inter.noise 2014
MELBOURNE AUSTRALIA
16-19 NOVEMBER

The effectiveness of particle damping for use on vertical surfaces

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ABSTRACT

Particle damping has been shown to effectively reduce vibrations on complex structures and systems. Particle damping relies on intimate contact with the vibrating surface to be effective. Previous studies have focused on applying particle damping to horizontal structures where gravity ensures contact with the surface. However, many applications would make use of this damping technology on vertical surfaces. This paper investigates the added challenge and methodology of packaging particle damping for vertical applications and studies the variables that impact performance. The outcome will establish the key system parameters which drive performance and allow the capability to optimize particle damping effectiveness for these types of systems.

Keywords: Particle, Damping, Structure I-INCE Classification of Subjects Number(s): 47.3

1. INTRODUCTION

Advanced Particle Damping (APD) is a vibration absorbing media comprised of uniquely prepared elastomeric particles. Previous studies on “Vibration reduction options for aircraft floors” (1), evaluated the effectiveness of APD applied to horizontal structures which yielded effective results non-typical of classical particle damping treatments. This work extends from the previous study by adding the complexity of applying APD to vertical structures and evaluating the proper method for yielding effective results and identifying the factors to predict performance.

On horizontal surfaces, the mode of vibration to be damped is collinear with the gravitational force. Therefore, the level of structural input excitation is directly related to the level of output particle movement. A 1G input acceleration will allow the particles to overcome their own weight and move freely. Less the 1G will allow restricted motion, while greater than 1G will allow excessive movement assuming a container with free space. Previous work has exposed an optimal level of input excitation where the particles become ‘fluid’, but remain in contact, which produces the highest damping result.

Applying APD to vertical structures becomes more complex as a result of the gravity force being 90 degrees from the vibration input direction. The input level required to excite the damping media is not as straight forward due to the restrictions imposed by the container.

This paper explores the methodology of proper placement of APD on a vertical plate, and how much APD is required to optimally provide a benefit.

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2. System Definition & Characterization

2.1 Plate Characterization

A carbon steel plate was selected with dimensions of 39.25cm high (The plots were carried out with 41.02 cm) x 21.02cm wide x 2.1mm thick. Simulations of the plate with free boundary conditions were conducted to identify the resonant mode shapes. The first mode resonance is 65 Hz. Subsequent modes were calculated at 88 Hz, 179 Hz, 190 Hz and 280 Hz. The mode shapes are shown in Figures 1 through 5. The blue lines are nodal lines, i.e. lines with zero displacement. Red areas indicate the maximum displacement.

