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Re-Active Passive devices for control of noise transmission through a panel

James P. Carneal[a,*], Marco Giovanardi[b], Chris R. Fuller[c], Dan Palumbo[c]

[a]*Vibration and Acoustics Laboratories, Mechanical Engineering Department, Virginia Polytechnic Institute & State University, Blacksburg, VA 24061-0238, USA*
[b]*Active Control experts, Inc. 215 First Street, Cambridge, MA 02142, USA*
[c]*NASA Langley Research Center, Hampton, VA 23606, USA*

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Abstract

Re-Active Passive devices have been developed to control low-frequency (<1000 Hz) noise transmission through a panel. These devices use a combination of active, re-active, and passive technologies packaged into a single unit to control a broad frequency range utilizing the strength of each technology over its best suited frequency range. The Re-Active Passive device uses passive constrained layer damping to cover relatively high-frequency range (>150 Hz), reactive distributed vibration absorber to cover the medium-frequency range (50–200 Hz), and active control for controlling low frequencies (<150 Hz). The actuator was applied to control noise transmission through a panel mounted in the Transmission Loss Test Facility at Virginia Tech. Experimental results are presented for the bare panel, and combinations of passive treatment, reactive treatment, and active control. Results indicate that three Re-Active Passive devices were able to increase the overall broadband (15–1000 Hz) transmission loss by 9.4 dB. These three devices added a total of 285 g to the panel mass of 6.0 kg, or approximately 5%, not including control electronics.

1. Introduction

The control of sound transmission through a panel has received widespread attention with the emphasis of producing increased attenuation by passive, reactive, or active means. This fundamental research has regained interest in the past 15 years as novel concepts such as active control, advanced constrained layers, and distributed reactive devices have been introduced. Throughout this research, it has been evident that no one technology can cover low, medium, and high relative frequency ranges. This is due to the physics of structural vibration and the structural–acoustic coupling that occur at each frequency range; therefore in any given frequency range a different technology will be the most efficient at addressing the physical mechanisms.

Passive sound control methods dissipate propagating acoustic and/or structural waves through various damping mechanisms that do not require an external supply of control energy. Foams and viscoelastic

*Corresponding author. Tel.: +1 540 231 3268.
E-mail address: jcarneal@vt.edu (J.P. Carneal).

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constrained layer damping are some of the examples. Usually, these methods work well at relatively high frequencies, where the wavelengths are short enough to produce significant strain in the damping device. At low frequencies, the amount of material needed for effective control of sound/vibration becomes economically infeasible, considering most of the applications are weight and volume sensitive, such as aircraft [1].

Reactive materials and devices, which include semi-passive or tuned absorbers/dampers, are devices that provide significant attenuation over limited spatial and frequency bands by transferring energy from the structure/acoustic field into a resonant system [2]. The damping of the system is chosen to determine the conflicting performance parameters of attenuation (related to the Q of the system) and device bandwidth. Acoustic cavities, spring–mass systems, and shunted piezoelectric ceramics are examples of tuned dampers. These devices work well in frequency ranges where the acoustic wavelengths are long enough to achieve a large zone of cancellation of the structure and/or acoustic space. However, for relatively low frequencies, the form factor of acoustic cavities and spring–mass systems requires significant space, and again implementation becomes economically infeasible. Conversely, there have been several commercial products based on shunted piezoceramics from the family of lead zirconate titanates (PZT), which use a combination of resistors and inductors to dissipate electrical energy from the piezoceramic that was induced by mechanical energy. They have been applied to skis, snowboards, and baseball bats [3].

Advances in smart materials, materials that change their mechanical properties by electrical, thermal or magnetic means, have introduced a new dimension to active sound control. The piezoelectric material that was mentioned above is one of the best candidates for active control due to its response speed, ease of integration and control authority. The practical implementation of these devices applied to active sound control has become relatively easier as the computing power and embedded integration of digital signal processors has progressed. Control is achieved by using a combination of actuators and error sensors to perform destructive interference with the sound field generated by the source. The piezoelectric ceramic has allowed researchers to demonstrate control of low-frequency noise with induced strain actuation and sensing, called active structural acoustic control. This technique has been successfully implemented to attenuate the sound generated by vibrating beams, plates and shells [4–6].

