replaced by a smaller number. Therefore at high frequency c is smaller than expressed by equation 3. The depth of penetration of the current into the armature is controlled by a parameter called the "skin depth", δ, which has the dimensions of length. Roughly speaking when $\delta > 2s$ equation 3 is valid. As we will see in Section 2.2 we can expect equation 3 to be valid below about 50 Hz. This frequency is far above expected LSS frequencies.

Equation 3 shows that the dimensions of the magnetic system (Figure 1) enter the calculation of c (the area, πr^2, of the permanent-magnet for example). What is not obvious from equation 3 is that the magnetic field B is also dependent upon the dimensions of the magnetic system as well as the type of magnetic material used. Optimized designs have a maximum c/m value which is dependent upon the magnetic system design. The maximum practical LSS c/m ratio, for an aluminium armature, is about 500 sec^{-1}. We will see in the next section that the c/m ratio is relevant when designing a AM-PED to damp more than one LSS mode. It is also important when comparing the equivalent weight of alternative passive damping concepts. In Section 3.2 we roughly compare the S-PED to VEM struts.

2.1 THE AUXILIARY MASS PASSIVE ELECTROMAGNETIC DAMPER (AM-PED)

The AM-PED concept is straight forward and very simply applied to a LSS. The idea is to continuously transfer LSS vibrational energy to a proof mass and dissipate the proof mass kinetic energy. In the case of the AM-PED the proof mass is essentially the magnetic system and the dissipative force is described by equations 1 and 3. The AM-PED is made lightweight and springs act as a linear restoring force. Relative motion causes currents to flow in the armature resulting in energy dissipation. For the present discussion it is only necessary to know that the dominant weight of the device arises from the magnetic system taken to be the mass m. For reference the amplitude of magnet motion will be about an inch (maximum LSS vibration amplitudes are fractions of an inch), the overall dimensions of the AM-PED designed for the TMT will be about 10 cm x 10 cm x 10 cm with a mass roughly equal to 4 lbs. This is a very compact device which is attached externally to the LSS (in this case the TMT) at positions of maximum vibration amplitude. (Note that many AM-PED designs are possible using different dimensions and magnetic materials.) As we will see AM-PED damping is expected to exceed 5% for very reasonable AM-PED masses and compete with active damping systems. 5% damping is approximately the requirement for many systems (Reference 7).

2.2 ELECTRODYNAMIC PED CONSIDERATIONS

As discussed in Section 2.0, the damping constant, c, depends upon the value of the current in the AM-PED armature and the magnetic field. The flow of current in the armature is affected by the development of electric fields which oppose the flow of current. These electric fields are manifested through the skin depth introduced in Section 2.0. The armature current flow also generates a magnetic field which may oppose the magnetic field of the magnet. An opposing magnetic field might reduce the force on the armature and demagnetize the magnet, so it must be considered in the analysis. If either the opposing electric field or the opposing magnetic field effects were substantial they could reduce the damping constant below that expected from equation 3. In the detailed analysis we find, as expected, that the parameter of greatest importance is the electrodynamic skin depth δ (meters)

$$\delta = (\mu_0 \sigma \omega / 2)^{-1/2} \quad , \tag{4}$$

where μ_0 is the permeability of free space (the armature is made of non-magnetic material) and equal to $4\pi \times 10^{-7}$ h/m, σ is the armature conductivity (mho/m), and ω is the angular frequency of motion. In order that the current in the armature reach its maximum value, δ must be larger than about twice the thickness of the armature. For expected PED armature dimensions, skin depth should not be a problem for frequencies less than about 50 Hz.

In order that the dimensions of the armature not affect the design of the system the detailed calculations suggest a minimum armature width to magnet diameter ratio. The corresponding length of the armature is determined by other design requirements. Large motion amplitudes can affect the high frequency content of the armature-current-generating electric field, however, if designed properly the damping system will be independent of amplitude.

2.3 PED SPACE ENVIRONMENT CONSIDERATIONS

Viscoelastic materials (VEM) are the recommended passive damping material for LSS, particularly for use within structural components of truss structures (diagonals for example). However, large temperature variations in space make designing passive VEM damping treatments difficult (Reference 8). The full temperature variation for an exterior spacecraft component could be 150°C, from about -50°C to about 100°C. Some VEM materials have a useful range of only 20 - 30 °C, making many materials and, depending upon the specific problem, temperature control elements necessary. One of the virtues of the PED designs is that, because they are made of metals, they are extremely stable with respect to temperature variations.

The Curie point (temperature at which magnetic properties change - Reference 10) of all common magnetic materials is many hundred degrees C, far above the highest expected space environment temperature. The most temperature dependent parameter in the damping constant "c" (equation 1) is the conductivity. "c" is proportional to the conductivity (equation 3). For temperatures near and above the Debye temperature (Reference 11) of the material, the conductivity varies directly with absolute temperature. -50°C is 223 °K and many metals have a Debye temperature near this value. The Debye temperature of silver, the armature material giving the largest c/m value is 226°K, for example. The ratio of absolute temperatures over the expected temperature range is 373/223 = 1.67 and so "c" is expected to vary by less than a factor of 2 over the full temperature range. A look at tables of material data supports this expectation. If a PED experienced the full temperature variation (an AM-PED at the end of a solar paddle, for example) it could be designed to operate most effectively at the mid temperature range (about room temperature) and then the expected variation in "c" would be less than ±30%.

Because the coefficient of thermal expansion for the materials under consideration is about 10-20 x 10^{-6} per degree C, the expected length or gap changes are only about .3%, too little to effect PED operation.

