Noise reduction in a launch vehicle fairing using actively tuned loudspeakers[a]

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(Received 17 April 2001; revised 19 November 2002; accepted 8 January 2003)

Loudspeakers tuned as optimal acoustic absorbers can significantly reduce damaging, low frequency, reverberant noise in a full-scale launch vehicle fairing. Irregular geometry, changing payloads, and the compliant nature of the fairing hinder effective implementation of a passively tuned loudspeaker. A method of tuning the loudspeaker dynamics in real time is required to meet the application requirements. Through system identification, the dynamics of the enclosure can be identified and used to tune the dynamics of the loudspeaker for reduction of targeted, high intensity, low-frequency modes that dominate the acoustic response in the fairing. A loudspeaker model with desired dynamics serves as the reference model in a control law designed to tune the dynamics of a non-ideal loudspeaker to act as an optimal tuned absorber. Experimental results indicate that a tuned loudspeaker placed in the nose cone of the fairing significantly reduces acoustic energy and verifies results calculated from the simulation.
[DOI: 10.1121/1.1558371]

PACS numbers: 43.50.Ki [MRS]

I. INTRODUCTION

The high-intensity, low frequency acoustic excitation that occurs in a rocket fairing at launch can induce structural-acoustic vibrations that damage the payload. As engineers endeavor to create larger, lightweight, cost-effective fairings, acoustic excitation becomes a more critical factor in payload launch survivability. Previous research in this area has investigated both active and passive methods for acoustic attenuation,[1–3] in addition to vibration control of the fairing itself.[4,5] At low frequencies, lightly damped modes dominate the interior acoustic response of the fairing, and the ineffectiveness of passive methods such as acoustic blankets, fiberglass, or acoustic foam impels consideration of active and hybrid control methods. This study presents the experimental results of an actively controlled, optimally tuned acoustic absorber implemented in a full-scale fairing testbed.

Previous work has shown that dissipative control can be achieved in acoustic enclosures using a collocated pressure sensor and a constant volume-velocity source.[6,7] Extension of this principle yielded significant attenuation of low frequency acoustic modes in a launch vehicle fairing by utilizing arrays of sensors and actuators working cooperatively to achieve global control. The transducer arrays were spatially weighted in order to selectively couple to low-frequency modes, which reduced the order of the controller and yielded significant modal control with minimal control spillover.[8] Global peak reductions (8–10 dB) of targeted acoustic modes required an extensive array of control sources, conditioning and amplification measures, microphones, control hardware, and wiring. Further review of these requirements has prompted investigation of less equipment intensive, ‘‘hybrid’’ active/passive controllers.

Previous research involving hybrid systems for sound absorption demonstrated the effectiveness of surface impedance control in rectangular enclosures[9,10] and impedance tubes[11] with low modal density. Both loudspeaker diaphragms and panels with tuned dynamics demonstrated attenuation through active impedance changes at enclosure boundaries, but none demonstrated acoustic absorption from within the enclosure. More relevant to this work, the authors joined an effort to develop a passive vibroacoustic attenuator based on a hybrid of a proof-mass actuator (PMA) and a shunted loudspeaker.[12] That study highlighted the possibility of combining a tuned mass damper for structural damping and a tuned diaphragm for acoustic damping. The attenuator was dynamically modeled and placed within a fully coupled structural-acoustic finite element model of the fairing. Results predicted a broadband noise reduction of 3 dB, but difficulties in the experimental development and application of such an attenuator left room for improvement.

As an extension of the hybrid principle, this work applies tuned mass damping theory to sound absorption applications in a flexible, irregular cavity that encloses changing payloads and demonstrates significant, three-dimensional modal complexity. This study proposes to attenuate low frequency modes in a launch vehicle fairing with loudspeakers tuned as optimal acoustic absorbers. The absorbers promote efficiency of the control system while demonstrating control results comparable to previous active studies. As a primary advantage, the significant logistical issues of typical active systems have been eliminated. The control law discussed herein actively tunes a loudspeaker to passively reduce the

[a]Portions of this work were presented in ‘‘Noise reduction in a launch vehicle fairing using actively tuned loudspeakers,’’ Proceedings of Joint 140th Meeting of the Acoustical Society of America and NOISE-CON 2000, Newport Beach, CA, 2000.

targeted mode(s), removing typical acoustic spillover in the control bandwidth, and allowing global attenuation with a single actuator. A parametric optimization scheme selects an optimal tuned absorber model which is compared to an identified model of the non-ideal loudspeaker in the fairing; the difference between them is minimized by a model reference control law, based on H_2 design. Loudspeaker tuning requires only low-order control in a limited bandwidth. These achievements increase the performance and stability of the controller, while reducing control energy and hardware by more than 90%.

Preliminary investigation of actively tuned acoustic absorber design facilitates application of that technology for noise control in acoustic enclosures, and more specifically, in an experimental, full-scale, composite launch vehicle fairing model at Duke University. In support of continuing efforts to improve payload survivability, the research sought definitive control results to promote further study of the effectiveness and efficiency of adaptive acoustic control within the fairing volume.

