

Global control of sound radiation from a plate using several adaptive vibration neutralisers with local control schemes

M. R. F. Kidner, R. I. Wright

Vibration and Acoustics Laboratory, Virginia Polytechnic Institute and State University, Blacksburg, Virginia, 24061

ABSTRACT

Adaptive tuned vibration neutralisers are common solutions to controlling a single but variable frequency disturbance, such as the interior sound field in a turbo prop aircraft. This paper presents a study of feedback control of several adaptive vibration neutraliser to minimize tonal sound radiation from a modally dense rectangular plate. It is shown that several adaptive vibration neutraliser's using local feedback loops can be managed by a simple global algorithm to minimize the sound radiation from a plate. As an adaptive passive approach is used, each individual adaptive vibration neutraliser can be constrained to be stable and the resulting global system is also stable. Spatially averaged single frequency reductions of up to 22dB are experimentally demonstrated in the radiated field.

INTRODUCTION

Vibration neutralisers have been used successfully for decades to control vibration due to harmonic disturbances. Although often used as a tuned device, (where the resonance frequency of the neutraliser matches the frequency of the disturbance), Fuller *et-al*, (Fuller 1995) showed that sound radiation from structures is minimised when the neutraliser is not resonant at the frequency of disturbance. This detuning of the neutraliser for control of sound fields was experimentally demonstrated by Carneal 1996.

Reviews of adaptive vibration neutraliser's including several examples of tuning algorithms are given by von Flotow 1994 and Wright and Kidner, (Wright 2003).

Kidner, (Kidner:1999 a-c), developed a beam-like type of adaptive vibration neutraliser that was tunable by changing the geometry of the beam cross section. A feedback control system was applied to actively remove damping via analog velocity feedback. Therefore, both the stiffness and damping of the neutraliser were independently controlled.

In the following section the adaption of the input impedance of the vibration neutraliser using velocity feedback is discussed. A non-gradient search algorithm is described in the third section of the paper. Experiments in which this scheme is used to minimise the sound radiated from a simply supported plate is discussed in section 4. Significant noise control was achieved. The final section contains conclusions on the use of adaptive vibration neutralisers for the control of sound radiation.

AN ADAPTABLE TUNED VIBRATION NEUTRALISER

The passive vibration neutraliser's input impedance is given in Equation (1).

$$Z = j\omega m \frac{c - j\frac{k}{\omega}}{c + j(\omega m - \frac{k}{\omega})} \tag{1}$$

The stiffness, damping and mass of the vibration neutraliser are indicated by k , c , and m respectively. The vibration neutralisers' effect on the structural response is determined by

the relative magnitude of its input impedance to the structural impedance. The velocity of the structure at the vibration neutraliser attachment location can be written as the superposition of its free velocity (x_o) and a coupled velocity due to the vibration neutraliser reaction force as written in equation (2).

$$\dot{x}_s = \dot{x}_o \left[1 + \frac{Z}{Z_s} \right]^{-1} \tag{2}$$

The effect of a neutraliser on the mobility of a stiff system is shown in Figure (1). The reduction at the tuned frequency depends on the damping of the neutraliser. It is important to note the increase in mobility that occurs. This is due to the resonance of the new mode of vibration created by the coupling of the two systems. The separation of the maxima and minima of the mobility is a function of the ratio of the reactive parts of the neutraliser and structure impedance (Kidner 1999).

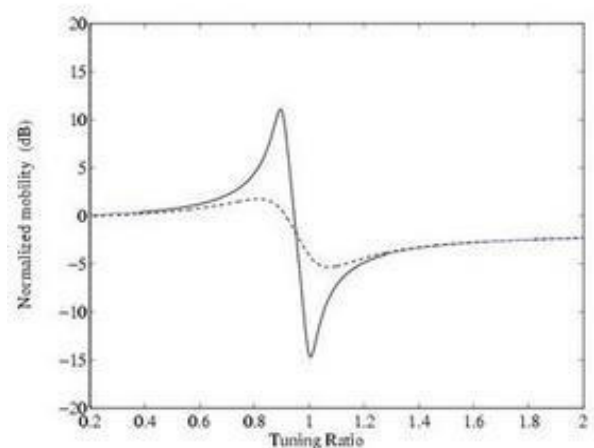


Figure 1 Illustration of the effect of a tuned vibration neutraliser on a mass spring system.

