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Minimization of sound radiation from plates using adaptive tuned vibration absorbers

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Abstract

Vibration absorbers have widely been used to minimize the vibration of structures, either globally for rigid structures or locally for flexible structures. This paper presents novel experimental work by applying the use of vibration absorbers to minimize sound radiated by a simply supported plate. Experimental results are presented comparing the effectiveness of sound attenuation by an adaptive tuned vibration absorber (ATVA) using two different tuning algorithms. The first algorithm "tunes" the ATVA to the excitation frequency of the plate. The second control algorithm adapts the ATVA tuning frequency to minimize the radiated sound field. Results indicate that "tuning" the ATVA to the excitation frequency can guarantee reduction of the vibration level near the base of the ATVA, but cannot guarantee reduction of the radiated sound field. In fact, it is shown that "tuning" the ATVA can actually increase the sound levels. Using an acoustic cost function to optimize the resonant frequency of the ATVA is a more effective procedure to guarantee a reduction of the radiated sound field. As a result of using this cost function, the ATVA was "detuned", i.e., the resonant frequency of the ATVA was not equal to the excitation frequency.
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1. Introduction

The application of tuned vibration absorbers (TVA) to efficiently minimize the vibrations of structures can be widely found in mechanical engineering books [1,2] and scientific articles [3,4]. A TVA is typically applied to single frequency disturbances, where its resonant frequency is typically tuned to be the excitation frequency of the structure. Recently, adaptive tuned vibration absorbers (ATVA), with a resonant frequency that can be adapted by a controller, have become feasible

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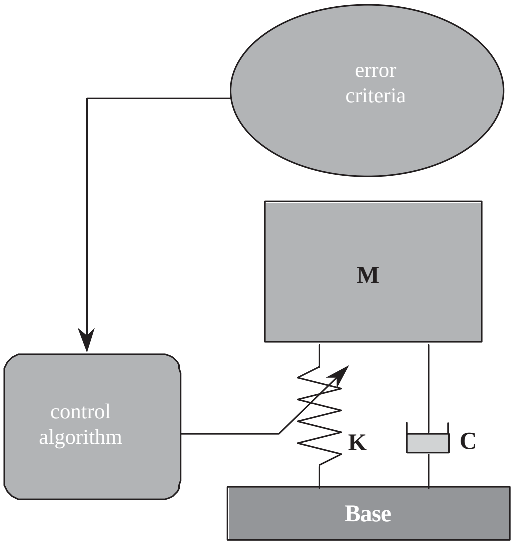

Fig. 1. Schematic of a single-degree-of-freedom ATVA.

with the proliferation of low cost computers and electronics. An example application for an ATVA would be to adapt its resonant frequency to track slight variations of excitation frequency with time. Fig. 1 presents a schematic of an ATVA.

Recently, TVA are becoming increasingly considered for sound reduction applications [5], especially for airplane applications [6–9]. The work presented by Jolly et al. [5] consisted of an analytical study for semi-infinite panels. The focus of this work [5] was concerned with the effect of a vibration absorber tuning (relative to the critical frequency of a simply supported semi-infinite panel) on the farfield radiated power and the radiation efficiency. The results presented showed that vibration absorbers were effective to reduce sound pressure level in airplane applications. Leigh et al. [6] and George [7] discussed the fact that vibration absorbers provide inexpensive tone reduction with fairly low weight penalty. In these references, the vibration absorbers were "tuned" to the excitation frequency, i.e., the resonant frequency of the absorber is equal to the excitation frequency. This implicitly means that the vibration absorber minimizes the vibration at its base. In contrast, Fuller et al. [8] and Guigou et al. [9] presented a control algorithm to adapt the vibration absorber resonant frequency in order to minimize a cost function that is directly based upon interior acoustic levels. The theoretical results show that the optimal solution of the controller "detuned" the vibration absorber (the resonant frequency of the vibration absorber is not equal to the excitation frequency of the system) to maximize the reduction of the interior acoustic levels. The results also demonstrated that detuning the vibration absorber gives improved attenuation over a tuned absorber. These results were consistent with previous work in acoustic structural active control (ASAC) [11] which show that minimizing the fuselage vibration does not perform as well as directly minimizing the coupled interior acoustic field for low-frequency disturbances.

