Integral Damping Treatment for Primary Aircraft Structures

by

Sal Liguore[1], Marty Ferman[2], Rudy Yurkovich[3]

McDonnell Aircraft Company
McDonnell Douglas Corporation
St. Louis, Missouri

ABSTRACT

The dynamic response of primary aircraft structure to buffeting flows, high acoustic levels, and shock boundary layer interaction has led to premature structural fatigue failures on current aircraft and is anticipated to be a continuing problem in the future. Increasing structural strength/stiffness can be a solution but this approach adds weight to the aircraft. Since the problem is dynamic response, increasing the amount of damping in the structure can also be a solution. If integral damping is considered as a part of the original design, a lighter weight design can result. The application of integral damping to primary aircraft structure was investigated and its effectiveness in controlling the primary structural modes was assessed. The findings show the approach is feasible. A simulated aircraft structure was tested with damping treatments applied. The most promising damping concepts were then analytically evaluated on the F/A-18 vertical tail.

1. Engineer, Structural Dynamics and Loads
P.O. Box 516, Saint Louis, Missouri 63166-0516
Mail Code 0341180, (314) 232-3109

2. Pr. Tech. Specialist, Structural Dynamics and Loads

3. Sr. Pr. Tech. Specialist, Structural Dynamics and Loads

BACKGROUND

Adding significant damping to reduce the primary structural mode response of lifting surfaces on aircraft can be challenging. The damping in these modes during flight can be high due to the aerodynamics present. For example, a 10 percent structural damping coefficient in the wing first bending mode is typical. Thus, in order to reduce the response by half, this aerodynamic damping level must be exceeded if the damping treatment is to be effective.

In 1987 as part of McDonnell Aircraft Company's Internal Research and Development (IRAD) program, a combined analytical and experimental program to explore the usage of viscoelastic damping in primary aircraft structure was initiated. As part of this study, the F/A-18 horizontal tail was selected for a primary structure damping treatment, Reference 1. The goal was to cut the stabilator response in half. The damping treatment consisted of a stiff graphite epoxy constraining layer adhesively bonded to the stabilator by 3M ISD-113 viscoelastic material. Modal loss factors as a function of temperature, as predicted by analysis and as measured by the experiment, are shown in Figure 1. As can be seen from the figure, the measured damping is considerably less than predicted from the analysis. The discrepancy between measured and predicted values was attributed to only having a 60 percent bond between the stabilator and patch. The difficulties of applying a stiff sheet to a sculpted surface produced poor bonding, thus limiting the effectiveness of the treatment.

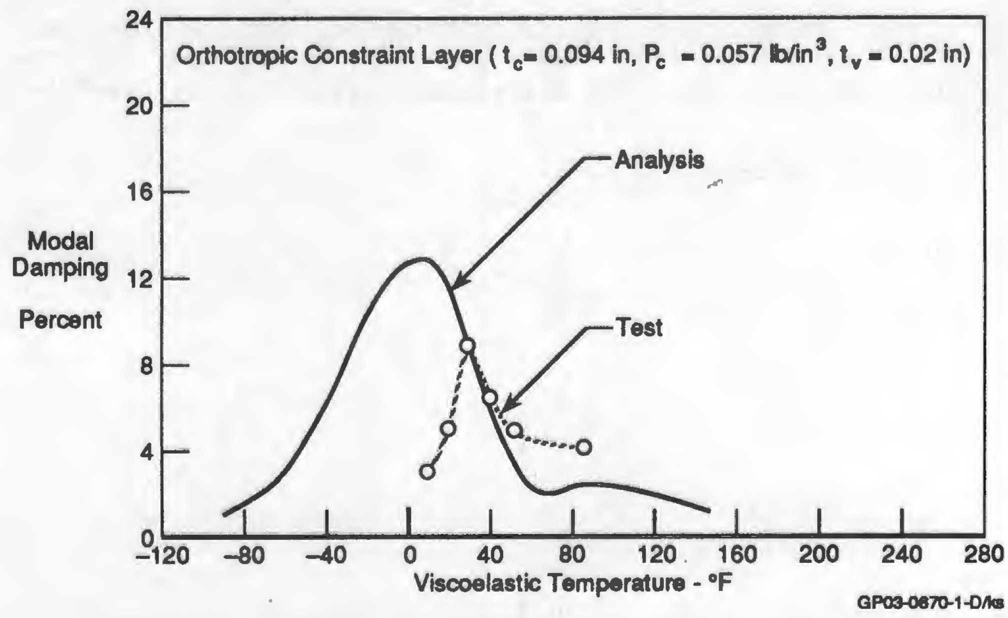


Figure 1. F/A-18 Horizontal Tail Constrained Layer Damping Treatment
Second Bending Mode

During 1988, as part of the above research program, several integral damping treatment concepts for the F/A-18 vertical tail were analyzed, Reference 2. The scope of the study was expanded to include not only the constrained-layer damping but also damped-link and tuned-mass damper concepts. In general, the modal strain energy (MSE), other than that concentrated at the root support was evenly distributed throughout the skin structure. This type of MSE distribution is inhibitive to layered damping treatments. The reason is that a constraining layer thickness that effectively extracts MSE from the first bending mode is unlikely to be successful on the first torsion mode. Also, the F/A-18's tail surface is not conducive to a constrained-layer damping treatment due to the unevenness of the composite skin. The conclusion from this study was that none of the constrained-layer damping treatments produce the desired levels of structural damping. The main reason cited was that global modes require a global treatment unless concentrations of MSE can be identified. The damped-link failed to produce the required levels of damping because there was not enough relative motion to add any significant damping. Damped-links are analogous to a shock-absorber and require that their end-points have large relative displacements. The tuned-mass-dampers (TMD) did offer some promise; however, the difficulty in practically applying this technology makes it the least favorable alternative. Some of the inherent problems in the construction of the TMD are creep and displacement control. For the F/A-18 vertical tail application, the most critical parameters are the control of modes over a wide frequency and temperature range. TMD designs are limited to one condition or one modal effect.

In a parallel effort to design a damping treatment for the F/A-18 vertical tail, Reference 3, a scaled test article was developed to quickly and economically demonstrate the viability of an add-on damping treatment concept. Using this test article, viscoelastic tuned beam damper concepts were demonstrated in controlling the primary modes. A response plot with and without the viscoelastic tuned beam damper is shown in Figure 2. There is a significant reduction in the response of the second mode with the damper installed. The tuned beam damper was found to be effective in controlling the important modes of the beam structure within the weight limitations.