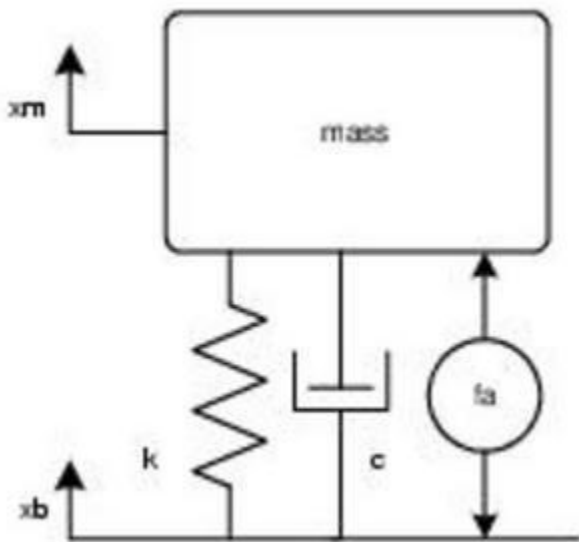


Figure 2 The inertial actuator configured as an adaptive vibration neutraliser

Figure (2) shows an inertial actuator configured as an adaptive vibration neutraliser. The active force is derived from the velocity of the inertial mass. A proportional-integral (PI) control strategy is used to produce the active force components. The transfer function, $H(s)$ of a PI controller and the closed loop response $T(s)$ is given in Equation (3).

$$T(s) = \frac{P(s)G(s)}{1 - G(s)A(s)H(s)} \quad (3)$$

where,

$$H(s) = K_p + \frac{K_i}{s} \quad (4)$$

$$G(s) = 1/(ms^2 + cs + k) \quad (5)$$

$$P(s) = cs + k \quad (6)$$

where $A(s)$ represents the sensor dynamics, (s for a velocity sensor and s^2 for an acceleration sensor).

Combining the three terms results in the closed loop characteristic equation, Equation (7). This characteristic equation is used to derive the tuning laws for the adaptive vibration neutraliser.

$$ms^2 + cs_k - A(s)(K_p + K_i s) = 0 \quad (7)$$

Feeding back the absolute velocity of the mass leads to the closed loop adaptive vibration neutraliser impedance expression in Equation (8).

$$Z_A = j\omega m \frac{cs + k}{ms^2 + (c - K_p)s + (k - K_i)} \quad (8)$$

The feedback gain couples only into the denominator of the input impedance expression. This configuration allows the feedback controller to modify the poles of the adaptive vibration neutraliser impedance, but has no effect on the zero Wright: phd . In the case of absolute mass velocity feedback, there is no longer an equivalent passive device, but an equivalent system can be conceptualised with a "skyhook" damper and spring of values K_p and K_i respectively.

THE SIMPLEX SEARCH ALGORITHM

A control strategy that requires very little knowledge of the system to be controlled is desirable as it would be most applicable to a wide variety of vibration problems. The search algorithm chosen to adapt the vibration neutralisers so that sound radiation is minimised is the downhill simplex search suggested by Nelder and Mead, (Nelder 1965). This algorithm performs a non-derivative based trial and error search which uses only four simple operations: *reflection*, *expansion*, *contraction*, and *shrinkage*. A function of N independent variables can be plotted as a hyper-surface in N dimensions. The minimum number of points required to define an area on this hyper-surface is $N+1$ points. This area enclosed by these vertices is a *simplex*, defined as the simplest possible geometric figure in a fixed dimensional space.