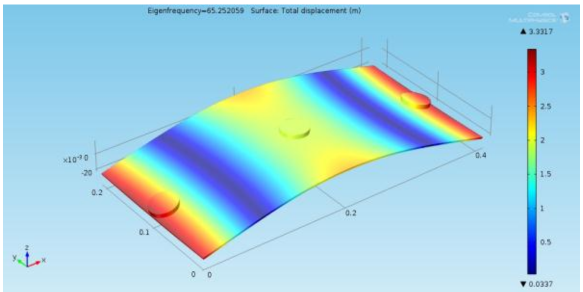

Figure 1 – First Mode Shape, 65 Hz

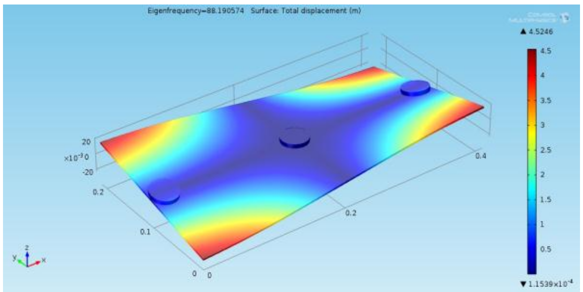

Figure 2 – Second Mode Shape, 88 Hz

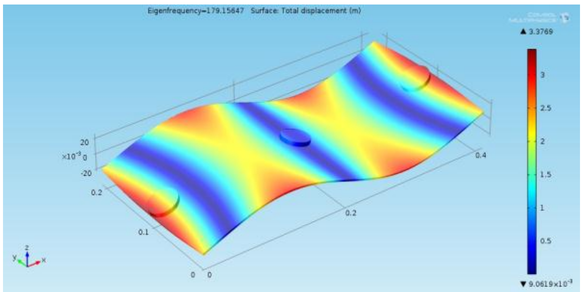

Figure 3 – Third Mode Shape, 179 Hz

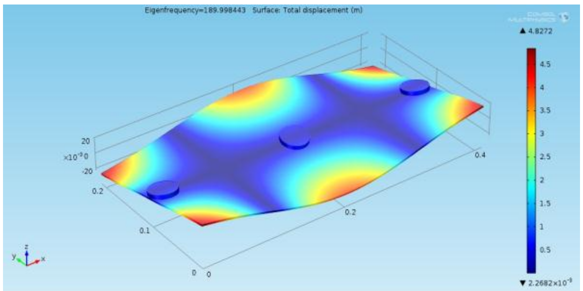

Figure 4 – Fourth Mode Shape, 190 Hz

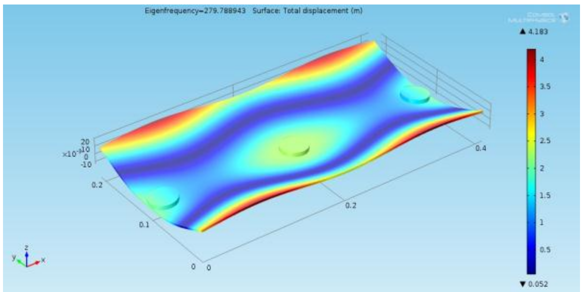

Figure 5 – Fifth Mode Shape, 280 Hz

The steel plate was vertically suspended using nylon cords from a support structure as shown in Figure 6. A shaker was attached to the lower right corner of the plate using a rigid stinger. A force sensor is attached in-line with the stinger to measure the input force. An accelerometer was used to measure the acceleration at the stinger attachment location. A second accelerometer was used to measure the response acceleration across 60 discretized zones as seen in Figure 6.

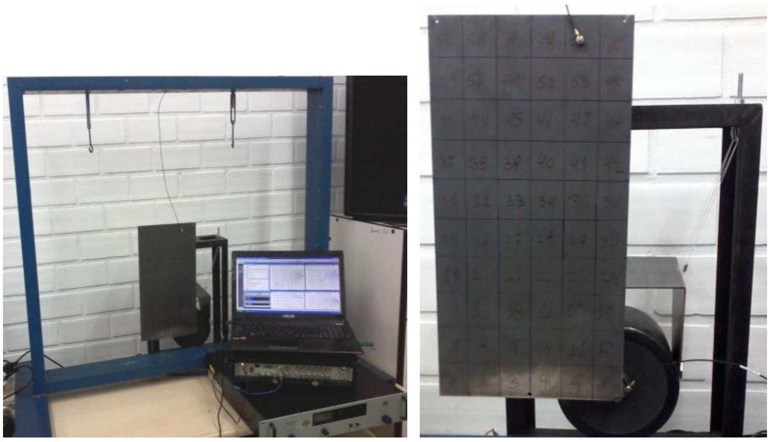

Figure 6 – Plate & shaker setup with 60 discretized zones

Testing of the plate resulted in a first mode frequency response of 60 Hz using a 0.25g input acceleration.

2.2 Spatial-average mean square inertance Curves

In order to quantify the plate acceleration level with a single representative curve, a spatial-average mean square inertance level was calculated (2). The spatial-average mean square inertance curve is obtained with the arithmetic addition of squared inertance levels in each of the 60 elements of the discretized plate. This curve in dB is given as:

$$\left\langle \overline{I^2} \right\rangle_{dB} = 10\log_{10}\left[\frac{1}{N}\sum_{i=1}^{N}\left(I_{rms}^2\right)_i\right] \tag{1}$$

Where: N = the number of spatial elements
I = output acceleration / input force for a specific frequency

2.3 Loss Factor (Damping) Calculation of First Modal Frequency

The method used to measure the damping loss factor of the first vibration mode of plate was the half-power bandwidth method (3). This method uses the frequency response function (FRF) of the structure, such as the inertance curve, and calculates the damping loss factor through the following equation (See Figure 7):

$$\eta = \frac{\Delta\omega}{\omega_0} \tag{2}$$

where η is the loss factor damping of a structural vibration mode, $\Delta\omega$ is the bandwidth defined by the frequencies corresponding to the half-power point (-3 dB from peak value), and ω_0 is the natural frequency of the structural vibration mode.

The half-power bandwidth method was applied in order to measure the damping loss factor of first vibration mode of plate with nothing, with 3 EPDs and with equivalent masses.