This paper presents an experimental study of the theoretical concept presented by Fuller et al. [8] and Guigou et al. [9]. An ATVA was developed and used to reduce the sound radiated by a finite simply supported plate excited by a single frequency harmonic disturbance. A control algorithm with an acoustic cost function was used to adapt the resonant frequency of the ATVA ("detuning"). Sound reductions were recorded with the ATVA resonant frequency tuned to the excitation frequency and are compared to the ones obtained by detuning the ATVA.

2. Experimental setup

The experimental setup is now detailed for the ATVA, the plate-ATVA system, and the controller followed by the experimental procedure.

2.1. "V" type ATVA

As mentioned previously, there are many different physical implementations of an ATVA [3,4]. However, due to the limited tuning range of current designs, a new type of ATVA was developed and implemented experimentally for this study and is called the "V" type ATVA (VATVA).

The VATVA, shown in Fig. 2, uses a stepper motor to act as its active mass. To adapt its resonant frequency, the stepper motor changed the flexural stiffness of the supporting shafts by moving the supporting ends. Changing the flexural stiffness of the ATVA is similar in principle to the Williams et al. design [10], however they used shaped memory alloys to change the stiffness. The control algorithm used two accelerometers to tune (or detune) the absorber. One accelerometer was located on the stepper motor (active mass) and one was located at the base. Fig. 3 presents the frequency response of the VATVA for different separation distances between

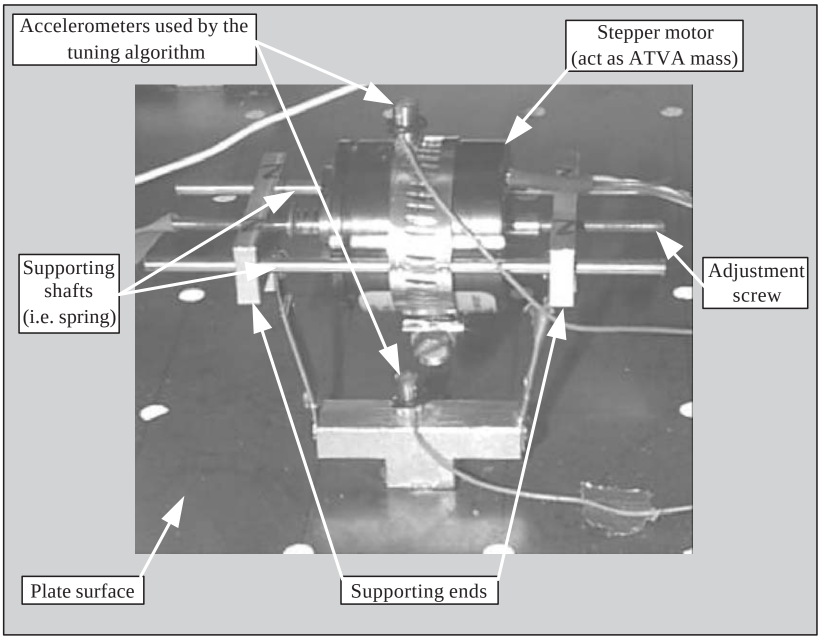

Fig. 2. "V" type ATVA.