In this application, the adaptive vibration neutraliser's are detuned sequentially so the acoustic cost is a function of only two independent variables, the stiffness and damping control gains. A simplex in 2-space is a triangle. Based upon the cost function associated with each of the vertices of the simplex a new "trial point" is computed in the direction most likely to be "downhill". The cost of the trial point is then compared to the original simplex points and its relative value determines the next algorithm step. By reflecting, expanding, contracting, and constantly replacing worse simplex points with better ones, the algorithm "walks" downhill towards the minimum value until a stop condition is met. For a complete description of the algorithm the reader is referred to the article by Nelder and Mead, (Nelder 1965).

As with any parameter based search algorithm, the speed and accuracy of the convergence is highly dependent on the choice of search parameters. The parameters used in this experimental work were chosen based on recommendations in the literature and refined empirically for this application. The stop conditions are a cost function convergence tolerance and a maximum number of iterations.

The controller is coupled via the global error, (the sum of the square of the microphone signals), but each adaptive vibration neutraliser is adjusted while the others remain fixed. In this way the controller is a very simple SISO-hierarchical scheme that is suitable for large problems such as the control of interior noise in aircraft.

CONTROL USING A SINGLE ADAPTIVE VIBRATION NEUTRALISER

A single adaptive vibration neutraliser was used to control the acoustic radiation from a simply supported plate driven at a frequency close to its first mode of vibration. This example shows that on a simple structure the adaptive vibration neutraliser can control the radiated noise.

Experiment set up

Figure (3) shows the experiment set up. A close up of the adaptive vibration neutraliser shows the base and mass accelerometers are vertically aligned. The plate is made of steel and measures 0.6m by 0.425 m by 0.2 mm. Simply supported boundary conditions are realized with thin aluminum supports.

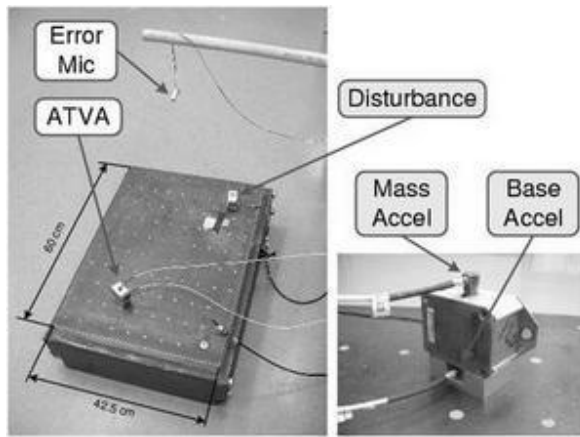


Figure 3 Test setup for single adaptive vibration neutraliser.

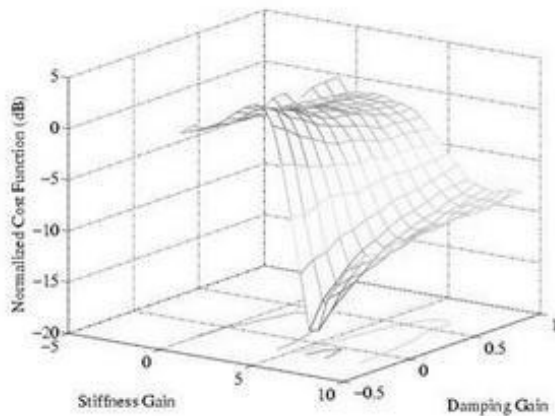


Figure 4 Cost function of the adaptive vibration neutraliser on a small plate at 135Hz.

Results

The plate was excited at 135 Hz, this is between the resonant frequencies of the first and second mode of the plate. The control gain stability limits were set to $-1 K_r$ and $-0.2 K_p$ and used to map the cost and adaptive vibration neutraliser phase surface. The surface is shown in Figure (4). It can be seen from Figure (4) that 18dB of sound reduction is possible.

CONTROL USING MULTIPLE ADAPTIVE VIBRATIONS NEUTRALISERS

Having demonstrated control using a single adaptable neutraliser on a structure whose response is dominated by only two modes, a more modally dense structure is now considered.

