SYNERGISTIC DESIGN OF PASSIVE DAMPING AND METAL MATRIX COMPOSITES

E. D. Pinson
Lockheed Missiles & Space Company, Inc.
Space Systems Division
1111 Lockheed Way
Sunnyvale, CA 94089-3504

E. M. Austin
CSA Engineering, Inc.
560 San Antonio Rd.
Palo Alto, CA 94306-4682

M. L. Zeigler
Flight Dynamics Directorate
Wright Laboratory, U.S. Air Force
Wright-Patterson AFB, OH 45433-5553

Advanced materials, such as Metal Matrix Composites, offer significant improvements in the areas of stiffness, weight, and thermal stability over most conventional materials. To further enhance the benefits afforded by these materials, integration of a passive damping concept should occupy a prominent place in the structural design process as early as feasible. To prove this synergistic approach, a simplified version of a flight-ready spacecraft attitude sensor platform has been chosen. Ultimately, two such platforms will be built, tested, and contrasted in several ways to demonstrate the performance improvements which are possible through this integrated approach. One structure will be fabricated using conventional materials and fabrication techniques, while the other will combine Metal Matrix Composite panels with a constrained-layer viscoelastic damping treatment. This paper describes the requirements, history, and design of the passively damped structure, and introduces plans for future testing and validation of analytical models.

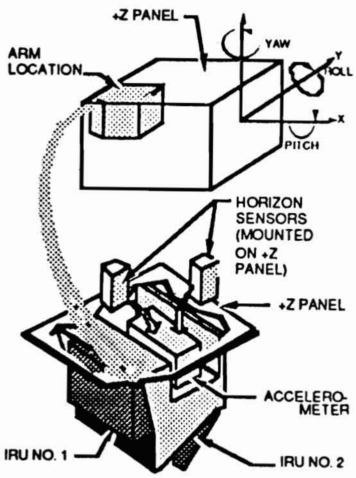
INTRODUCTION

Present military aerospace systems, as well as those planned for future development, require increasingly stringent dimensional precision which is unattainable with reasonable extrapolations of current technology. In particular, the capabilities of systems involving radar, optical, communications, and infrared surveillance are becoming limited by structural and attitude control considerations. Deformations induced by thermal excursions and gradients make attainment of required dimensional precision extremely challenging. In addition, dynamic motions due to operational or residual vibrations often limit the ability to establish and/or maintain required pointing accuracies. These challenges are especially critical when the attitude control system itself is involved, because even small errors in sensor alignment can lead to significant misalignments of the satellite.

This paper describes the Damping and Metal Matrix for Precision Structures (DAMMPS) program, an Air Force Wright Laboratory program being performed by Lockheed Missiles and Space Co. (LMSC). The objective of this program is to demonstrate the synergistic benefits of metal matrix composites (MMC) and viscoelastic materials (VEMs). MMC technologies provide high specific stiffness, high thermal conductivity, and a low coefficient of thermal expansion, and VEM technologies provide passive vibration damping (PVD). The approach of DAMMPS is to design, fabricate, and test two Demonstration Structural Articles (DSAs) that functionally duplicate a satellite Attitude Reference Module (ARM) [1]. One of these structures will be constructed of aluminum, providing baseline data from which improvements will be measured, and the other will incorporate PVD and MMC in the design. Both structures will contain mass simulators representing inertial reference units (IRUs), horizon sensors, and an accelerometer unit.

The process of selecting an appropriate DSA began with five potential candidate structures and was based on an evaluation of several weighted criteria. Because this is an advanced development program for the Air Force with the goal of technology transition into hardware programs, the two highest weighted criteria were probability of insertion and cost. Other criteria, in descending order of weighting, were definition of environment, PVD treatment, and testing complexity. Based on these criteria, the ARM, which is currently in the hardware phase of an ongoing Air Force program, was chosen as the DAMMPS DSA because it possessed the greatest design maturity and detailed knowledge of the two environments of interest. The most unattractive aspect was the relative complexity of the structural configuration, but with subsequent simplifications, preserving all salient features, a modified version of the ARM emerged as the final choice.

