Active Damping and Vibration Control for Aircraft Fin and Appendage Structures

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Numerous vibration control techniques, employing both passive and active methods, have been developed and tested over the last several decades. Some of these techniques are in widespread use, while others have rarely or never left the laboratory. This paper considers the value that vibration damping and control of aircraft fins and appendage structures can have in reducing loads and subsequent fatigue and possible failure. These structures often are subject to high loads resulting from wakes of upstream external stores. The vibration control methods were considered as part of a larger study focused on active flow control. The options for passive or active vibration control on a class of fin-type structures are reviewed, and one approach – active and passive damping using piezoelectric materials – is covered in greater detail. Piezoelectric transducer sizing for expected pressure loading and modeling of piezoelectric-based active damping control systems are discussed. Motivation for another possible techniques coupling active flow and vibration control is presented using arguments from adaptive filtering and feedforward control. Results are presented for bench tests with simulated disturbances, for low speed wind tunnel tests, and for high speed wind tunnel tests.

I. Introduction

During the lifetime of any given aircraft program, the addition of numerous external pods and stores can be required to meet the needs of unanticipated mission profiles. One means of mitigating the flow-induced loads on structures is active flow control (AFC). The feasibility of AFC has been demonstrated in previous studies, including that described in Ref. 1. An ongoing effort is aimed at determining the complementary or even collaborative application of AFC and active structural control (ASC) to reduce structural loads on fins and other appendage structures. A separate paper describes the overall program and the system-level results from wind tunnel testing, with more information on the AFC portion. This paper concentrates on the ASC and attempts to consolidate design methods and assess general applicability, while also reporting on laboratory and wind tunnel tests that demonstrated vibration and load reduction.

The paper begins with a review of vibration control methods and considers the benefits and issues associated with a number of passive and active methods including constrained layer damping, tuned mass dampers, and approaches using smart materials. The greater flexibility afforded by active systems is discussed in the context of possible information transfer, by wire, from an AFC system to an ASC system. Design and sizing of transducers for two particular fin models is reviewed using available information for expected flow disturbances. A laboratory setup and measurements showing vibration reduction in the presence of simulated disturbances are described. Results from low speed wind tunnel testing and special considerations for high speed testing are reviewed. The laboratory tests supporting the planned high speed wind tunnel testing also include passive vibration control using piezoelectric materials. The ASC for vibration control when velocities approach Mach 1 considers gain limiting,

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and aeroelastic effects, principally the stiffening of the first bending mode with increasing Mach number. The control system for high speed wind tunnel testing also addresses the time-limited (short duration) nature of each test.

II. Load Alleviation by Structural Vibration Control

Aircraft structures can tolerate some vibration, but excessive vibration, actually excessive cyclic stress, can degrade and damage components, potentially resulting in their failure. Vibration typically either is a direct result of forcing inputs, or is a result of broadband forcing inputs driving structural resonances. In the former case, the vibration can be reduced by methods that include stiffening of the structure in question. In the latter case, vibration damping can reduce resonant response.

Vibration damping has been installed in numerous aircraft to reduce fatigue stress. In general, damping is implemented as viscoelastic material treatments using constraining layers, or in some cases as tuned mass dampers. Occasionally more exotic means are used in high temperature environments including engine structures. Active vibration damping was investigated in detail in the 1980s, and has since been tested in several ground demonstration systems. While the performance of these technologies showed promise, their extension to full-scale structures such as fighter aircraft vertical tails has challenges.

Passive damping of bending and torsion modes for cantilevered, or approximately cantilevered, structures is often best implemented using a tuned mass damper (TMD). Sometimes called a vibration absorber, a TMD is a resonant system that includes a moderate amount of damping. It is tuned near in frequency to the natural frequency of the troublesome mode and installed in or on the structure at a location of relative large displacement. The TMD dynamically amplifies the structure motion producing larger relative motion and drawing energy from the structural mode and dissipates it locally as heat.