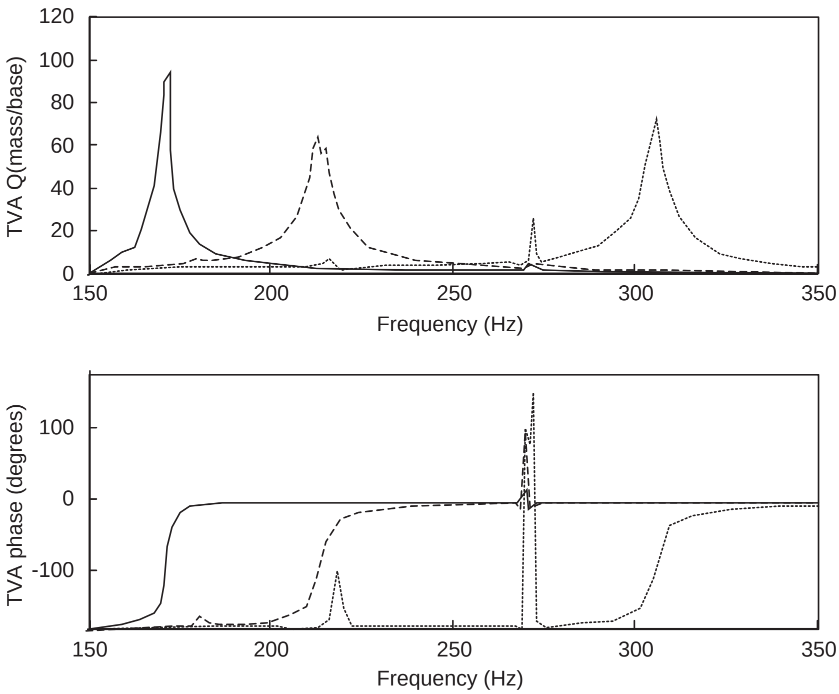

Fig. 3. Frequency response functions for varying separation distances of "V" type ATVA. ——, 25 mm; - - -, 15 mm; · · · · · · ·, 3 mm.

the stepper motor and the supporting ends. A 25 mm separation distance yields a resonant frequency of about 170 Hz, while a 3 mm separation distance corresponds to a resonant frequency of 310 Hz. The maximum range of the VATVA was 95–310 Hz giving it a tuning range of approximately 200 Hz $\pm$ 50%.

2.2. Plate-ATVA system

Fig. 4 shows the experimental set-up for the plate—ATVA system. A simply supported plate, whose characteristics are described in Table 1, was mounted in a hemi-anechoic frame and located in an anechoic chamber. The disturbance is generated by a piezoceramic actuator positioned at $x/L_x = 0.7$, $y/L_y = 0.3$, and having the characteristics given in Table 2. The ATVA is positioned at $x/L_x = 0.65$, $y/L_y = 0.45$. The plate flexural vibration was measured by a laser vibrometer at 100 points on the plate surface indicated by the circular reflective tape. The ATVA has a total active mass of approximately 200 g. The plate mass is 1.86 kg, which gives a *mass absorber/mass plate* ratio of around 0.11. In general, the goal is to minimize the mass absorber/mass plate ratio since it is desired to mainly use the dynamic effects of the ATVA, not the added mass effects.

2.3. Global detuning

Again referring to Fig. 4, seven monitoring microphones were distributed in the far field of the plate on a half-sphere radius of 0.61 m. The sum of the square of the amplitude of the signals from

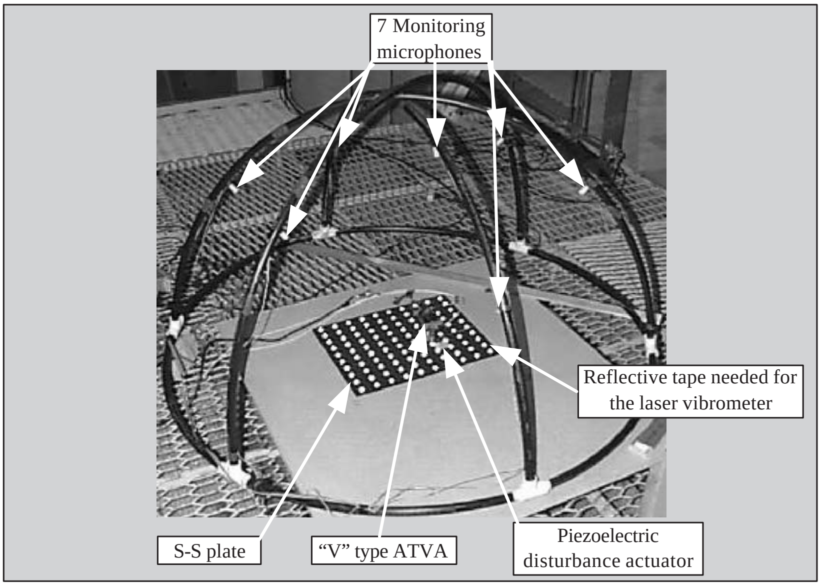

Fig. 4. Experimental set-up used for the global detuning test.