II. THEORY

A. Fairing acoustics

A rocket fairing is essentially a flexible-walled structure surrounding an acoustic enclosure that protects the payload during launch. Fairings have typically been constructed of metal, such as aluminum, but composites have been used more recently. Although composites offer many advantages, they are less massive and therefore allow increased transmission of exterior disturbances. Vibration of the fairing excites acoustic resonances within the enclosure, which detrimentally couple to the payload. Data from previous satellite launches indicate that the overall acoustic levels inside the fairing during launch can exceed 140 dB. At low frequencies (10 to 200 Hz), the levels reach approximately 120 to 130 dB.[13]

The fairing considered in this work is based on the Orbital Sub-orbital Program Space Launch Vehicle (OSPSLV) minotaur. The shroud is 5.3 meters long with a maximum diameter of approximately 1.3 meters. Therefore, the low-frequency modes are longitudinal rather than radial. The fairing is excited by a number of disturbances, including aerodynamic buffeting during flight,[14] structural vibrations induced by the rocket motors, and pyrotechnic shocks during stage separations. These sources act in addition to the explosive noise produced by the motors themselves.

The most harmful vibration occurs in the first 30–45 seconds after ignition, as both acoustic transmission and aerodynamic buffeting decrease with decreasing atmospheric pressure and density. While on the launch pad, the earth acts as an acoustic baffle, reflecting enormous waves of energy at the payload. As the vehicle lifts away from earth, increased speed causes significant aerodynamically induced noise. When the payload exhibits a structural resonance near the acoustic resonances of the fairing, the acoustic loading can result in appreciable damage to the payload. The Department of Defense reports that at least 40% of first-day satellite failures result from vibration damage incurred at launch.[15] Flat surfaces of solar panels and light-weight structures such as thin films, membranes, and precision optics are particularly susceptible to damage from the low frequency excitation. The presence of multiple, random, low-frequency disturbance sources precludes passive and feedforward control, compelling the use of an innovative feedback control scheme.

B. Tuned acoustic absorbers

The concept of loudspeakers tuned as optimal acoustic absorbers grew from the familiar use of tuned mass-spring systems for vibration absorption in structures. Here, active tuning controls enclosed acoustics. Departing from previous acoustic approaches which attempted to change the enclosure boundary, this work develops the ability of production loudspeakers to act in reverse, or as self-contained tunable absorbers. Essentially, a loudspeaker is actively tuned to respond passively as though it were an ideal absorber for a targeted modal frequency.

Realization of the ideal absorber begins with development of a theoretical model that couples the response of the fairing enclosure to a disturbance source and a typical loudspeaker. An optimization scheme then selects an ideal absorber for reduction of global acoustic energy at a targeted modal frequency. A model reference control law can then minimize the response difference between the ideal model and the actual loudspeaker, forcing that loudspeaker to act as an ideal absorber.

III. MODELING, DESIGN, AND OPTIMIZATION

The combined behavior of the fairing disturbances is unpredictable, but the modal acoustic response in the enclosure due to the vibration of the fairing structure can be easily modeled. A rigid walled cylinder model of equal dimension provides a strong prediction of the magnitude, phase, and frequency of the lowest modal frequencies, and is constructed here for the purposes of absorber design and optimization. The development follows Morse and Ingard[16] and Cheng and Nicolas.[17] Application of the appropriate boundary conditions allows solution of the homogeneous wave equation in a cylinder of length L with a Hankel function of order m:

$$p(r,\theta,z,t)=\sum_{m=0}^{\infty}\sum_{n=1}^{\infty}\sum_{\mu=1}^{\infty} A_{mn\mu}\sin\left(m\theta+\gamma\frac{\pi}{2}\right) \times J_m(k_r r)\cos\left(\frac{\mu\pi}{L}z\right)e^{i(\omega t-k_\theta\theta-k_{z_{mn\mu}}z)}, \quad (1)$$

where p represents the pressure in the fluid, c is the local speed of sound, and r, z, and θ are the cylindrical coordinates within the cavity. Here, m, n, and μ, and γ are the azimuthal, radial, longitudinal, and symmetric (rotational) modal indices, respectively. Here, k_θ is the azimuthal wave number and k_z is the longitudinal wave number. The radial wave number k_r is the nth root of the derivative of the Bessel function of the first kind evaluated at the cavity wall.

The model includes a vibrating rectangular panel, curved and placed at the enclosure wall to simulate the trans-

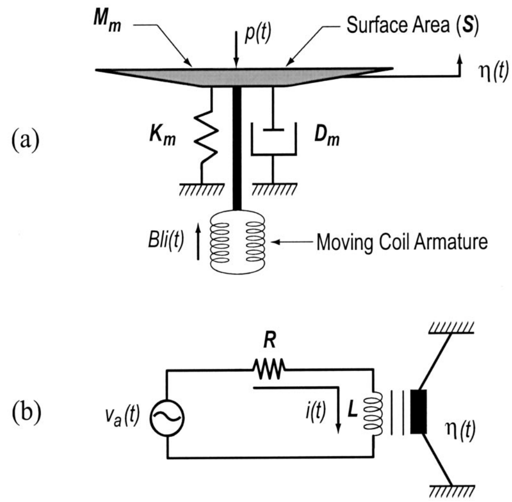

FIG. 1. (a) Mechanical schematic of loudspeaker system. (b) Electrical schematic of loudspeaker system [after Clark *et al.* (Ref. 20)].