The basic difficulty in application of TMDs to alleviate aircraft appendage and fin vibration is the limited volume available in the locality where the TMD can be most effective, the tip or outboard part of the structure. That location is unavailable because the structure, sensor, or other component is designed so that it is thin or comes to a point. A compromise is to locate the TMD as far outboard as possible. Even the TMD of Figure 1, designed to fit within available space in an F-15 or F/A-18 vertical tail had limitations. The damper height limit meant that the highly effective damper (Figure 1, right) had greatly reduced effectiveness at the highest motion amplitudes, where there was inadequate rattle space.

![Figure 1: Tuned mass damper (TMD) designed to damp the first bending mode of A-15 or F/A-18 vertical tail was effective but limited by required rattle space](image)

Piezoelectric transducers provide a means of introducing significant damping without requiring space near the tip of the structure. It is possible to mimic the effect of a TMD using a passive transducer shunting network, or more generally, active damping can be implemented with more arbitrary control approaches. Passive damping via shunting was implemented on a fin as part of this study, and results are noted at the end of the paper. But the main focus here was on active damping using piezoelectric wafer or plate transducers located near the base of fin structures.

What level of damping is to be expected using piezoelectric materials, sensors and a control system? That depends on two main factors. First is the level of force that can be generated by the actuation system, in the frequency range of interest, including the piezoelectric transducer limits and amplifier current capabilities. Importantly, the amount of force must be related to the amount of disturbing force on the system. Second is the
control algorithm, and in particular the coupling to modes other than the one(s) of interest, the order of the compensator, and the amount of robustness built into the controller.

Here, the goal is 5-10% damping in the fundamental structural mode of the fin, implemented with a system that is compact, structurally sound, and able to withstand exposure to high speed flow.

III. Actuator Sizing Based on Expected Aero Loading

Sizing of piezoelectric actuators for effective vibration control depends on three things: the input disturbance levels, the structure, and the required performance levels. Of these three, the first was not known well in this case, the second was known, and the third was stated as a goal to reduce vibration (and therefore stress) by at least a factor of 2.

The expected wind tunnel levels of pressure loading were not known exactly. However, an analysis was developed to predict fin response. A finite element model of the fin was created, and pressure loading computed from an aerodynamic analysis was applied over 6x7 grid on the fin surface.

Figure 2: Predicted tip displacement response to distributed pressure loads on a fin structure

Figure 2 shows the predicted response at 100 Hz. Predicted zero-to-peak displacements exceed 0.35 inches, at least twice what was eventually tested in the bench tests, and higher than the actual levels eventually measured in the wind tunnel. However, the model predictions provided a conservative basis for sizing actuators. Subsequently, with the wind tunnel data available, and in the future, with flight test data available, some conservatism can be removed in the actuator sizing. In the most general terms, the total actuator volume is related to the total force needed to produce significant damping, and the paper will illustrate this to be true for the particular test structure.
IV. Laboratory/Bench Testing of Fin Under Simulated Aero Loads

Figure 3 shows the most basic representation of the tests that were conducted. Lacking a wind tunnel, an electromagnetic actuator was used to introduce motion near the tip. The presence of this actuator also introduced some mass and lowered the frequencies of the various modes.

The bench tests were successful in achieving damping and response reduction in the first bending mode near 100 Hz (Figure 4). The controllers implemented on the bench top tests utilized both classical and modern design methods with the feedback sensor being a strain gage nearly collocated with the piezoelectric actuators. The controller illustrated utilized a classical approach. The strain response of the first mode was reduced, while the second and third mode responses were relatively untouched. There was roughly a 7.5 dB reduction in the strain signal at the fundamental.

![Figure 4: Effect of control on the transfer function from tip force to strain in bench tests](image)

V. Wind Tunnel Testing

Wind tunnel testing was completed in the summer of 2004. Tests were conducted in an existing low speed wind tunnel. Figure 5 shows a photograph of the fin installed in the low speed wind tunnel with the flow from right to left. The fin is on the left, downstream from the pod. The AFC actuator (Ref. 2) is mounted at the blunt downstream end of the pod. The two actuator patches are clearly visible on one side of the fin. In between the two actuators is a strain gage at the root of the fin and used for feedback control. Also attached to the fin are several low profile accelerometers and pressure sensors, and a tip mass used to tune the frequency of the fundamental mode of the fin. The piezoelectric actuators and strain sensors are configured to be actuate/sense the spanwise bending of the fin.