Table 1
Simply supported plate characteristics

Length	38.0 cm
Width	30.0 cm
Thickness	2.1 mm
Young's modulus	2.0×10^{11} Pa
Density	7750 kg/m^3

Table 2
Piezoelectric actuator characteristics

Length	3.79 cm
Width	3.16 cm
Thickness	0.27 mm
Pos. of the center along the X-axis	26.6 cm
Pos. of the center along the Y-axis	9.0 cm
Young's modulus	6.3×10^{10} Pa
d_{31}	2.1×10^{-10} m/V

the seven microphones provided an estimate of the radiated power of the plate and was used as the cost function for the global detuning algorithm. The global detuning algorithm was based upon the least mean squares (LMS) steepest descent gradient algorithm, which will find a minimum of any defined cost function. If the cost function is defined as the square of a linear function of the control inputs and the disturbance, the LMS algorithm will find the global minimum. However, if the cost function is defined as the square of a non-linear function, the LMS algorithm will find a local minimum. Changing the resonant frequency of a device is inherently non-linear; therefore the LMS algorithm will find a local minimum of the cost function. Due to this limitation, a starting point for the LMS algorithm had to be chosen. Since this study was concerned with the effects of detuning an absorber compared to a tuned absorber, the global detuning algorithm was applied after the ATVA resonant frequency was tuned to excitation frequency.

Using a similar form of the well-known update equation used for the LMS control algorithm, the update for the tune frequency can be defined as

$$f_{k+1} = f_k + \mu(-\nabla_k). \tag{1}$$

For this algorithm, the gradient (∇) at iteration k is defined as the finite difference approximation of the change in the time averaged cost function (J) with respect to the actuator tune frequency (f), written as

$$\nabla_k = \frac{\Delta J}{\Delta f} = \left(\frac{J_k - J_{k-1}}{f_k - f_{k-1}}\right), \tag{2}$$

where the cost function is an approximation of radiated sound power defined as the sum of the microphone mean square voltages

$$J_k = \sum_{t=1}^{N} |V_t(\omega)|^2. \tag{3}$$

N is the total number of error signals and $V_t(\omega)$ is the single frequency Fourier transform of the error (microphone) signals.

Implementation of the detuning algorithm used a time averaged gradient LMS (TAG-LMS) algorithm. The controller determines the cost function (J) at time step $k-1$. The controller then perturbs the ATVA changing its tune frequency by $\mathrm{d}f$ (delta frequency). The controller re-measures the cost function determining the cost function (J) at time step k. The controller calculates the gradient $\mathrm{d}J/\mathrm{d}f$ to determine the gradient and steps in the direction that minimizes J. Once the controller reaches a minimum, the gradient will be positive in both directions and the algorithm will stay at the minimum. As with the LMS algorithm, the convergence parameter (μ) must be chosen carefully to ensure proper convergence. The global detuning algorithm was programmed on a digital signal processor board (DSP) mounted in a personal computer (PC).

2.4. Experimental procedure

In order to compare the results obtained when the ATVA is tuned to those obtained with the ATVA detuned, the following experimental procedure was used for each single frequency disturbance tested. First, the ATVA was set to its high limit (i.e., maximum resonant frequency possible, which is around 310 Hz), where the ATVA simply acts as an added point mass on the plate. These measurements are referred to as "baseline" case. Then, measurements of the microphone signals were performed and the plate flexural displacements were measured at 100 points regularly spaced over the plate surface with a laser vibrometer. Once the baseline measurements were completed, then the ATVA was electronically tuned to the excitation frequency and a second series of measurement, are performed, which are referred to as the tuned case. The algorithm for tuning the ATVA is the classic algorithm of driving the signals from the two accelerometers on the ATVA (see Fig. 2) to be in quadrature (i.e., 90° out of phase). Finally, a third series of measurements was made after the global detuning algorithm achieved convergence, which is referred to as the detuned case.

3. Experimental results

Results are presented for two single frequency disturbances. The first frequency of 156 Hz was off-resonance between the (1,1) mode and the (2,1) mode. The second frequency of 256 Hz was on-resonance of the (1,2) mode. The microphone measurements, plate flexural modal amplitudes, and plate flexural displacement measurements are presented in Fig. 5–7 for the 156 Hz case, respectively, and in Fig. 8–10 for the 256 Hz case, respectively.