Experiment set up

Five adaptive vibration neutralisers are used to minimize the average of six microphone signals. As seen in Figure (5), the five adaptive vibration neutralisers were located randomly on the plate. The disturbance actuator is located near a corner of the plate allowing it to spatially couple to a large number of vibrational modes. The six error microphones are hung around the perimeter of a rack suspended above the plate at a distance of 1.8 m. The average spectrum of the six error microphones is shown in Figure (6).

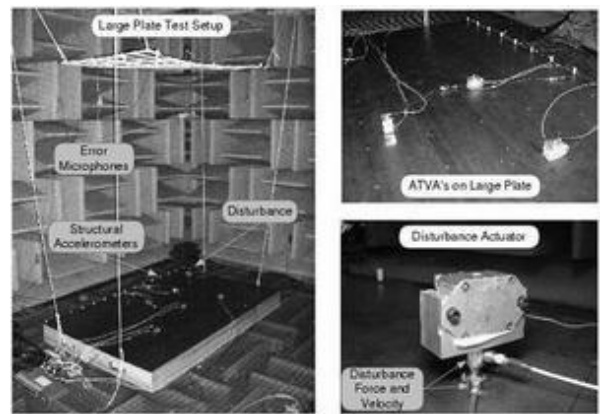


Figure 5 Large plate used to assess noise control using adaptive vibration neutralisers

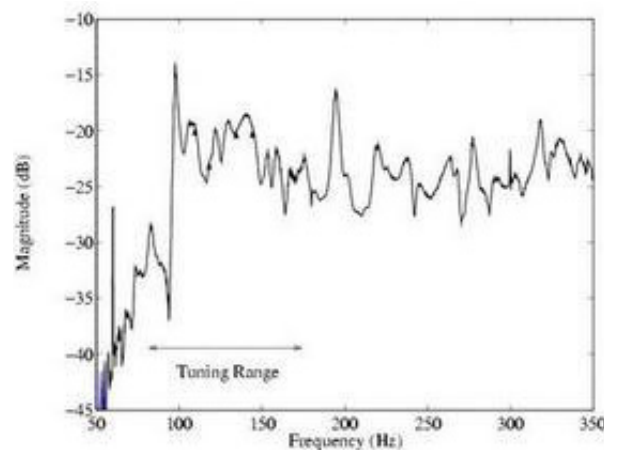


Figure 6 Cost function: frequency response from the disturbance input force to the averaged error microphones

Results

The no control case was chosen to include the passive (uncontrolled) adaptive vibration neutraliser dynamics rather than just the bare plate. The uncontrolled response is indicated by the dotted line in Figure (7). The adaptive vibration neutraliser's are then tuned to the excitation frequency while a damping control gain of -0.50 is applied to increase their authority, the response in this case is shown by the dashed line in Figure (7). The simplex algorithm is then sequentially applied to each one, The controlled response is shown by the solid line in Figure (7). A reduction of 22dB is achieved at the drive frequency.

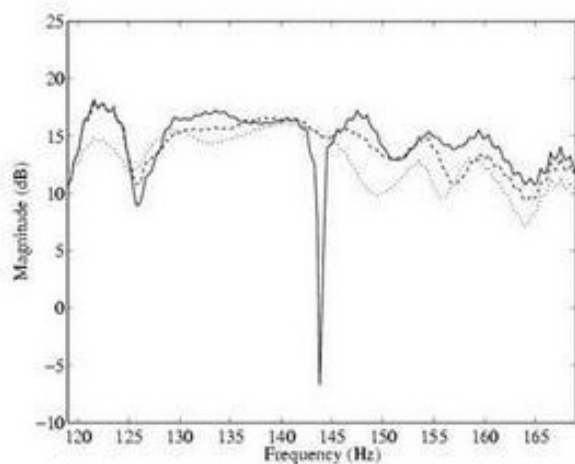


Figure 7 Sum of frequency response functions from disturbance input to radiated pressure with and without control.