![Figure 5: Photograph of low speed wind tunnel test, with flow right to left](image)
Two types of control algorithms were implemented in the wind tunnel. Feedback control for active damping, as implemented in the bench tests, was the primary algorithm. Feedforward control was later added to tie the ASC to the AFC. Feedforward control can be implemented with or without the feedback control operating. A diagram of the combined control system is shown in Figure 6.

![Diagram](image)

Figure 6: Feedforward/feedback implementation integrating active flow and vibration control; for most of the tests, the feedforward portion was not used

The active vibration control was effective at reducing response to the basic flow and excitation resulting from the pod and other structures upstream in the wind tunnel (Figure 7). This response is zoomed in on the fin fundamental structural resonance at 100 Hz. During the wind tunnel tests, the tunnel Mach number was approximately 0.1.

![Figure 7](image)

Figure 7: Autospectrum of strain gage signal on fin in wind tunnel without jet excitation (zoomed in)

Figure 8 shows the open and closed-loop autospectra of the strain gage on the fin in the wind tunnel with the tunnel on and a jet control excitation at 100 Hz. Note that the active damping control behaves almost exactly as before. The difference with the jet excitation is seen in the small peak at 100 Hz. This peak is due to the jet excitation and the controller doesn’t do much about it. The fact that the fin response is attenuated in that region is due to the active controller acting on the broadband strain gage signal and less on the narrowband 100 Hz spike. Table 1 shows the change in RMS of the strain gage autospectrum from 0 to 500 Hz. The table shows that a 6.7 dB reduction was achieved in the strain gage signal with both the control turned on and the jet excitation set to 100 Hz.
Figure 8: Measured autospectrum of strain gage on fin in wind tunnel with 100 Hz jet excitation and a zoomed view from 80-120 Hz

Table 1: Table of strain gage RMS for various control cases

<table>
<thead>
<tr>
<th>Case</th>
<th>Output RMS (V from 0 to 500 Hz)</th>
<th>Reduction</th>
</tr>
</thead>
<tbody>
<tr>
<td>Jet OFF, Control OFF</td>
<td>0.4197</td>
<td></td>
</tr>
<tr>
<td>Jet OFF, Control ON</td>
<td>0.2404</td>
<td>4.8 dB</td>
</tr>
<tr>
<td>Jet at 100 Hz, Control OFF</td>
<td>0.4966</td>
<td></td>
</tr>
<tr>
<td>Jet at 100 Hz, Control ON</td>
<td>0.2287</td>
<td>6.7 dB</td>
</tr>
<tr>
<td>Jet at 200 Hz, Control OFF</td>
<td>0.6084</td>
<td></td>
</tr>
<tr>
<td>Jet at 200 Hz, Control ON</td>
<td>0.3076</td>
<td>5.9 dB</td>
</tr>
</tbody>
</table>

Additional tests were done using feedforward control from the AFC to the ASC system. The results are reported in the companion paper.

Figure 9: Simulated performance for a 19-state modern controller based on wind tunnel data
After completion of the low speed wind tunnel tests, additional control system design using the measured data predicted that with more sophisticated feedback control utilizing modern state space control methods, even greater vibration and stress reduction could be achieved on the first and second bending modes (Figure 9).

