Proceedings of IMECE04
2004 ASME International Mechanical Engineering Congress and Exposition
November 13-20, 2004, Anaheim, California USA

IMECE2004-61205

WEIGHT OPTIMIZED STRUCTURAL AND ACOUSTIC ACTUATORS FOR THE CONTROL OF SOUND TRANSMISSION INTO ROCKET PAYLOAD COMPARTMENTS

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ABSTRACT

The reduction of sound transmission into rocket payload compartments is a challenging application for active control due to the broadband nature of the disturbance, the large structural and acoustic space and the very high acoustic levels required. The exterior acoustic field that drives the payload fairing at liftoff is typically in the order of 145dB and the active control system must be able to counteract this high drive level using lightweight actuators. This paper is concerned with the development of structural and acoustic actuators for this application with the emphasis on maximum output level in the 60-200Hz bandwidth for a given actuator weight.

The electromagnetic structural actuators are based on powerful rare earth magnets in a two degree of freedom arrangement. It is shown that a two degree of freedom arrangement allows the output in the bandwidth of interest to be increased over a simple one degree of freedom arrangement. The design is termed a distributed active vibration absorber or DAVA as the second degree of freedom is provided by a light and distributed foam element that allows easy attachment and low stress concentration on the structure. The two degree of freedom arrangement also acts as a natural low pass filter to naturally remove unwanted spillover at higher frequencies.

The acoustic component is also based on powerful rare earth magnets, however the two degree of freedom arrangement used for the structural actuator is no longer of interest. The main concern is in the reduction of the speaker and cabinet weight. It is shown that careful design of the speaker and cabinet can lead to large reductions in weight without loss of performance.

Data taken from an active control experiment on a large composite cylinder, coupled with data from the characterization of the actuators will be used to determine the total actuator weight needed for control in a typical launch environment.

INTRODUCTION

The motivation behind this work is the reduction of sound transmission into rocket payload fairings. High noise levels during rocket launch [1] (typically >140dB) can cause damage to payloads and is in fact the cause of many first day payload failures. The adoption of light composite materials for fairing construction has led to a tremendous increase in the acoustic energy transmitted into the payload compartment and exacerbated this problem. Much of the fairing interior noise is at low frequencies where the attenuation provided by absorptive blankets is poor. Virginia Tech in co-operation with the Boeing, VaSci and the Air Force have been developing both passive [2],[3] and active control systems [4] in order to reduce the low frequency sound transmission into payload compartments using structural and acoustic control devices. Active control has been previously been investigated as a solution to this problem [5],[6],[7] but there are few experimental studies using realistic drive levels, fairing structures or control actuators.

Both feedback [7] and feedforward [4] control architectures have been investigated as potential control solutions. In general feedback controllers are limited to improving the effective damping in the system and perform similarly to passive Helmholtz resonator treatments [8]. On the other hand feedforward control architectures are limited by the availability of suitable reference signals, although this problem may be overcome through the use of a large number of external reference microphones [9].

Both feedforward and feedback control systems are also limited by the availability of the lightweight and powerful actuators required to counteract the high noise levels experienced at launch. This paper outlines the development of both structural and acoustic actuators for the active control of sound transmission into a payload compartment. Both devices are tested on a large cylinder, constructed in a similar manner to modern payload fairings, and the total actuator mass required for good active control is calculated based on active control experiments using a multichannel feedforward active control system.

ACTUATOR DEVELOPMENT

In this section the development of both structural and acoustic actuators will be presented. The performance metric for both of these actuator types is the maximum output in the 60-200Hz range divided by the actuator mass.

Structural Actuator Development

In order to create an effect structural actuator both the active source and the passive dynamics of the actuator need to be considered. This subsection will first outline the design and optimization of the active component or "motor" and then detail the development of a 2 degree of freedom arrangement that optimizes the passive dynamics of the actuator.

Lightweight motors

In order to control the sound transmission into a payload compartment the actuators must be of an inertial type as there is no "ground" available for the actuator to react against. Therefore inertial actuators require an inertial mass against which to react and this is typically designed as the heaviest component in the system, namely, the magnet. This also implies that the force output at low frequencies will always be limited by the natural frequency of the inertial mass (with support spring) below which the active force falls off.