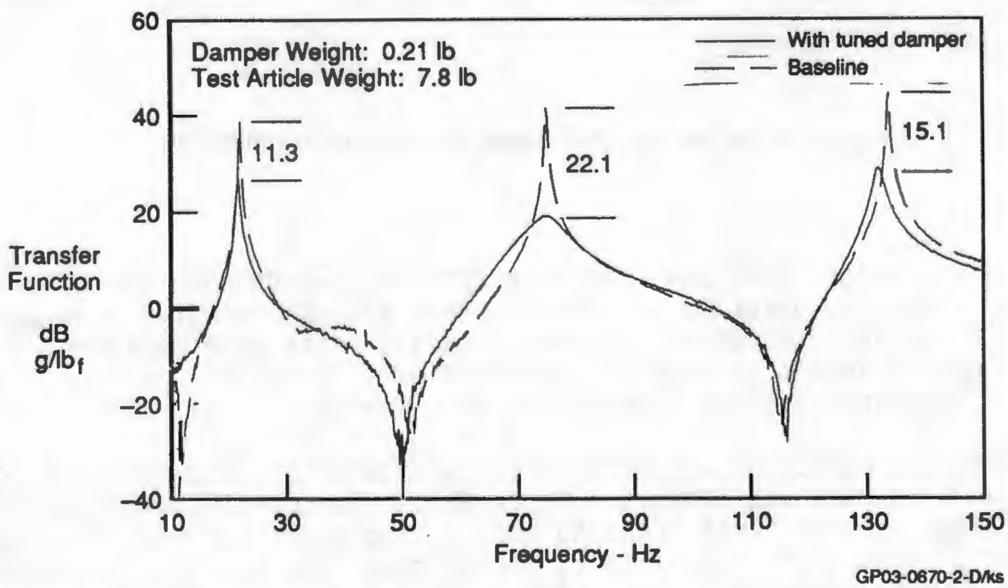
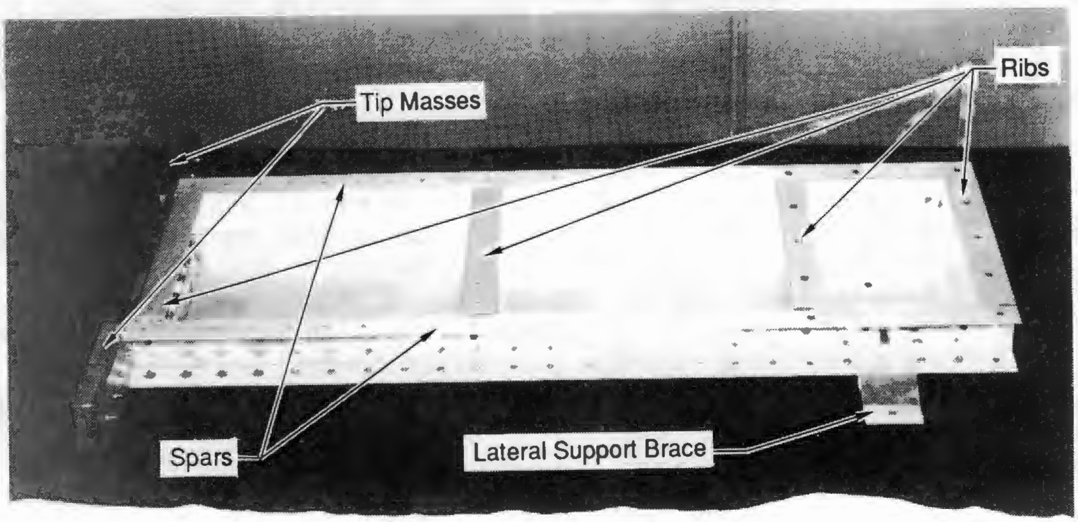


Figure 2. Response Data for Cantilever Beam With and Without TMD

SUBSCALE STRUCTURE STUDY

The previous studies had identified many damping concepts and provided data on the damping of empennage structural response. In continuing this research, our thrust was to investigate, in a more controlled manner, the previously identified damping concepts. We used a subscale structure that simulated the vibrational characteristics of the F/A-18 vertical tail. Two damping treatments were tested using this subscale structure which is a simple box beam shown in Figure 3. The box beam was of a single cell construction with all aluminum structure. The box beam is 48 inches long (of which 12 inches is a support root) by 18 inches wide and 3 inches deep. A tip mass was added to simulate the lowest frequencies of the F/A-18 vertical tail.