Researchers have been working on various hybrid approaches for noise and vibration control including active constrained layer damping [7–9] and "smart foam" [10]. Most of the research centered on constrained layer damping integrated with a piezoelectric actuator, allowing the damping layer to control high-frequency regions and active control for the low frequencies. To achieve broadband global control of complex structures, multiple sensors and actuators will have to be implemented, leading to modeling and co-linearity problems that are an inherent result of MIMO control systems. Therefore, it is best to reduce the bandwidth and complexity of the controller. One method to achieve this goal is the inclusion of a reactive device to further reduce the bandwidth covered by the active device and augment the performance of the active constrained layer damping devices.

This paper details an experimental investigation of the performance characteristics of a new device, called Re-Active Passive, which combines active, re-active, and passive technologies packaged into a single unit. This device utilizes the strength of each technology over its best-suited frequency range to achieve broadband performance. In this paper, the Re-Active Passive was implemented to reduce the low-frequency (<1000 Hz) noise transmission through a panel mounted in a transmission loss test facility. The Re-Active Passive device uses passive constrained layer damping to cover relatively high-frequency range (>150 Hz), reactive (distributed vibration absorber) to cover the medium-frequency range (50–200 Hz), and active control for controlling low frequencies (<150 Hz). Experimental results are presented for the bare panel, and combinations of passive treatment, reactive treatment, and active control.

2. Experimental setup

The experimental setup for the panel, panel modal testing, transmission loss testing, Re-Active Passive device design, and feedforward controller configuration are discussed in this section.

2.1. Panel mounted in transmission loss facility

The 1.21×0.55 m steel panel was mounted in the transmission loss test facility in a common wall between two reverberation chambers. Since the panel mounting frame was 1.24×1.24 m, a panel adaptor was constructed using two 19 mm medium density fiberboard (MDF) boards with a 1.19×0.53 m rectangular hole cut in the center. Since the panel had flared edges, a third piece of MDF board was used as a frame to offset the panel from the adapting mount. See Fig. 1. The MDF panels were bolted to the panel adaptor with foam rubber weather stripping in between to provide a soundproof seal. The panel has the following characteristics: dimensions between clamped edges $1.19 \times 0.53 \times 0.001$ m thick, 200 GPa modulus and 6.0 kg total weight.

To provide clamped boundary conditions, the panel was bolted to the MDF panels with the MDF frame providing a clamping edge on one side and an aluminum angle providing a clamping edge on the other side. All bolts were tightened in a cross pattern with a torque wrench. The bolts and the aluminum angle were match drilled to make a soundproof seal.

2.2. Modal testing

Modal testing of the panel was performed using a shaker with a force transducer and roving accelerometer. Acceleration measurements were taken on the panel on a 5 by 11 grid to determine the panel response, including the response of the edges. A schematic for the modal test is presented in Fig. 1. The sampling parameters are as follows: 4000 Hz sampling frequency, 4096 samples per average, 1000 Hz anti-aliasing filter and a Hanning time window. The shaker was excited with broadband random noise, band-pass filtered from 10 to 500 Hz.

To prove the shaker was not mass loading the panel, a transfer function was taken with a modal hammer to one accelerometer location and compared to the transfer function of the shaker to the same accelerometer. The results are presented in Fig. 2. As can be seen, the transfer functions are similar in modal content and trends. Comparing the two transfer functions, it can be seen that the natural frequencies do not shift, indicating that there is no mass-loading from the shaker. There is an obvious factor of 10 gain in the transfer function due to a factor of 10 difference in sensitivity between the modal hammer and the force transducer used by the shaker.

2.3. Re-Active Passive performance testing

Several tests were performed on the panel with various passive, reactive and active configurations. Due to the large number of tests, only the most relevant results will be presented. The transmission loss of the panel was tested as follows:

1. Baseline configuration with only the piezoelectric actuators mounted.
2. Baseline with passive distributed vibration absorber.

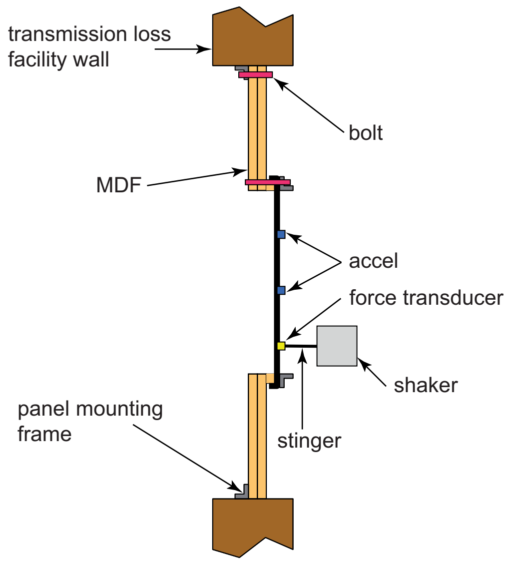

Fig. 1. Schematic of panel mounting configuration and modal testing setup.