The DSA design further evolved during the preliminary design phase from a simplified representation of the ARM to a detailed dynamic model with better traceability to satellite system configuration and requirements. The original design placed the two horizon sensors on the "top" of the DSA along with the accelerometer unit. Further, the size and mounting details of the simulated sensors were not precisely replicated. Following a meeting between ARM and DAMMPS personnel, it was decided to move the horizon sensors to the "bottom" of the DSA, and to scale and more closely represent actual mounting characteristics of the mass simulators. In this way, the DSA will more accurately reflect the dynamic response of the ARM. Figure 1 depicts the original ARM and the current DSA.



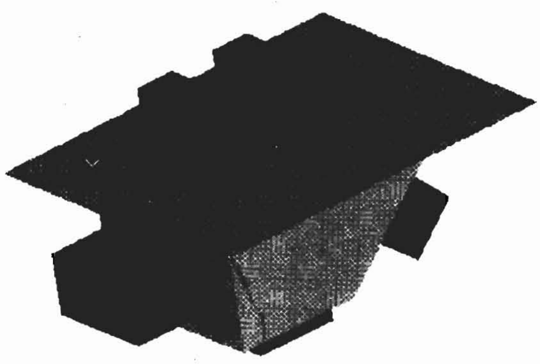


Figure 1. Traceability of the DSA to ARM

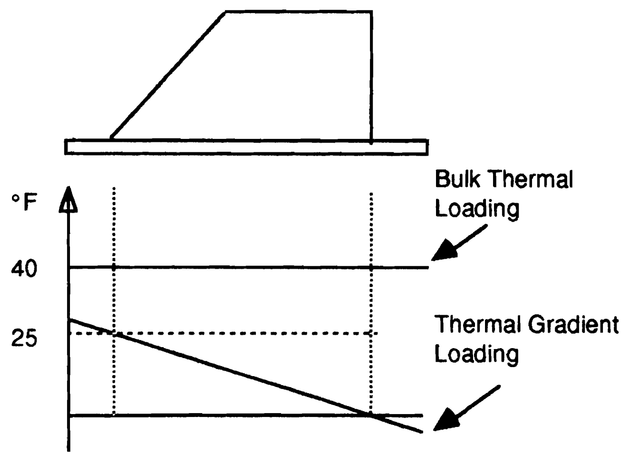
This paper provides an overview of the DAMMPS program at LMSC, including goals, technical progress, and plans. Particular emphasis will be placed on the passive damping aspects.

ARM AND DSA OPERATIONAL ENVIRONMENTS

One of the principal reasons for selecting the ARM for detailed study under the DAMMPS contract was the maturity of knowledge regarding the thermal and dynamic environments. In most instances, the level of detail available greatly exceeded that which could be efficiently utilized in the planned test program, therefore conservative simplfications were employed. Figure 2 is a comparison between the ARM thermal environment specification and the simplified temperature distributions which are designed to simulate a nominal operational loading.