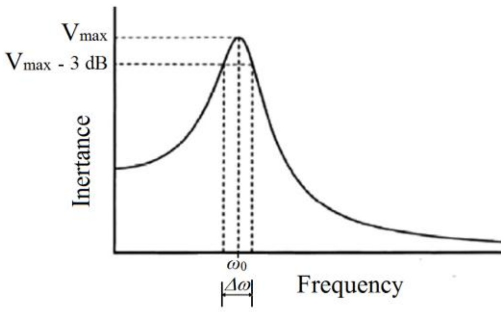

Figure 7 – Half-power bandwidth method for damping loss factor measurement

(Vmax is the maximum value of the inertance curve)

The drive point and response point were chosen within the areas of greater displacement of the first vibration mode (see Figure 1). The drive point was localized in the number element six, as show in Figure 6, and the response point was localized in the number element 58, this is in the center of opposite side. Thus, an inertance curve was obtained from the acceleration in the response and the force applied from the drive point.

The system was excited using a swept sine signal with a bandwidth of 20 Hz centered in the first natural frequency mode. The input acceleration has a peak value in this natural frequency mode and was set to be equal to 0.25 g.

Figure 8 shows the spatial-average mean square inertance of the plate with no additional damping treatment applied. This curve is used as the baseline curve for subsequent comparisons of damping treatments.

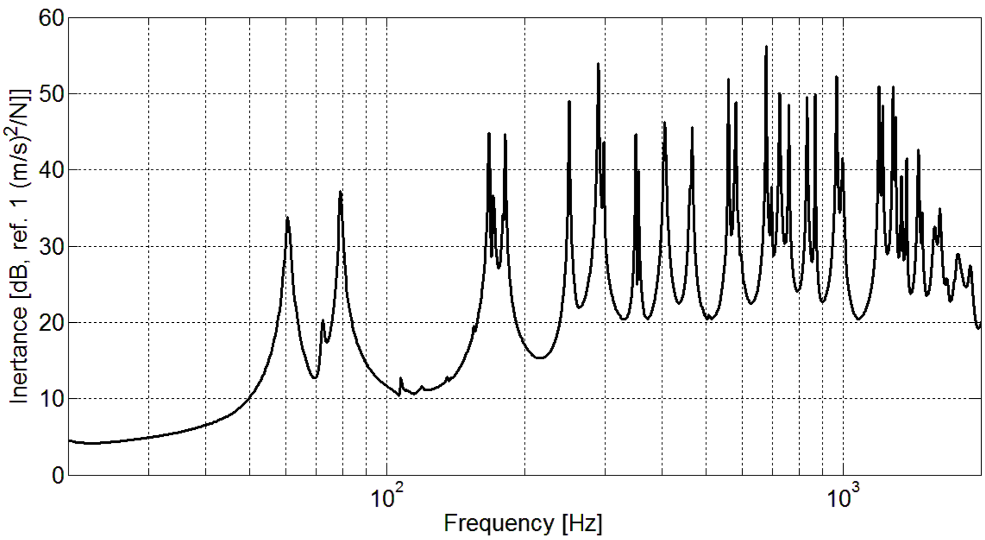

Figure 8 – Spatial-average mean square inertance curve of the plate

Figure 9 shows the zoomed inertance curve of the plate about the first natural frequency mode with no additional damping treatment applied. The damping loss factor obtained for this vibration mode using the half-power bandwidth method of Equation (2) was 1.8 %. This curve is used as the baseline curve for subsequent comparisons of damping treatments.

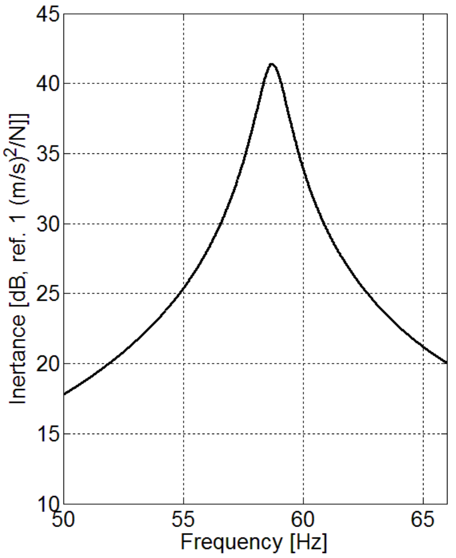

Figure 9 – Zoomed inertance curve of plate about the first natural frequency mode

3. Advanced Particle Damping (APD) Characterization

3.1 APD Container

A cylindrical aluminum container having a first natural frequency above 800 Hz was selected to house the APD media. This frequency is well outside the frequency range of interest, and therefore can be regarded as a rigid body. In addition, the container mass of 4 grams is very small compared to the mass of the plate at 1,357 grams which will minimize any mass effects or frequency shifts in the study.