mission path in the actual fairing. Following fluid-structure coupling techniques, the modal interaction model by Fahy,[18] and recent work by Henry and Clark,[19] the model results. From the inhomogeneous wave equation for the pressure in the cavity, the equations of motion for the enclosure as disturbed by the curved panel can be written as

$$\Lambda_{n_p}\ddot{v}_{n_p}(t)+\Lambda_{n_p}\omega_{n_p}^2 v_{n_p}(t)=-\rho_0 c^2 S_0\sum_{k=1}^{K} C_{n_p k}\ddot{w}_k(t), \qquad (2)$$

where ρ_0 is the density of the fluid in the cavity, and $v_{n_p}(t)$ is a generalized modal coordinate. The panel undergoing out-of-plane displacement, w, is modeled as a distributed volume velocity source just inside the enclosure boundary. The modal volume, Λ_{n_p}, is represented by the integral of spatial mode shape functions, and the subscripts m, n, and μ have been replaced in by the generalized modal index n_p. The coupling coefficient $C_{n_p k}$ is the surface integral of the product of the mode shape functions of the panel and the enclosure.

A. Tuned absorber coupling

Modeling and design of the tunable acoustic absorber requires coupling the equations of motion for a typical loudspeaker to the acoustics of the enclosure model developed above. The passive interaction of the tuned loudspeaker with the fairing acoustics forms the tunable boundary condition for control application. In preliminary tests, the nose cone seemed ideal for installation of the tuned absorber: the primary active control targets are the modes of lowest frequency, so the absorber should interact with longitudinal, not radial or azimuthal, wave forms. These wavelengths are most easily targeted spatially by observation and control at the end caps of the fairing. Applied damping at the boundary condition responsible for longitudinal modal response will reduce acoustic energy at the modal frequency, thereby reducing overall acoustic energy propagation within. Additionally, placement of the tuned absorber in the nose cone minimizes the control system impact on available payload volume.

A model of a loudspeaker, radiating in its piston mode, was therefore incorporated in one end cap of the enclosure. The electromechanical coupling and structural acoustic coupling are described by assimilating the typical loudspeaker equations of motion[20] with the enclosure model above. Figure 1(a) provides a schematic representation of the mechanical system, and the respective equation of motion can be written as

$$M_m\ddot{w}(t)+D_m\dot{w}(t)+K_m w(t)=Bli(t)-S_0 p(t), \qquad (3)$$

where M_m, D_m, and K_m are the mechanical mass, damping, and stiffness of the speaker, and $\dot{w}(t)$ represents the displacement of the speaker coil. The force input has two components: electromotive force is the product of the field strength of the inductor B, the conductor length l, and the current $i(t)$; and the pressure input is the product of the surface area of the loudspeaker diaphragm, S_0, and the acoustic pressure, $p(t)$. This pressure input, or radiation resistance, is the most crucial coupling term for this application. Figure 1(b) provides a schematic representation of the electrical system, governed by the following equation:

$$L\frac{\mathrm{d}i(t)}{\mathrm{d}t}+R_s i(t)=v_a(t)-Bl\dot{w}(t). \qquad (4)$$

Note that the voice coil (Bl) couples the electrical and mechanical systems in the speaker. These equations of motion can now be coupled through diaphragm displacement w and acoustic pressure p to the acoustic model of the enclosure for full system modeling and optimal design.

B. Optimization

A constrained optimization scheme determines the optimal mechanical and electrical parameters of the loudspeaker for acoustic energy absorption at a targeted mode in the enclosure. Within reasonable constraints, the routine could select candidate speakers by arbitrary variation of each significant property of the loudspeaker system: B, l, R, M_m, K_m, D_m, and S_o. However, to enhance the feasibility of actual loudspeaker construction and tuning, the parameter field was narrowed to those available for physical tuning: the resistance R of the circuit, and the apparent spring stiffness, K_m, of the mechanical system. The cost functional was calculated as the H_2 norm of the targeted modal acoustic output v_n of the coupled loudspeaker/enclosure system. The routine utilized a constrained optimization function provided in the optimization toolbox for MATLAB. The function, *fmincon.m*, follows a sequential quadratic programming method in which a quadratic programming subproblem with linearized constraints is solved at each iteration as the routine searches for the minimum of the given cost functional. A positive definite quasi-Newton estimate of the Hessian of the Lagrangian function is updated at each iteration to provide the direction and magnitude of the next iteration. This estimate is calculated with the BFGS (Broyden, Fletcher, Goldfarb, and Shanno) formula which enhances the efficiency of the scheme.[21]

To assure the accuracy of the optimization results, several sets of initial conditions were passed to the routine for

