Simultaneous noise and vibration control using active structural acoustic control inside an enclosed stiffened cylinder with floor structure.

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ABSTRACT

The concept of using an active structural acoustic control system attached to an airframe structure to address the problem of excessive tonal interior noise levels inside vibration sensitive aerospace applications can on occasion produce unwanted simultaneous increases to the structural vibration levels. In this paper, the trade-off in global interior noise reduction performance required to accommodate the additional requirement of there being no net increase to the global structural vibration levels, under the control action of an active structural acoustic control system, is examined. The potential for controlling both interior noise and structural vibration levels simultaneously will be discussed across different frequency cases (representing scenarios of different coupling complexity) for the example of a stiffened cylinder with floor structure and enclosed cavity. Simulation results support the notion that simultaneous control of interior noise and prevention of structural vibration increase (under controlled conditions) is both plausible and reasonable with a relatively small trade-off in the interior noise reduction performance. This occurs when measures for both global interior noise levels and global structural vibration levels are considered integrally into the cost function that guides the selection of the optimum vibration actuator locations using a genetic algorithm search optimisation procedure.

1 INTRODUCTION

One area of Active Structural Acoustic Control (ASAC) research that has received significant attention over time is that of the attenuation of low frequency interior noise in propeller aircraft. Aerospace applications are typically vibration sensitive, due to significant issues associated with the fatigue of aluminium load-bearing components. Strict certification requirements are applied to maintain appropriate safety margins for the continuing safe operation of the airframe structures. These same regulatory requirements place restrictions on the form of noise and/or vibration control measures that can potentially affect the fuselage/airframe fatigue life. This presents a significant issue for the retrofit installation of ASAC systems to fuselage structures. Therefore, it is reasonable to assume that any ASAC system designed to reduce interior noise levels in aircraft needs to do so without causing any unintended increase in the structural vibration levels. However, a problem often encountered with ASAC systems designed specifically for reducing the coupled interior noise responses inside the cavity, is the spillover of the control vibration energy into structural modes not well coupled acoustically.

In this paper, the trade-off required in terms of global interior noise reduction performance to accommodate the additional requirement of there being no net increase in structural vibration levels is examined across different frequency cases for the example of a stiffened cylinder with floor structure and enclosed cavity. Simulation results support the notion that simultaneous control of interior noise and prevention of structural vibration increase (under controlled conditions) is both plausible and reasonable with a relatively small trade-off in the interior noise reduction performance.

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to global structural vibration levels, is examined. The potential for controlling the interior noise while eliminating any structural vibration increase is discussed for selected frequency cases on the example of a stiffened cylinder with floor structure and enclosed acoustic cavity.

2 PREVIOUS WORK

Early investigations of the use of active structural acoustic control to minimize aircraft interior noise used simply supported-cylinder structure models coupled together with cylinder cavity acoustic models [1-3]. The structural and acoustic modal interactions proved highly selective, limiting the way in which ASAC control is achieved [4]. More advanced cylinder models alleviated this selectivity [5]. However, the presence of an internal floor attached integrally to the cylinder also proved effective in broadening the nature of the response of the coupled system [6-9] and appeared more intuitively similar to realistic aircraft fuselage structures. The effect of the floor on the response of the structure, and the shape of the cavity introduced opportunity for less-selective coupling behaviour to occur [9], producing more realistic levels of coupling complexity between the structural and acoustic modal responses. This model configuration has proven effective for fundamental analysis of ASAC mechanisms and behaviour, enabling further understanding of coupled system behaviour [4,10] and exploration of optimisation techniques [11].

The need for greater predictive accuracy in assessing the feasibility of ASAC in realistic aircraft applications led to the development of many experimentally-based prediction models [12-18] incorporating measurement data in the form of transfer function analysis and system identification techniques. With these models, the effectiveness of discrete point control forces, piezoceramic actuators and tuned vibration absorbers, arranged with varying degrees of optimisation, were trialled to assess the effectiveness of the ASAC noise reduction in specific fuselage structures. Although providing improved accuracy for determining interior noise reduction predictions, it proved difficult to elicit detailed insights into the underlying ASAC mechanism behaviour using this form of predictive modelling. Therefore, understanding of the fundamental system behaviour has advanced little since the advent of such models. In this paper, preference is given to a coupled fuselage model configuration that is fundamental in nature, allowing more detailed insights into the coupled system behaviour when under the action of ASAC.