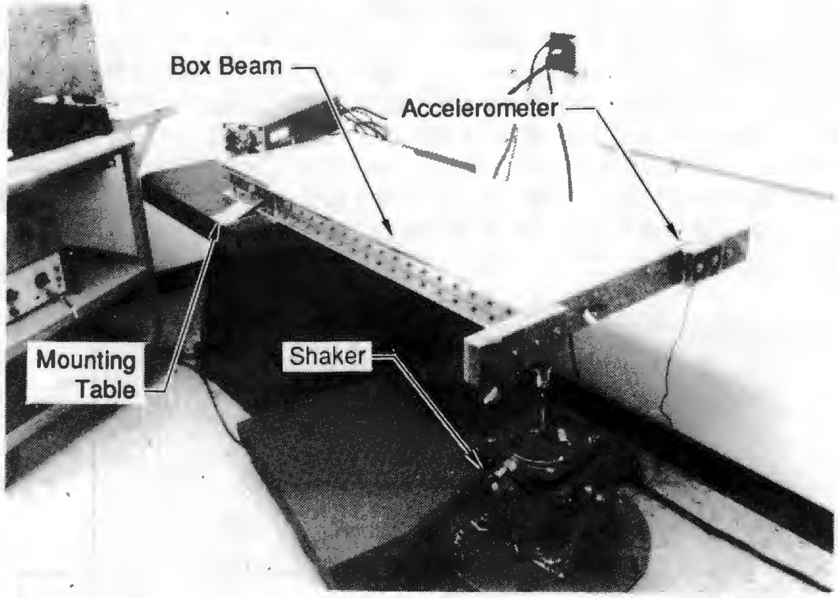


GP03-0670-3/kas

Figure 3. Single Cell Box Beam Structural Configuration

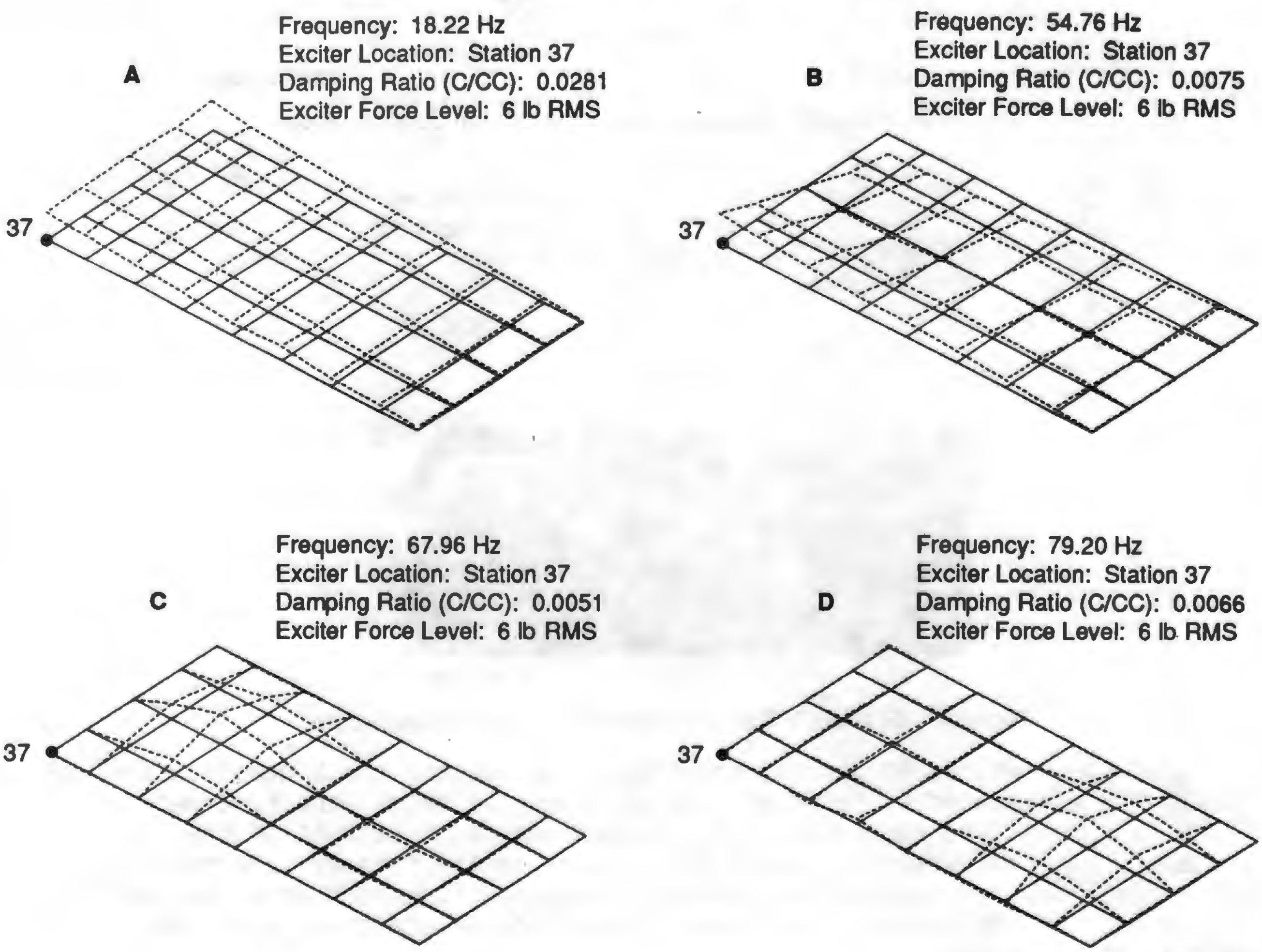
Using the single cell box beam test article, two damping concepts were evaluated: a partial exterior add-on treatment and an integrally damped interface treatment. Modal and dynamic response tests were performed to verify increased levels of damping in the primary (first bending and first torsion) and secondary (panel) modes of the box beam.

The box was tested in a cantilevered configuration, Figure 4. A complete baseline modal survey was conducted to establish frequencies, mode shapes, and damping values for the first bending and torsion modes and the first panel modes of the skins. The mode shapes, frequencies, and damping are shown in Figure 5. Forced vibration tests using random and sine excitation were conducted. During these tests transfer functions were measured at various locations on the untreated structure to provide a baseline from which the response of the damped structure could be compared.



GP03-0670-4/kas

Figure 4. Box Beam Vibration Test Setup



GP03-0670-5/kas

Figure 5. Mode Shape Plots of the Primary Modes of the Baseline Box Beam

The damping treatments tested consisted of: a stand-off damping treatment applied to the outer skin, and an adhesive or "interface" damping layer applied between the skin and spar-caps. The stand-off treatment consisted of an 0.080 inch thick syntactic foam layer adhesively applied to the skin with a 0.005 inch thick layer of 3M 468 and a double application of 0.005 inch thick 3M ISD-112 and 0.010 inch thick soft aluminum constraining layer, Figure 6. The treatment was applied in 12 by 15 inch sized patches to all four exposed panel areas (top and bottom) on the box beam, Figure 7.