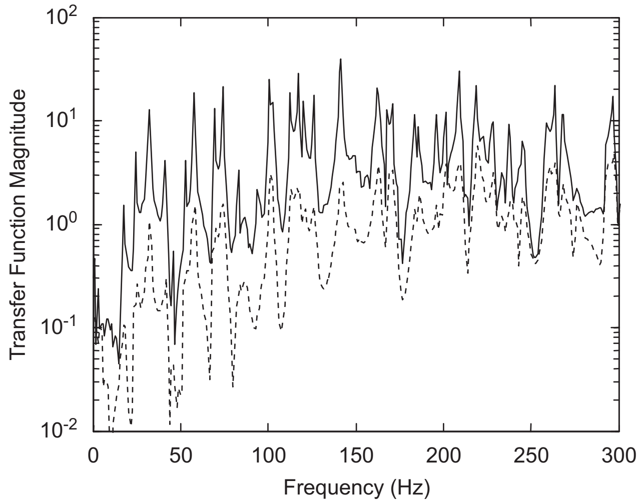

Fig. 2. Comparison of panel response due to hammer (solid) and shaker (dashed) excitation.

3. Distributed vibration absorber with passive constrained layer damping.
4. Constrained layer damping with least mean squares (LMS) adaptive feedforward control.

The specifics of the transmission loss testing and the controller configuration are now discussed.

2.4. Experimental procedure

2.4.1. Transmission loss testing

Since the frequency range of interest was 10–1000 Hz, and the cutoff frequency of the reverberation chamber (the frequency below which the chamber exhibits modal behavior) is approximately 300 Hz, anechoic inserts were placed on the incident and radiating chambers to approximate free field conditions. A schematic for the transmission loss (TL) test is presented in Fig. 3.

For this experiment, the incident acoustic field was provided by a speaker positioned inside an anechoic insert and adjacent to the panel at a distance of 0.25 m. This configuration has been shown to provide an effective approximation of a plane wave [11]. A broadband signal of 10–1000 Hz was input to the speaker providing excitation of the panel. Incident pressure measurements were taken by a single microphone positioned near the center of the panel. Radiated pressure measurements were taken by seven microphones positioned at several points on a hemisphere in an anechoic room. The hemisphere was divided into equal areas and one microphone was placed at the center of each area. From the microphone measurements and associated area, an approximation of transmission loss (TL) can then be calculated by

$$\mathrm{TL} = 10\log_{10}\left(\frac{\Pi_i}{\Pi_r}\right) \approx 10\log_{10}\left(\frac{p_i^2 A_i}{\sum_{r=1}^{N} p_r^2 A_r}\right), \tag{1}$$

where Π_i is the incident power, Π_r is the radiating power, p_i is the blocked pressure, p_r is the radiated pressure, A_i is the incident area, A_r is the partial area of the hemisphere covered by each microphone in the radiating field. All pressure measurements were processed by custom software written for a National Instruments data acquisition system where the auto-correlation and cross-correlation of the disturbance signal and the pressure measurements were computed. This information was saved on a PC compatible computer and analyzed using a MATLAB code that yielded the transmission loss data as per calculations detailed previously.

The sampling parameters are as follows: 4000 Hz sampling frequency, 4096 samples per average, 1000 Hz anti-aliasing filter and a Hanning time window. The speaker was excited with broadband random noise band-pass filtered from 10 to 1000 Hz.

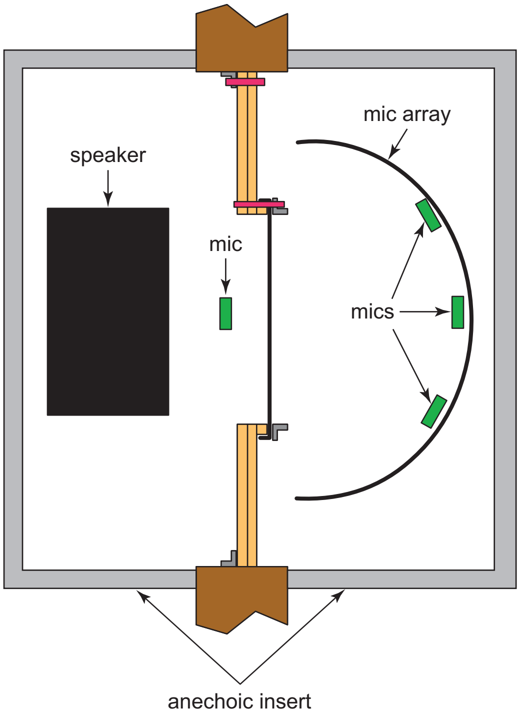

Fig. 3. Schematic of transmission loss testing configuration.