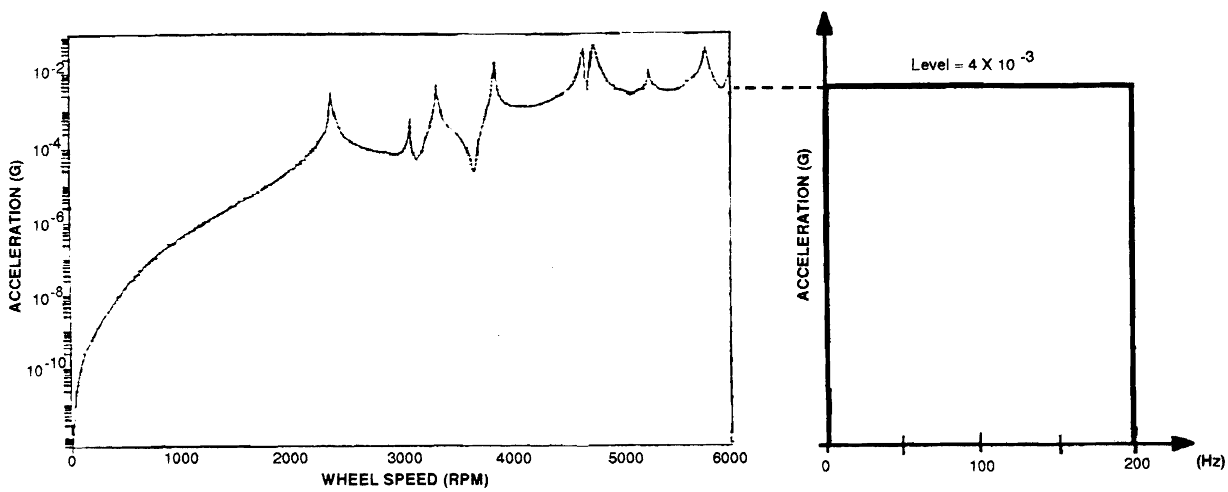
Location	Hot °F	Cold °F
Horizon Sensors	82	47
IRUs	107	72
Accelerometer	98	61
ARM -Roll Axis End	103	68
ARM +Roll Axis End	101	69
ARM -Pitch Axis End	96	66
ARM +Pitch Axis End	98	62
ARM/IRU-1 Support Plate	107	67
ARM/IRU-2 Support Plate	107	68

ARM Thermal Environment Specification



DSA Thermal Environment Simplification

Figure 2. Thermal Environments of the ARM and DSA



ARM Dynamic Environment Specification

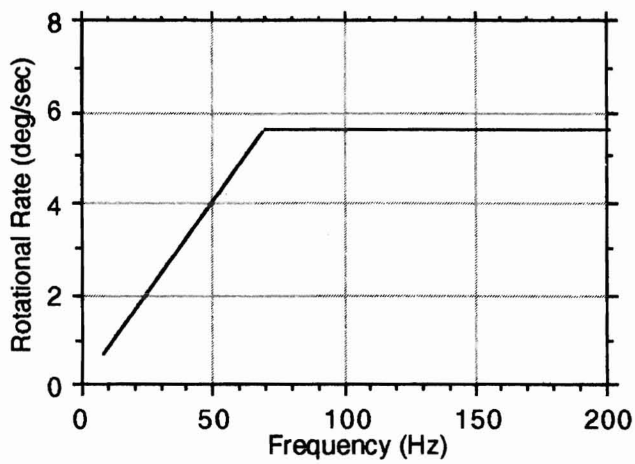
DSA Dynamic Environment Simplification

Figure 3. Dynamic Environments of the ARM and DSA

The dynamic environment of the ARM is dominated by inputs from the reaction wheel assembly (RWA) which is mounted to the spacecraft bus a short distance away. The spectrum of this excitation source is known to be dominated by the first harmonic, and may excite the bus (hence, the ARM) in any direction. The test-derived envelope of RWA spectra, representing maximum vibration levels observed in any direction at specific wheel speeds, is shown in Figure 3. The RWAs are presently used at speeds under 3000 rpm, but growth plans involve operation to 3500 rpm (58.3 Hz., assuming first harmonic dominance). Based on these considerations, the peak dynamic excitation from the wheels (below the growth speed) is 0.004 g – assumedly steady-state, due to the nature of the excitation source. A more conservative dynamic environment (shown for comparison purposes in Figure 3) has been assumed for the DSA. This simplified spectrum is based on the peak acceleration level of the current RWAs, but extends to 200 Hz.

ARM AND DSA PERFORMANCE REQUIREMENTS

Another of the principal reasons for selecting the ARM for DAMMPS contract studies was the maturity of system-level requirements flow-downs with regard to thermally and dynamically induced error budgets. These requirements fall into two general categories: those designed to protect the sensors themselves, and those designed to yield consistent information from all sensors (alignment tolerances). Requirements from both categories are shown in Figure 4.