Figure 1 shows both a schematic and a picture of the inertial actuator design used. The magnet is suspended inside a thin steel casing using two spiders at either end. This arrangement allows the relatively heavy magnet to be supported rigidly in the middle of the casing while allowing motion in the axial direction. This also ensures that most of the actuator mass is part of the moving portion of the actuator (i.e. inertial mass). Unlike speakers which have a single flux gap this design has two flux gaps with two electrical coils wound in opposite directions (i.e. magnetic flux crosses in opposite directions in the two gaps). A current running through the coil creates an opposing axial force between the casing and the suspended magnet. In order to improve efficiency, a rare earth magnet (NdFeB) whose energy product (J/m^3) can be over ten times larger than typical ferrite magnets, was used. The casing can then be attached to a structure in order to supply a force to the structure. The actuator will only be effective above the mass spring resonance of the device.

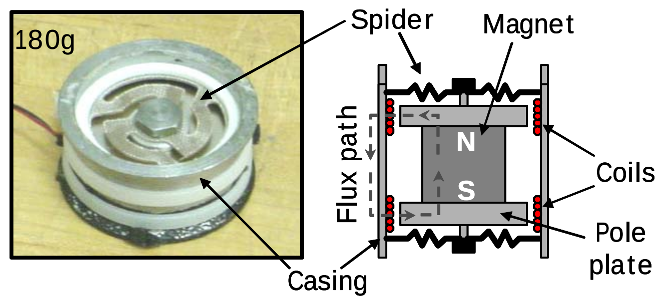

Figure 1: Picture and schematic of a lightweight inertial actuator (motor)

Figure 2 shows both a schematic of the measurement apparatus and the measured blocked force output of this device. The device was clamped between two rigid boards one with a hole cut in the middle large enough to allow freedom for the moving parts of the actuator but small enough such that the casing remained clamped. The motion of the magnet and core was then monitored using an accelerometer. The force output of the actuator is then determined by multiplying the moving mass by the acceleration. This force output per unit input voltage is plotted in dB on the graph. As can be seen the actuator output is characterized by a single resonance at approximately 100Hz below which the output falls off. This natural frequency can be adjusted by changing the thickness, and hence stiffness, of the spiders.

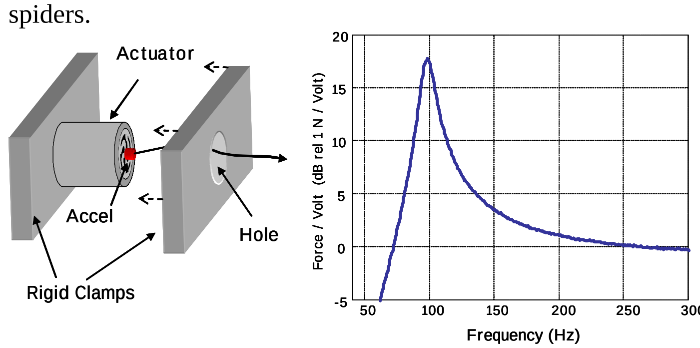

Figure 2: Measurement setup and resulting measured blocked force output of actuator

2DOF Passive arrangement

Previous work by Burdisso and Heilmann [10][11] demonstrated that the output of an actuator over a limited bandwidth could be improved using a two degree of freedom arrangement. Specifically, the design consisted of an active element positioned in between two reaction masses, which are attached to a primary structure through elastic elements. In this work the lightweight motors were integrated into the passive acoustic foam typically used for high frequency noise control inside of fairings. This is achieved by attaching the motor to a thin lightweight and stiff panel and then attaching this panel to the acoustic foam layer. The actuator therefore acts on the structure through the extra single degree of freedom system (i.e. mass layer and foam stiffness). A schematic and a block diagram representation of this design are shown in figure 3. This device is termed a Distributed Active Vibration Absorber or DAVA. This name is an extension of the similarly designed Distributed Vibration Absorber (or DVA) used for passive vibration control [8].