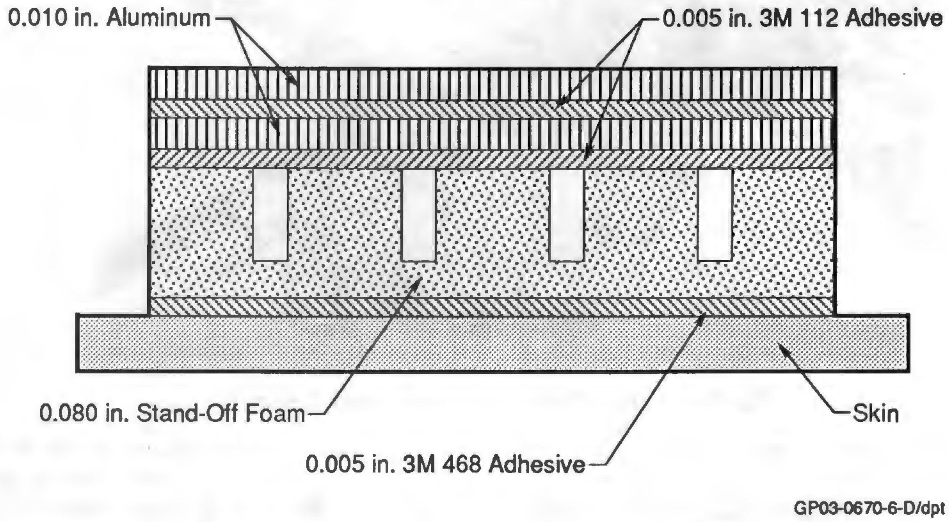
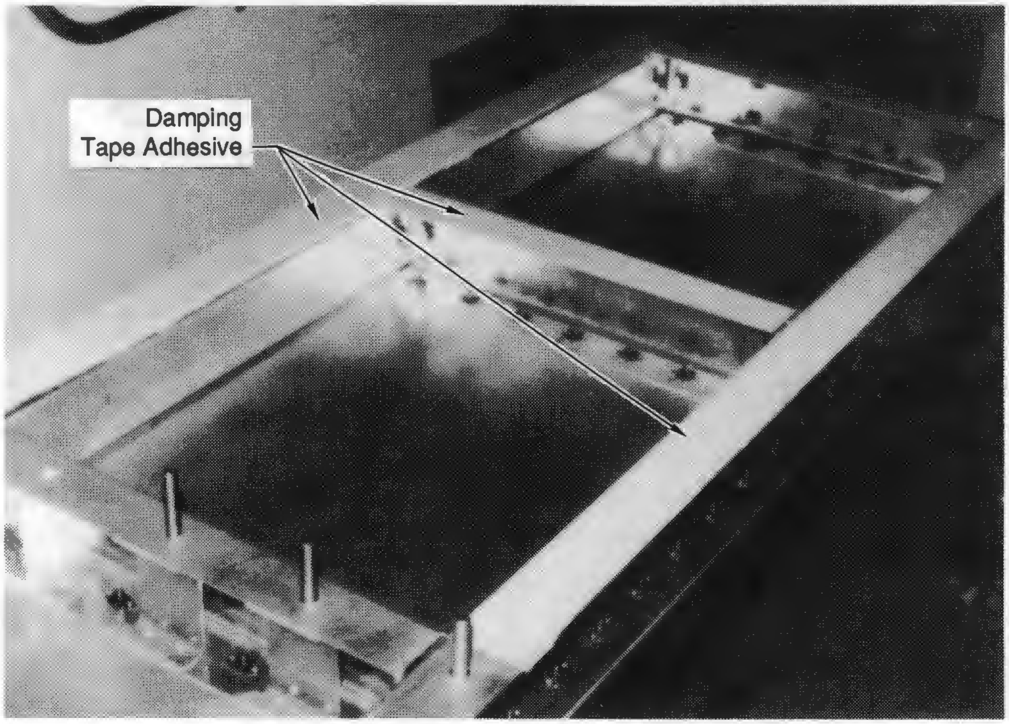


Figure 6. Stand-Off Damping Treatment Design Configuration

Figure 7. Stand-Off Damping Treatment on the Box Beam

For the "interface" damping treatment, a 0.02 inch thick damping layer of 3M ISD-113 was bonded to the spar caps and then the skins were fastened in place with adhesive, Figure 8. The previous damping treatment of the stand-off material was not removed (the effect on the primary mode response was minimal and the accelerance frequency response functions were less noisy.) The effects of the two treatments were assumed to be additive, with the initial effect known.



Box Beam With Top Skin Removed to Show Adhesive

GP03-0670-8/kas

Figure 8. Spar/Skin Joint Damping Treatment on the Box Beam

The vibration test results are summarized in Figure 9. These results indicate that the exterior damping treatment is effective in reducing the response of the primary structure, but it is even more effective in reducing response in the local panel modes of vibration, Figure 10. An increase in damping of 29 percent in the first bending mode was observed. This is a significant increase in damping, considering that the baseline first bending modal loss factor was 0.064. A 94 percent reduction and a 77 percent reduction from baseline response in the first and second panel modes was measured.

Mode Shape	Baseline		Stand-Off		Interface	
	Freq (Hz)	Modal Loss Factor	Freq (Hz)	Modal Loss Factor	Freq (Hz)	Modal Loss Factor
First Bending Primary	18.22	0.056	17.87	0.092	19.30	0.103
First Torsion Primary	54.76	0.015	54.77	0.026	44.09	0.082
First Panel (Front Panel)	67.69	0.010	78.89	0.163	Not Determined	
Second Panel (Back Panel)	79.20	0.013	89.03	0.052	Not Determined	

GP03-0670-9-D/dpt

Figure 9. Box Beam Damping Test Results Summary

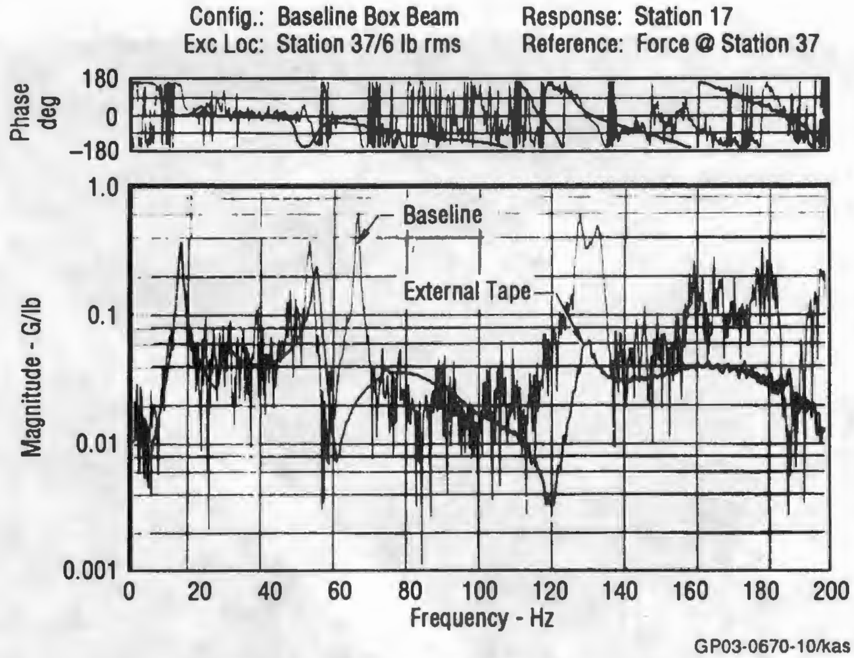


Figure 10. Overlay of the Box Beam Baseline and External Tape Random Response Transfer Function