COMPONENTS	Axis	IRUs Pitch	IRUs Roll	IRUs Yaw	HSs Pitch	HSs Roll	HSs Yaw
Accelerometer	Pitch				120		
	Roll					120	
	Yaw			120			
IRUs	Pitch				34		
	Roll					34	
	Yaw						
HSs	Pitch	34					
	Roll		34				
	Yaw						

Units: Arc-second

Figure 4. ARM and DSA Rotational Rate and Alignment Requirements

DSA RESPONSE ANALYSES

During the preliminary design phase of the cotract, several finite-element models of DSA were constructed and subjected to appropriate thermal and dynamic environments. All models share the same general geometry as shown in Figure 5, which is a representative finite-element model used for dynamic response studies.

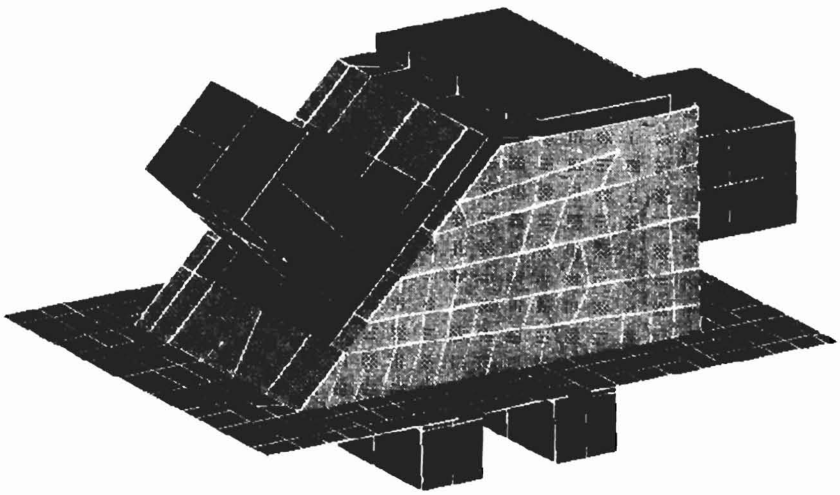


Figure 5. Representative Finite-Element Model of the DSA

For the purpose of comparison, several different material properties were substituted into geometrically similar models (see Figure 6). Standard aluminum characteristics were used to establish a baseline and coarse-tune panel thicknesses. Current plans involve the use of an aluminum sheet, folded to produce the general shape of the DSA above the baseplate. All joints will then be fastened using either sheet-metal screws or welded connections.

Two different MMC materials and shapes are required to produce the other DSA. In this case, SiC/Al structural angles (0.25 VF) and several flat P100/Al quasi-isotropic plates will be joined together to form a precision sensor base. For connections between the "upper" DSA and the baseplate, angles will be placed in exterior corners for ease of construction. Due to mounting constraints, angles to fasten the top plate to the side panels will be located on interior surfaces.

Material	Young's Modulus [psi]	Density [lb/in^3]	Coefficient of Thermal Expansion [in/in/°F]
SiC/Al (0.25 VF)	17.0	0.102	7.0 X 10^{-6}
P100/Al (Quasi-isotropic)	20.0	0.090	1.9 X 10^{-6}
Aluminum	10.0	0.098	13.0 X 10^{-6}

Figure 6. Selected Mechanical Properties of DSA Materials

The aluminum DSA was used to establish a baseline thermal performance level which was compared with that of the MMC DSA. Misalignment results for the Aluminum DSA are shown in Figure 7. Figure 8 presents the same results for the MMC DSA. While these results should be compared to each other to determine the level of improvement attainable by changing the material of the structure, these results should also be compared to Figure 4 – the alignment requirements. This latter comparison indicates that neither structural configuration produces the required results, but the MMC DSA deviates from allowable levels in only two areas (both involving IRU-2), whereas, the aluminum DSA exceeds acceptable values in several cases involving the horizon sensors and IRUs. Slight modifications to the configuration, presently under investigation, are available to the designer and analyst to improve these results even further.