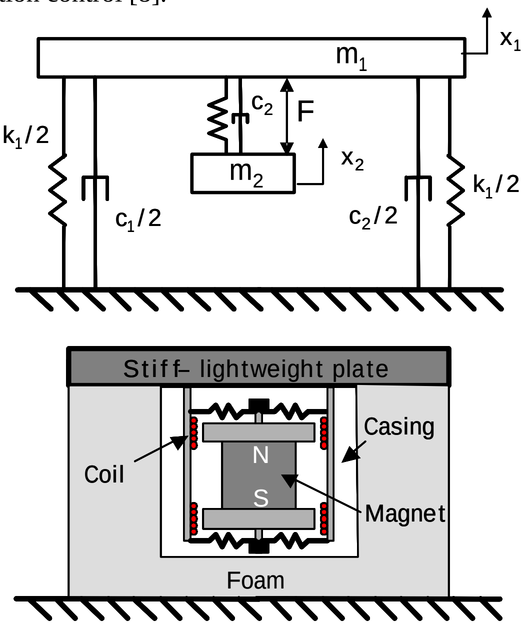

Figure 3: Free body diagram and schematic of a DAVA

The force F_{DAVA} transmitted to the structure is directly related to the motion of mass m_1 through the stiffness k_1 and damping c_1 provided by the foam layer [12]. The overall force transmitted to the based due to the force F from the actuator (proportional to input current) is therefore given by,

$$\frac{F_{DAVA}}{F} = (k_1 + j\omega c_1)\left[K + j\omega C - \omega^2 M\right]^{-1}\begin{bmatrix} 1 \\ -1 \end{bmatrix} \quad [1]$$

where the matrices K, C and M are given by,

$$M = \begin{bmatrix} m_1 & 0 \\ 0 & m_2 \end{bmatrix} \quad C = \begin{bmatrix} c_1 + c_2 & -c_2 \\ -c_2 & c_2 \end{bmatrix} \quad K = \begin{bmatrix} k_1 + k_2 & -k_2 \\ -k_2 & k_2 \end{bmatrix} \quad \textbf{[2]}$$

In order to optimize the output of the device over the frequency range of interest a cost function E was created that is defined as the square root of the integrated transfer function squared of the frequency range of interest ($f=60Hz$ to $200Hz$), or effectiveness, divided by the total mass of the DAVA.

$$E = \frac{\sqrt{\int_{60}^{200} \left| \frac{F_{DAVA}(f)}{F(f)} \right|^2 df}}{m_1 + m_2} \quad \textbf{[3]}$$

The damping was chosen to be representative of the damping in the motor and the damping typical of the acoustic foam. This cost function could then be plotted over a range of design mass and stiffness (m_1 and k_1).

Figure 4 shows the cost function (Effectiveness/mass) for a range of plate mass and foam stiffness. It would appear from this plot that the optimal mass is zero along with a high stiffness (i.e. actuator directly mounted to structure). However the casing of the actuator has a mass of approximately 70g that is not part of the moving mass and hence the minimum mass m_1 is 70g. This implies that an intermediate mass spring layer, designed to lie along the diagonal ridge in Figure 4, will enhance the performance of the actuator within the frequency range of interest.

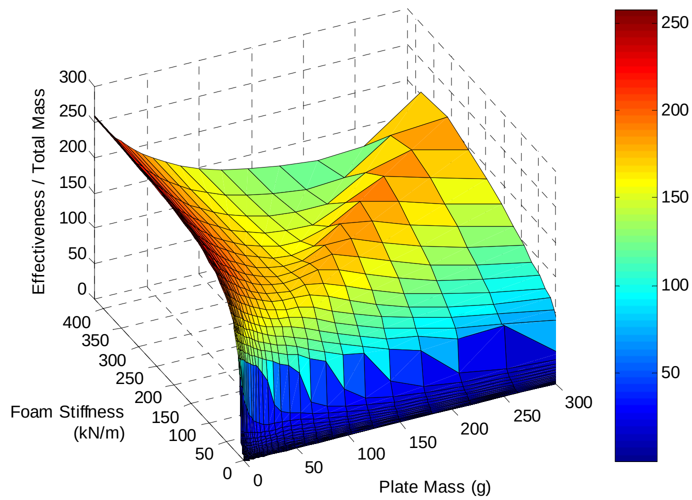

Figure 4: The cost function *E* (or effectiveness/mass) for a range of stiffness and mass values

In practice the intermediate mass spring system has an uncoupled natural frequency near to the top of the frequency range of interest where the motor had a natural frequency near the bottom of the frequency range of interest. Coupling these two systems together pushes the natural frequencies apart such that the two natural frequencies span the frequency range of interest. The final design chosen used a 260g total mass m_1 and was in practice a 9.5'' by 9.5'' honeycomb panel mounted on 2'' of lightweight melamine acoustic foam [12]. Figure 5 shows a picture of this device and Figure 6 shows the measured and predicted force output of the DAVA system. It is interesting to compare Figure 6 with Figure 2 where both the output is improved and the response outside the bandwidth of interest falls off. This means that the intermediate layer acts as a natural lowpass filter that reduces any spillover from the actuators at higher frequency.