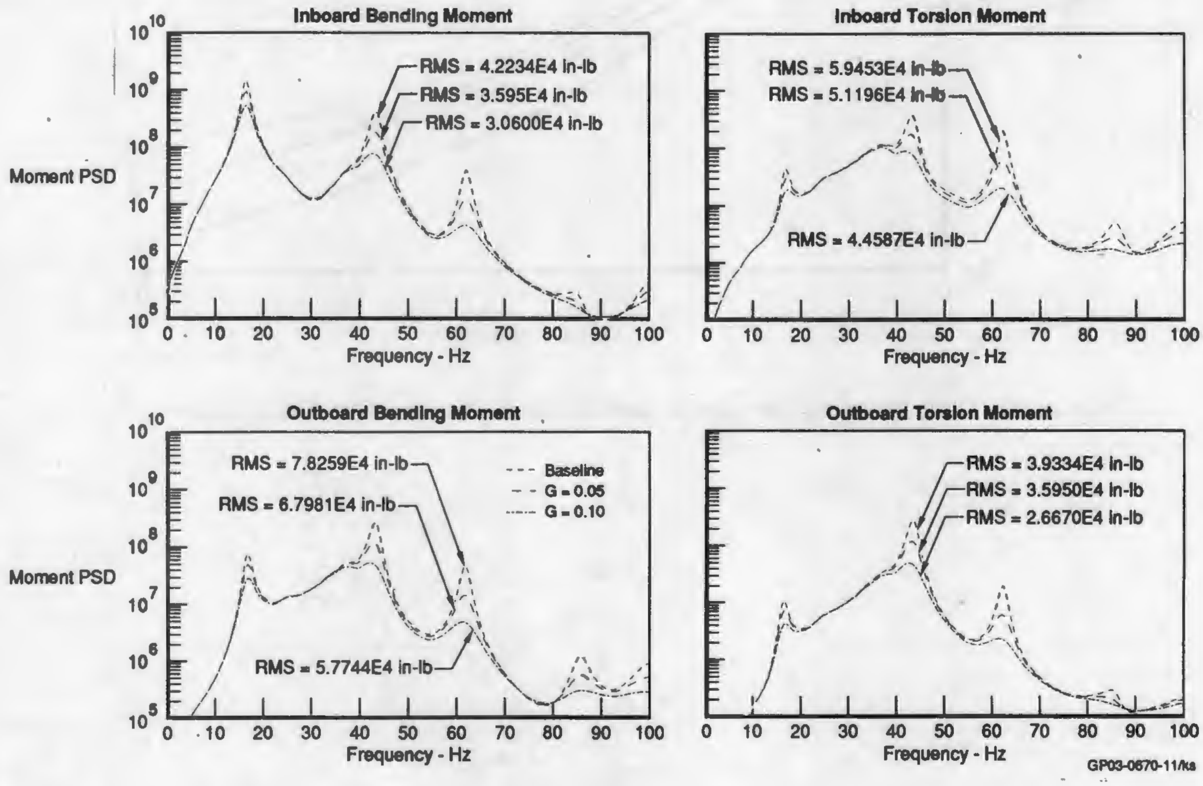
The interface damping concept increased the damping in the first bending mode and caused an additional 13 percent reduction in response or a 39 percent reduction overall from the baseline response. The damping treatment was more effective in damping the first torsion mode. The combined effects of the standoff and the interface damping treatments caused an 82 percent reduction from the baseline torsion mode response. A decrease in stiffness of 19.5 percent from the baseline torsion mode frequency was observed which was observed by the decrease in the modal frequency from 54.77 Hz to 44.09 Hz.

F/A-18 VERTICAL TAIL APPLICATION

For the major case of interest here, the F/A-18 vertical tail is subjected to severe buffeting forces at angles of attack above 20 degrees. These buffeting forces cause very high dynamic response in the primary modes of the tail; i.e., zero to peak amplitudes in excess of 500g have been observed in flight. If the objective is to cut the buffet response in half, then the level of structural damping in the vertical tail needs to be significantly increased.

In order to investigate the effectiveness of damping to control the vertical tail response during buffeting flow conditions, buffet response calculations were made using simulated levels of structural damping. The simulated damping levels are assumed to come from the inclusion of the damping treatment to the structure. Unsteady pressures during buffet were measured during the wind tunnel program described in Reference 4. These pressures were scaled to aircraft size and were used as the forcing function in the response calculation. The scaling method and calculation approach are also described in Reference 4. The results of the calculations are shown in Figure 11. The

F/A-18 Vertical Tail Buffet Response Predictions With/Without Damping



data are presented in the form of bending and torsion moment PSD's for a 14 percent and 66 percent span station. For the condition of Angle of Attack of 32 degrees, Dynamic Pressure of 347 psf and Mach Number of 0.6, the first bending mode dominates inboard bending moment responses, and the second bending mode dominates the outboard bending moment responses. The overall RMS response reductions, (Figure 12), suggest that 50 percent is the maximum that can be obtained from a damping increase alone.

As previously mentioned, analytical studies for primary structure damping treatments for the F/A-18 vertical tail, Reference 2, had concluded that constrained-layer damping could not be effectively included because the structure itself was well designed with no areas of major strain energy concentrations. For the F/A-18 vertical tail application two treatments were analyzed. These consisted of a "hybrid" design of the solid spacer treatments identified in Reference 2 and the interface concept which was tested using the subscale structure. Analyses of these two treatments required extensive modification of the existing F/A-18 dynamic finite element model, shown in Figure 13, in order to examine the damping treatments. The damping concepts were each individually modeled and extensively analyzed using the MSE Method, Reference 5, for various damping configurations.