2.4.2. Feedforward controller configuration

To achieve active control, a feedforward LMS control algorithm was implemented using a 2 input 2 output configuration. The two inputs were microphones in the far-field microphone array used for transmission loss (TL) measurements. Two control channels were used: (1) the center Re-Active Passive actuator and (2) the left and right Re-Active Passive actuators wired in-phase. A reference channel was provided to the controller from the signal generator used to excite the speaker. A system identification over the frequency range of interest was performed prior to the control test.

3. Re-Active Passive device design

The Re-Active Passive device was designed to use three technologies packaged into one device to provide increased transmission loss of a panel covering a frequency range of 10–1000 Hz. Each technology is known to work for a specific frequency range: piezoelectric active control actuators for low frequencies (<200) Hz, distributed vibration absorbers for medium frequencies (75–250 Hz), and constrained layer damping for high frequencies (>200 Hz). By combining these technologies and packaging them into a single device, control over an extended bandwidth can be achieved. The individual design of each technology will now be discussed.

3.1. Piezoelectric active control actuators

The active actuator was made from two Active Control Experts (ACX) QP40 piezoelectric actuators. Each actuator was made from PZT material packages in a phenolic substrate with copper traces to provide actuation voltage. The actuators were bonded to the plate with a typical "five minute" epoxy. The three Re-Active Passive devices were positioned near the antinodes of the most efficient acoustic radiators, the (1,1), (3,1) and (5,1) modes of the plate, to achieve effective modal coupling. The modal decomposition of the plate, presented in the results section, indicates that these modes cover a frequency range from 20 to 60 Hz, which is the approximate design frequency range of the actuators (<150 Hz). A photograph of the Re-Active Passive devices mounted in the panel is presented in Fig. 4.

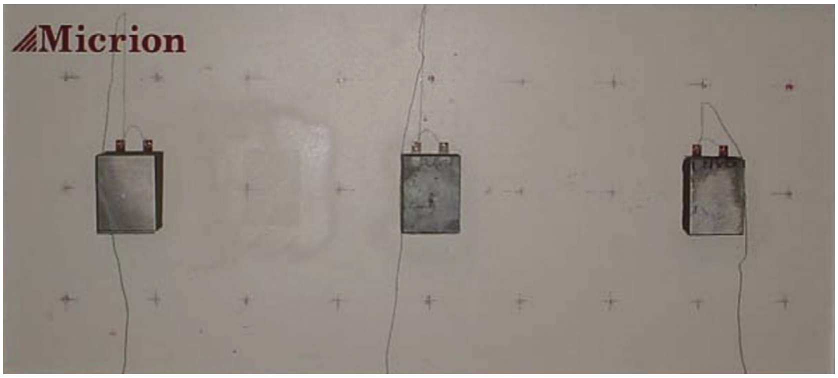

Fig. 4. Panel with three Re-Active Passive devices tuned to 60 (center), 72 (left), and 92 (right) Hz.

3.2. Distributed vibration absorbers

The distributed vibration absorbers are fabricated from metal plates of varying mass mounted to open cell foam [12]. These devices provide an optimally damped, easily manufacturable vibration absorber that can be made in any reasonable size and shape. Therefore, the distributed vibration absorber mass and foam were designed to cover the same area as the Active Control Experts' QP40 piezoelectric actuators. To further specify the design constraints, the distributed vibration absorbers part was designed to cover the frequency range of approximately 50–200 Hz, therefore the distributed vibration absorbers were tuned to the distributed frequencies of 60, 72, and 92 Hz. As will be shown the results section, the 72 Hz distributed vibration absorbers and the 92 Hz distributed vibration absorbers were tuned to specific modes of the panel near these frequencies to provide maximum reduction in vibration. A typical transfer function measured from a base to the mass accelerometers for the 92 Hz distributed vibration absorbers is presented in Fig. 5. Note the distributed vibration absorbers has a Q of about 16 dB.

3.3. Viscoelastic constrained layer damping

The viscoelastic constrained layer damping part of the Re-Active Passive device was made from 3 M 112P05 material which is a 1.6 mm (1/16″) thick tar-like material with a 0.1 mm thick aluminum sheet attached. The device was made to cover the same surface area as the ACX QP40 actuators. This particular material was chosen since the thickness of the material was best suited for low-frequency damping control, i.e. >150 Hz. A picture of the 3 M 112P05 on the panel is shown in Fig. 6.