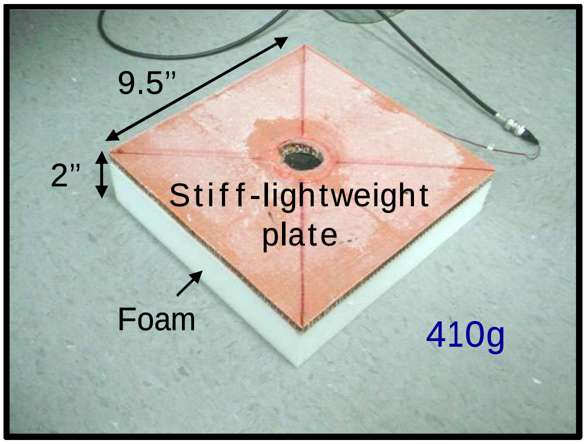

Figure 5: Picture of the final DAVA design using a 9.5" square honeycomb plate and 2" thick melamine foam.

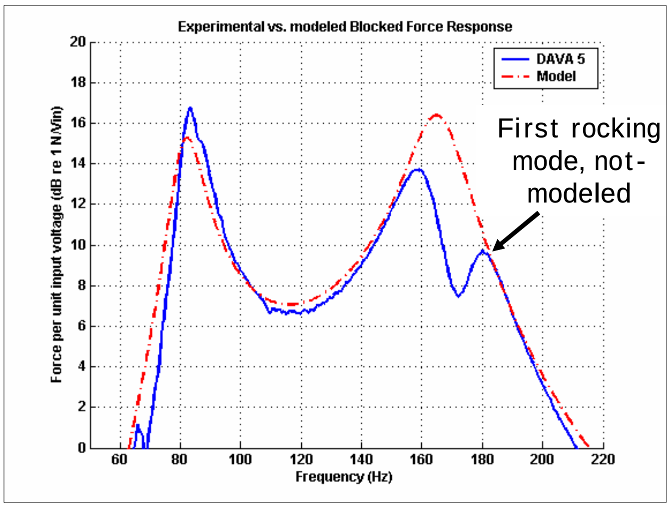

Figure 6: Measured and predicted actuator output.

Acoustic Actuator Development

In order to develop a lightweight acoustic actuator both the speaker design and the enclosure design must be considered. The strategy pursued was to take a typical speaker system using a reference 12'' Pioneer loudspeaker mounted in a typical speaker cabinet (3/4'' Medium Density Fiberboard) and to redesign the system in order to make it lighter without loss of performance.

Lightweight speaker

Figure 7 shows a schematic of a typical loudspeaker. The lightweight diaphragm is connected to an electrical coil that sits in a single flux gap. The distribution of weight in the reference speaker is dominated by the weight of the ring magnet and the steel pole pieces (or yoke assembly) that make up the flux circuit (see Table 1). The rest of the weight comes from the frame assembly that is used to hold the magnet in place and whose mass is dependent on the mass of the magnet which it supports. Therefore the mass can be substantially reduced by changing the magnet from a ferrite material to a rare earth neodymium magnet (shown as a centrally located cylindrical magnet) with smaller pole pieces.

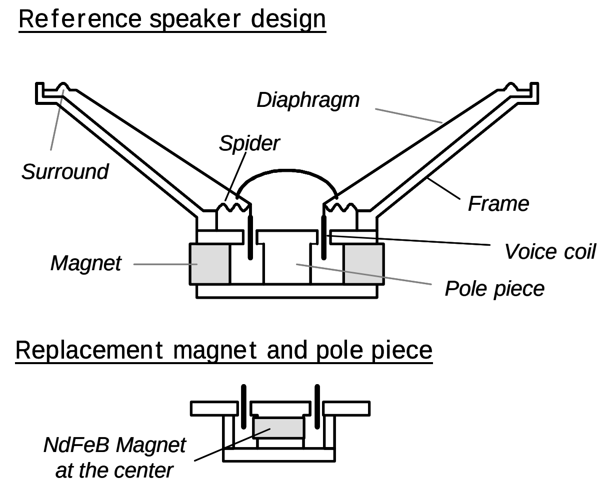

Figure 7: Schematic of a typical loudspeaker and a suggested replacement magnet and pole piece