		IRU-1			IRU-2			HS-1			HS-2		
COMPONENTS		Pitch	Roll	Yaw	Pitch	Roll	Yaw	Pitch	Roll	Yaw	Pitch	Roll	Yaw
Accelerometer	Pitch							18/2			6/6		
	Roll								109/36			123/6	
	Yaw			0/0			0/0						
IRU-1	Pitch							18/2			6/6		
	Roll								123/58			137/15	
	Yaw												
IRU-2	Pitch							18/2			6/6		
	Roll								34/4			21/38	
	Yaw												
HS-1	Pitch	18/2			18/2								
	Roll		123/58			34/4							
	Yaw												
HS-2	Pitch	6/6			6/6								
	Roll		137/15			21/38							
	Yaw												

Notes: 1. Units: arc-seconds.
2. Bulk Response/Gradient Response.

Figure 7. Alignment Deviations due to Thermal Loading of the Aluminum DSA

		IRU-1			IRU-2			HS-1			HS-2		
COMPONENTS		Pitch	Roll	Yaw	Pitch	Roll	Yaw	Pitch	Roll	Yaw	Pitch	Roll	Yaw
Accelerometer	Pitch							28/7			0/3		
	Roll								21/10			23/30	
	Yaw			0/0			0/0						
IRU-1	Pitch							28/7			0/3		
	Roll								11/10			10/0	
	Yaw												
IRU-2	Pitch							29/8			1/3		
	Roll								38/6			40/16	
	Yaw												
HS-1	Pitch	28/7			29/8								
	Roll		11/10			38/6							
	Yaw												
HS-2	Pitch	0/3			1/3								
	Roll		10/0			40/16							
	Yaw												

Notes: 1. Units: arc-seconds.
2. Bulk Response/Gradient Response.

Figure 8. Alignment Deviations due to Thermal Loading of the MMC DSA

The DSA was designed such that salient dynamic characteristics are traceable to the ARM. To achieve this, mass simulators and panels were sized and placed to yield the first four modal frequencies near those of the ARM, though information concerning the associated mode shapes is minimal. While the finite-element models contain more element detail than needed for most basic eigenvalue analyses, there is appropriate detail for performing trade studies and predicting damping performance. Fixed boundary conditions were imposed at the edges of the baseplates to simulate mounting on a relatively stiff spacecraft bus or test fixture. The first four modes predicted for the aluminum DSA are shown in Figure 9. Their frequencies are 66.8 Hz, 77.6 Hz, 78.4 Hz, and 109.8 Hz. – acceptably close to those of the ARM.