The effects of environmental temperature variation on the spring of the AM-PED are also expected to be manageable. Considerable information exists about the effects of temperature on the mechanical properties of metals (Reference 12). This information suggests that strength may change by ±10% over the applicable range. This can easily be addressed in the detailed design of the spring. The small displacements, ±.17% imposed by thermal expansions can be considered similarly. Finally, the stiffness may vary by ±5% which should not significantly detune the device.

3.0 PRELIMINARY DESIGN OF PED SYSTEMS

In this section we will be concerned with analytic evaluations of PED performance and the impact of performance upon design parameters. The major focus is upon the AM-PED, considered in Section 3.1. The analytic approximations and discussion in Section 3.1 are a background to the consideration of another PED configuration, the Strut-PED (S-PED). Preliminary estimates do suggest that the S-PED may to be very competitive in performance with VEM damping strut configurations. In addition the S-PED does not have any of the VEM temperature dependence, outgassing, nonlinearity and frequency dependence problems.

3.1 AM-PED PRELIMINARY DESIGN FOR THE 12 METER TRUSS (TMT)

One of the objectives of this section is to compare the predicted performance of an AM-PED with that of an equal-weight actuator, active-damping system. Active-damping computer predictions have been made for the TMT in its low frequency cantilevered position (References 2 and 3). These predictions are compared with an analytic, continuous-beam model of the AM-PED/TMT combination. In Section 4 AM-PED performance is compared with the active damping calculations utilizing a ADINA computer model of the TMT. This later comparison is important because the actual TMT is not continuous and the beam-model resonant frequencies differs from the experimentally measured TMT frequencies. The measured ratio of the second bending to the first bending, TMT mode frequency is 10.72/2.26 = 4.74. The frequency ratio of a one end clamped beam is 6.27 so the analytic model is reasonable but differences should be expected between the analytic model predictions and the more accurate ADINA model. As we will see in Section 4, AM-PED damping is actually more effective with the ADINA truss model. This is in part due to the fact that the truss does not adjust its modal shape to external forces in the same manner as the continuous beam. The analytical modeling provides a framework for understanding how to design an AM-PED and is used to make preliminary estimates of AM-PED performance. (In the original study AM-PED effectiveness on a free-free TMT was also computationally simulated. The analysis is not presented in this paper. The damping was found to be 23% less than the cantilevered beam, for equivalent weight AM-PEDs.) The general dynamical problem is considered first.

What is required is to solve the dynamical equations of motion of the AM-PED system coupled to the LSS. The design requirements are that the percent of LSS modal damping should be about 5% so the damping can be solved for by a perturbation analysis. In addition the mass of the LSS, M, is much greater than m so m/M can be treated as a small parameter. As the detailed analysis shows, if $F_s(\omega, x)$ is an expression for the force exerted on the AM-PED by the LSS, where x is the amplitude of motion for the frequency ω, then the frequencies of the system can be obtained from the equation

$$F_s(\omega,x)/M = \beta\omega^2 g(\omega)x, \tag{5}$$

where

$$\beta \equiv m/M, \tag{6}$$

$$g(\omega) \equiv (-2ri + W^2(W^2 - 1) + 4r^2)/(4r^2 + (W^2 - 1)^2), \tag{7}$$

$$r(\omega) \equiv c/2m\omega, \tag{8}$$

$$W(\omega) \equiv \omega_0/\omega, \tag{9}$$

$$\omega_0^2 \equiv k/m, \tag{10}$$

and k is the AM-PED spring constant. In the limit that the new LSS/AM-PED modal frequencies are very near the old frequencies, Ω_{0n}, (that is β is small) we find that percent of critical damping, γ_n, given for each of the LSS modes is

$$\gamma_n \approx -\mathrm{Imag}\{\beta\, \Omega_{0n} g(\Omega_{0n})x/(M^{-1}\, \partial F_s(\Omega_{0n},x)/\partial\omega)\}. \tag{11}$$

Assuming the TMT can be modeled as continuous cantilevered beam and the AM-PED is mounted on its free end, F_s can be analytically defined and the operations required by equation 11 performed. The result is

$$\gamma_n = \{2\beta\}[2r(\Omega_{0n})][4r(\Omega_{0n})^2 + (W(\Omega_{0n})^2 - 1)^2]^{-1} \equiv 2\beta d_n, \tag{12}$$

where reference is to be made to equations 8 and 9. Assuming the mass m is fixed at the value of the active damping actuator, we can choose the AM-PED parameters c/m and ω_0 to either maximize the damping for a particular mode or damp more than one mode. We also note that if the LSS were a one dimensional system of mass M, the factor in curly brackets would be 1/4 of the equation 12 result. This means, at least in the limit of small frequency changes, that the effective mass of the cantilevered beam is 1/4 of its actual mass, for all modes when an AM-PED is attached to its free end.