The spillover of control vibration energy into structural modes not effectively coupled with cavity modes was identified as a significant issue from the earliest investigations of ASAC in aircraft [19-20]. The spectrally white structural vibration excitation introduced via the action of applied control forces produces excitation of structural modes previously unexcited by the primary source. Focussing exclusively on minimising the interior noise levels can result in significantly excited but poorly coupled structural modes. The prevalence of this issue in the various ASAC applications reviewed varies significantly. For many cases, excitation at the fundamental blade passage frequency is sufficiently low, allowing the dominance of single coupled structural/acoustic modal pairs. The simultaneous reduction of the offending primary noise and vibration appear easily achieved by applying ASAC in these cases [15,17]. However, excitation at slightly higher harmonic frequencies generally produces less encouraging results in comparison. This is due to higher modal
densities contributing to more complex coupled energy transmission through to the interior cavity from the primary excitation of the structure.

In one study [18], a genetic algorithm was used to arrange ASAC actuator grouping configurations on an aircraft fuselage with the goal of maximising noise and vibration reductions simultaneously. This was achieved by sequentially swapping the specified cost functions (that guide the genetic algorithm search) between measures for interior noise reduction and structural vibration reduction over successive search generations. This technique demonstrated that the range of individual actuator arrangement solutions identified (in the final generation) achieved improved compromise between noise and vibration control objectives, than if either cost function measure were considered alone. However, the analysis offers little insight into the underlying reasons why the compromise between noise and vibration reductions is achieved. In addition, at the excitation frequency corresponding to the blade passage frequency, simultaneous noise and vibration reduction appears easily achieved. This occurs regardless of the actuator configurations selected, or even the cost function method used to guide the genetic algorithm search, meaning that the issue of control vibration spillover does not occur and has not been addressed in this study.

It is the objective of the present investigation to explore further the issue of control vibration spillover and determine potential trade-offs that are needed in terms of absolute noise reduction potential for there to be no net increase in the structural vibration levels under ASAC conditions. The cases selected for analysis are designed to address problematic spillover of ASAC control vibration energy, regardless of the number of vibration actuators used. A genetic algorithm search method is implemented to optimise the locations of the control vibration actuators. However, this is undertaken using a new approach, where a single numeric composite cost function is developed by combining numerical predictions of the noise and vibration reductions to guide the progress of the search for optimum actuator configurations. The effectiveness of this method is compared with that of optimising the location of control vibration actuators to minimise either noise or vibration alone. Finally, analysis of the optimised actuator configurations so identified is required to gain insight into the characteristic behaviours and mechanisms associated with the control of the interior noise without any corresponding increase in structural vibration levels.

3 MODEL DESCRIPTION

3.1 Stiffened Cylinder with Floor Model

The example coupled vibro-acoustic system selected to investigate the characteristics of simultaneous noise and vibration control, using active structural acoustic control, consists of a longitudinally stiffened cylinder of length, 3m, and radius, 0.45m, with a rigidly-attached internal stiffened floor structure, joined to the cylinder at an angle of \( \theta_f = 40^\circ \) (Figure 1). The analytical model of the structure combines the modal response of a cylinder and plate structure together and represents the effects of thirty longitudinal stiffeners along the length of the cylinder, and six longitudinal stiffeners along the length of the interior floor. Both the cylinder and floor structures are assumed to have shear-diaphragm boundary support conditions and rigid end-caps. The cavity modal properties are determined using a 2-D finite difference implementation of the Helmholtz equation to account for the irregular cross-section shape, extended axially by assuming an axial model shape function response of a rigid-walled 1-D enclosure [21]. Modal coupling theory [22] is used to describe the nature of
the interactions between the structure and the interior acoustic cavity under the action of an external monopole source excitation, located midway along the cylinder length at location \((r, \theta, L) = (0.9\text{m}, 0^\circ, 1.5\text{m})\). Harmonic point forces are used to represent the effect of active structural acoustic control actuators, with the optimum control force settings determined using quadratic optimisation theory [23].