4. Experimental results

4.1. Modal testing experimental results

Modal testing of the panel was performed to identify the mode shape and natural frequencies of the panel. This information was then used to determine to location(s) of the Re-Active Passive devices to obtain effective control over the bandwidth, as well as to design the resonant frequencies of the Re-Active Passive distributed vibration absorber natural frequencies. The autospectra of the accelerometers from the modal test were previously presented in Fig. 2. As can be seen in the figure, there are 11 modes below 100 Hz with the (1,1) mode being at approximately 19.5 Hz. The efficient acoustic radiators, the (1,1), (3,1) and (5,1) modes have frequencies of 19.5, 35.6, and 63.2 Hz, respectively, which determined the placement of the piezoelectric actuators as previously discussed. A comparison of the experimental and theoretical [13] modal frequencies is presented in Table 1. The theoretical natural frequencies of the panel were calculated using the plate dimensions and properties given previously, with clamped boundary conditions. As can be seen in the table,

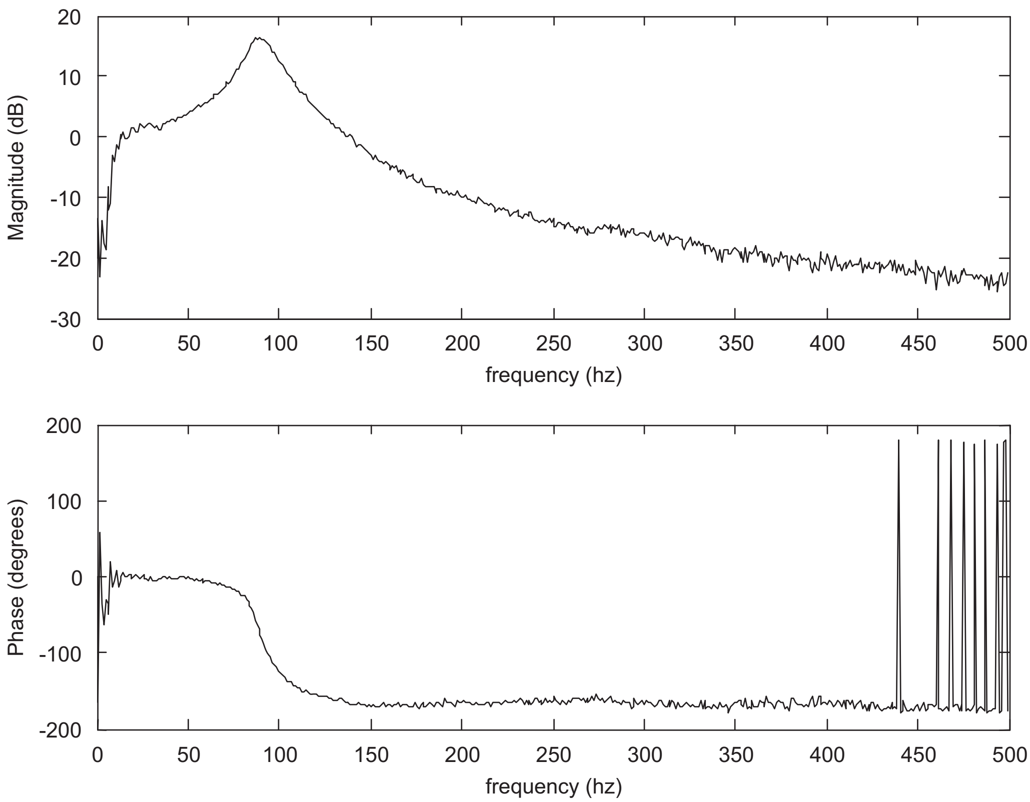

Fig. 5. Frequency response of 92 Hz distributed vibration absorber.

the experimental frequencies agree well with the theoretical frequencies indicating that the boundary conditions of the plate act as expected, like clamped boundary conditions.

4.2. Transmission loss experimental results

The transmission loss results of the panel with the various Re-Active Passive technologies are presented. Transmission loss (TL) was calculated as previously presented in Eq. (1). Note that peaks in the autospectra of the panel vibration response will be minima in transmission loss response, and increases in transmission loss will be increases in the minimum values.