VI. Implications for Scaling and Higher Speed Tests

The end goal of the research effort is to know whether the ASC and AFC can be implemented at flight speeds for flight structures. The structures, such as the F16 ventral fin are larger (~25 times the area), and speeds are obviously higher than Mach 0.1. Following the successful tests in the low speed tunnel, attention turned to preparation for tests in a high speed tunnel. The fin structure design was influenced by the desire to incorporate ASC much higher speed, compressible flows. With this in mind, the following items were part of the fin design:

- The fin was scaled to have a first span-wise bending mode frequency in the range 500-550 Hz. This frequency was compatible with AFC synthetic jet capabilities and the physical space available in the wind tunnel.
- Trailing and leading edge shaping was minimal, allowing a relatively large flat surface for the integration of higher numbers of piezoelectric actuators.
- The overall area of the fin was decreased and the thickness increased to allow modest strain energies at resonance allowing for control authority with reasonable numbers of actuators.
- The mechanical boundary condition at the base presented a well-defined transition between a stiff structure and the fin.

![Figure 10: Comparison of high and low speed fin actuator layouts](image)

With the higher speeds, it was expected that larger vibrations would be excited by the flow. In order to control these vibrations with the ASC, the actuator authority had to correspondingly increase. Using piezoelectric actuators, two means can be used to increase the actuator authority: increasing the thickness of the actuator and increasing the number of actuators attached to the fin. Both means were utilized. For the high speed fin, both the actuator area and actuator thicknesses were doubled. Table 2 provides a comparison of the actuators for the low and high speed fins. Figure 10 shows a side-by-side comparison of the high and low speed fins and actuator-sensor layouts. Note that the layouts are identical on both sides of the fin and that the actuators and sensors are configured to be sensitive to spanwise bending.
Table 2: Comparing Low and High Speed Actuators

<table>
<thead>
<tr>
<th></th>
<th>Low Speed Fin</th>
<th>High Speed Fin</th>
</tr>
</thead>
<tbody>
<tr>
<td>Actuators per side</td>
<td>2</td>
<td>8 (2 layers)</td>
</tr>
<tr>
<td>Surface Area Covered</td>
<td>15.2%</td>
<td>52.4%</td>
</tr>
</tbody>
</table>

Once the high speed fin had been designed and built, it was tested in a series of bench top and wind tunnel tests, similar to those for the low speed fin. Because of time constraints, most of the ASC testing was performed in the compressible flow wind tunnel at speeds up to nearly Mach 1.0. Figure 11 shows photographs of the fin Lockheed’s Compressible Flow Tunnel in Smyrna, GA. The left hand photo shows a close up of the fin with the nylon tip mass attached. Clearly visible are the structural actuators. Also attached to the fin are several low profile pressure sensors and accelerometers. The red material is wax used to protect the transducer wires from the effects of the flow by streamlining the profile of the fin. The right hand photo shows the fin down stream from the pod. This set up is essentially the same as that tested in the low speed wind tunnel.

In order to provide tunability of the fundamental mode of the fin, a series of tip masses of various materials were used on the fin. Figure 12 shows the open loop transfer function between the control voltage and resulting strain for the bare fin and each of the four tip masses, with no air blowing. Note the increase in the modal peak with increased tip mass. As the tip weight increases beyond the modal mass of the fin itself, the system behaves more like a spring and a lumped mass rather than a distributed system. As the ratio of distributed damping over mass in the system decreases, the magnitude of the modal peak (the Q of the system) increases. For each of the transfer functions, separate classical controllers were designed that placed the high frequency rolloff at the proper point. Table 3 summarizes the resonant frequencies.

Table 3: Resonant Frequencies for Various Tip Masses

<table>
<thead>
<tr>
<th>Tip Mass</th>
<th>Resonant Frequency (Hz)</th>
</tr>
</thead>
<tbody>
<tr>
<td>None</td>
<td>541</td>
</tr>
<tr>
<td>Nylon</td>
<td>499</td>
</tr>
<tr>
<td>Aluminum</td>
<td>411</td>
</tr>
<tr>
<td>Steel</td>
<td>302</td>
</tr>
<tr>
<td>Tungsten</td>
<td>208</td>
</tr>
</tbody>
</table>
Control tests were performed for a series of Mach numbers: 0.25, 0.67, and 0.95. Figure 13 shows a set of typical control results for the nylon tip mass and Mach 0.67. In this case, the classically designed controller achieves 35 dB of attenuation without saturation of the actuators. In fact, one of the biggest surprises of the wind tunnel testing was that actuator saturation was not achieved for even the highest tunnel speed. It was expected to encounter actuator saturation limits at some speed for the highest gain controllers, but this did not occur.