3.1. Results for the 156 Hz excitation frequency

The 156 Hz case shows that simply tuning the ATVA to the excitation frequency can actually increase the sound pressure levels radiated by the plate. The microphones measured a spatially average increase of 12 dB in this case. By applying the global detuning algorithm, the detuned ATVA reduced the sound pressure levels by an average of 16.5 dB compared to the baseline case, as seen in Fig. 5. Fig. 6 shows that by tuning the ATVA, the modal contribution of the strong radiator modes [11,12] (i.e., odd–odd order modes) was increased, and the (1,1) mode was increased by a factor of 4. This is consistent with a 12 dB increase in the sound pressure levels. However, detuning of the ATVA led to a decrease of the modal contributions, especially for the

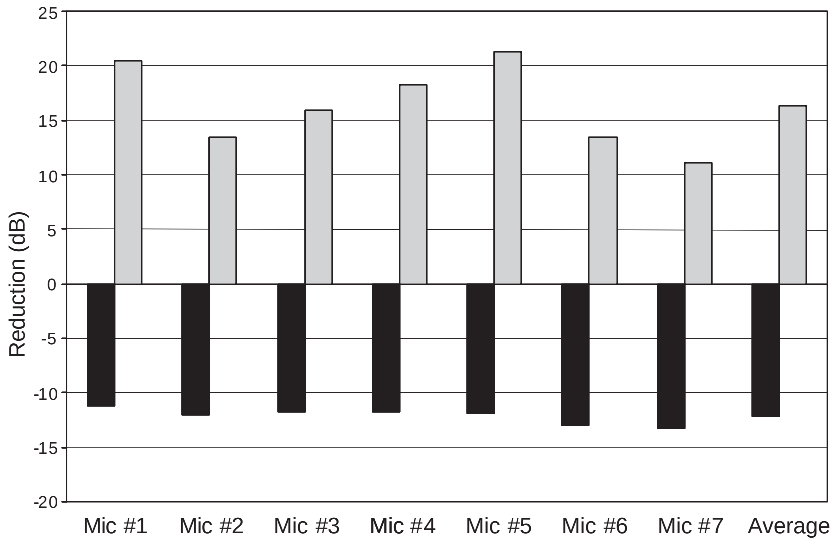

Fig. 5. Microphone results at 156 Hz. ■, ATVA tuned, □, ATVA detuned.

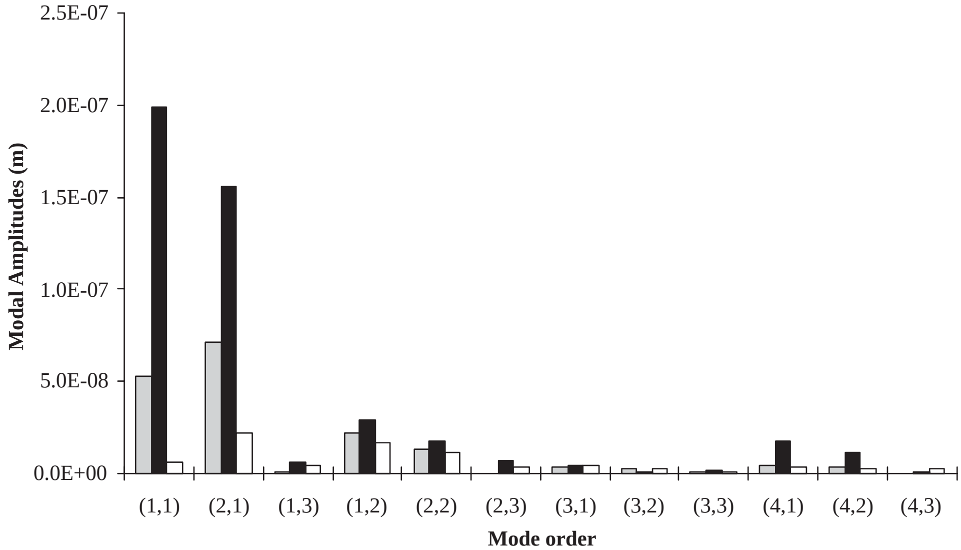

Fig. 6. Modal amplitudes at 156 Hz. ATVA resonant frequency: ▢, high limit (baseline), ■ ATVA tuned; □, ATVA detuned.