3.2 APD Characterization and Selection

Experimental performance curves of the APD damping loss factor were plotted as a function of frequency and acceleration for various types APD media and quantity as shown in Figures 10 to Figure 13. The various APD media included 20 grams of a soft elastomer (EEC163001), 20 grams of a stiff elastomer (EEC166501), 40 grams of the soft elastomer (EEC163001), and 40 grams of the stiff elastomer (EEC166501).

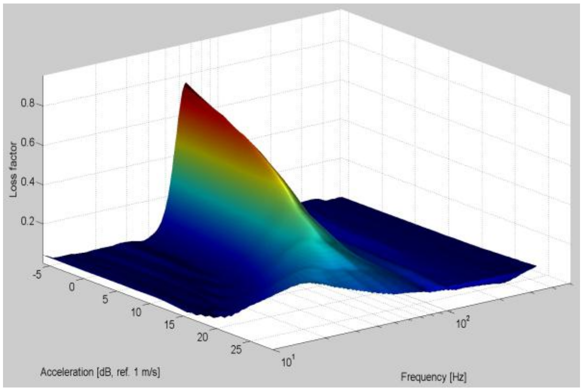

Figure 10 – 20 grams, Soft Elastomer

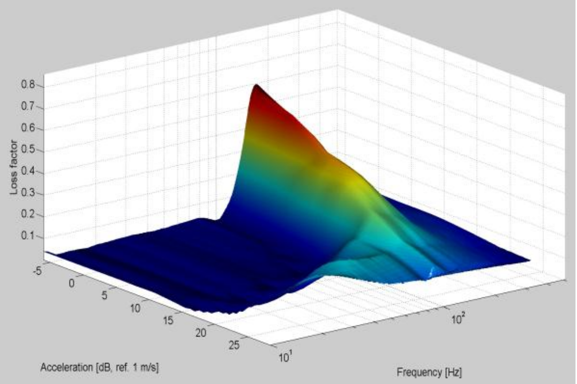

Figure 11 – 20 grams, Stiff Elastomer

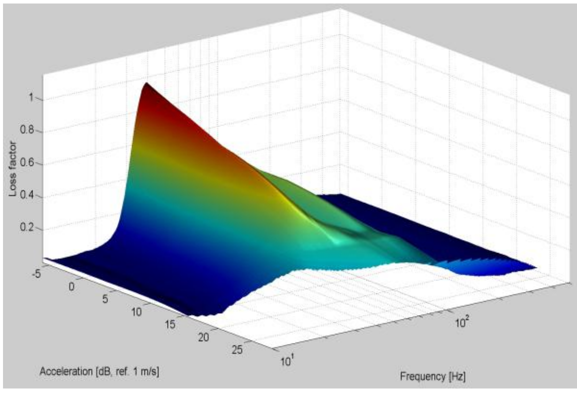

Figure 12 – 40 grams, Soft Elastomer

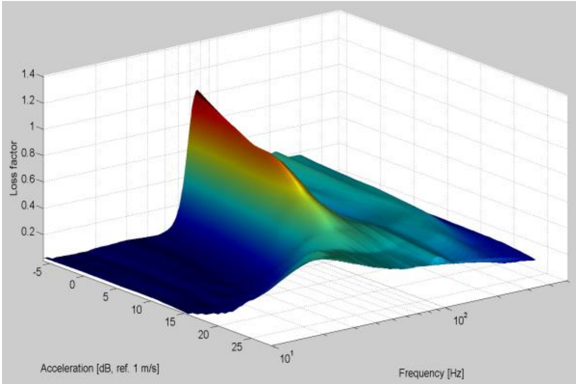

Figure 13 – 40 grams, Stiff Elastomer

As seen in these plots, APD has various damping levels dependent on the input frequency and acceleration levels, η (f , a). In general, the lower the input acceleration, the higher the damping level, but with a narrow frequency response range. The higher the input acceleration, the lower the damping level, but with a wide frequency response range.