3.2 Genetic Algorithm Search

The genetic algorithm optimisation method works by encoding the critical search variables (such as the location of each vibration control actuator on the structure) into a single numerical string. The string is then assigned a cost function value, which provides a numerical indication of how well the represented solution satisfies some predetermined performance criterion (such as the acoustic potential energy reduction inside the bounded cavity, or the vibrational kinetic energy reduction). Using a process of chance (which is biased towards the “survival of the fittest”), two strings are selected from an initial breeding population of potential candidates. The contents of these two strings are randomly swapped and recombined (crossover) to produce two (hopefully unique) offspring with the eventual goal of producing further improvement to and displacing the worst search solutions already present in the breeding population. The genetic algorithm developed for this analysis is based on the principle of a steady-state genetic algorithm and is the culmination of previous work undertaken to develop and select a range of appropriate genetic algorithm operators for optimum convergence performance in vibration actuator placement problems of this type [11]. The surface of the structure and the internal floor are discretised into 1925 separate grid points, representing potential vibration actuator locations. The number of control sources to be optimised using the developed genetic algorithm is varied between one and ten in the present investigation.

Choosing an appropriate cost function to guide the progress of the genetic algorithm search forms a critical part of achieving the desired characteristic behaviour displayed by the optimised ASAC vibration actuator configurations. Commonly, the cost function is a single numerical quantity directly related to the sound pressure levels inside the
enclosed cavity. For this purpose, the acoustic potential energy reduction inside a bounded enclosure is defined as:

\[ E_{p}^{\text{red}} = E_{p}^{\text{primary}} - E_{p}^{\text{control}} \]  

(1)

where:

\[ E_{p} = \frac{1}{4 \rho_{o} c_{o}} \int_{V} |p(\vec{y})|^{2} d\vec{y} \]  

(2)

and \( p(\vec{y}) \) is the sound pressure at some arbitrary location \( \vec{y} \) in the volume \( V \), \( \rho_{o} \) is the ambient air density and \( c_{o} \) is the speed of sound in air.

Alternatively, if structural vibration reduction is desired, then an alternative cost function measure is the vibrational kinetic energy reduction, which is defined as:

\[ V_{p}^{\text{red}} = V_{p}^{\text{primary}} - V_{p}^{\text{control}} \]  

(3)

where:

\[ V_{p} = \frac{1}{2} \int_{S} \rho_{s} h_{s} \left| \dot{w}(\vec{x}) \right|^{2} d\vec{x} \]  

(4)

and \( w(\vec{x}) \) is the out of plane displacement of the structure of area \( S \), material density \( \rho_{s} \), and thickness \( h_{s} \).

In either case, these numerical measures are useful to track trends associated with interior noise and structural vibration reductions as a result of the application of active structural acoustic control to the coupled stiffened cylinder with floor system.

4 MEASUREMENT OF ASAC SYSTEM RESPONSE

4.1 Acoustic potential energy reduction vs. vibrational kinetic energy reduction

To better understand the coupled system response of the longitudinally stiffened cylinder with floor (in response to the external monopole excitation) a single control source is exhaustively trialled at each of the 1925 potential candidate actuator positions on the structure (referred to as a full structural scan). At each excitation frequency (between 10 – 250 Hz), a full structural scan is performed, enabling the identification of actuator locations that achieve either maximum acoustic potential energy reduction, or maximum vibrational kinetic energy reduction. Figure 2 illustrates the best acoustic potential energy reduction achieved at each frequency. The corresponding vibrational kinetic energy reduction (when the control source is positioned to maximise acoustic potential energy reduction) is also shown for comparison. Alternatively, Figure 3 illustrates the best vibrational kinetic energy reduction achieved at each frequency, with the corresponding acoustic potential energy reduction shown also for comparison.