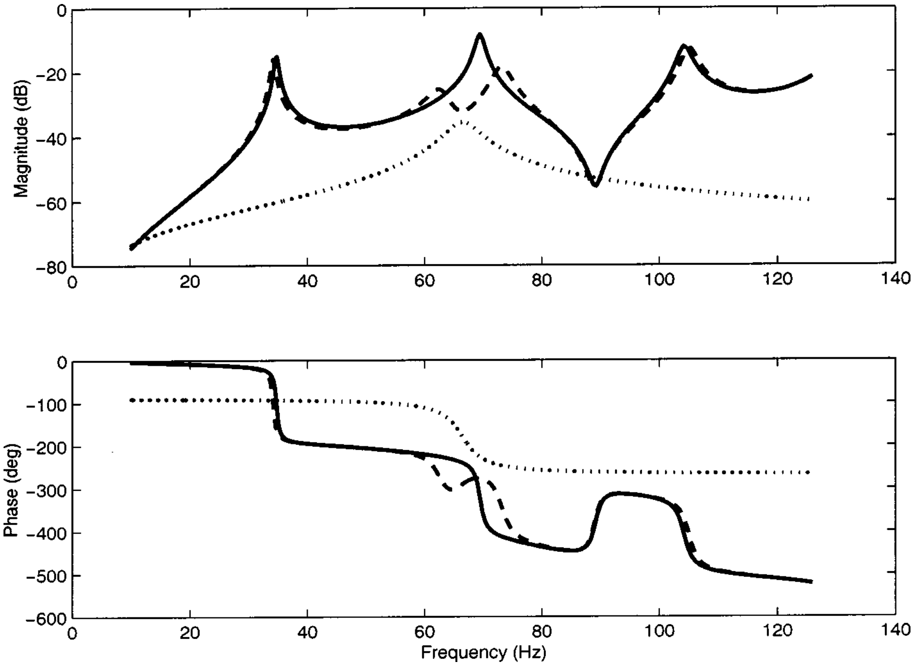

FIG. 2. Frequency response of enclosure without (—) and with (— — —) reduction due to coupled tuned absorber (···).

each targeted mode. The routine repeatedly selected similar models within one hundred iterations. The dynamic response of a typical optimal absorber demonstrates increased response at a frequency slightly above or below the targeted modal frequency. This observation confirmed initial hypotheses about the parallels between optimal acoustic absorbers and tuned vibration control devices. Figure 2 displays the predicted acoustic response of an optimal loudspeaker coupled to the acoustic enclosure developed above. Note the significant reduction in response at the second mode. This pressure response, taken in the acoustic far field of the tuned loudspeaker, demonstrates significant global reduction from a single tuned actuator. This ideal actuator serves as the reference model in a control scheme designed to tune the actual loudspeaker to act as the ideal absorber.

IV. EXPERIMENTAL IMPLEMENTATION

Development of the tuned absorber continues with application of model reference control to the loudspeaker in the fairing. Based upon work by Ogata,[22] the development of a model reference plant involves evaluation of a performance metric calculated as the response difference between the actual system and the model reference system, given identical inputs.

An approach based on the eigensystem realization algorithm[23,24] identifies low-order, discrete, state-space realizations of the untuned speaker for comparison with the optimal loudspeaker model developed above. A common source then disturbs both the realized system and the optimal model through the respective voltage inputs, ensuring that the controller affects the loudspeaker regardless of acoustic pressure input. The characteristics of the disturbance and its coupling to the fairing are irrelevant to the controller, broadening the scope of possible application. The outputs of the two systems are compared and the difference between them serves as the performance variable in the control law.

The augmented system[25,26] was then assembled and modeled using Simulink,[27] and H_2 control laws were computed using MATLAB. Filters designed to shape the controller are added for stability but increase the order (number of states) of the augmented plant and the resulting control law. The controller, now of unnecessarily large order, is reduced

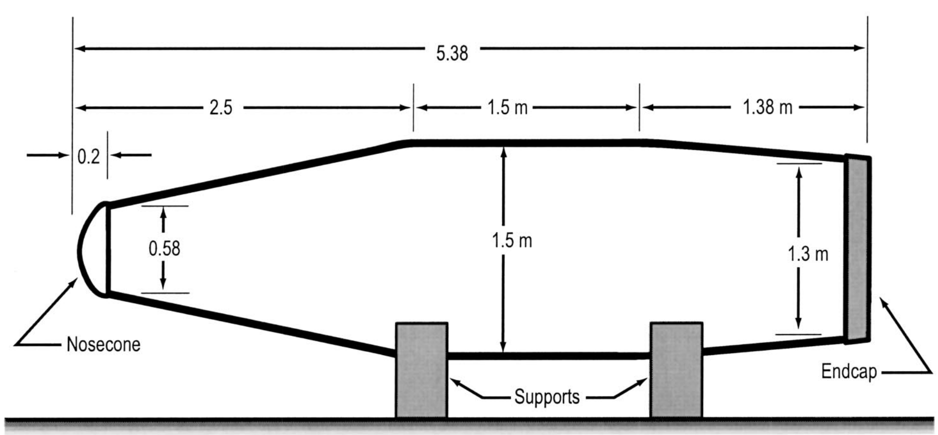

FIG. 3. Illustration of fairing testbed.

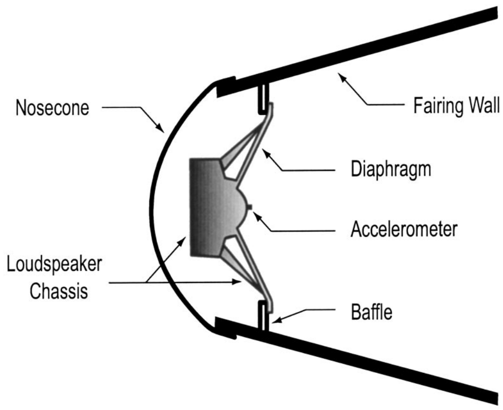

FIG. 4. Schematic of absorber system mounted in nose cone.

through model reduction techniques, such as balanced residualization or truncation.[26,28] After computation, a Tustin routine transforms the reduced-order control law into the discrete domain and the control law is downloaded to a digital signal processor (DSP) for implementation.