For example, the test with 40 grams of the soft APD particles works efficiently when submitted to frequencies close to 40 Hz. The test with 40 grams of stiff APD particles works efficiently when submitted to frequencies close to 80 Hz, However, the optimal frequency response of the APD media varies significantly according the acceleration input. When submitted to -30 dB of acceleration, the damper with 40 grams of stiff particles works efficiently at close to 85 Hz, and when submitted to 18 dB of acceleration work efficiently at 59 Hz.

These trends with the APD media follow what is expected in terms of frequency and strain sensitivity of the base elastomer material.

To determine the appropriate APD type and amount, two conditions need to be satisfied:

1. The two damping regions must be separated with no overlap to prevent interference with the measurements. Previous studies (1)(4)(5) identified a fluidizing region which appears up to approximately 50 Hz. This fluidizing region occurs when the APD particles begin to move enough as to allow loss of contact, and this damping, at higher input displacements. Therefore, the frequency range of maximum damping of the appropriate APD should be above 50 Hz to avoid the fluidized region of the particle response. The APD studied that met this condition are those with 20 and 40 grams of stiff elastomer particles (EEC166501).

2. The second parameter to be determined was the amount of APD. Previous studies showed that, as the mass of APD particles grow (while maintaining the individual particle mass), the damping also increases. Consequently, the APD with 40 grams of stiff elastomer (EEC166501) particles was chosen. Addition to its greater damping, this APD presents a lower frequency variation of the maximum damping compared with the APD with 20grams (see Figure 14).

The frequency of maximum damping for the APD with 40 grams of stiff elastomer (EEC166501) corresponding to an acceleration equal to 0.25 g (2.4517 m/s or 7.8 dB ref: 1 m/s) is equal to 65 Hz (see corresponding black lines in Figure 14). This maximum frequency response matches well with the steel panel’s first frequency response mode.

4. APD Placement

4.1 Where to apply the APD damping treatment?

The first frequency mode shape of the plate with free boundary conditions presents two nodal lines and three areas of displacement. Therefore it is desirable to place at least one EPD in each of these areas (see Figure 15).

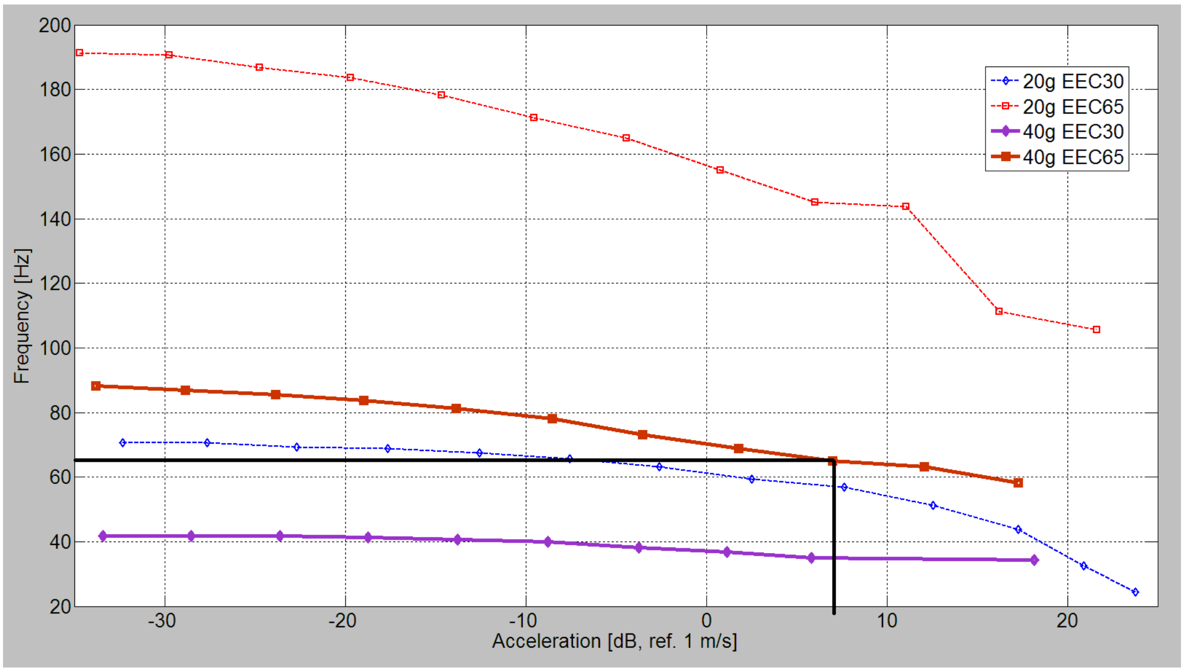

Figure 14 – Maximum damping loss factor as a function of acceleration and frequency