The active damper is effective for both the first and second bending modes (Ω_{00} and Ω_{01}) respectively so we design the AM-PED to compete with it and also damp the first and second bending modes. We are interested in obtaining the best damping we can for the lowest mode and still obtain reasonable damping for the higher modes. As the detailed analysis shows, for a given r and Ω_{00} in equation 12 the numerator can be minimized by choosing

$$W(\Omega_{00}) = 1, \text{ or } \omega_0 = \Omega_{00}.$$

The damping of Ω_{00} is then maximized with the choice

$$c/m = \Omega_{00}. \tag{13}$$

As discussed in Section 2 this c/m ratio is easily achievable with the magnetic system. The choice of AM-PED parameters defined by equations 12 and 13 imply that for Ω_{0n} >> Ω_{00}

$$\gamma_n \approx \gamma_0[\Omega_{00}/\Omega_{0n}], \tag{14}$$

when

$$\gamma_0 = 2\beta. \tag{15}$$

The TMT active damper predictions were actually made with two 4 lb dampers at the free end <u>and</u> two additional 4 lb dampers, one at the center and one one-quarter of the length from the free end. A worst case comparison is made by using only one 8 lb AM-PED (equivalent to two 4 lb dampers) at the free end. Since the TMT is 220 lbs and the ratio of Ω_{00}/Ω_{01} = 4.74, as stated above, we find for the TMT that

$$\gamma_0 \approx 2 \times 8/220 = 7.3\%, \text{ and } \gamma_1 \approx 7.3\%/4.74 = 1.5\%. \tag{16}$$

Table 1 shows the TMT active damper predictions as a function of four velocity feed back schemes. The AM-PED is therefore expected to be very competitive with active damping systems. In Section 4 we will see that there is reason to suspect that the AM-PED may, in some circumstances, be a better damper than the active system. (The overall damping ratio with an 8 lb AM-PED is 11% for the computer simulated TMT, 50% greater than the analytic, continuous beam result.)

Table 1 TMT Active Vibration Control
(From Reference 2, x Bending)
Closed-Loop Modal Damping Predictions

	Open-Loop	LQG, LTR output Feedback	MEOP	Overlapping Decomp	Component Synthesis
1st Bending	.80	9.36	4.49	8.02	7.24
2nd Bending	.16	1.45	1.38	3.19	2.97

One can also estimate the TMT modal damping by using a one dimensional analog. For driven, single degree of freedom system the damping is $(2T)^{-1}$, where T is the transmissibility. Using this relationship, where the cantilever is base driven, we obtain, 6.6% damping for the first mode and 1.4% damping for the second. These numbers are consistent with the results in equations 16. But again, we have here used a continuous beam model for the TMT and differences are expected for the real structure.

3.2. STRUT-PED (S-PED) CONFIGURATION

A S-PED would be used very much like VEM damper struts used in LSS truss structures. For example, experiments were performed with the TMT using VEM diagonal strut dampers (Reference 2 and 3) in all bays (see Figure 2). The resulting damping was 4.2% for the first bending mode and 7.0 % for the second bending mode but the weight of the TMT was increased by more than 100 lbs (45 kg). It is clear that the struts could be removed from those bays experiencing the lowest modal strain energy and the damper weight would be reduced. However, the damping would be reduced somewhat as well. The TMT with strut dampers in all bays probably represents the maximum TMT damping possible with VEM.

A rough comparison of what is possible with a S-PED can be made by employing two diagonal S-PEDs in the first TMT bay (see Figure 2) and choosing a c to maximize damping for the first mode. A transmissibility analysis is then used to evaluate the damping for the second mode. The detailed analysis indicates that under these circumstances the strut dampers operate very much like the Isolator-PED of Reference 13. The Reference 13 analysis showed that the damping of the first cantilever mode was maximized at about 30% when

$$c/m = 1.5\,\Omega_{00}. \tag{17}$$

This equation is very similar to the AM-PED design equation (equation 13) except that in the case of equation 17 the mass is the effective mass of the TMT in its first mode and not the mass of the AM-PED. As discussed in Section 3.1 the effective mass of the

TMT is 1/4 the total mass or 55 lbs (25 kg). We saw that about 7% damping was expected with two 4 lb AM-PED masses, larger masses producing greater damping. In the S-PED design the moving mass is the system itself. The larger mass implies greater damping.

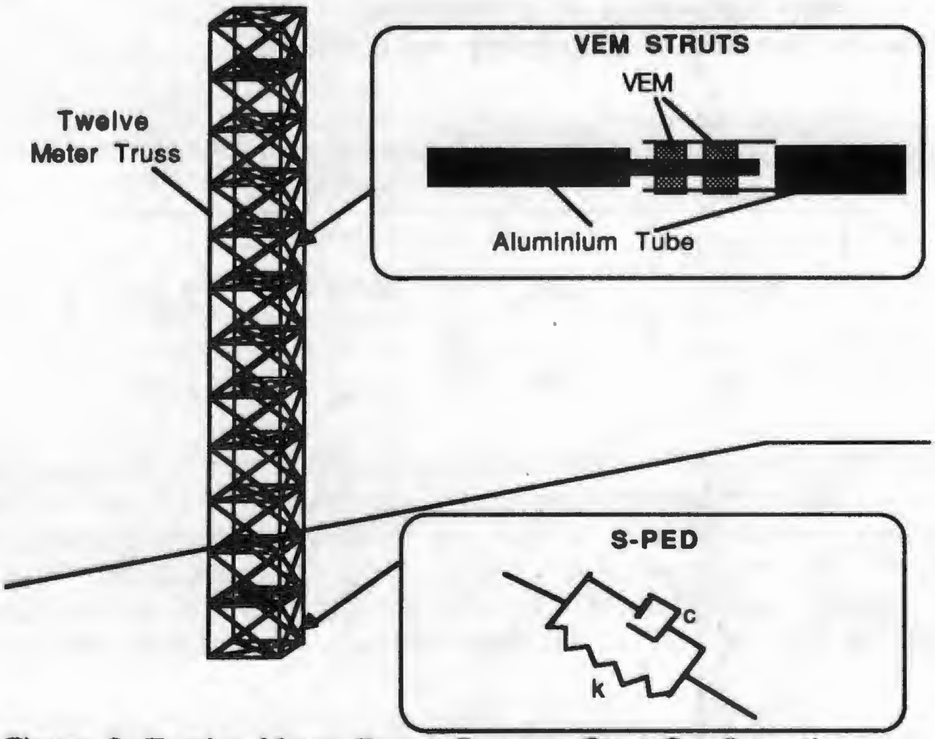

Figure 2. Twelve-Meter-Truss, Damper Strut Configurations.