A comparison of the transmission loss between the panel baseline configuration (Baseline) and the panel with distributed vibration absorbers is presented in Fig. 7. As can be seen, there are transmission loss minima at approximately 20, 38, 72 and 92 Hz, which corresponds to the frequencies of the efficient acoustic radiators of the (1,1), (3,1), (5,1) and (6,1) modes. The distributed vibration absorbers were specifically tuned to the (5,1) and (6,1) mode frequencies of 72 and 92 Hz, while the third was tuned to 60 Hz. As can be seen, the distributed vibration absorber acts as a rigid mass below their natural frequency by shifting the (1,1) mode to a lower frequency. Specifically, the (1,1) mode is moved from 21 to 18 Hz. It is interesting to see that the transmission loss of the (3,1) mode is increased significantly from 5 to 18 dB, which is due to the highly damped 60 Hz distributed vibration absorber. As seen in Fig. 5, the distributed vibration absorber has a broad resonance peak due to high damping, and can have an effect at $\pm 40\%$ of its tune frequency. Therefore, the 60 Hz distributed vibration absorber can affect the panel response at 38 Hz.

Above the natural frequency, the distributed vibration absorbers have a more pronounced effect. By distributing the resonance frequencies and utilizing high damping ratios, the distributed vibration absorber increased the transmission loss of all of the panel resonances from 60 to 150 Hz. The overall increase in

Fig. 6. Viscoeleastic constrained layer-damping material (3 M 112P05).

Table 1
Comparison of theoretical and experimental modal frequencies

Mode order	Frequency (Hz) Experimental	Frequency (Hz) Theoretical
1,1	19.5	21.0
2,1	26.8	25.8
3,1	35.6	34.4
4,1	45.5	47.1
1,2	55.4	55.3
2,2	–	60.2
5,1	63.2	63.6
3,2	–	68.4
2,4	–	80.3
6,1	92	96

broadband (15–1000 Hz) transmission loss was 4.7 dB. The distributed vibration absorber's added 0.15 kg to the panel mass of 6.0 kg, or approximately 3%.

As seen in Fig. 8, the effect of adding 3 M 112P05 constrained layer damping to the distributed vibration absorber's was to reduce several of the transmission loss minima above 150 Hz. In the figure, the axes have been plotted on a linear scale and the scales have changed to more clearly illustrate the effect of the constrained layer damping. There was minimal effect of the constrained layer damping treatment below 150 Hz due to low strain (and therefore is not shown). From 150 to 500 Hz, the strain was sufficient and the

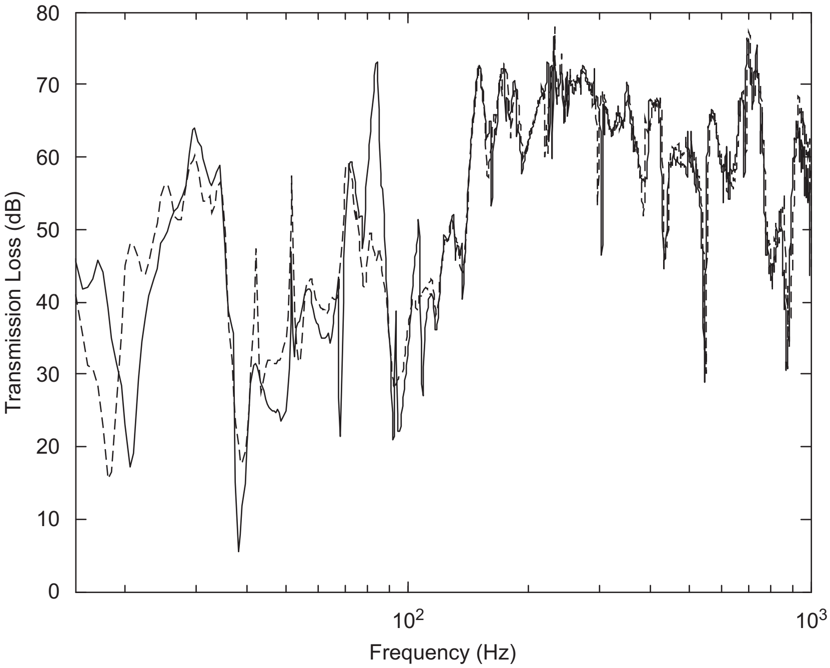

Fig. 7. Transmission loss of panel (solid) compared to panel with three distributed vibration absorbers (dashed) tuned to 60, 72, and 92 Hz mounted on center, left, and right actuator, respectively.

actuators were positioned such that increases in transmission loss of 4–8 dB are seen. Above 500 Hz, the constrained layer damping treatment had little effect (and therefore is not shown) since the positions of the devices were not optimized over that frequency range.