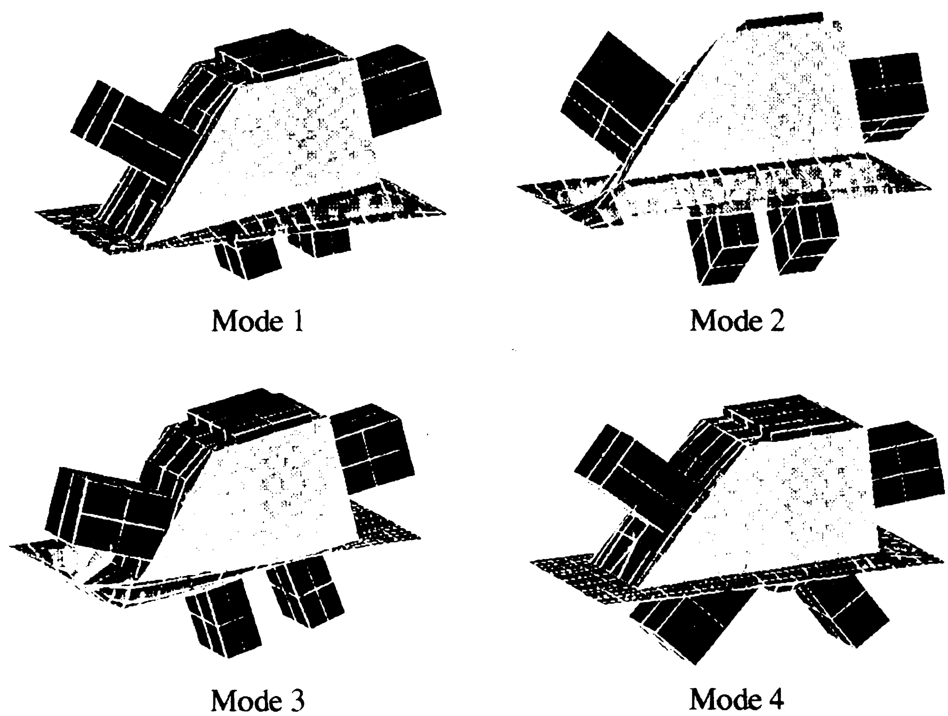


Figure 9. Four Lowest Frequency Mode Shapes of the Aluminum DSA

A baseline undamped MMC finite-element model was constructed and used as a starting point for the analogous damped structure. In this model, MMC panels were sized such that natural frequencies of the first four modes were as close possible to those of the aluminum baseline, further maintaining traceability.

Standard damping treatment design methods require identifying offending modes of the structure and performing trade studies involving various damping solutions. The target modes were those which contributed the largest percentage to base rotation rates at the IRUs and sensor misalignments – the two classes of performance requirements. At this point, the damping level normally associated with a jointed aluminum structure (e.g., 0.5% of critical) was used as a baseline for all modes to determine the level of additional damping required from a passive damping treatment. Since, by replicating known frequencies of the "as-mounted" ARM, the DSA model was designed to simulate the dynamic load path from the RWA to the ARM, excitation was applied as a base shake of the DSA from the constrained rim of the base plate. Using the assumed 0.5% damping ratio, transfer functions were calculated which related base shake inputs in the x, y, and z directions to each of the pertinent response quantities (i.e., relative alignments and IRU base rotational rates). By definition, these transfer functions represent the dynamic responses to steady-state, unit-amplitude, sinusoidal base shake inputs. Data thus obtained were scaled to 0.004 g to determine the responses at the actual input levels.

As expected, several different analytical modes produce dominant responses, depending on the excitation frequency and direction. To determine modal damping requirements, each of the peak responses in each of the transfer functions were tabulated. A total of nineteen response quantities and three input directions yielded 57 plots which were examined in this manner. Final results indicated that the lowest four modes dominated all critical response quantities. Therefore, by linearly scaling the baseline damping level, the required damping levels for these four modes (in order) were determined to be 4.0%, 3.5%, 4.0%, and 1.2%. Implicitly, the assumption was made that only damping is used to bring the responses to a level consistent with the requirements.

PASSIVE DAMPING TRADE STUDIES

The most common classes of passive damping treatments are constrained-layer, free-layer, and discrete. Free-layer treatments consist of a layer of viscoelastic or other damping material applied directly to the base structure. In general, these treatments are heavy as well as inefficient for bending modes; thus the concept is not being considered for DAMMPS. Discrete damper concepts such as tuned-mass dampers and damped links, usually used to damp a few specific modes, are not applicable to this structure because the basic design is not readily amenable to such treatments. Constrained-layer treatments, however, are both effective and weight-efficient for damping multiple bending-dominated modes, if the bulk of the strain energy in these modes distributed over the treated area of the structure.

There are five primary characteristics of a constrained-layer damping treatment that can be varied during a trade study: location, constraining-layer (CL) thickness, CL modulus (assumed equal to that of the base material in this case), VEM thickness, and VEM modulus. The trade study undertaken for the DSA addresses each of these with an overriding goal of achieving the levels of damping outlined in the previous section while minimizing the weight of the damped platform.