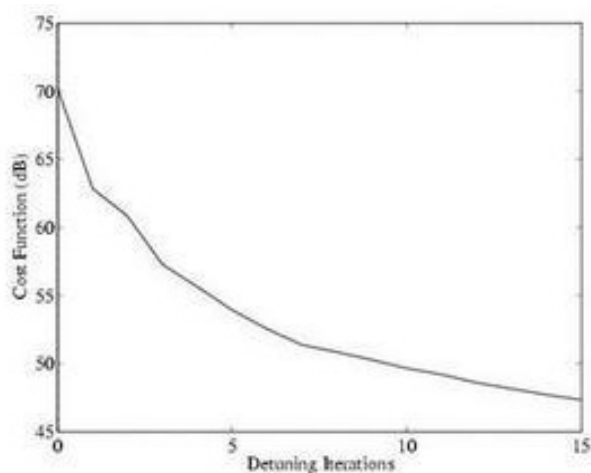


Figure 8 Convergence of cost function for 144Hz test case.

Structural attenuation is observed at the location of every adaptive vibration neutraliser when they are tuned. After adaption, although the sound radiation has been minimised the structural vibration increases at three out of five adaptive vibration neutraliser locations. This indicates that minimising vibration does not always control sound radiation.

The reduction in the cost function versus control iteration is shown in Figure (8), the iteration was stopped because all of the adaptive vibration neutraliser's reached their negative damping control gain limit. There was a decrease in structural impedance on the order of 2-3 dB for both the tuned and detuned control cases, a finding supported by broadband impedance measurements shown in Figure (9). This led to an increase in disturbance input power. An increase in total plate vibrational energy was also measured in the wavenumber domain in both control cases.

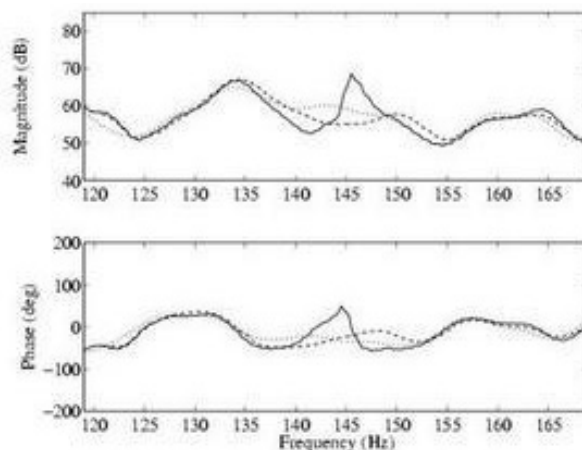


Figure 9 Input impedance at source for 144Hz test case with and without control

Wavenumber Analysis

Measured changes in the supersonic plate energy are consistent with observed noise control results in a qualitative sense only. This is because the frequency resolution under which the wavenumber analysis data was acquired was too coarse to capture the extremely narrow control bandwidth observed in Figure (7) which required resolution 4 greater.

The wavenumber domain data remains a valid source of information for revealing whether the plate undergoes modal restructuring as the adaptive vibration neutraliser's are tuned and globally detuned. Indeed, it is clear from the wavenumber transform contours in Figure (10, 11) that some degree of modal restructuring occurs in the plate vibration and this is concluded to be the primary mechanism of noise control at this test frequency.