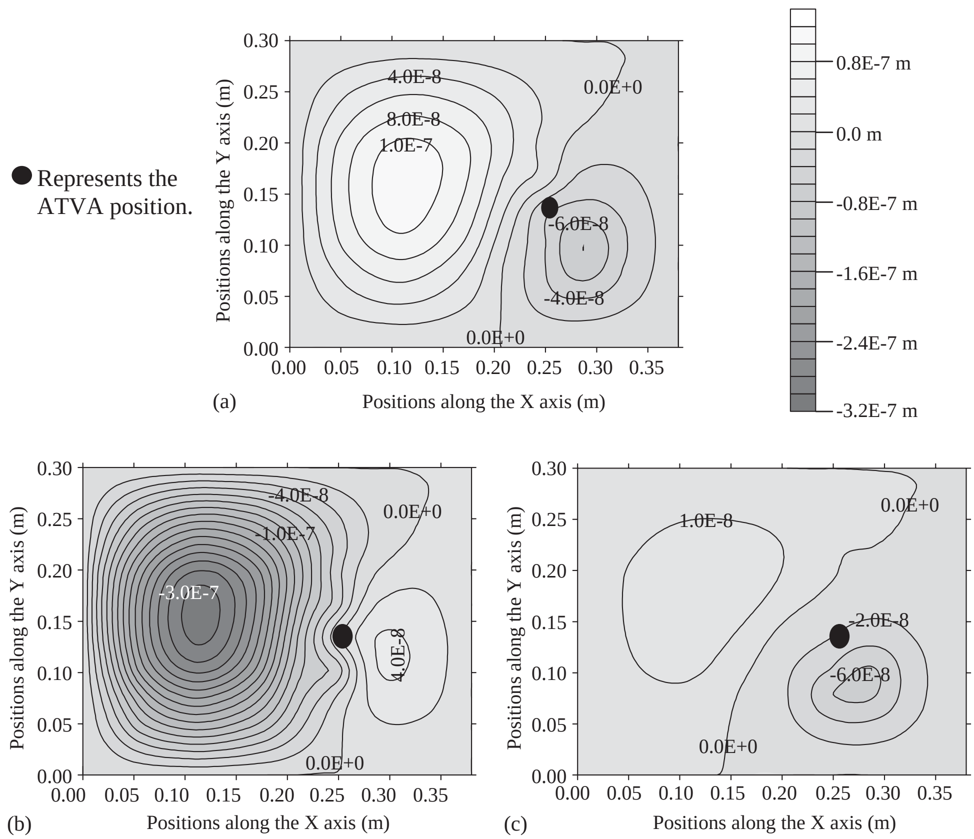

Fig. 7. Plate flexural displacement at 156 Hz: (a) baseline (ATVA set at high limit), (b) ATVA tuned, (c) ATVA detuned.

(1,1) mode, which is the most efficient radiator [11,12]. Comparing the modal amplitudes of the detuned case to the tuned case in Fig. 6, it is seen that the detuned modal response decreased the modal amplitudes of the efficient acoustic radiator modes ((1,1) and (2,1)) while increasing the levels of the less efficient radiator modes ((1,3) and (2,3)). The plate flexural displacement, shown in Fig. 7(a), indicates the plate acts as a dipole type radiator for the baseline case. Note the circle on the plate indicates the position of the ATVA. Tuning the ATVA resonant frequency to the excitation frequency forces the plate displacement to be zero at the base of the absorber, as indicated in Fig. 7(b). For this case, the plate deformation is similar to a monopole type radiator compared to the baseline case. Note the plate flexural displacements are in agreement with the increased contribution of the strong radiator modes discussed previously. In addition, the close spacing of the contour lines clearly show the increased vibration levels. Detuning the ATVA reduced the vibration levels over the surface of the plate, as seen in Fig. 7(c). It should be noted that the global detuning algorithm found the optimal resonant frequency of the ATVA to be 110 Hz for an excitation frequency of 156 Hz. This represents a tune ratio (*excitation frequency/ ATVA resonant frequency*) of 1.4.