Comparison of the open-loop and closed-loop frequency response functions from a disturbance input (band-limited random noise) to an array of performance microphones yields a controller performance metric. Averaging these frequency response measurements provides representative open-loop and closed-loop frequency responses and facilitates evaluation of the overall effects of the controller as a function of frequency. From this evaluation, the occurrence of spillover, the degree of coupling to the target modes, and the average amount of local and global attenuation is observed.

A. Fairing testbed

Experiments are conducted on a full scale composite fairing model containing the loudspeaker/absorber, the accelerometer sensor, the acoustic disturbance source, and sixteen arbitrarily positioned performance microphones. The fairing (shown schematically in Fig. 3) is approximately 5.3 meters in length, 1.3 meters in diameter (maximum), and tapered at both ends. A plywood end cap is attached to the base of the fairing and sealed appropriately. A hemispherical aluminum end cap completes the nose cone and seals the acoustic chamber behind the loudspeaker, as shown in Fig. 4. The loudspeaker and baffle are rigidly attached to the nose cone. The loudspeaker is a manufactured 18 inch subwoofer, occupying roughly 60% of the area of the nose cone baffle.

The accelerometer is fixed at the diaphragm center for maximum signal and focus on the piston mode of the speaker. A small conditioning device located within the fairing amplifies the accelerometer signal. A disturbance loudspeaker in the corner of the fairing near the base end cap excites the interior cavity modes. The performance microphones were distributed throughout the interior at arbitrary positions to measure the overall controller effects. All cabling was connected through a panel at the base of the fairing. The controller, spectrum analyzers, power amplifiers, microphone conditioners, and other required hardware were housed external to the fairing. A block diagram of the setup is presented in Fig. 5.

V. RESULTS

A. Tuned absorber

The experiment begins with an investigation of the coupling effects between the loudspeaker and the fairing acoustics. Figure 6 supports the predicted effects of this coupling. The frequency response of the loudspeaker is measured both in open air and after installation in the fairing nose cone baffle. Both were measured as frequency response functions between a band-limited random disturbance applied to the voltage input of the loudspeaker and the voltage signal from the accelerometer mounted on the loudspeaker diaphragm. Note the strong modal response when the loudspeaker is coupled to the fairing acoustics. The modes apparent here are coupled mechanoacoustic modes of the loudspeaker and fairing, not merely a reflection of the fairing acoustics. This coupling is crucial to the success of the tuned absorber, as discussed in Sec. III, and verifies conjecture about this pressure input to the dynamic system modeled herein.

Modeling of the loudspeaker dynamics for the purpose of control is accomplished through system identification techniques. Following identification, the control law is applied to tune the loudspeaker as an optimal absorber. The controller effectively changes the dynamic response of the loudspeaker, as shown in Fig. 7. The loudspeaker response is

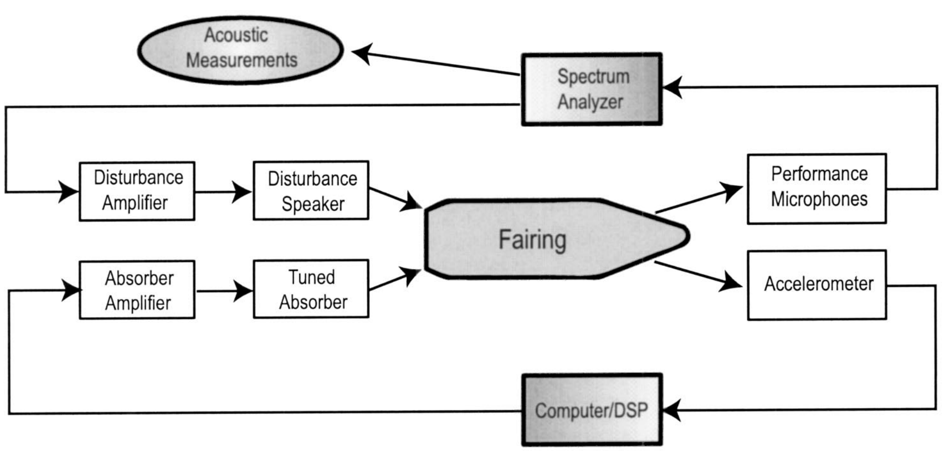

FIG. 5. Schematic of experimental setup.