The c that we need in order to obtain this large damping is given by inserting the correct parameters into equation 17. We need

$$c = 25 \times 1.5 \times 2\pi \times 2.25 = 530 \text{ kg/sec}, \tag{18}$$

or if two struts are used per bay c = 265 kg/sec for each strut. For a particular design we can achieve the required c with a total magnet mass of 2.1 kg (4.6 lbs). With this S-PED system the expected damping, predicted from a base driven transmissibility analysis of a continuous cantilever beam, is 19% for the first bending mode and 2.5% for the second bending mode. We have tuned the S-PED system for the first bending mode and it is most effective for that mode. The 19% damping of the first mode differs from the 30% expected from the single degree of freedom Reference 13 analog, but given the difference in the approximation methods numerical differences are expected. In addition, experience with comparing the AM-PED, continuous beam, analytic results with the TMT computer simulations suggests that the analytic damping estimates are a conservative lower bound.

It is difficult to directly compare the analytical S-PED analysis with the TMT VEM damping data. Theoretically, with a factor of 22 less weight (excluding the weight of the struts and armature) we have a factor of 4.5 more damping in the first bending mode. This seems to be a definite advantage. The S-PED damping in the second mode is, however, about a factor of 3 less than the VEM system. It is clear that by detuning the

S-PED the second mode damping could be increased at the expense of the first mode, if that were desirable. Although the analysis has not been performed, the expectation is that the S-PED would be superior to the VEM damping strut system. Replacing the VEM system with an S-PED would have a number of advantages: (1) there would be no outgassing problems, (2) designing a damping treatment would be simpler since the PED system is linear with respect to amplitude and c does not depend upon frequency, (3) heating coils would be avoided because the PED system performance changes very little with temperature, (4) the same PED system could be used anywhere on the LSS because of the near temperature independence of the PED.

4.0 STRUCTURAL DYNAMICS

In Section 1, it was noted that the AFWAL 12m Truss (TMT) dynamically represents large space structures of generic interest. To suppress the vibration of such systems, the application of the PED as an auxiliary mass damper was discussed to be very promising. In Section 3, a preliminarily design AM-PED for the cantilevered 12m Truss was discussed. It was observed that its effectiveness, reliability, and weight compare favorably to actively controlled and mechanical passive damping alternatives.

The structural dynamics of the TMT with AM-PED is now comprehensively analyzed to further investigate these promising possibilities. The modal analysis is considered first to gain insight and then realistic transient excitations are considered. Next, the effect of AM-PED parameters on performance is examined.

4.1 MODAL CHARACTERISTICS

The study begins with the natural frequencies and mode shapes of the system. These were obtained through the ADINA finite element model of Figure 3 (Reference 1). This describes each of the 16 bays of the TMT with a 2-noded beam element. Complete restraint against translation and rotation is assumed at the support. The AM-PED is modeled as a lumped mass connected to the free end through a general element having concentrated damping and stiffness. In all, 33 degrees of freedom describe the planar flexural vibration of this system. The modal characteristics of this response were found through a determinant search algorithm.

The undamped TMT was first considered without the AM-PED. The total length was taken to be 471 in. and the total weight 220 lbs in accordance with reported data. The stiffness parameters of the beam elements are adjusted to match the first two frequencies measured by the AFWAL. These are given in Table 2 and reflect the influence of shear as well as flexural deformation. The corresponding shapes, Figures 4 and 5, contain one and two lobes in the first and second modes, respectively, as one would expect.

Next, the influence of the AM-PED on these characteristics was studied. For this, the parameters of the preliminary design are considered which are repeated in the first row of Table 3. The AM-PED design causes the system to have two modes corresponding to the undamped fundamental mode of the cantilever. These natural frequencies and shapes appear in Table 2 and Figure 4, respectively. The first of these modes has a slightly lower frequency than the undamped fundamental and is characterized by auxiliary mass motion in phase with the beam. The second has a frequency slightly higher than the fundamental and an auxiliary mass motion in opposition to the beam. In both modes, the large amplitude of the auxiliary mass motion will be effective in dissipating the beam's vibration. Note in Table 2 that the second flexural frequency of the beam is minimally influenced by the design. Neither is the corresponding mode shape, Figure 5, in which the auxiliary mass experiences little displacement.

4.2 TRANSIENT RESPONSE

With the benefit of the foregoing modal insight, the response of the system to a transient excitation is considered. A uniform, unit, initial velocity of the beam is specifically chosen. This approximates the excitation of an impulsive maneuver by the spacecraft from which it would be cantilevered. It may also represents the loading produced by the fluence of a hostile impulsive laser attack on the platform. The response to this initial disturbance was calculated using the ADINA model through a direct time integration with a step of 0.010 sec.

The resulting tip deflection for the undamped case is shown in Figure 6. This is dominated by the fundamental mode at 2.26 Hz. With no dissipative mechanism in the system, the oscillations continue indefinitely. Such behavior is not consistent with the precise stability requirements for many space platforms.

Fortunately, the situation improves dramatically in the response with the preliminary AM-PED design which is superimposed in Figure 6. This response is initially dominated by the fundamental bending modes. However, these are effectively damped in a few cycles. A least squares fit of the response indicates that it decays with an exponential envelope corresponding to 6.2% damping. This is almost twice as large as predicted in Section 3.1 for the continuous beam. (Note that in Section 3.1 we considered an 8 lb AM-PED, here the simulation was for a 4 lb AM-PED. Equations 6 and 12 show that damping is expected to be linearly proportional to AM-PED mass.)