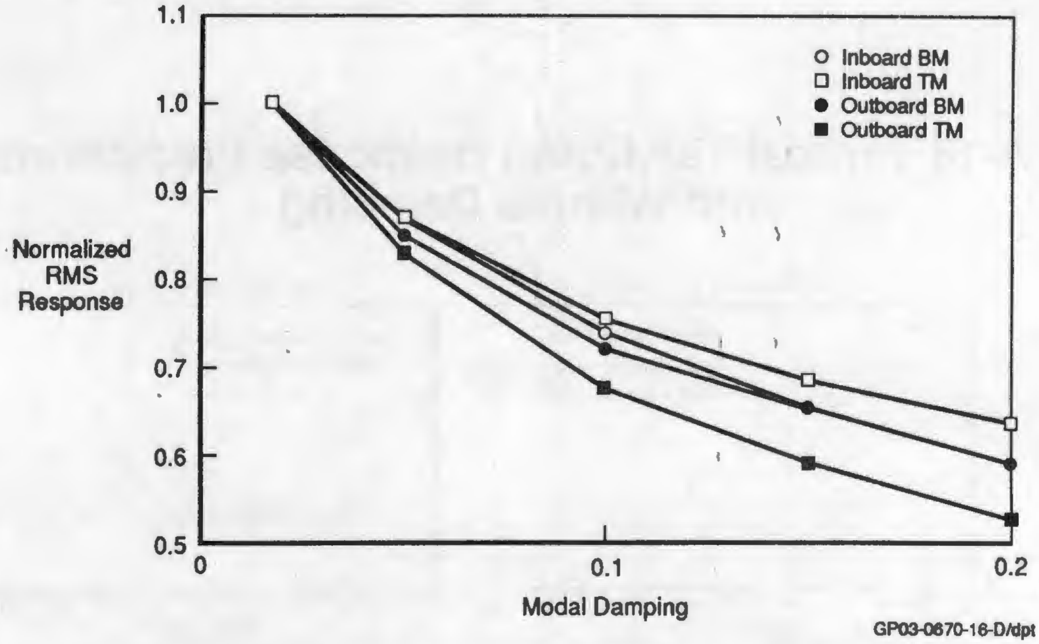


Figure 12. RMS Damping Response Normalized to the Baseline RMS Response

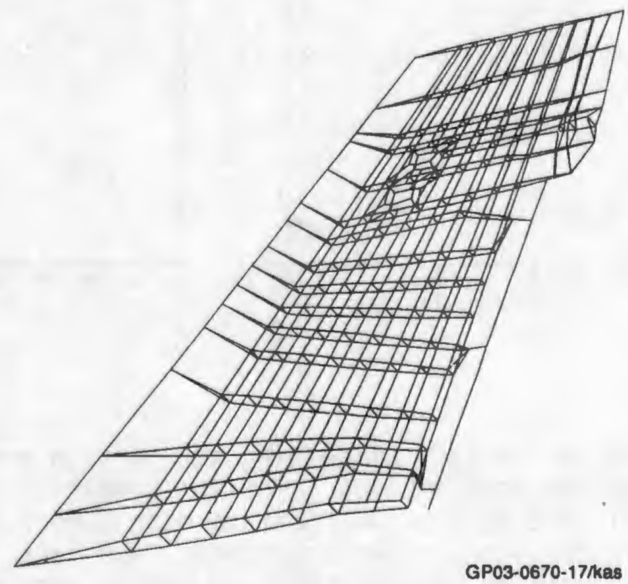


Figure 13. MSC/NASTRAN Dynamics Model of the F/A-18 Vertical Tail

INTERFACE DAMPING CONCEPT

The interface damping treatment required that a layer of shear deformable elements be included between the spar cap and skin. Hence, an extra set of nodes was placed underneath the existing spar câp nodes. This model reflects the detail in the primary load path (skin through fastener to spar) needed to analyze the problem sufficiently. In this concept, Figure 14, a portion of the beam shear load is transferred through the VEM located between the moldline skin and substructure. The remaining load is carried through by the fasteners. In the study, fasteners were assumed to be either widely spaced or were excluded from the model. Both variations of the interface concept will be discussed.

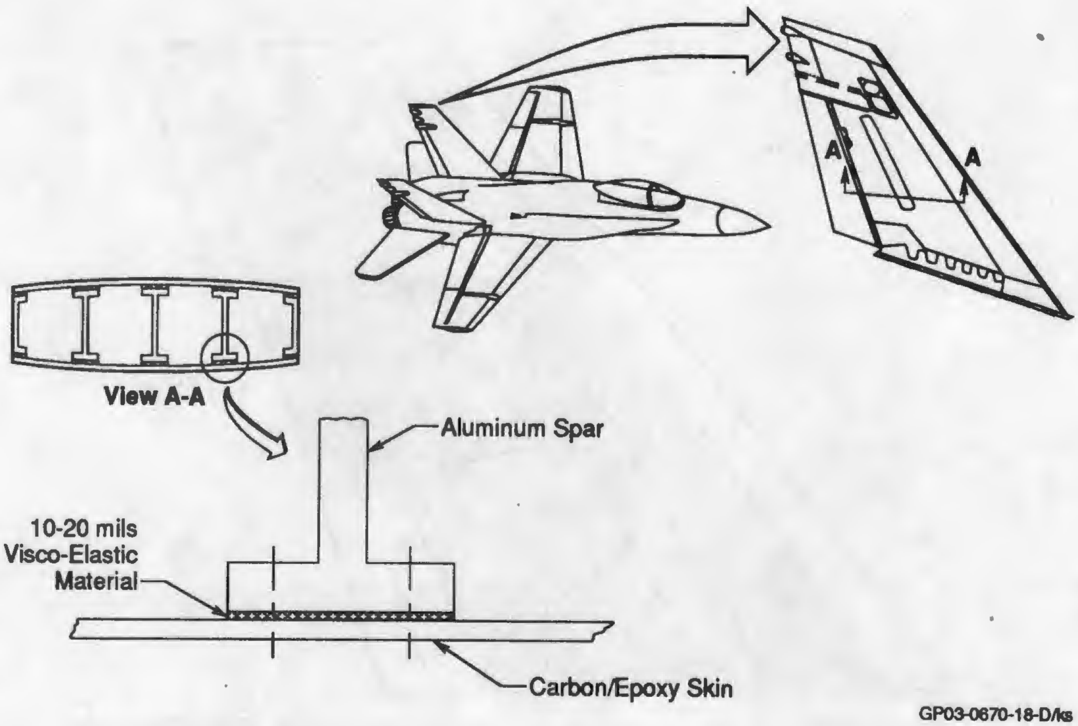


Figure 14. A Structurally Integrated Passive Damping Concept

The interface layer was included into the detailed F/A-18 vertical tail as a shear panel with the transverse degree-of-freedom (DOF) rigidly constrained between the spar cap and skin element DOF. When the fastener effects were included, they were modeled with rigid bar-type elements in all DOF. The treated areas of the structure are shown in Figure 15. With fasteners, this treatment only produced 1.5 percent and 2.0 percent MSE in the first and second bending modes and nearly 8 percent MSE in the first torsion mode of the vertical tail. Without the fasteners, 3 percent and 4 percent MSE were produced from the first bending and second bending modes, respectively. When no fasteners were assumed to be in place, the modal strain energy produced in the first torsion mode increased to a peak MSE of 12 percent (Figure 16), but at a subsequent loss in stiffness of the structure noticed as a decrease in frequency, Figure 17.

SOLID SPACER DAMPING CONCEPT

Previously, this concept had been analyzed in two solid spacer arrangements, Figure 18, and neither concept showed any significant benefit for further evaluation. However, it was thought that a combination of the two concepts, Figure 19, would show the necessary levels in damping that would make this concept a candidate for future design application. Thus, the damping treatment was evaluated.