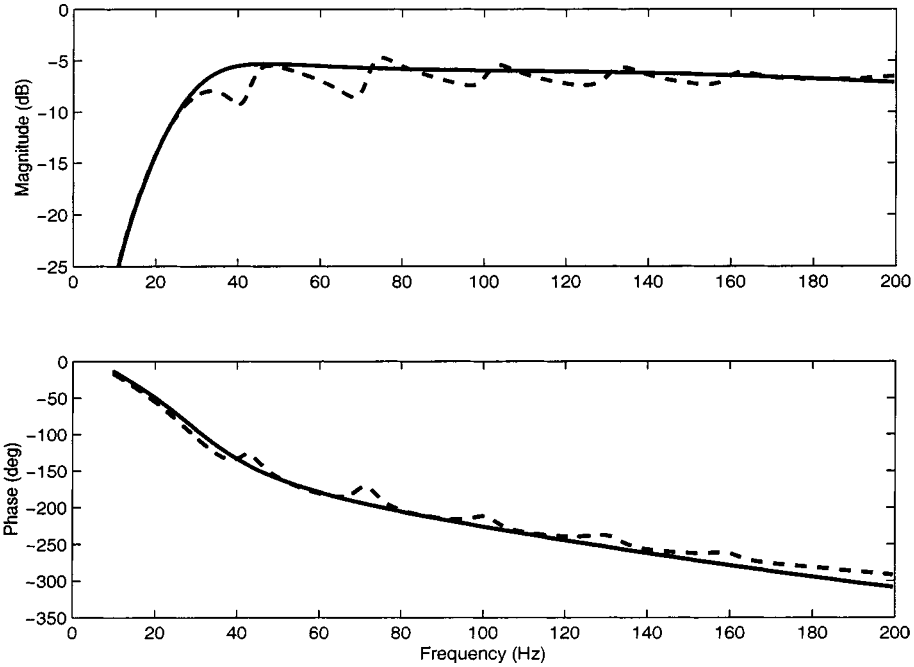

FIG. 6. Acoustic coupling demonstrated by loudspeaker responses measured from voltage input to accelerometer in open air (—) and after mounting in nose cone baffle (— — —).

measured by the accelerometer relative to an acoustic disturbance applied to the fairing volume. Note the significant peak introduced at the targeted frequency of 65 Hz. Figure 7 details the effects of the controller when the disturbance is purely acoustic; the modes of the fairing are apparent. The controller does not retard response of the loudspeaker at the non-targeted first acoustic mode, so the response of the speaker at that frequency remains unchanged.

Model reference control demonstrates authority over the dynamics of an actual loudspeaker. By design, the control law can tune the loudspeaker to absorb at any frequency, though some frequencies are more conducive to tuning than others. The efficiency of the controller and the magnitude of resulting effects are dependent on the dynamics of the speaker, as expected. If the desired tuning frequency is proximal to a coupled mode of the loudspeaker and fairing, then the controller is more successful, as the effort required to move a system pole is significantly less than that required to introduce a new pole. Additionally, the modal targeting introduced by the noise filters in the control law prevents placement of control effort out of the most efficient bandwidth. An inherently efficient controller results from this combination of control strategies.

B. Acoustic control in the fairing

Demonstration of a stable tuned loudspeaker allows implementation in the fairing volume. Key to the stability and minimal energy consumption of the controller is the separation of the controller from the acoustics of the fairing. The control law seeks only to tune the mechanics of the loudspeaker, not to actively control the fairing acoustics. This separation precludes the development of typical acoustic instabilities found in most active acoustic control systems,

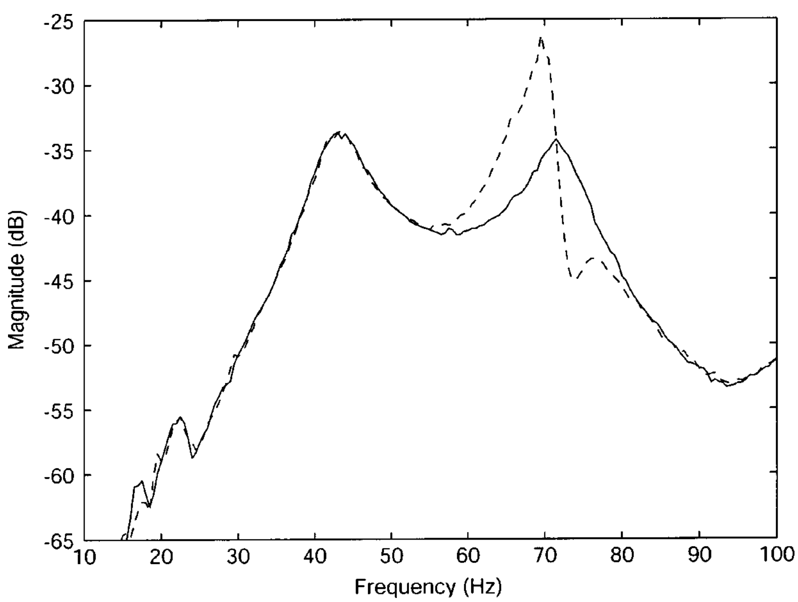

FIG. 7. Loudspeaker acceleration response measured from acoustic disturbance to accelerometer without (—) and with active tuning (— — —).