The overall increase in broadband (15–1000 Hz) transmission loss was −0.6 dB compared to the distributed vibration absorber's. This result is expected since the average power radiated by the panel is dominated by the radiated power in the low-frequency region and the effect of the constrained layer damping is minimal in this range. The distributed vibration absorber + constrained layer damping treatment reduced the overall broadband transmission loss by 4.1 dB compared to baseline. The constrained layer damping treatment added 15 g of weight to each actuator for a total of 45 g to the panel.

The effect of adding the LMS adaptive feedforward control is presented in Fig. 9. The feedforward controller was able to increase the transmission loss of the (1,1) mode by 18 dB (from 18 to 36 dB) at 18 Hz. For the (3,1) mode, the controller increased transmission loss 14 dB (from 16 to 30 dB) at 39 Hz. The controller was able to reduce the rest of the modes by approximately 10 dB in the range of 45–100 Hz. Note that there was slight spillover in the 66–82 Hz range. The overall increase in broadband (15–1000 Hz) transmission loss was 5.4 dB compared to the distributed vibration absorber + constrained layer damping test. The QP40 actuators added 30 g of weight to each actuator for a total of 90 g to the panel, not including control electronics.

Finally, the effect of the Re-Active Passive device is compared to the baseline panel in Fig. 10. As can be seen, the Re-Active Passive device has increased transmission loss over a frequency range of 15–300 Hz. To extend reductions to 1000 Hz, an optimization should be run to determine the best locations to cover that range. As stated previously, the constrained layer damping treatment would not be effective on even modes since all three Re-Active Passive devices were placed on the antinodes of the low-order odd modes. However, the current actuator placement performed effectively in achieving this goal. The overall increase in broadband (15–1000 Hz) transmission loss was 9.4 dB compared to baseline as seen in Table 2. The Re-Active Passive devices added a total of 285 g to the panel mass of 6.0 kg, or approximately 5%, not including control electronics.

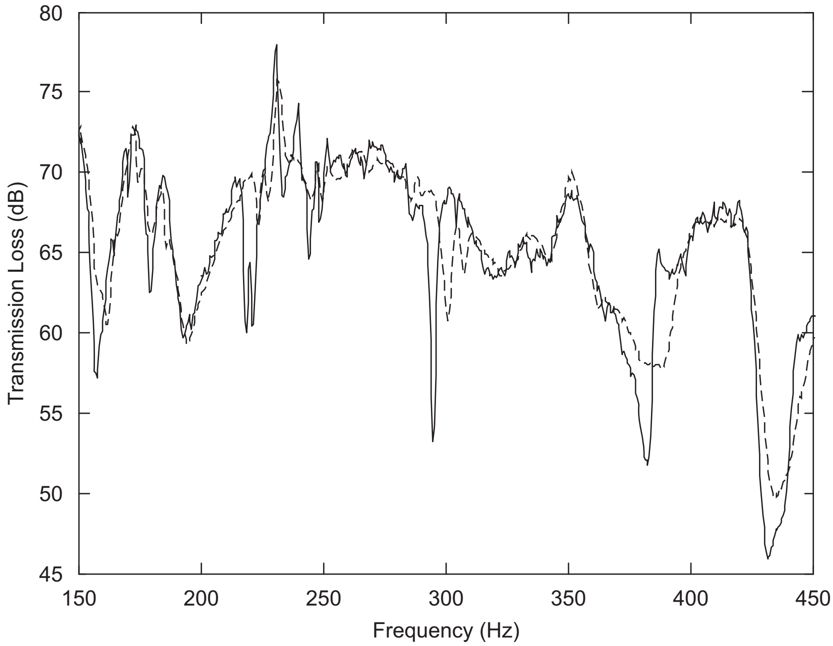

Fig. 8. Transmission loss of distributed vibration absorber panel (solid) compared to same with visco-elastic constrained layer damping (CLD) material added (3 M 112P05) (dashed).