Having identified the target modes damping from the dynamics requirements, the first step in the PVD design is to determine which areas of the DSA participate most in these modes. It is conclusive from both the distribution of modal strain energy (MSE, reference 2, Figure 10) and the mode shapes that damping the baseplate is necessary to achieve the goals.

	Mode 1	Mode 2	Mode 3	Mode 4	Mode 5
Frequency [Hz.]	66.77	77.60	78.45	109.8	124.9
Base Plate	81.8	61.3	82.2	13.1	51.9
Vertical Housing Plates	6.7	10.2	5.7	20.1	8.0
Sloping Housing Face	1.5	17.0	3.8	45.9	7.5
Other	10.0	11.5	8.3	20.9	32.6
Total	100.0	100.0	100.0	100.0	100.0

Figure 10. Modal Strain Energy Distribution for Target Modes

Three concepts were chosen for study to take advantage of the bending characteristics of the baseplate:

1. Make the baseplate a sandwich section, consisting of a layer of VEM between two MMC facesheets;
2. Bond the housing portion of the DSA to the baseplate with a high-loss VEM instead of a structural adhesive; and
3. Attach a pair of extruded MMC angle sections to the underside of the baseplate with a VEM.

A variation of the second candidate treatment was also studied, in which the longer two attachment brackets between the housing and baseplate were extended to the outer edge of the baseplate.

As partially enumerated before, each concept has strengths and weaknesses. In order to evaluate the relative merits of the treatments, goals were set with regard to added weight, modal frequencies, and damping. The goals for weight and damping were simple: minimize the weight of the treatment and maximize the damping provided. For frequency, the goal was to fall within 15% of the baseline aluminum DSA frequency. Figure 11 summarizes the best effort arising from each concept, along with the relative score and overall ranks. The scores were normalized on a 100-point scale – higher scores being the best. For conciseness, the frequency and percent MSE of only the first mode are listed in this table, though each of the first four modes was considered in the final score. The VEM thickness was maintained at 0.006 in., with a shear modulus of 140 psi for each of these runs. This corresponds to the 3M Y-966 tested under DAMMPS evaluated at room temperature and 70 Hz. [3]. The loss factor of the Y-966 at room temperature is greater than 1.0 for frequencies in the approximate range of 5 to 1000 Hz., so the MSE values in Figure 11 correspond to roughly equal levels of structural loss. (In terms of structural loss, the goals for the first four modes are twice the goals for viscous damping, e.g., 8.0%, 7.0%, 8.0%, and 2.4%, respectively.)

CONCEPT	WEIGHT*	CONSTRAINING ELEMENT	MODE 1 FREQ. [HZ.]	MODE 1 MSE	FINAL SCORE	FINAL RANK
Baseline Aluminum	14.2	N/A	66.8	N/A	N/A	N/A
Sandwich Baseplate (1)	12.5	0.06" Face-sheets	49.8	8.6%	30.5	4
Structural Adhesive (2)	13.2	Housing	63.5	10.4%	56.1	3
Variation on Structural Adhesive (3)	9.3	Housing with Extended Angle Brackets	109.0	9.6%	62.5	2
Constrained Angles on Bottom (4)	8.8	1 X 1 X 0.08 Extruded Angles	48.2	20.7%	100.0	1

* Weight does not include the 62.1 lbs for the mass simulators

(1) VEM sandwiched between two 0.060"-thick MMC facesheets

(2) VEM replacing structural adhesive between housing and baseplate

(3) VEM replacing structural adhesive between housing and baseplate as well as long-direction attachment brackets extending the length of the baseplate.

(4) VEM sandwiched between baseplate and extruded angle section on the underside of the housing

Figure 11. Relative Ranking of Candidate Damping Treatments

Each of the concepts could be altered so that it met some of the goals, but adding the "constrained stiffener" treatment to the underside of the baseplate proved to be the best compromise. It provides very high levels of damping (10% viscous in the first two modes and 5% in the third and fourth) and results in a weight reduction of approximately 40% over the aluminum baseline structure.