4.3 PARAMETRIC ANALYSIS

The above transient analyses indicate that the preliminary AM-PED design should be quite effective in suppressing the vibration of large space structures. Accordingly, the influence of its design parameters on this effectiveness is studied. Auxiliary mass and frequency tuning is specifically addressed.

To examine the effect of auxiliary mass, this parameter is doubled above the 4 lb preliminary design. In accordance with the preliminary design procedure, we also double the stiffness and damping values to maintain the same tuning relative to the cantilevers fundamental mode. The response with this 8 lb device is compared to that previously calculated for the 4 lb design in Figure 7. With the additional mass, the vibration is suppressed even more rapidly. The equivalent damping, Table 3, is now 10.9 %. Thus the damping effectiveness increases almost linearly with the size of the auxiliary mass, as suggested by our first order perturbation analysis.

To examine the influence of AM-PED tuning, the auxiliary mass is returned to the initial value of 4 lb. In lieu of the preliminary design of Section 3, an alternative exists which attempts to limit the response of the primary system over a range of frequencies in the neighborhood of its fundamental mode (Reference 14). The parameters of this "optimal" design for the cantilevered TMT are given in the third row of Table 3. The response of this system is compared to that previously calculated for our preliminary design in Figure 8. The effectiveness of the AM-PED is seen to be a function of frequency tuning. For the uniform initial excitation imposed, the "optimal" design achieves 3.6 % damping and is less effective than the preliminary concept.

TABLE 2 NATURAL MODES OF 12M TRUSS

Beam Character	Undamped Frequency Hz	AM-PED Frequency, Hz
Fundamental	2.26	1.98 2.56
Second	10.70	10.71

TABLE 3 AM-PED EFFECTIVENESS

12m Truss Configuration	mg lb	c lb-sec/in	k lb/in	Damping ratio
Cantilevered	4	0.1464	2.069	0.062
Cantilevered	8	0.2928	4.138	0.109
Cantilevered	4	0.0487	1.577	0.036
Free	2*	0.422*	34.4*	0.048

*Values for each of two AM-PEDs. Free-Free analysis not presented.