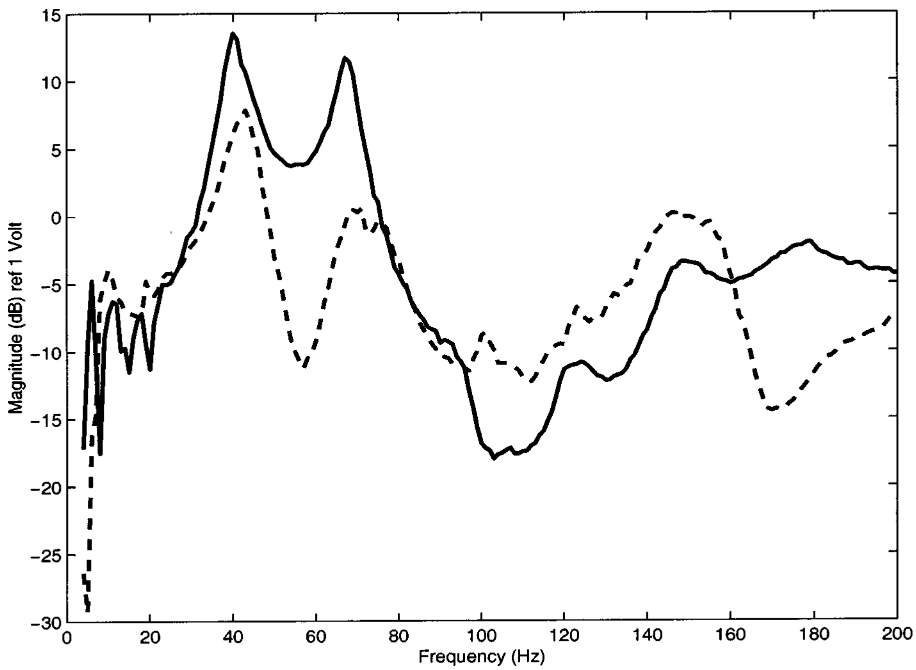

FIG. 8. Local acoustic pressure response measured from acoustic disturbance to a single microphone (No. 3) without (—) and with active tuned absorber (— — —) targeting second acoustic mode (69 Hz).

allowing significant control signal gain without fear of unstable acoustic feedback. In other words, the controller actively tunes the loudspeaker to act as a passive acoustic absorber, as discussed in the preceding sections.

Optimization efforts revealed significant control capability at the second mode of the fairing, 69 Hz. Proximity to an actual coupled modal frequency of the loudspeaker and fairing promotes effective control of loudspeaker dynamics. Candidate absorber models with natural frequencies slightly above or below the targeted 69 Hz were chosen by the optimization routine. The best results, both predicted and actual, utilized an absorber tuned to 80 Hz. The optimal tuning frequency is roughly 15% above the targeted modal frequency, a result typical of tuned mass absorber design. Figure 8 displays the local pressure response for this control configuration, measured at a microphone located in the cross-sectional center of the fairing, approximately 0.8 meters from the tuned diaphragm. The 12 dB reduction at the second mode (69 Hz) is apparent; the resonant peak is split and of greatly reduced magnitude. This microphone represents the response of the fairing in the acoustic near field of the tuned loudspeaker.

An evaluation of global performance in the fairing yields the acoustic response shown in Fig. 9. This response was measured as the simple average of the pressures recorded at 16 arbitrary positions within the fairing. Significant global peak reduction (4 dB) is apparent; control is achieved.

Note that the global reduction is less significant than the local control shown in Fig. 8, as expected. To investigate the performance at individual microphones throughout the space,

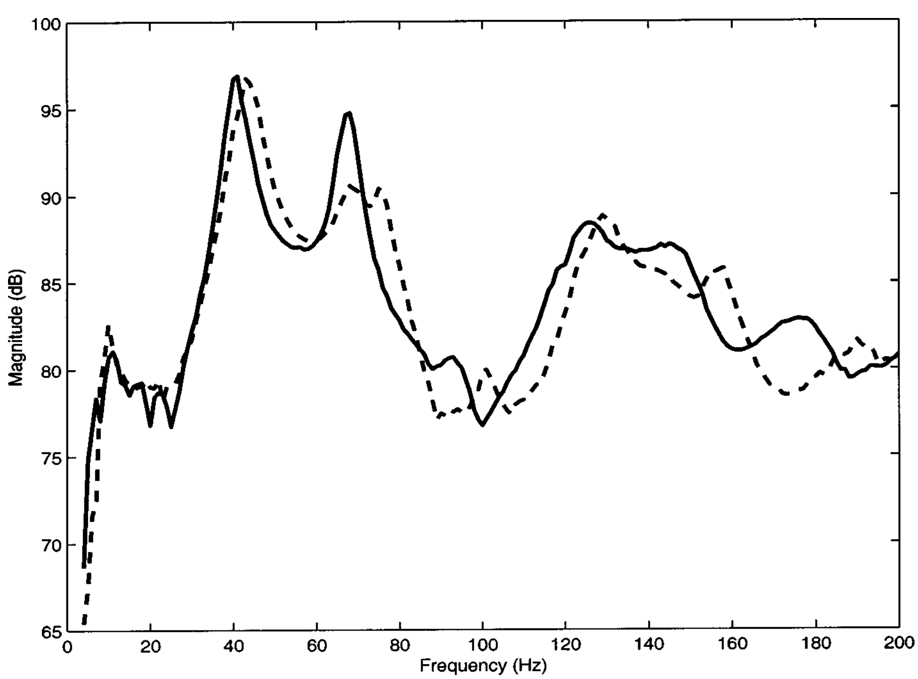

FIG. 9. Global acoustic pressure response, as averaged from acoustic disturbance to 16 microphones, without (—) and with active tuned absorber (— — —) targeting second acoustic mode (69 Hz).

TABLE I. 69 Hz mode peak pressure reductions achieved with an actively tuned loudspeaker, calculated from measurements of frequency responses between acoustic disturbance and each microphone.