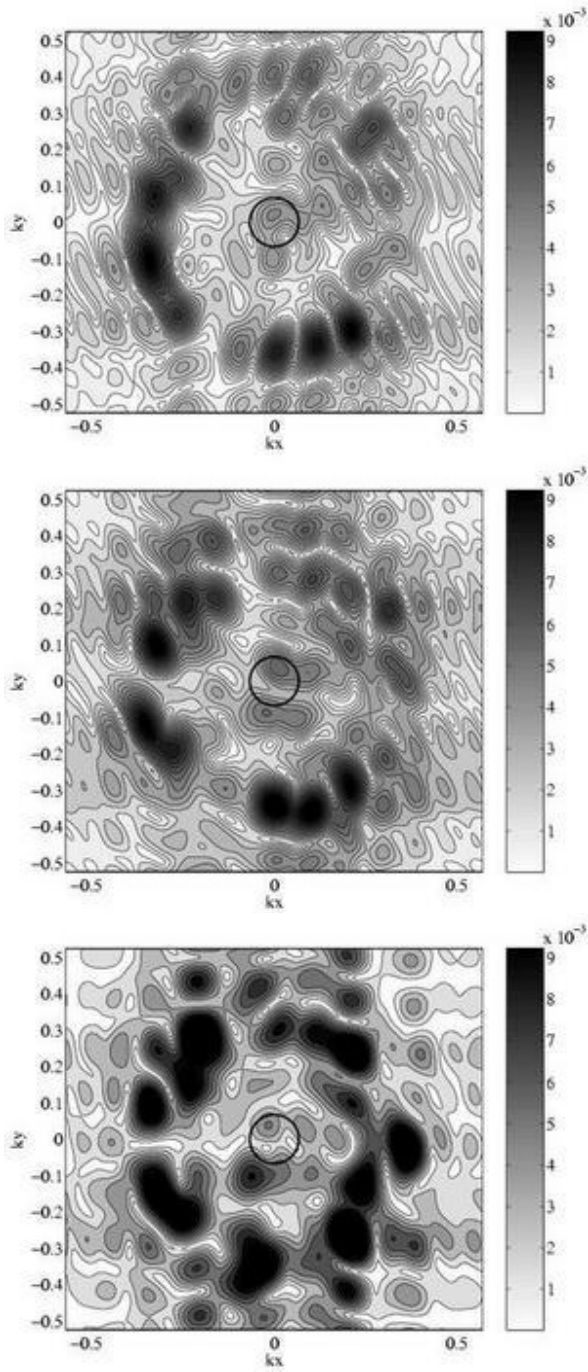


Figure 10 2-D Wavenumber transforms of the plate acceleration at 144Hz.

MECHANISMS OF NOISE CONTROL

When a structure's vibration field becomes highly complex due to the participation of a large number of structural modes, the effect of the adaptive vibration neutraliser's on the structural mobility is not as apparent as the cases of the simply supported plate described above. In these cases, it is convenient to look at the energy in the system to assess the effect of the adaptive vibration neutraliser's. A reduction in sound radiation can be achieved by reducing the amount of energy entering the structure, absorbing some of the energy that does enter the system, or redistribute the energy within the structure such that the structure couples less efficiently to the acoustic medium. A brief discussion on each of these mechanisms is included next.