Component	Weight (g)
Magnet	852
Magnet assembly (magnet + yoke)	2380
Moving parts (voice coil + diaphragm)	61
Frame (including moving parts)	685
Total	3065

Table 1: Weight distribution in the 12" Pioneer reference loudspeaker

The objective was to find a magnet that maximized a cost function similar to that in equation 3 where instead of force output the term to be integrated was pressure output due to voltage input. It was also necessary to consider the mass of the yoke assembly in the choice of magnet dimension. A model of the speaker system was developed [13] by considering the mechanical, electrical and acoustic radiation components (see Figure 8) of the system. The pressure output due to an input voltage is given by,

$$\frac{P(\omega)}{V(\omega)} = \frac{-\omega^2 \rho \left(\frac{a^2}{4r}\right)\left(\frac{Bl}{R_E}\right)}{-\omega^2 M + j\omega\left(b + \frac{B^2 l^2}{R_E}\right) + \left(k_S + k_B\right)} \quad \textbf{[4]}$$

Where ρ is the density of air, a is the radius of the diaphragm, r is the distance from the speaker, B is the flux density in the flux gap, l is the length of coil in the flux gap, R_E is the resistance of the coil, M is the moving mass of the diaphragm and coil, k_S is the stiffness of the diaphragm supports (spider and surround) and k_B is the stiffness provided by the speaker cabinet (or Box). The flux density B can be calculated as,

$$B = \sqrt{-\frac{\mu_o (BH) V}{V_g}} \quad \textbf{[5]}$$

Where μ_0 is the permeability of free space, (BH) is the maximum energy product of the magnet, V is the volume of the magnet and V_g is the volume of the flux gap. Equation 4 assumes that the speaker is baffled and radiates as a monopole (which is a reasonable assumption in the low frequency range of interest).

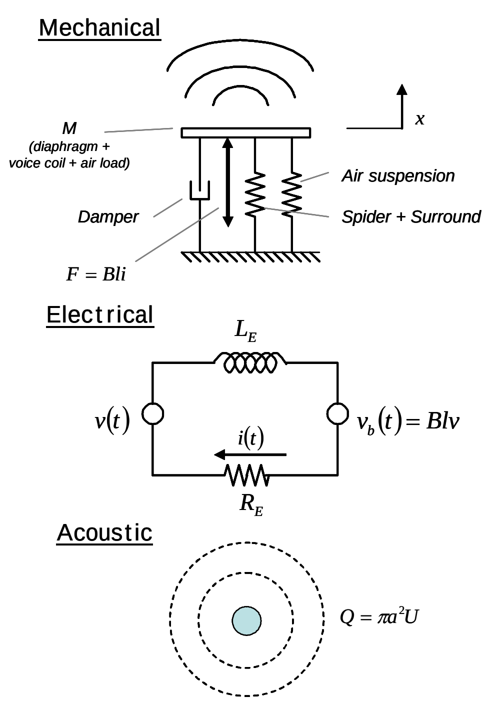

Figure 8: The mechanical, electrical and acoustic components of a loudspeaker. *Q* is the volume velocity of the source.

Using equation 4 and assuming all other parameters from the reference speaker remained the same, the size of the magnet (and hence magnet assembly) were investigated in order to maximize the sound pressure output over the frequency range of interest. A cylindrical magnet close to the optimal was chosen and used to replace the standard ferrite magnet.

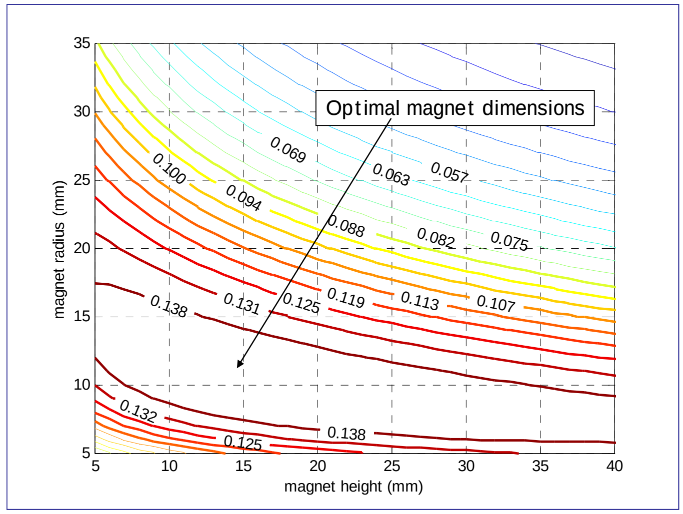

Figure 9: Contour plot of the cost function used to optimize the dimensions of the magnet