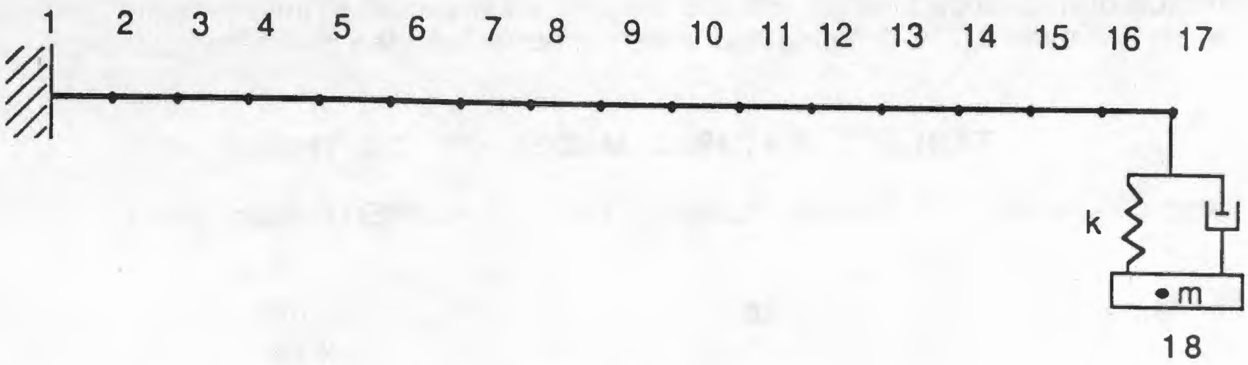

Figure 3. ADINA Model of Cantilevered TMT

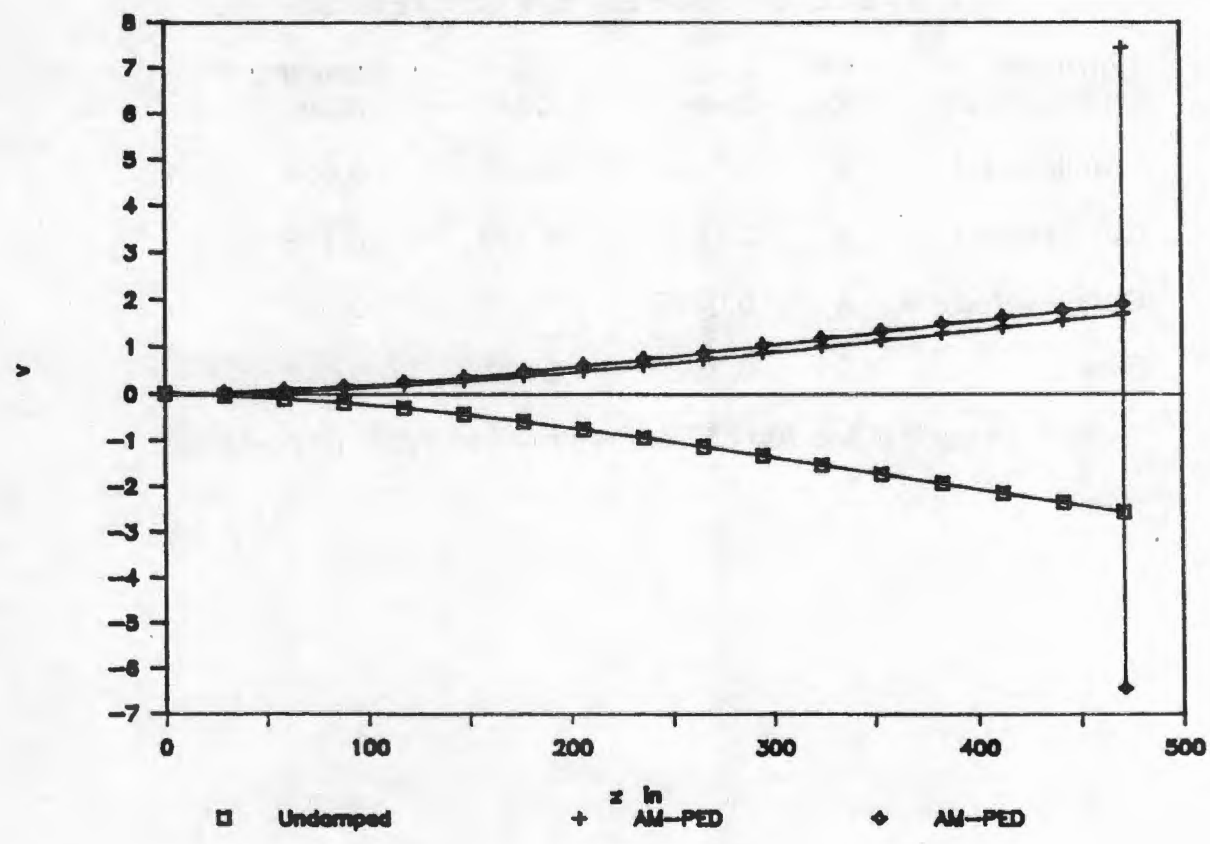

Figure 4. Fundamental Mode of TMT

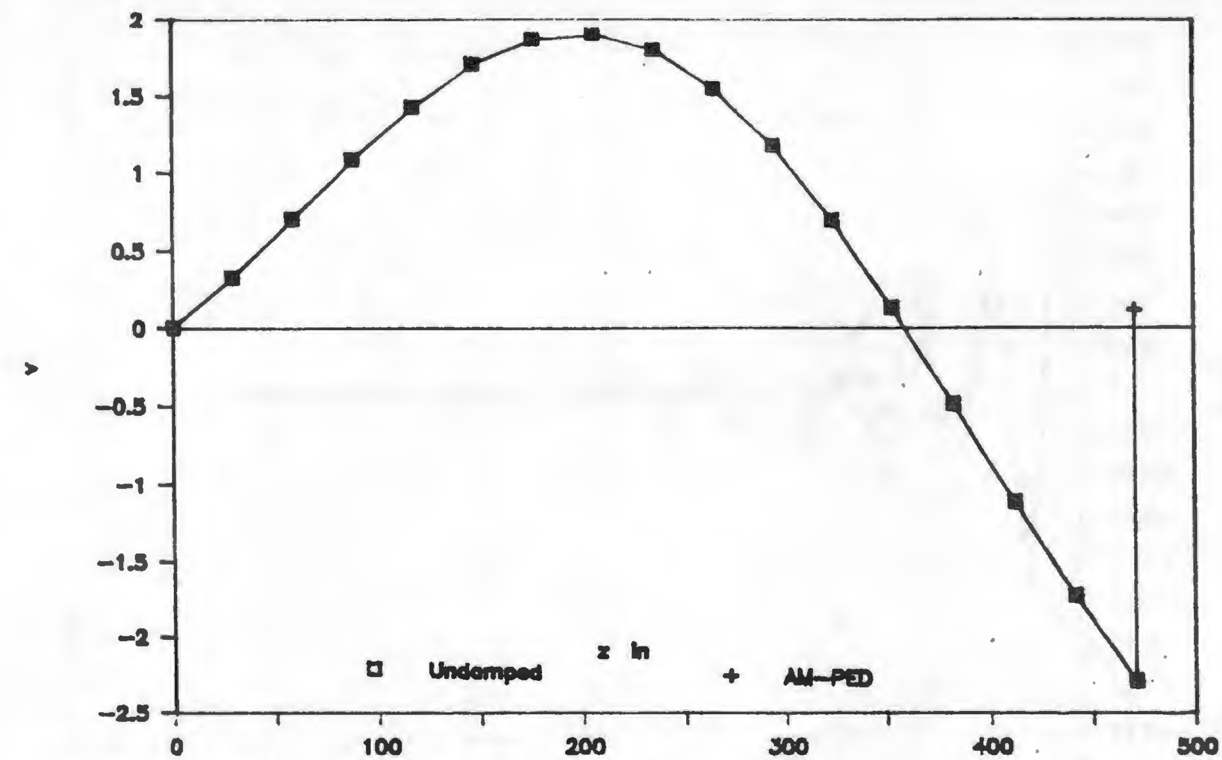

Figure 5. Second Mode of TMT

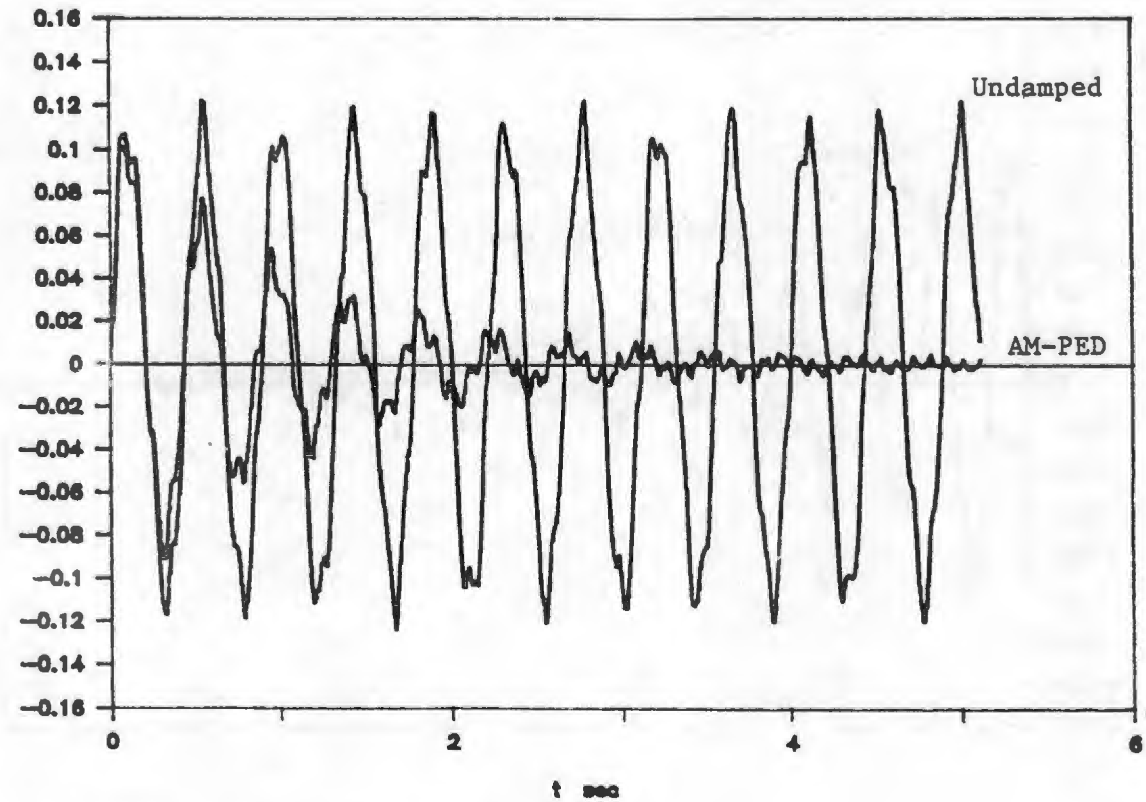

Figure 6. Transient Response of TMT

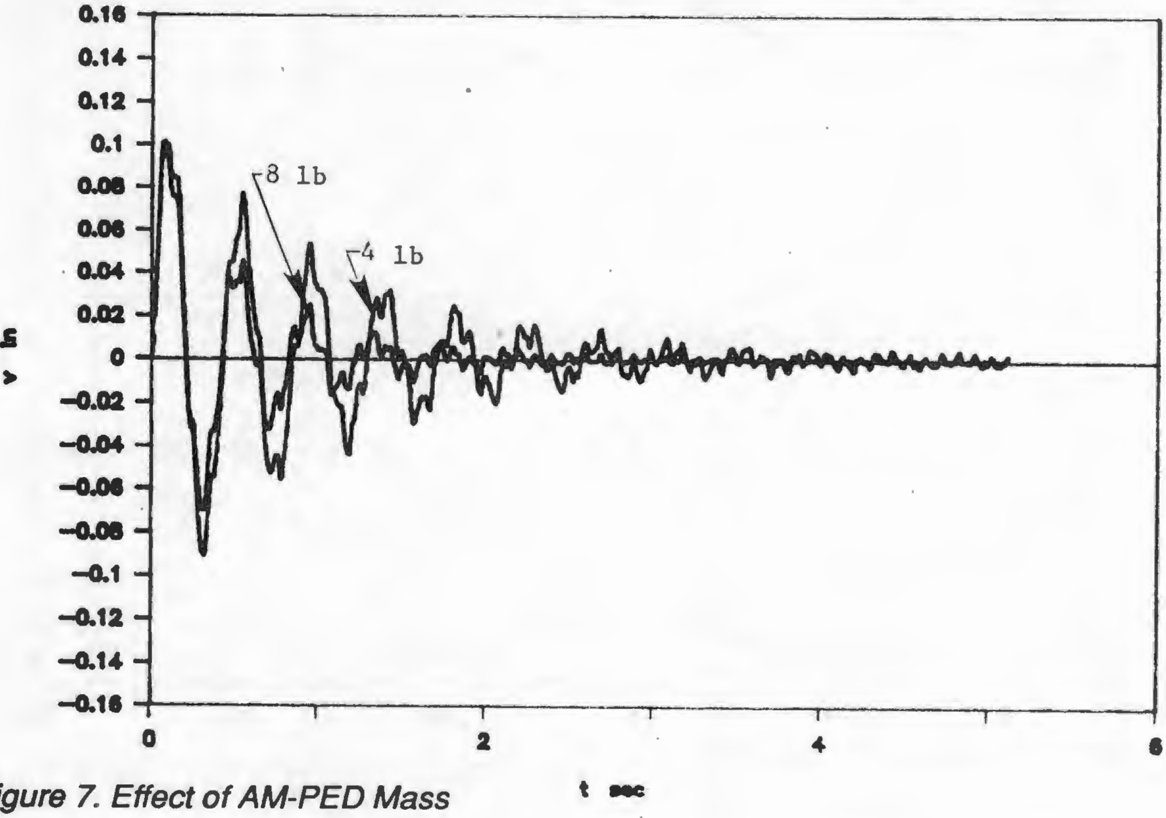

Figure 7. Effect of AM-PED Mass

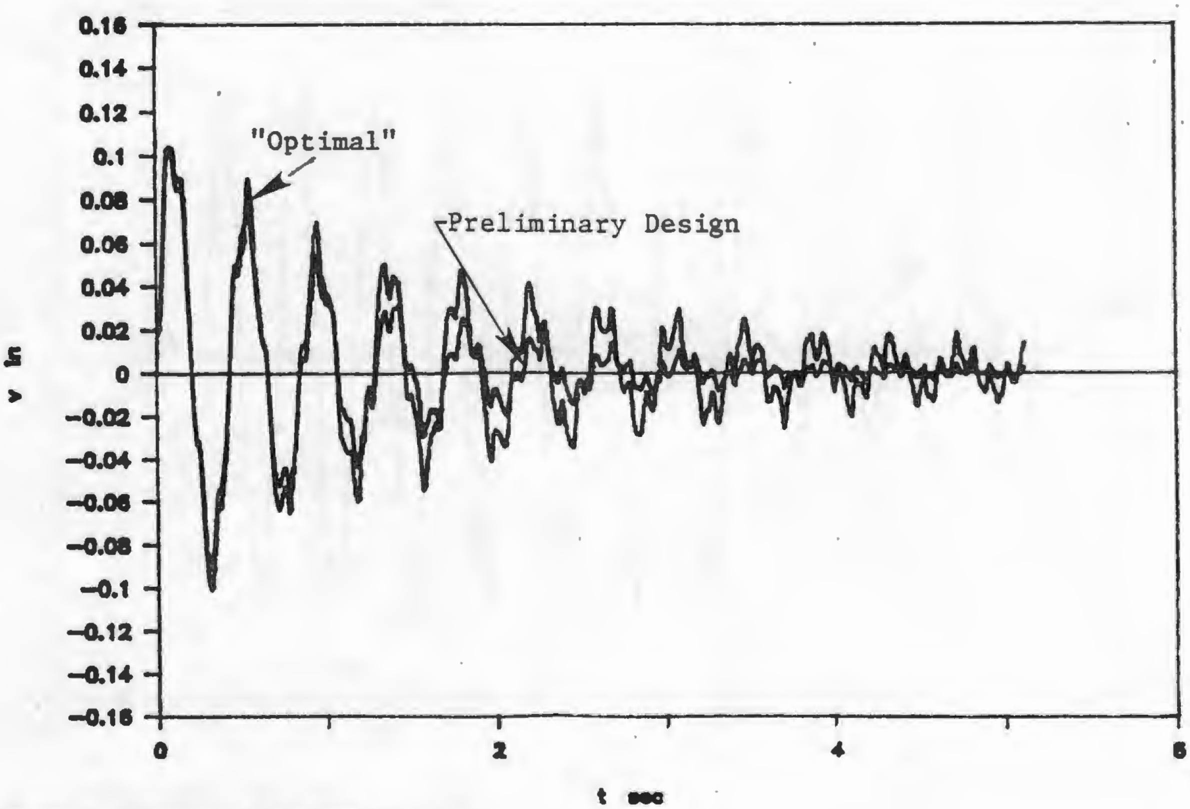

Figure 8. Effect of AM-PED Tuning

5.0. CONCLUSIONS AND RECOMMENDATIONS

In this study the feasibility of Passive Electrodynamic Dampers (PEDs) for Large Space Structures (LSS) has been investigated. The overall conclusion is that preliminary PED designs appear very promising, being competitive and in many ways superior to current active damping and passive damping LSS technologies. The overall recommendation is that a detailed design and experimental test program be undertaken to verify the conclusions of the study. The detailed conclusions of this study are presented below.