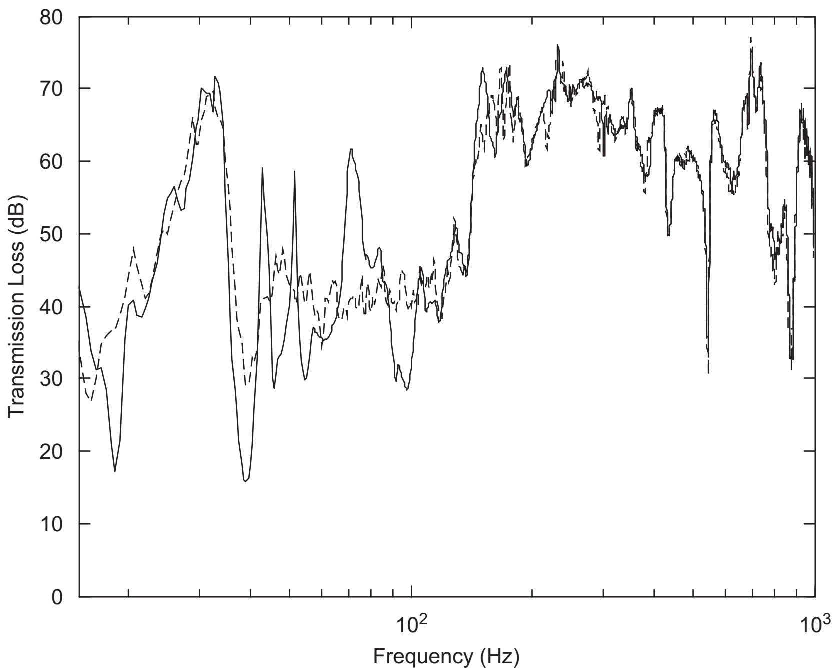

Fig. 9. Transmission loss of distributed vibration absorber + constrained layer damping panel (solid) compared to same with 2I2O LQG feedback controller using 2 microphones as error sensors and 2 control actuators (1 ACX QuickPack40 mounted in center of panel and 2 QP40 ganged together at mode 3 antinodes) (dashed).

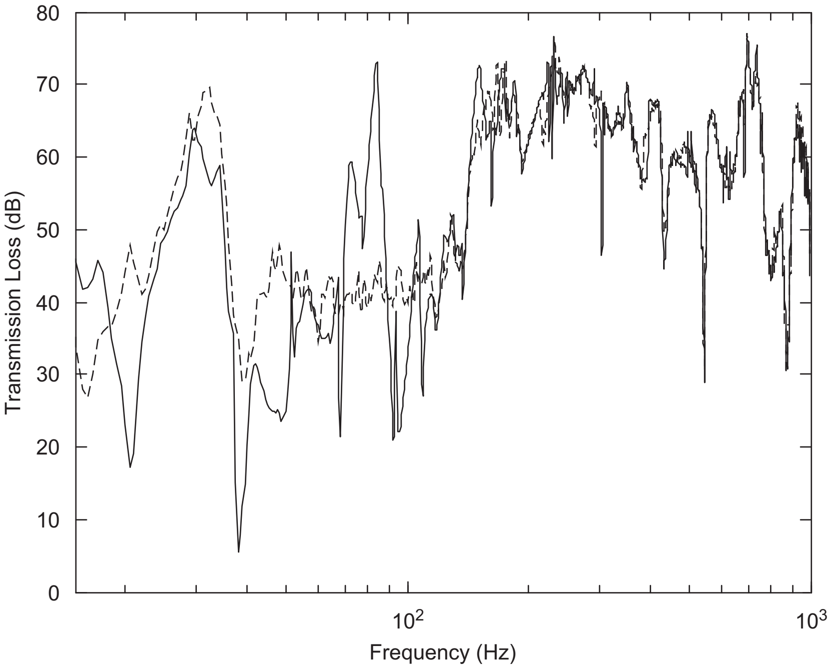

Fig. 10. Transmission loss of panel (solid) compared to panel with Re-Active Passive device (dashed).

Table 2
Broadband transmission loss from 15 to 1000 Hz

Panel configuration	Increase in broadband transmission loss (15–1000 Hz)
Baseline	–
DVA	4.7
DVA + CLD	4.1
RAP (DVA + CLD + Active)	9.5

5. Conclusions

Re-Active Passive devices were designed and tested to increase transmission loss (TL) of a panel mounted in a transmission loss test facility. The cumulative effect of the individual technologies on transmission loss of a panel was measured. Individually, the distributed vibration absorber, constrained layer damping, and active control technologies reduced the transmission loss in the frequency range where they were most effective. Together, the Re-Active Passive device delivered performance over a broader range of frequencies than either technology alone. Active control was applied to the low-frequency range (< 150 Hz) and worked quite well due to low modal density of the structure. When the modal density increases, distributed vibration absorber's were effective by adding dynamic mass to the structure. Once you increase the frequency range above 150 Hz, constrained layer damping became effective. Overall, the Re-Active Passive device increased broadband (15–1000 Hz) transmission loss by 9.4 dB. The three Re-Active Passive devices added a total of 285 g to the panel mass of 6.0 kg, or approximately 5%, not including control electronics.

Acknowledgments

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