A comparison of results presented in Figures 2 and 3 gives a degree of insight into the nature of the coupled structural/acoustic system response and the frequencies at which active structural acoustic control displays the greatest potential for noise and vibration reduction. For example, simultaneous noise and vibration reduction is easily achieved with only one control source at the excitation frequency of 87Hz. This is characteristic of a single dominant coupled structural/acoustic mode pair, which is responsible for the majority of the interior sound transmission. The mechanism by which this ASAC system behaviour is controlled is referred to as “modal control” [4], and has been exploited in low frequency ASAC applications investigated previously [14,16]
Figures 2 also contain frequencies at which the best level of noise reduction achieved using one control source is low. Many of these are observed to produce increase to the structural vibration levels at the same time (as indicated by negative values of vibration kinetic energy reduction). This suggests a coupled system response that is more complex, as one control source is not able to simultaneously reduce the increased number of structural and acoustic modes contributing significantly to the coupled system response. Applying a single point control force to the structure induces a spectrally white vibration excitation of all structural modes, many of which appear not well coupled to the dominant acoustic mode response. Although good levels of reduction of the induced acoustic excitation are possible, this can be at the cost of increasing the structural vibration levels. This ASAC system behaviour implies a simultaneous and complex mix of the control mechanisms “modal control” and “modal rearrangement” [4], which work together to produce the appropriate reductions of structural modes responsible for the coupled interior acoustic response. It is interesting to note that this particular system behaviour is characteristic of the difficulties facing the application of ASAC where control vibration spillover is problematic. Consequently, the frequencies at which this occurs are of particular interest to the present analysis.

4.2 Candidate Excitation Frequencies

Ideal candidate frequencies have reasonable levels of $E_p^{\text{red}}$ with the application of a single vibration control source, but also experience simultaneous increases in vibration levels (as indicated by negative values of $V_p^{\text{red}}$). Increasing the number of optimally located vibration control sources (to maximise $E_p^{\text{red}}$) can introduce significant benefit in reducing the levels of control vibration. However, frequency scenarios where this does not occur, characteristically demonstrate the need to implement an optimisation strategy beyond that of simply increasing the number of vibration control sources to reduce the spillover of the control vibration energy.

The first candidate frequency to be considered is 68 Hz. In Figure 3, this corresponds to a point in the controlled system response where positive values of $E_p^{\text{red}}$ are accompanied by negative values of $V_p^{\text{red}}$. To further characterise the ASAC system behaviour at this frequency, a series of separate genetic algorithm searches are performed where a different number of control sources are specified for each case. The
genetic algorithm search is repeated ten times for each different number of vibration control sources used, to allow reasonable assurance that the near-best actuator configuration has been found in each case. This yields ten separate individual $E_{p}^{\text{red}}$ and $V_{p}^{\text{red}}$ results for each case where the number of control actuators is varied. The results are plotted in Figure 4.

Using just one optimally located vibration control source yields $E_{p}^{\text{red}}$ approaching 18 dB, but is at the cost of increasing the vibrational kinetic energy levels by 10 dB. Adding additional optimised vibration control sources steadily and incrementally improves the levels of $E_{p}^{\text{red}}$ up to levels approaching 40 dB, but tends to produce marginal and occasional reduction in the level of vibration increase (as indicated by slightly less negative values of $V_{p}^{\text{red}}$). Even with the maximum of ten control sources, a degree of control vibration spillover still exists (a minimum of 3 dB is observed). Therefore, although improvement is observed with the spillover of control vibration as control actuators are added, control vibration increase remains an issue at this frequency when the cavity acoustic potential energy levels are minimised.

The second candidate frequency is 136 Hz. The observed response reflects a more complex coupling scenario where a higher number of significantly excited coupled modal pairs exist, resulting in ASAC having greater difficulty in achieving large reductions using a finite number of control sources. Despite this, the characteristic behaviour of the optimised ASAC system is similar to that of before, where the number of optimally located control sources has little incremental influence on the degree of control vibration spillover experienced (Figure 5). It is clear that additional strategies for guiding the actuator location optimisation process are required in these cases to explore further the potential for reductions to the acoustic potential energy without causing a corresponding increase to the vibrational kinetic energy levels.