3.2. Results for the 256 Hz excitation frequency

In the case of 256 Hz, an average reduction of sound pressure levels of 4.9 dB was achieved by tuning the ATVA to the excitation frequency. The global detuning algorithm was able to reduce the sound pressure levels by an average of 25.1 dB. This represents an increased reduction of 20.2 dB compared to the tuned case, as seen in Fig. 8. Fig. 9 indicates that the further reduction obtained by detuning the ATVA can be mainly associated with an increase in vibration reduction of all the modes. The plate flexural displacements, seen in Fig. 10(a), show the plate acts mainly as

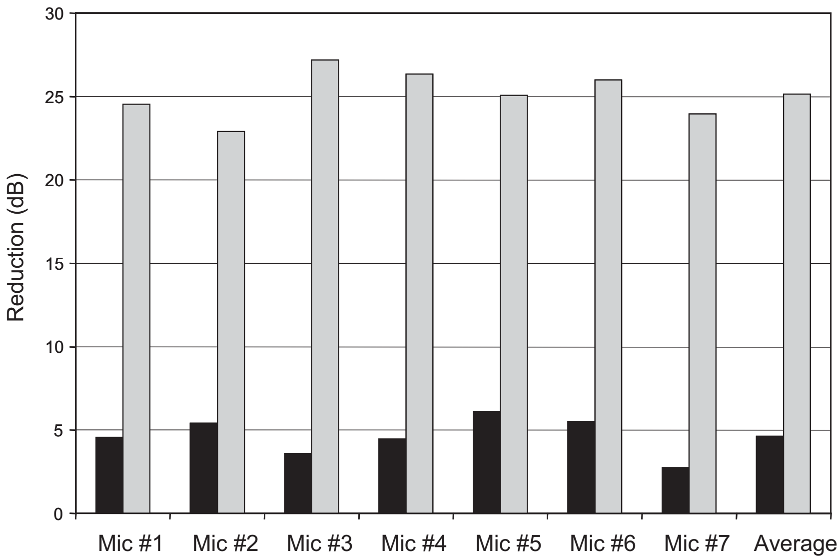

Fig. 8. Microphone results at 256 Hz: ■, ATVA tuned; □, ATVA detuned.

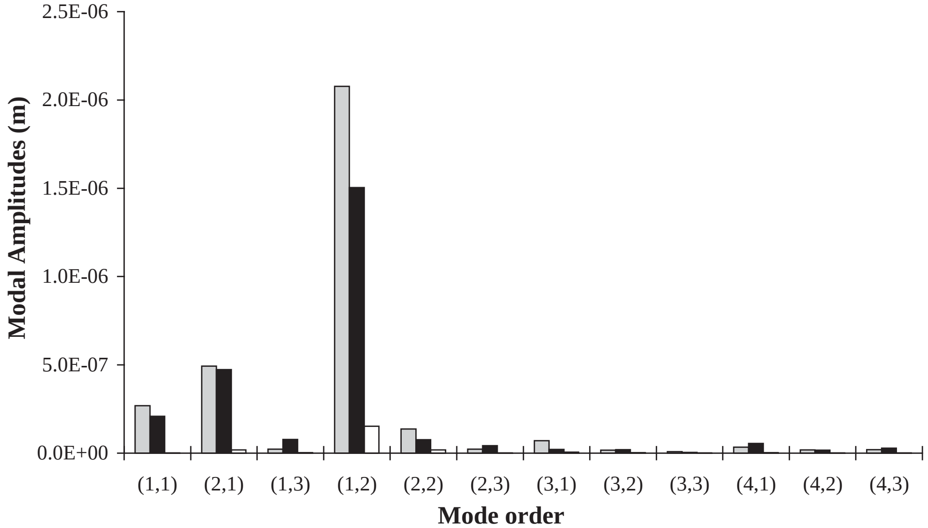

Fig. 9. Modal amplitudes at 256 Hz. ATVA resonant frequency: ▨, high limit (baseline); ■, ATVA tuned; □, ATVA detuned.