There was an additional concern that coupling of the air and fin dynamics at high wind speeds may shift the torsional mode of the fin closer to the first bending mode making control of this mode difficult without inducing unwanted flutter of the fin. In order to examine the change in the system dynamics with wind speed, transfer functions were taken at all three tunnel speeds. Due to the short time per blow of the tunnel, the number of ensembles was limited to one making the data noisy overall. The results however, show that there was no appreciable change in the system with wind speed. The same controllers were used for all the wind speeds with equal success. The transfer functions are shown in Figure 14.
VII. Passively Shunted Fin

A secondary goal of this effort was to examine the effectiveness of a passive inductive shunt on the high speed fin. There have been many papers that address the theory behind inductive shunting of piezoelectric actuators. The details of the theory are not important here. The basic damping mechanism is described below.

1. Mechanical energy from the vibrating structure is converted to electric charge via the piezoelectric effect.
2. An inductive shunt sized to the mechanical resonance provides low impedance for this charge to be converted to voltage between the capacitance of the piezo and inductance of the shunt.
3. Energy is dissipated by a series resistor or the distributed resistance of the network itself.

The resonant frequency and damping of the system are controlled by the resistance $R$, inductance $L$ of the shunt and the capacitance $C_p$ of the piezoelectric. The calculated optimal resistance was below 1 Ohm allowing us to use the natural resistance of the wires in the shunt itself thereby simplifying the system. A bench top test was used to explore the effectiveness of a passive shunt on system damping. Unlike active approaches that can be used to produce damping over a broader frequency range, the inclusion of an inductive impedance across the terminals of the piezoelectric can produce damping only at a specific frequency. As shown in Figure 15, the passive shunt was able to reduce the modal peak by 12dB.
VIII. Possible Scaling

Can this active damping approach be scaled to the F-16 ventral fin? Much depends on the amplitude of the disturbance inputs. That structure has a higher ratio of chord to span than the test fins, and of course it is substantially larger. Based on strain gage flight data, and an estimated root moment of 2450 in-lb, with most of the energy between 80 and 100 Hz, we sized a system conservatively to do active damping.

Actuation would consist of piezoelectric actuators having 24 times the total volume used in the high speed fin tests. There would be 8-16 independent groups that could be used to control different modes of vibration. Sensing would use 16 strain gages, again grouped to emphasize the key mode or modes. A flight demonstration would use 150-200 pounds of piezo drive amplifiers and a 200 VDC bus. Control algorithm implementation is a lower risk than actuation. Finally, an inductive shunt system could be viable with the same number of piezoelectric transducers, along with 10-20 inductors, each having a mass of 4-6 lbm.

IX. Summary

Laboratory and wind tunnel tests demonstrated the role that active structural control (ASC) can play in reducing stresses in aircraft components, where those stresses result from flow-induced loads. ASC also can be used in conjunction with active flow control (AFC) in ways that may offer reduced total power, and even direct electronic information transfer from the AFC to the ASC system.

The particular ASC approach studied here used piezoelectric actuators. This system proved effective at active vibration damping. For wind tunnel testing, the electric field and voltage limits were monitored closely, and the control system gains were adjusted to accommodate output saturation without introducing excessive conservatism. This is particularly important in achieving the highest possible efficiency in vibration control without adding excessive actuator volume or mass, or in the presence of severe constraints on the locations of vibration control actuators.

Two ASC systems were tested, one for a low speed wind tunnel and one for a high speed wind tunnel. In both cases, the ASC performed extremely well and actuator saturation was not encountered even for the highest controller gains and highest tested Mach number (0.95). Additionally, a passive resonant shunt targeting the fundamental structural mode was tested on the high speed fin in a laboratory setting. It also achieved good attenuation performance.

References