4.3 Composite Cost Function Selection

The potential for a genetic algorithm search to optimise the locations of four vibration control actuators in a manner that maximises the acoustic potential energy reduction inside the cavity, while maintaining zero net increase in the vibrational kinetic
energy levels, is explored. A combined cost function is formulated, which is designed to incorporate measures for $E_{p,\text{red}}$ and $V_{p,\text{red}}$ together in the form of a single numerical measure that indicates relative success in achieving noise and vibration control simultaneously. In this context, a cost function that combines these measures is intended only to be loosely indicative of relative performances of actuator layout solutions, so that when maximised, it guides the optimisation process towards layout configurations displaying the desired characteristics of noise reduction without vibration increase. The simplest way of achieving this is to apply some form of performance penalty to the levels of interior noise reduction achieved, in direct proportion to the amount of vibration increase induced. It is logical therefore to consider the simple summation of the predicted values for $E_{p,\text{red}}$ with $V_{p,\text{red}}$, producing a new cost function of the form: $(E_{p,\text{red}} + V_{p,\text{red}})$. Maximising this function is intended to shift the focus of the optimisation process away from actuator configurations which may achieve high cavity noise reductions, but only at the cost of substantially increasing the structural vibration levels as well. Instead, the composite form of these performance measures is to promote the ranked importance of those configurations which have slightly less cavity noise reduction but which may produce no vibration increase at all.

5 RESULTS

Figure 6 displays the results for the 68 Hz external acoustic monopole excitation of the structure, with separate genetic algorithm searches for the optimum placement of four vibration control actuators performed for each cost function case where either $E_{p,\text{red}}$, $V_{p,\text{red}}$ or $(E_{p,\text{red}} + V_{p,\text{red}})$ are maximised. The clusters of best results determined from each of ten separate genetic algorithm runs performed for each cost function case are plotted. The search results that focus on maximising the $E_{p,\text{red}}$ and $V_{p,\text{red}}$ levels show significant and characteristic differences. When $E_{p,\text{red}}$ is maximised, global vibration increase is observed due to the effects of control vibration spillover. The highest $E_{p,\text{red}}$ achieved is 30dB, but this is at the cost of $V_{p,\text{red}}$ approaching -12dB (negative values of $V_{p,\text{red}}$ indicating vibration increase). If the cost function of $V_{p,\text{red}}$ is maximised, global vibration decrease is observed, where a best value of $E_{p,\text{red}}$ of 19dB is achieved for a corresponding $V_{p,\text{red}}$ of 5dB. Maximising the composite cost function $(E_{p,\text{red}} + V_{p,\text{red}})$ at this frequency achieves a maximum $E_{p,\text{red}}$ of over 25dB for a $V_{p,\text{red}}$ of around 3dB. This demonstrates the possibility of eliminating issues associated control vibration spillover for just a 17% reduction in the maximum noise reduction performance achieved. This contrasts with a 37% reduction required if a cost function based on $V_{p,\text{red}}$ alone were to be used.

Figure 7 displays the best optimised actuator location configurations for each cost function case used. It is interesting to observe that for greatest $E_{p,\text{red}}$, all four control source actuators align at various angular locations around the cylinder at the same axial mid-point location ($L = 1.5m$). This is due to the symmetry of the applied primary pressure distribution of the monopole source (about the axial location, $L = 1.5m$) inducing excitation in structural modes of odd axial mode order. All such modes have an anti-node positioned at $L = 1.5m$, and are readily controlled from this location. Further investigation reveals the dominance of a coupled structural/acoustic mode pair (structural mode resonant at 87 Hz - see Figure 3) exerting significant influence on the transmission of the primary source excitation at 68 Hz. The optimised control source positions seek to control the response of this structural mode, with little apparent
consideration for the unintended excitation of remaining weakly coupled structural modes.