The combination of the three damping layers allows for shear deformation to take place in all three layers. If only the center-plane layer existed with the two rigid spacers rigidly attached to the skins, then no relative shearing could take place in that layer. This is because the vertical tail is

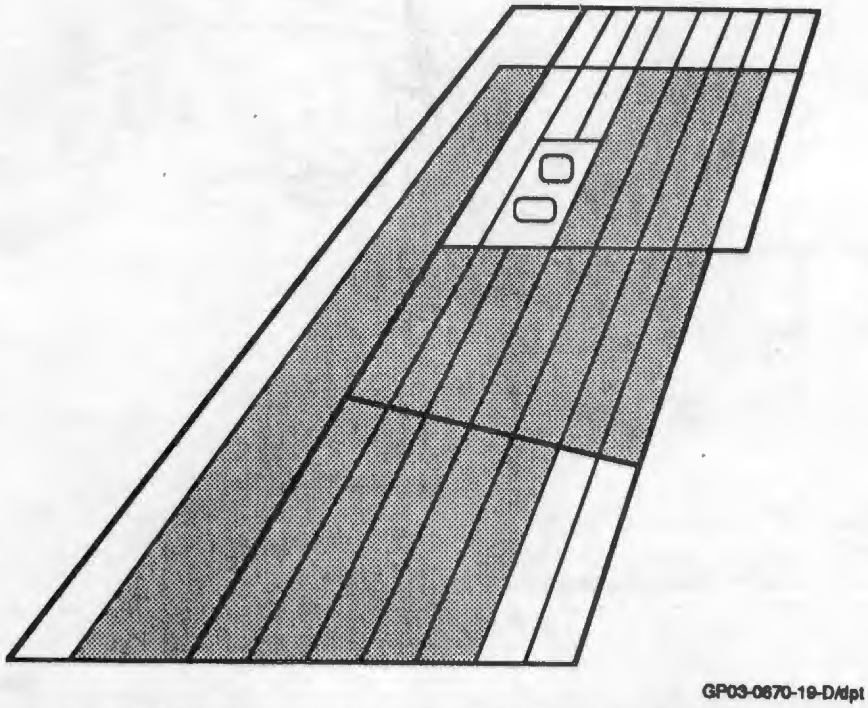


Figure 15. Solid Spacer Damping Treatment Coverage on the Vertical Tail

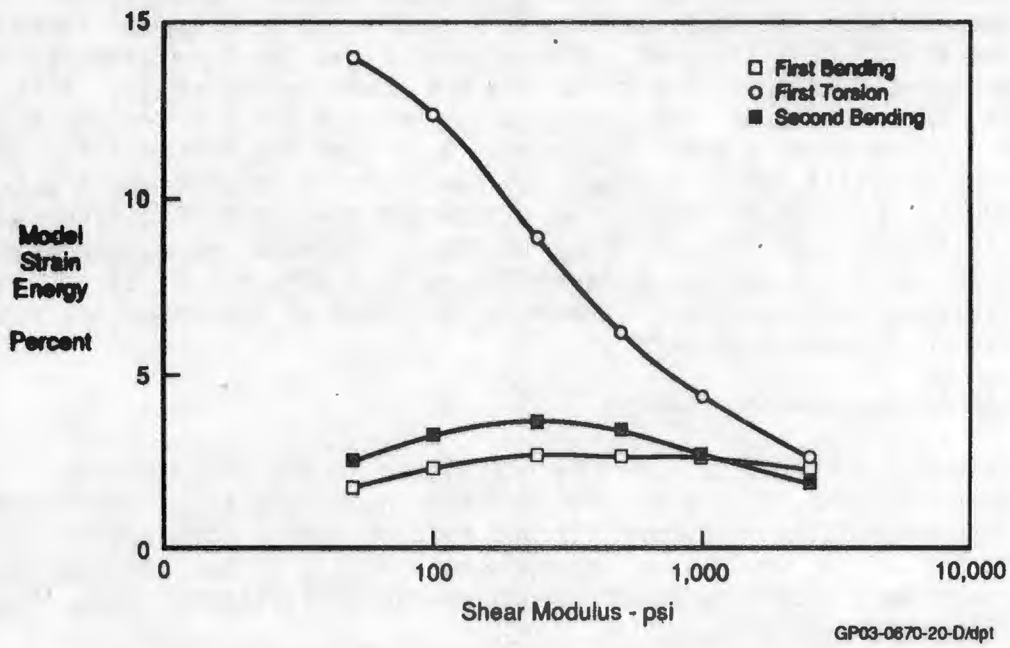


Figure 16. Interface Damping Treatment With No Fasteners

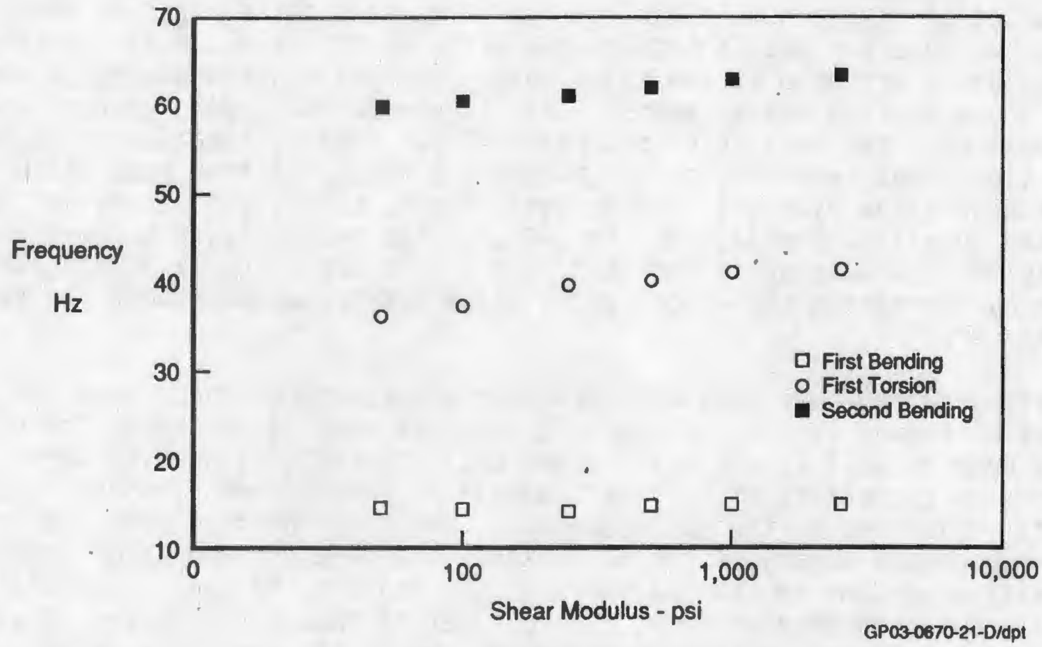


Figure 17. Interface Damping Treatment With No Fasteners Modal Frequency Dependence on Shear Modulus