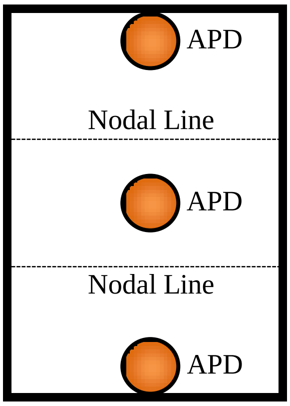

Figure 15 – Nodal lines of the first mode and proper placement of APD dampers for damping the first mode of vibration.

5. Test Results

5.1 Test Treatments

Three test configurations were tested; the plain steel plate (baseline), APD media in aluminum containers and equivalent APD masses (see Figure 16).

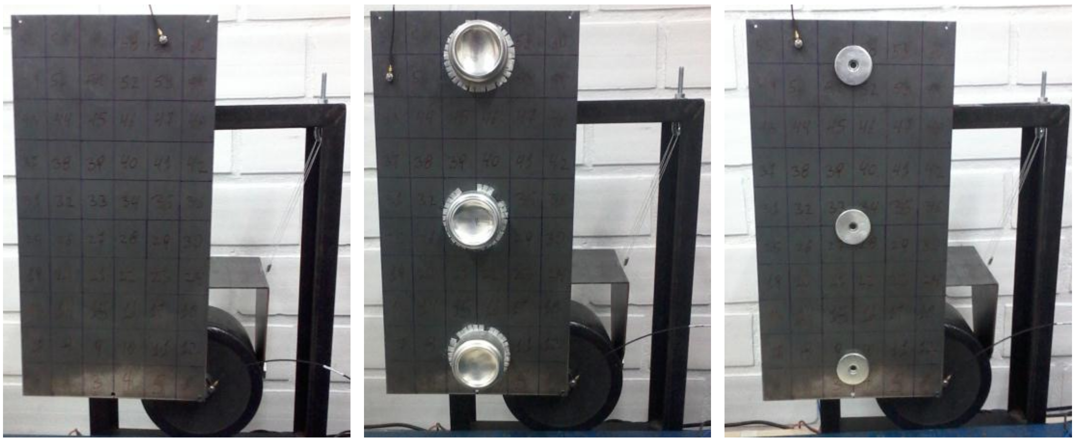

Figure 16 – Plain steel plate, APD damper, and equivalent APD masses, respectively.

5.2 APD Test Results

Figure 17 shows the spatial-average mean square inertance curves of the plain steel plate, and with 3 APDs attached at the modal frequency zones. The attenuation of peak values produced by adding damping to plate by the APD particle dampers is clearly observed. This attenuation is present from the first natural frequency up to approximately 1300 Hz. This demonstrates the high frequency filter effect classically seen in other types of damping treatments.

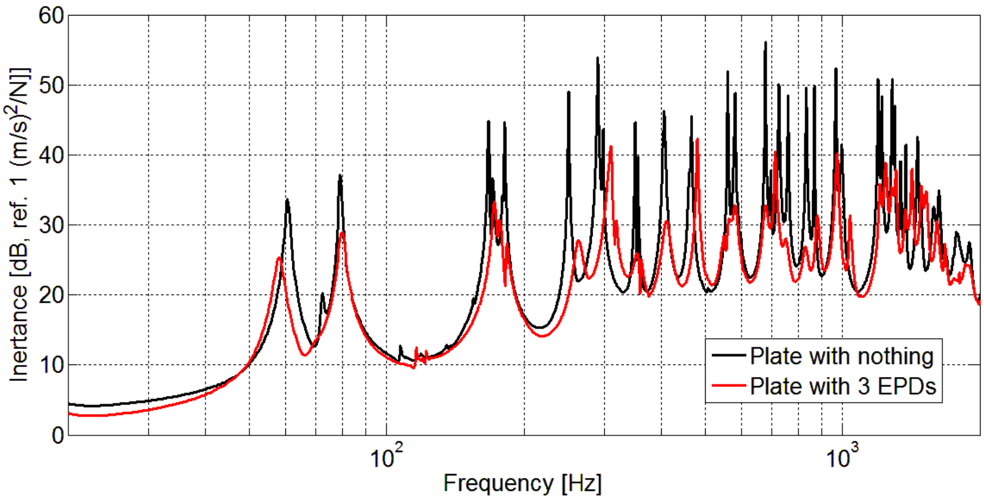

Figure 17 – Spatial-average mean square inertance curves of the plain steel plate and with 3 APDs

5.3 Equivalent Mass Test Results

Figure 18 shows the spatial-average mean square inertance curves of the plain steel plate, and with 3 masses attached which were equivalent to the APD masses, located at the modal frequency zones. The curves are similar. The added mass did not increase the damping with only a slight decrease in natural frequencies.