The AM-PED operates by converting LSS vibrational energy to the kinetic energy of the magnetic-system mass. This energy is then dissipated through ohmic heating in the armature. Mechanical springs are used as a restoring force and the system is "tuned" to damp over a range of LSS modal frequencies. Because of the simple PED force relationship, analytic LSS damping estimates can be made when analytic LSS modal solutions exist. Computational solutions are required for realistic LSS truss structures which only roughly approximate continuous, analytically-tractable systems.

Besides computationally evaluating the effectiveness of the PEDs it was considered important to compare PED effectiveness with experimental data and pretest predictions for other damping systems. The AFWAL 12 Meter Truss (TMT) experiments were chosen for comparison. The majority of analyzed TMT data is for the low-frequency cantilevered position. The AM-PED, designed according to the analytic analysis, and the TMT were ADINA modeled.

The conclusions of the study are the following:

(1) The maximum practical LSS c/m ratio is about 500 sec^{-1} in mks units.
(2) The maximum c/m ratio dependents on magnetic system size.
(3) The dominant effect which reduces c is the dependence of skin depth on the frequency. PED designs should be independent of frequency below 50 Hz.
(4) PED damping should be independent of amplitude for expected LSS vibrational amplitudes.
(5) PED damping should vary by only about ±30% over the full 150°C space environment temperature variation.
(6) Compared to TMT bending-mode, active-damping predictions, for a roughly equivalent weight damping system (8 lbs - actually the total active damping actuator weight was twice the AM-PED weight), the AM-PED is more effective than active damping. AM-PED damping is expected to be about 11%. The largest active damping is expected to be 9.4% for the first bending mode. The AM-PED is not only expected to be superior to active damping in performance but more reliable and cost effective. The AM-PED doesn't require a power supply, motion sensors or a computer control system.
(7) The effectiveness of the AM-PED damper is only slightly decreased (23%) below the cantilever TMT, for the free TMT (analysis not presented in this paper).
(8) A Strut-PED system is expected to be comparable or superior to VEM strut damping systems in performance, should weigh less and be far superior with respect environmental stability, outgassing, and calculability.

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