A picture of the new magnet and the old reference magnet assemblies are shown in Figure 10. The new assembly weighed 755g, a reduction by a factor of 3.2. These speakers were placed in a 1ft^3 rigid cabinet and placed in an anechoic chamber. The sound radiation from each speaker was monitored using a *1m* sphere of microphones. The averaged squared transfer function from the speaker input voltage to the output of each microphone was monitored and is also plotted in Figure 10. As can be seen, the output from the new speaker is actually 2.8dB higher (averaged across the bandwidth of interest). Both speakers have a similar maximum input voltage ($30V_{RMS}$) and under these conditions the new speaker still performs 1.7dB better than the reference speaker. It should also be noted that the new speaker, because of its slightly higher B value, is also flatter across the bandwidth of interest. This is because the B term influences the back EMF and hence damping term in equation 4.

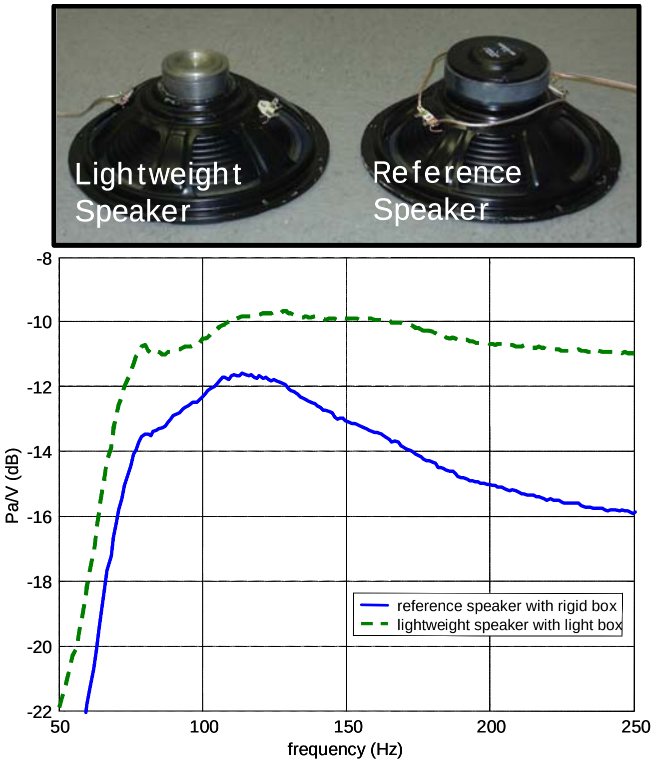

Figure 10: Picture and performance of the output of the new lightweight speaker and the reference speaker.

<u>Lightweight cabinet</u>

At low frequencies it is essential that the speaker be effectively baffled in order to efficiently radiate sound and any compliance in the speaker cabinet will lead to a reduction in speaker performance. In related research [13] it has been shown that very lightweight Helmholtz resonators, made from 0.6mm thick plastic tubes, can achieve near ideal rigid performance. Similarly the speaker cabinet can be made of very lightweight material (Figure 11) without substantial loss of performance. As an example a speaker cabinet made from very lightweight honeycomb material and stiffened with braces was constructed. The reference speaker performance in the lightweight 0.85Kg cabinet was compared against the performance of the same speaker made from ¾'' Medium Density Fiberboard (7.4Kg). The performance was measured in an anechoic chamber using the spherical array of microphones and is presented in Figure 11 . The lightweight box showed a performance loss of only 0.2dB while achieving large reductions in mass.

The final actuators design including the magnet, yoke, diaphragm, support structure and speaker cabinet weighed only 2.29Kg.