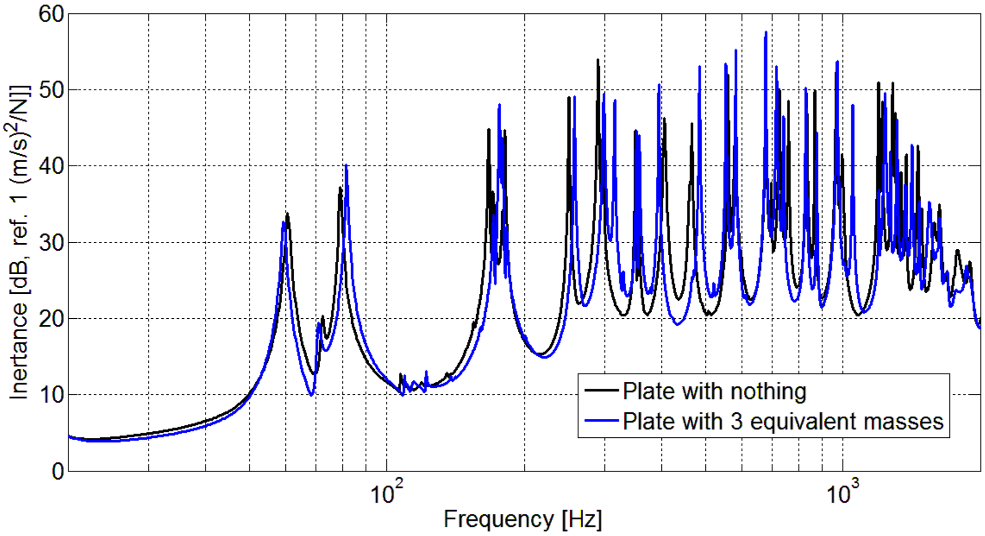

Figure 18 – Spatial-average mean square inertance curves of the plain steel plate and with 3 equivalent masses to the APDs

Figure 19 shows the comparison of the spatial-average mean square inertance curves of the plate with 3 equivalent masses and with 3 APDs. A great attenuation of peak values of the plate is observed.

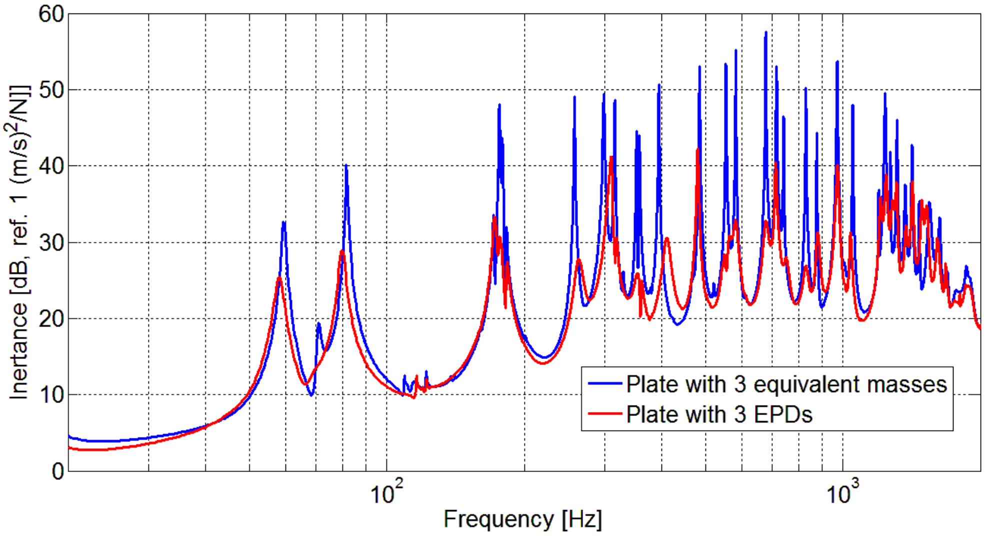

Figure 19 – Spatial-average mean square inertance curves of the 3 APDs and equivalent masses