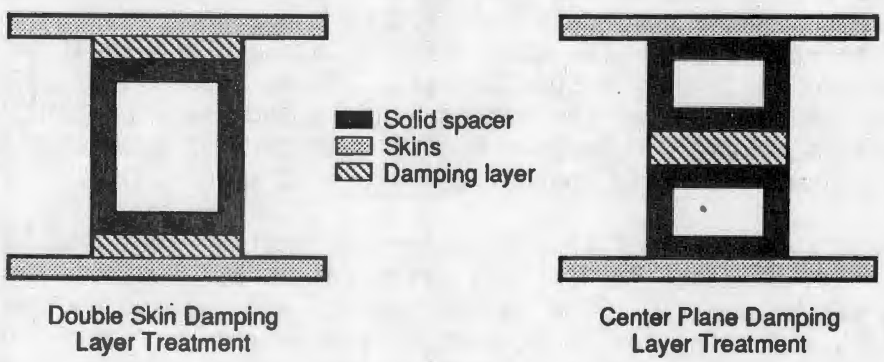


Figure 18. Solid Spacer Damping Treatment Concepts
Reference 2

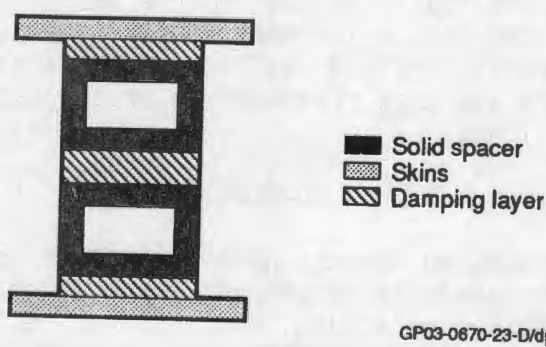


Figure 19. Hybrid Solid Spacer Damping Concepts

constructed of skin and multiple spar which is thus quite rigid in shear with no relative shearing motion between the skins of the tail. When two skin damping layers are held by one solid spacer, only a minimal amount of damping can be produced from these layers. This is similar to trying to damp a very thick beam with two very thin constrained-layer damping treatments applied to either side. But, when the solid spacers are decoupled from both skins and at their center plane by a soft viscoelastic layer, dramatic shear relief is exhibited in all of the layers. In fact, as the center layer becomes weaker and thicker, the damping in the skin layers will begin to maximize shear effects by increasing the amount of relative displacement between the skin and solid spacer.

This treatment was applied continuously between the spars over the shaded portions of Figure 15. It includes an integral damping treatment for the leading edge as well as the main torque box. Sensitivity studies were performed on the effect of the shear moduli in the different layers including the shear stiffness of the solid spacers. The first bending mode has very little dependence on any of the parameters considered. Damping of this mode is heavily dependent on the stiffness at the root of the tail. For the first torsion and second bending modes, the parameters that optimize the strain energy in these modes is opposing. For instance, the torsion mode yields 9.2 percent modal strain energy at a G_{skin}/G_{core} = 500 psi/1000 psi, where G_{skin} is the shear modulus of the skin side damping layer and G_{core} is the shear modulus for the damping layer at the center plane, and the second bending mode maximizes at 7.7 percent at G_{skin}/G_{core} = 20 psi/100 psi. Finally, more strain energy can be produced in the bending modes when the solid spacer is assumed to be very rigid. The above studies assumed the shear modulus of the solid spacer to be G_{spacer} = 500,000 psi. As an upper limit, 22.0 percent MSE was produced in the VEM for the second bending mode, 4.7 percent MSE for the first bending mode and 6.0 percent MSE for the torsion mode when G_{skin}/G_{core} = 20 psi/100 psi with a rigid spacer, $G_{spacer} = 50 \times 10^6$ psi.

The negative aspect of this treatment is that it adds nearly 40 pounds per tail. This does not reflect any optimization by placement or geometry to reduce the weight penalty. The weight penalty was imposed by the use of the spacers which accounted for 85 percent of the weight increase. These spacers were modeled with solid finite elements which were assumed to represent hollow tubes made of composite materials and very stiff in shear. The overall weight of the damping treatment could be reduced by removing the treatment from certain areas of the structure that had little effect on the modes of interest. For instance in the first bending mode, the leading edge and lower to mid tail regions contribute the most to the damping increase. In the second bending mode, the mid region contributes the most to the damping increase. In the present vertical tail structural arrangement, it would not be practical to try to use this treatment in areas obstructed by wire bundles, hydraulics, and fuel lines.

CONCLUSIONS

Viable integral damping concept have been shown to merit further full scale evaluation. The analysis of the interface damping concept shows that it can be tailored for specific damping, strength and stiffness requirements by altering the structure fastener spacing. Evidence from the study shows that a reduced number of fasteners is required for the interface concept because aircraft standards for fastener spacing along a spar results in an overly rigid structure which inhibits any shear relief through the VEM. The analysis of the solid spacer concept proved the proof-of-concept and showed that it would be a candidate for future aircraft. However, a damping concept of this sort will need to be considered in the initial design phase in order to make the concept more weight efficient.

ACKNOWLEDGMENTS

The research studies reported in this paper were conducted under the McDonnell Aircraft Company's Independent Research and Development Program. Special assistance from the McDonnell Aircraft Structural Dynamics Lab Group was given in the set-up and conduct of lab tests. Also, the damping materials were graciously supplied by the 3M Company Industrial Specialities Division and Wright Laboratory "Damping Clearing Center."

REFERENCES

[1] Yurkovich R.N., "Application of Constrained Layer Damping to the F/A-18 Horizontal Tail," Damping 89, West Palm Beach, FL., February 1989.

[2] Gibson, W.C., Maly J.R., and Austin E.M., "Conceptual Design of the Damping Treatments for the F/A-18 Vertical Tail," CSA Report No. 88-11-02, November 1988.

[3] Parin, M. and Nashif, A.D., "Development of Vibration Control Measures for the F/A-18 Vertical Tail," Anatrol Report No. 88024, February 15, 1989.

[4] Ferman, M.A., and Zimmerman, N.H., "Prediction of Tail Buffet Loads for Design Application," NADC-880434-60, July 1987.

[5] Johnson, C.D., Keinholz D.A., and Rogers, L.C., "The Modal Strain Energy Finite Element Analysis Method and its Application to Damped Laminated Beams," 51st Shock and Vibration Symposium, San Diego, CA, October 21-23, 1980.