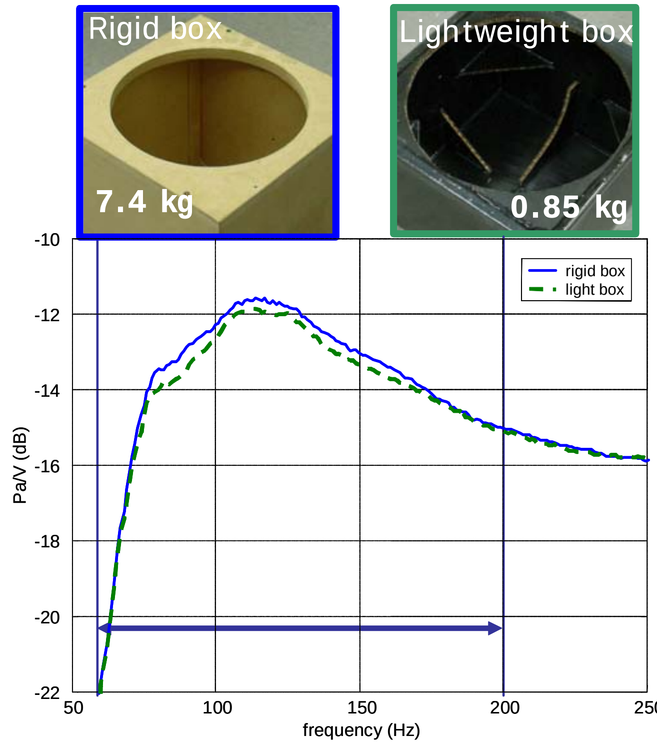

Figure 11: Performance of the reference speaker in the lightweight honeycomb box compared with a rigid box.

ACTIVE CONTROL EXPERIMENTS

The focus of this research was to experimentally determine the actuator weight that would be required to achieve good active control in a rocket payload compartment using realistic systems and structures. In order to achieve this two active control results will be briefly described here but are presented in more detail in other publications [4],[12], [14]. It is of course very difficult in practice to achieve the external drive levels experienced at rocket launch and an unbiased extrapolation of the maximum drive levels is required.

Two different active control experiments, one using just DAVAs and one using DAVAs and speakers, were conducted on the same cylinder that was built using a sandwich composite design similar to modern payload fairings. The cylinder was constructed by Boeing and is housed at Virginia Tech (see Figure 12).

The two experiments presented here were conducted a year apart but used the same control architecture with slightly different primary excitation and actuator configurations. A multichannel feedforward active control system [15] developed at Virginia Tech was used to drive the actuators in order to minimize the sound at an array of 12 internal microphones. Figure 13 shows a feedforward control architecture where the reference signal x is used to drive a set of actuators (plant) through a set of control filters H. The error signals are the output signals from the error microphones which are the linear combination of the noise from the primary disturbance *d* and the sound from the actuators *z*. The error signal signals and the reference are then used to find the optimal control filters. In the experiments, reference signals were taken from both an external microphone (as shown in Figure 13) and also the drive signal to the external speakers. Three additional microphones, not included in the control loop, were used to monitor the pressure away from the error sensors and the results presented here are the average over all 15 internal microphones (12 error, 3 monitor).

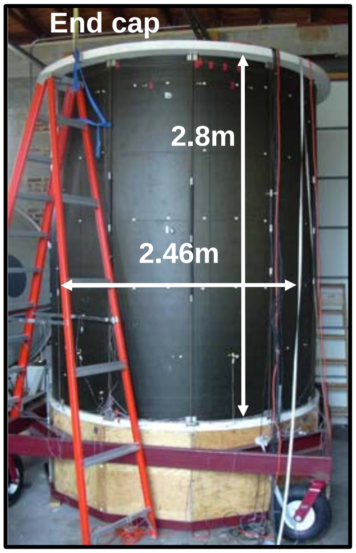

Figure 12: Boeing Composite cylinder housed at Virginia Tech.

Active Control Using DAVAs

In the first experiment eight DAVAs weighing approximately 3kgs were attached to the inside of the cylinder (no speakers). The cylinder was placed outdoors between two large banks of speakers set either side of the cylinder. The speaker banks were driven with incoherent bandpassed white noise. With good reference signals the 8 DAVA treatment was able to achieve over 8.5dB of control over the 60-200Hz range and was able to maintain good control up to the maximum external drive level of 131dB (measured at the cylinder's external surface). The DAVA input voltage during control at these levels was approximately $10V_{RMS}$ and the actuators were shown in the lab to be able to perform up to a level of $20V_{RMS}$ before becoming non-linear. Assuming a 6dB increase in level per doubling of input voltage this implies an external level of approximately 137dB could be controlled using this actuator treatment. However in practice the value is typically somewhere between 3dB (doubling of power) and 6dB (doubling of pressure).