For greatest $V_{p_{\text{red}}}$, the control actuators adopt axial positions away from the axial mid-point location ($L = 1.5$ m) with preference for high impedance locations at or near the cylinder/floor junction, or, on the floor structure itself. This introduces significant levels of excitation of structural modes of even axial mode order, which were not previously excited with the primary monopole source excitation (located at $L = 1.5$ m). However, increases in the levels of the structural excitation of modes of even axial mode order appears offset by reductions in the dominant coupled structural/acoustic mode pair (structural mode resonant at 87 Hz), yielding a net decrease in structural vibration levels. The structural mode responsible for the dominantly coupled modal excitations appears to be one of a small number that have finite displacements at the location of the cylinder/floor junction position; therefore experiencing lower impedance levels to the input of control source vibrations, relative to many other structural modes. Achieving control over this mode by applying the control forces at the cylinder/floor junction position appears to be an effective method for the present coupled system example in limiting the amount of control vibration spillover into other uncoupled structural modes.

The best overall compromise between noise and vibration control is achieved using the composite cost function ($E_{p_{\text{red}}} + V_{p_{\text{red}}}$). This compromise is also reflected in the control actuator layouts, where positions are adopted both symmetrically and non-symmetrically about the axial location, $L = 1.5$ m, with one actuator also placed on the location of the cylinder/floor junction.

Figure 8 displays the genetic algorithm search results for the placement of four vibration actuators to control the 136 Hz external acoustic monopole excitation of the structure. Again, the trends associated with the optimised results show clustering of the solutions on the plot of $E_{p_{\text{red}}}$ against $V_{p_{\text{red}}}$, similar to those identified in the 68 Hz case. A noticeable reduction in the magnitude of the overall $E_{p_{\text{red}}}$ reductions is observed, due to the higher modal density of the structure decreasing opportunities for dominance by single coupled structural/acoustic modal pairs. The results show that the issue of
control vibration spillover is eliminated with only 18% reduction in the maximum noise reduction performance. This contrasts to a 70% reduction in the maximum noise reduction performance if the cost function identifying the optimum actuator locations were to be based on $V_{p_{red}}$ alone.

Figure 9 shows the best actuator location configurations found on the structure for each cost function case at 136 Hz external acoustic monopole excitation. The trends associated with maximum $E_{p_{red}}$ are similar to those identified previously. Each vibration actuator is aligned at various angular locations around the axial mid-point plane at $L = 1.5\text{m}$, on the cylinder. However, as before, this appears to produce considerable control vibration spillover into weakly coupled structural modes. When positioning the vibration actuators to minimise the $V_{p_{red}}$ levels, actuator locations are again chosen in close proximity to the cylinder/floor junction, and are not restricted to positioning around the axial mid-point plane. However, the best compromise between noise and vibration reduction is achieved using the composite cost function ($E_{p_{red}} + V_{p_{red}}$). The vibration actuators are positioned symmetrically around the axial mid-point plane of the structure, at or near the cylinder/floor junction positions. The selection of these locations again appears strategic in this example for providing an appropriate level of control vibration to coupled structural modes excited by the primary source without introducing unnecessary amounts of spillover vibration to the structure under ASAC conditions.

6 CONCLUSIONS

Significant levels of control vibration spillover in two selected frequency scenarios were effectively controlled for the example of a longitudinally stiffened cylinder with floor structure excited by an external monopole source. A genetic algorithm search was implemented using a composite cost function measure that combined predicted measures for global interior noise and global vibration levels into a single numerical performance indicator, which was used to optimise the vibration actuator locations to achieve the highest interior noise reductions without increasing the structural vibration levels. The trade-off required in terms of maximum interior noise reduction
performance to achieve zero net increase in the structural vibration levels when under the action of ASAC, was less than 18%.

The seemingly intuitive selection of actuator locations around axial mid-point locations on the structure produced the highest levels of interior noise reduction, and also the highest levels of vibration control spillover. However, minimum levels of control vibration spillover were added to the structure when control actuators were at or near the junction between the cylinder and floor structures. To maximise interior noise reductions with zero net increase in structural vibration levels, a practical mix of locations at or near the cylinder/floor junction and locations around the cylinder not limited to the axial mid-point plane (at L = 1.5m) were found to be effective in minimising the induced coupled excitation of the longitudinally stiffened cylinder with floor structure.

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8 REFERENCES