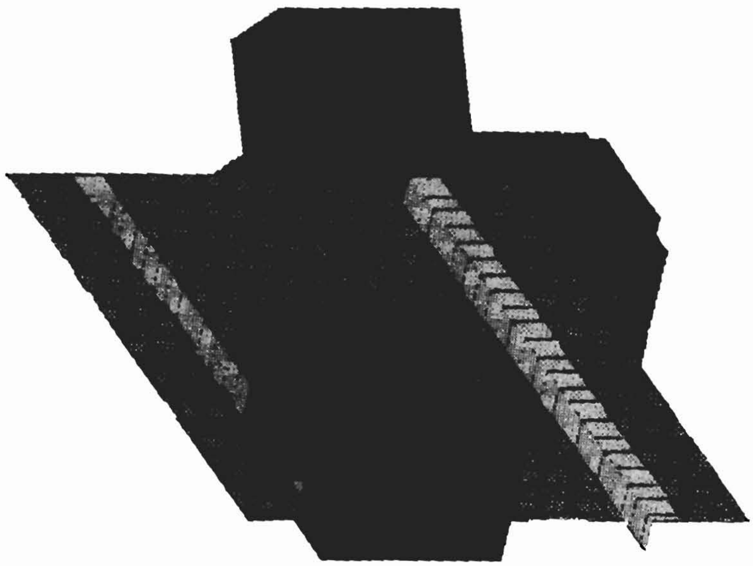


Figure 12. Selected PVD Concept

SUMMARY AND FUTURE PLANS

At this point in the study, a baseline structural configuration has been selected and modeled. Thermal analysis shows that some minor modifications and/or analytical refinements must be made to meet the requirements in the case of the undamped DSA. Further analysis will be undertaken to determine whether the damped DSA exhibits greater or lesser tendency to achieve the alignment goals for the system. If thermal conduction properties of the VEM are shown to hinder this performance, several novel concepts for optimizing this property will receive further investigation.

Purposely omitted from this paper are the ongoing activities involving design and testing of MMC coupons. This topic could not be treated adequately in conjunction with the damping studies in a discussion of reasonable length. Since the stiffness characteristics and the associated MSE distribution of the DSA are contingent on the accurate mechanical properties of MMC, the current thrust of efforts in this area is directed toward proving the accuracy and repeatability of these properties. Results from these studies will be utilized in future analyses of both the thermal and dynamic performance of the DSA.

As knowledge of MMC properties is obtained and the details of the PVD treatment converge on a viable design, component testing will begin, to prove the analytical assumptions and predictions and to fine-tune the finite-element models. Following this basic groundwork, the DSAs will be fabricated and tested at the system level.

ACKNOWLEDGEMENTS

The work described in this paper is being performed under the Damping and Metal Matrix for Precision Structures (DAMMPS) contract, sponsored by the Flight Dynamics Directorate of the Wright Laboratory. Though the efforts of several others were very important to achieving the successes summarized here, special thanks is given to Dr. Y.C. Yiu for his technical advice and service as a focal point for this phase of the program. The technical and administrative assistance of Mr. J. Warren Hoskins, Program Manager, is also gratefully acknowledged. Special thanks are also due to Mr. Kevin Napolitano for his expertise in modeling the DSA and performing several of the passive damping trade studies described herein.

REFERENCES

1. Yiu, Y.C., and Pinson, E.D., "Damping and Metal Matrix for Precision Structures - Semi-Annual Report, April 1990," Lockheed Missiles & Space Company, Inc., Document F294320-6.

2. Johnson, C.D., Kienholz, D.A., "Finite Element Prediction of Damping in Structures with Constrained Viscoelastic Layers," AIAA Journal, Vol. 20, September, 1982, pp. 1284-1290.

3. Allen, B.R., Pinson, E.D., "Complex Stiffness Test Data for Three Viscoelastic Materials by the Direct Complex Stiffness Method," presented at the Damping '91 Conference, San Diego, CA, February 13-15, 1991.