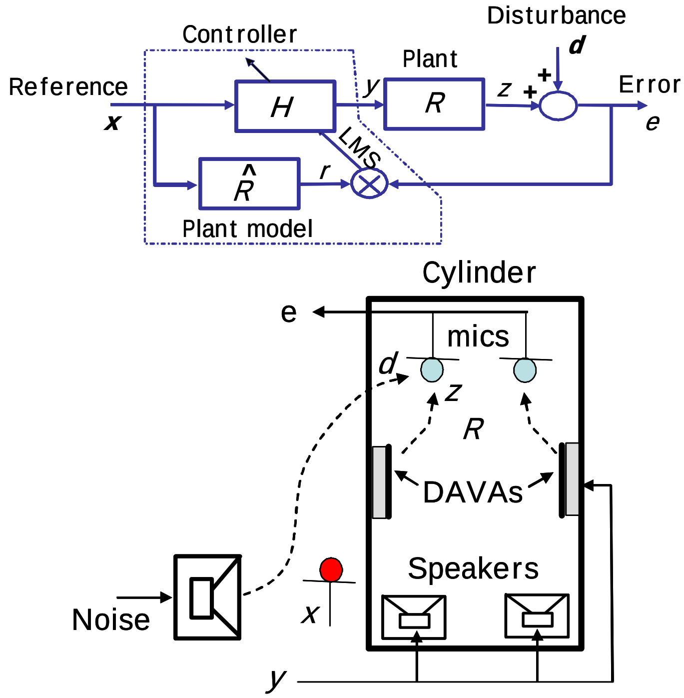

Figure 13: Feedforward active control system applied to the test cylinder

Active Control Using Speakers

A similar experiment was conducted on the cylinder using four DAVAs and four speakers. The cylinder in this case was housed indoors and driven using a single primary speaker. The objective of the test was to investigate the performance of this hybrid system [14] and to determine the mass of a speaker based treatment. The sound reductions achieved using the hybrid speaker/DAVA system was 9.1dB in the 60-200Hz range using an ideal reference and is comparable to the performance achieved using a purely 8 DAVA system.

In this case the maximum controllable external field was calculated differently than that of the previous test. The following is the experimental procedure followed in order to give an accurate measure of the maximum controllable external levels using only loudspeakers (since the disturbance is not powerful enough to achieve ≈140dB externally):

1. The control system using just 4 speakers was allowed to converge with relatively low external drive levels
2. The exterior sound level was measured (in this case at 94.4dB)
3. The controller was locked and the primary speaker turned off (i.e. control speakers still driving at same level inside cylinder)
4. Interior sound level at microphone array recorded
5. All control speaker levels increased simultaneously up to their maximum drive levels ($30V_{RMS}$). The increase

in interior sound level was monitored (+38.7dB in this case) i.e. headroom

6. Maximum external controllable level calculated by adding the headroom to the initial exterior sound level (94.4+38.7=133.1dB)

This method takes into account the performance of the speakers as they *act together with the same relative drive signals* as would be necessary during active control. In addition the final speaker design presented in Figure 10 has a maximum output level 5dB better (as measured in the anechoic chamber) across the bandwidth of interest than the prototype speakers used in the experiments. This implies that approximately 138dB could be controlled using the final speaker design which weighed 9.2Kg.

CONCLUSIONS

Lightweight DAVA actuators and lightweight speakers (with light cabinets) can be designed using rare earth magnets in order to greatly reduce actuator weight. The designs presented here were optimized in order to maximize output in the 60-200Hz frequency range. These designs were built and tested in the lab. Their maximum performance authority was then tested using a large cylinder constructed in a similar manner to modern payload compartments. It was shown that DAVAs provided a more lightweight treatment than speakers but there is further weight reduction to be achieved on the speaker cabinet, by using a cylindrical enclosure, and on the speaker support frame. The actuator weight required to achieve control of 145dB external levels is within the 10-20Kg range for a fairing of this size. The construction of this cylinder is extremely light [8] with a total mass of only 75Kg and this may explain why structural actuators perform so well. With heavier construction, the transmission loss increases and it would be expected that the interior levels would fall and reduce speaker requirements.

ACKNOWLEDGMENTS

We would like to acknowledge Boeing for the support of this work and more broadly VaSci and the Air Force for the support of this program area.

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