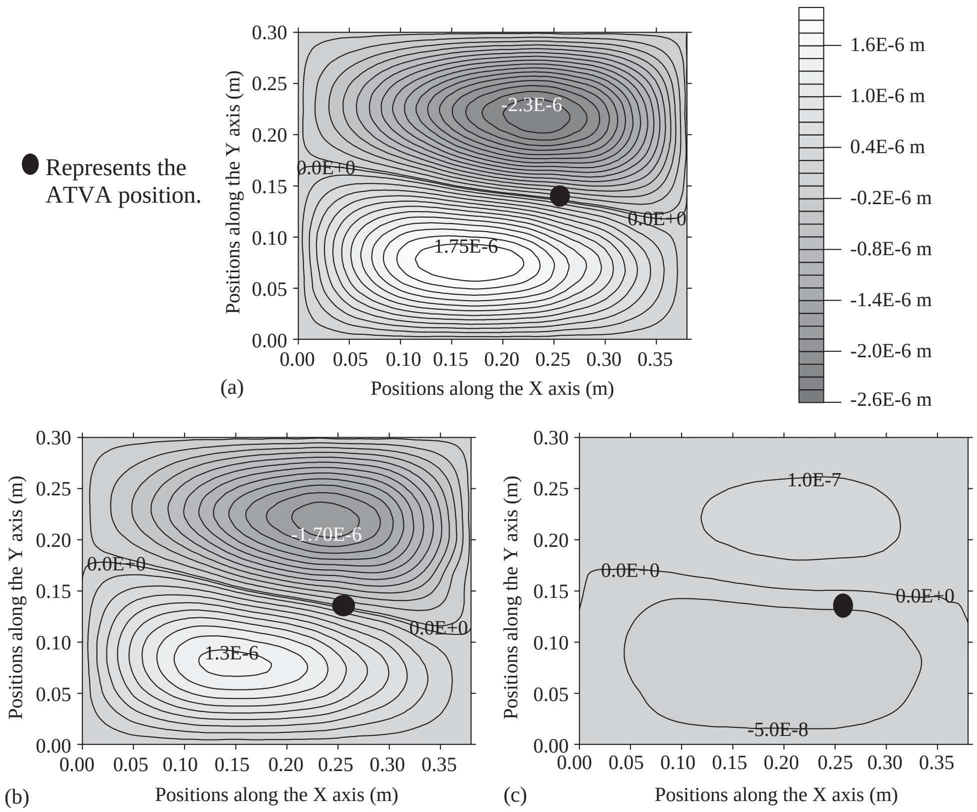

Fig. 10. Plate flexural displacement at 256 Hz, (a) baseline (ATVA set at high limit), (b) ATVA tuned, (c) ATVA detuned.

a dipole type radiator when the ATVA is at its high resonant frequency limit, i.e., baseline case. Due to the position of the ATVA, the tuning of the ATVA to the excitation frequency practically does not change the deformation of the plate, Fig. 10(b). It only reduces the vibration levels by a small amount. Due to the small quantity of contour lines, Fig. 10(c) clearly shows that by detuning the ATVA the plate vibration levels were greatly reduced. The optimal resonant frequency of the ATVA determined by the global detuning algorithm for this excitation frequency of 256 was 165 Hz. This represents a tune ratio of 1.55.

3.3. Summary

The experimental results for the two single frequency disturbances presented in this section demonstrated that the concept of adjusting the ATVA characteristic using an acoustic cost function yield much larger sound attenuation than simply tuning the ATVA to the excitation frequency. The absorber tune ratio of 1.4 and 1.55 obtained from the experimental results was surprising since the analytical work [8,9] demonstrated tune ratios close to 1.0 for high Q absorbers. It is believed that the tune ratio will be closer to 1.0 as the number of vibration absorber is increased and/or the frequency of interest is also increased.

4. Conclusions

An experimental study to minimize the sound radiated by a simply supported plate using an ATVA was presented. A new and original ATVA was developed and implemented experimentally. A comparison of the sound attenuation obtained by tuning the ATVA versus the detuning of the ATVA was performed. Tuning the ATVA to the excitation frequency represents a vibrational cost function and does not guarantee the best possible attenuation of sound radiated by the plate. In some cases, tuning the ATVA can actually increase the radiated sound levels, as shown for the 156 Hz off-resonance excitation frequency. A detuning algorithm was programmed using an acoustic cost function implemented by summing the sound pressure levels at different points in the far field of the plate. The experimental results clearly showed that using an acoustic cost function to detune the ATVA can achieve better control of the radiated acoustic field compared to a vibrational cost function.

Acknowledgements

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