Microphone ID	Peak reduction (69 Hz)
1	7 dB
2	8 dB
3	12 dB
4	8 dB
5	0 dB
6	1 dB
7	0 dB
8	6 dB
9	6 dB
10	5 dB
11	12 dB
12	4 dB
13	−3 dB
14	3 dB
15	5 dB
16	5 dB

Table I details the second mode (69 Hz) peak reduction measured at each location. The microphone locations are numbered consecutively from forward to aft. The local frequency response plotted in Fig. 8 corresponds to microphone number 3, as the first two positions were closer to the loudspeaker but not in the center of the fairing. Note that attenuation is achieved at 13 of the 16 microphone locations, the exceptions being two microphones where no attenuation was noted and a 3 dB increase measured at microphone number 13. However, the overall RMS pressure level in the 200 Hz bandwidth was still reduced by 0.61 dB at microphone No. 13. Additionally, two separate locations, one in the acoustic near field (No. 3) and one in the far field (No. 11) measured peak reductions of 12 dB. The previous active system requiring 16 actuators did not achieve any peak reduction greater than 12 dB.[8]

A promising observation throughout the tests is the voltage necessary to achieve this level of control. A frequency spectrum of the control voltage is provided in Fig. 10. This controller, by design as merely an enabling device for passive absorption, required 0.216 Volts RMS (1.180 V peak) for the duration of the testing. These levels are at least an order of magnitude less than the voltage required per actuator by active systems achieving similar levels of control with multiple actuators.[8] Power requirements are also appropriately reduced. Each of the above results demonstrates the effectiveness of a low-order model-reference controller used to tune a loudspeaker to act as an optimal tuned acoustic absorber in a full-scale payload fairing.

VI. CONCLUSION

Preliminary results presented here demonstrate the effectiveness of a tuned loudspeaker acting as a self-contained optimal acoustic absorber in a launch vehicle fairing. Application of structural tuned mass absorber theory to the realm of acoustics has allowed efficient acoustic reduction with active technology but without the logistical cost of previous active and passive control apparatus.

Following development of a theoretical model of the enclosure and of a typical loudspeaker, an optimization scheme selects the dynamic properties of an ideal absorber for a chosen acoustic mode of the enclosure. This optimal absorber then serves as the reference model in a control law designed to actively tune an off-the-shelf loudspeaker to act as an optimal tuned absorber. Placement of this tuned loudspeaker in the nose cone of a full-scale launch vehicle fairing allows experimental verification of the tuned absorber concept. A tuned loudspeaker acting passively in the fairing enclosure absorbs significant acoustic energy at targeted modal frequencies.

The feasibility of a launchable application remains the quest of this technological development. The advantages here are simple: the actively tuned absorber minimizes controller impact on payload weight and volume. The system requires only one actuator per acoustic mode to be controlled, rather than the array used for active studies. With each actuator goes an accompanying amplifier, microphone, and signal conditioner. Further, control signal voltage and actuator power requirements are orders of magnitude less

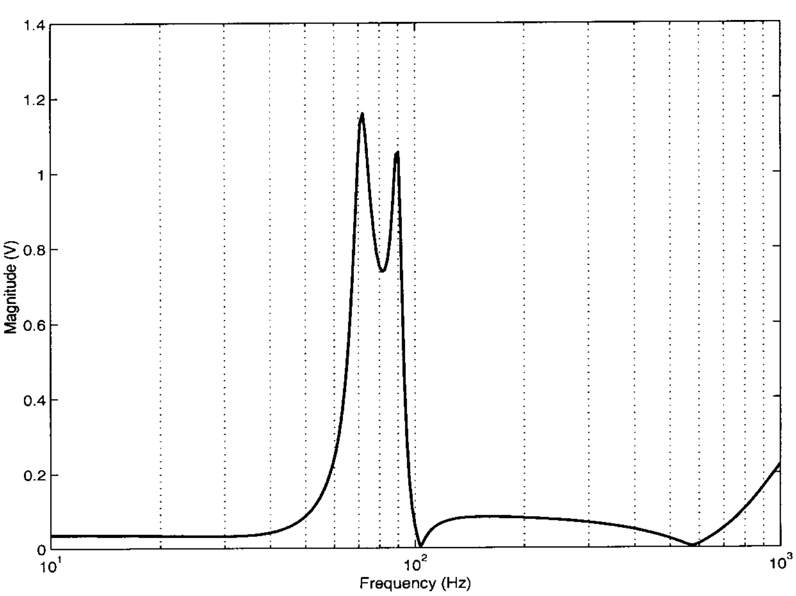

FIG. 10. Frequency spectrum of the model reference control signal voltage for the actively tuned loudspeaker.

than active counterparts, relieving battery weight. This hybrid controller accomplishes efficient, lightweight noise attenuation in a frequency range where both active and passive control have significant logistical and performance problems.

Future work will address challenges that arise in the application of any controller to a launchable enclosure. Acoustic response of the fairing, and therefore optimization and modal targeting parameters, will change with payload and atmospheric conditions, so the tuned system must be able to react accordingly. The tuned absorber system offers feasible solutions. Because the tuning parameters are based solely on a targeted frequency, they could be updated without continuous system identification. In fact, these tracking adjustments could be incorporated in the control law. Additionally, further studies should include the use of multiple tuned loudspeakers targeting single and multiple frequencies in the enclosure. Development of these concepts holds promise for future investigations, as the actively tuned loudspeaker has demonstrated acoustic control in a full-scale fairing.

ACKNOWLEDGMENTS

The authors would like to gratefully acknowledge the partial support for this research provided under the AFOSR Presidential Early Career Award for Scientists and Engineers, Grant No. F49620-98-1-0383, monitored by Major Brian Sanders and Dr. Dan Segalman. We would also like to express our gratitude to the Air Force Research Laboratory for providing the rocket fairing model used in these experiments.

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