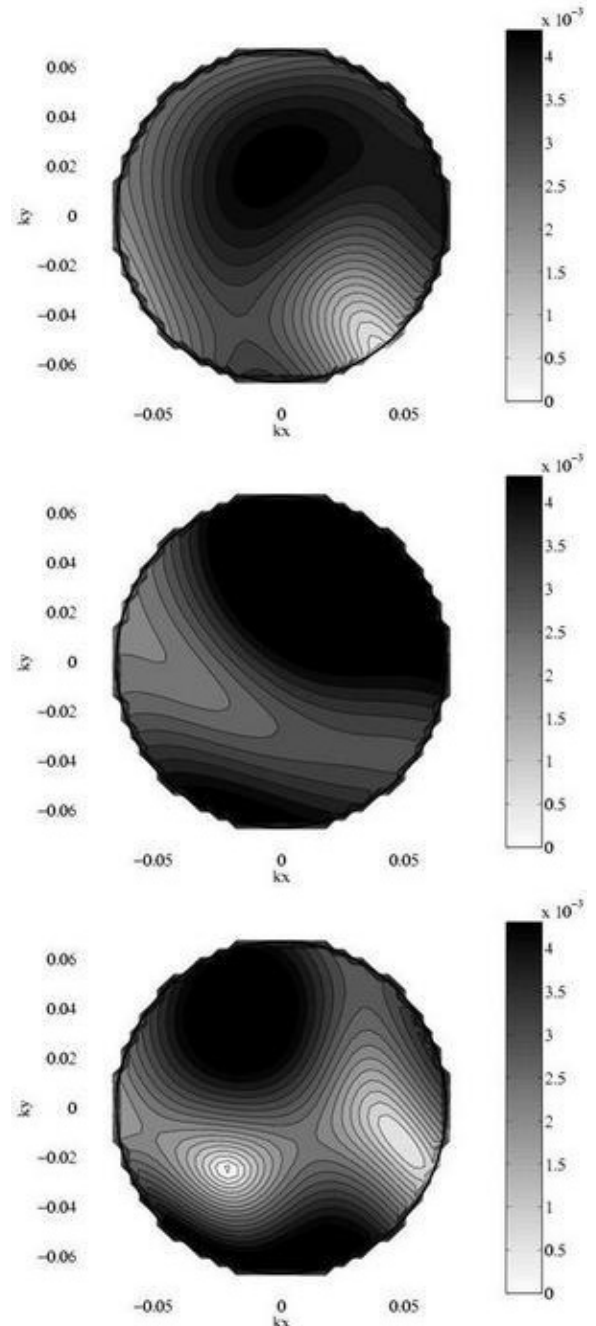


Figure 11 2-D Wavenumber transforms of the plate acceleration at 144Hz. Only the supersonic wavenumber components are shown.

Disturbance Rejection

Reduction of radiated noise can be achieved by reducing the energy input to the structure. This can be done by mismatching the structure and source impedances. Disturbance rejection is simpler when the disturbance is localized, but is also possible with distributed disturbances. Experimental results presented here suggest that this is not the dominant mechanism in this case, because an increase in input power was measured.

Energy Absorption

To reduce the sound radiation from a vibrating structure the energy can be dissipated to heat or transferred to another system. Dissipation to heat is accomplished by attaching devices or materials with high damping characteristics. Vibration neutralisers typically contain very little damping in order to maximize their affect on a vibrating host structure.

The neutralisers do act as reactive loads and as such can store energy in their mass and spring, this will act as an energy sink for the plate.

Modal Restructuring

An adaptive vibration neutraliser can reduce structural radiated sound by redistributing the energy within a vibrating structure. Vibrational energy is redistributed from modes that radiate efficiently to modes that do not radiate as efficiently.

CONCLUSIONS

Cost function surfaces, predicted by an analytical model of the simply supported plate with an adaptive vibration neutraliser, are experimentally verified.

The cost function surfaces are found to be smooth with a single global minimum a single global maximum and no local minima. Further, if the structural response is dominated by a single mode, tuning the adaptive vibration neutraliser to the excitation frequency guarantees that the response will be between the global maximum and minimum. This condition is crucial to guarantee the success of global detuning when a single mode is dominant. The global detuning algorithm is initiated from the tuned case. Global stability is addressed by constraining the adaptive vibration neutraliser control gains.

The downhill simplex search algorithm is chosen to perform the global detuning of the adaptive vibration neutraliser's. This algorithm requires no initial information about the structure or the adaptive vibration neutraliser and is well suited to this smooth cost function surface with no local minima.

A multiple adaptive vibration neutraliser noise control system is experimentally demonstrated on a modally dense structure. Attenuation of 22 dB of the structurally radiated noise was demonstrated at a single frequency. Further, globally detuning the adaptive vibration neutraliser's outperformed perfectly tuned adaptive vibration neutraliser's on a structure whose response is due to many modes.

A qualitative energy analysis is presented to identify the potential mechanisms by which the adaptive vibration neutraliser's can achieve structurally radiated noise control. Modal restructuring is concluded to be the primary adaptive vibration neutraliser noise control mechanism on this modally dense test structure. Evidence of energy dissipation was not found based on the damping control gains and disturbance rejection evidence was insignificant and inconsistent.

Wavenumber transforms of the plate vibration have shown reorganization of structural energy between the uncontrolled, tuned, and detuned adaptive vibration neutraliser. The presence of modal restructuring is concluded by significant change observed in the energy distribution among wavenumber components. A decrease in supersonic wavenumber energy has been observed every time the adaptive vibration neutraliser's are detuned.

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