5.4 Half-Powered Damping Results

Figure 20 shows the results of the half-power bandwidth method applied at the first natural frequency of vertical plate without treatment, with 3 equivalent masses, with 3 APDs and horizontal plate with the same 3 APDs. The inertance curve of the horizontal plate with 3 APDs is below other curves due to the force necessary to move the plate is greater. The inertance is the acceleration in the response divided by the force applied to the drive point. However, this situation did not influence the damping loss factor η obtained. The damping loss factor η of the plate with nothing and with 3 equivalent masses are equal to 1.8%. The damping loss factor η of the plate with 3 APDs present a greater value equal to 4.3%.

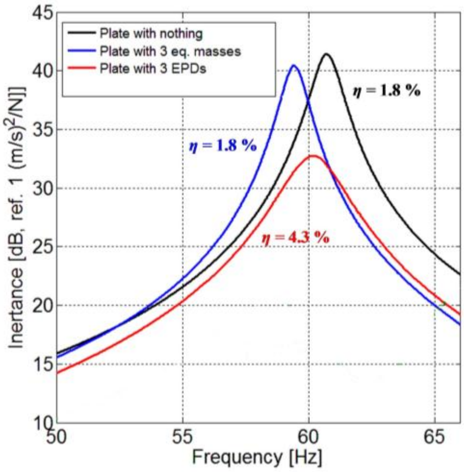

Figure 20 – Comparison of inertance curves about the first natural frequency mode

6. Conclusions

Three variables were evaluated to optimize the use of Advanced Particle Damping (APD) on vertical systems. These choices included the type of APD, the amount of APD, and where to apply the APD.

The frequency response of APD is affected by both type and amount of material used. The soft elastomer material responded to lower frequencies than the stiff elastomer media. The higher the amount of APD used, the lower the frequency response performance and vice versa. By combining these two types of variables, an appropriate frequency response of APD type and amount can be targeted to match the primary modes of an attached structure.

The location of the APD was determined from modal simulations to identify the zones of greatest displacement. The predicted modal responses of the plate suggest that the first mode should produce the highest input level where the APD’s are located. Conversely, the second mode should produce almost no input since the APD’s are coincident with modal node zones for this response. An interesting result seen in Figure 17 shows the amount of damping to be similar between the first and second modes. This suggests that the localized input level where the APD’s are attached is not of significant consequence.

The APD did increase the structural damping loss factor of a vertically oriented steel plate at the first modal frequency from 1.8% to 4.3%. The added damping does not appear to be a result of the increased mass since the test results from adding equivalent solid masses to the plate yielded the same damping value as the untreated plate.

Results from these tests were compared to that of the same system turned in a horizontal orientation. Damping values for the horizontal system increased from 4.3% to 6.6% from the vertical system. This result suggests that more intimate contact between APD and the vibrating source is desired to maximize damping performance of the vertical application, as opposed to relying on a rigid container to transmit the forces. Future work will consider inclining the APD containers where gravity will improve contact and more closely represent a horizontal configuration.

REFERENCES

1. Bustamante M, Gerges S, Cordioli J, Azevedo G, Weisbeck J. Vibration reduction options for aircraft floors. Proc INTER-NOISE 12; 19-22 August 2012; New York City 2012.
2. F. Fahy and P. Gardonio, Sound and Structural Vibration (Second Edition) Radiation, Transmission and Response, 2nd ed. Oxford: Academic Press, 2007, p. 633. Chapter 3.)
3. Ralph E. Blake, Basic Vibration Theory, Chap 2 in Harris’ Shock and Vibration Handbook, edited by Allan G. Piersol and Thomas L. Paez, Mc Graw Hill, 6th ed., New York, (2010).
4. M. Bustamante, S. Gerges, J. Cordioli, O. Martin, J. Weisbeck and M. Ott. “Experimental study on some parameters that affect the performance of an elastomer particle damper”. ICA 2013, Montreal, Canada, 2-7 June, 2013, Vol. 19
5. M. Bustamante, S. Gerges, V. Soares, O. Martin, J. Weisbeck and M. Ott. "Experimental curves for damping as a function of acceleration and frequency of an elastomer particle damper called EniDamp™". Inter Noise 2013, Innsbruck, Austria, 15-18 September, 2013.