ANALYSIS OF THE APPLICATION OF THE POWER INJECTION METHOD TO DAMPING EVALUATION OF REINFORCED STRUCTURES

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Abstract. *Ribbed-stiffened panels are structures widely used in the construction of aircraft, rockets, satellites, ships and other applications where you need light weight and good resistance. The vibro-acoustic analysis has great importance in the design phase of such structures, as well as in the project of appropriate treatments to reduce noise and vibration. However, one of the most important dynamic properties of a structure is the damping loss factor, and one of the most used methods in its determination is the Power Injection Method. In this work the Power Injection Method (PIM) is used to evaluate the damping loss factor of reinforced structures. This method is based on the analysis of the steady state vibration response of the structures. Results from many controlled experiments are presented and they provide means to investigate the better positions to locate the excitation and response points, either on the skin or on the beams. Additionally, an investigation of the necessary number of such points is carried out. Some new results concerning the sound radiation of fuselage panels, with and without viscoelastic materials, are presented. Finally, based on the obtained conclusions, some recommendations viewing the better application of the PIM are presented.*

Keywords*: vibration, reinforced structures, power injection method, damping loss factor*

1. INTRODUCTION

A good estimate of the damping loss factor is an important design parameter allowing the creation of efficient damping treatments. For Statistical Energy Analysis (SEA) purposes, the damping loss factor is usually estimated through the Power Injection Method (PIM). For many subsystems the overall response level is inversely proportional to the damping level. This is represented in the SEA power balance equation for an isolated subsystem by (Lyon and DeJong, 1995):

$$\Pi_{in} = \Pi_{diss} = 2\pi f \eta E_{tot} \,, \tag{1}$$

where Π_{in} is the input power, Π_{diss} is the dissipated power, and E_{tot} is the total kinetic energy of the subsystem at frequency f (Hz). The quantity $2\pi\eta$ is the ratio of the energy dissipated per cycle of oscillation to the total energy in the subsystem.

Three methods are widely known to evaluate the damping loss factor of the structures: *the decay rate*, *half-power bandwidth* and the *power injection method*. But no method is applicable to all situations, because each method has limitations and requirements. The decay rate method is limited for highly damped structure because the source excitation and response electronics have a finite response time that will put on an upper limit on the measurable decay rate. Half-power bandwidth method can be applied only to the determination of damping of a single mode. For this reason its use is limited to simple structures that do not have overlapping modal responses (Lyon and DeJong, 1995). The PIM, otherwise, constitutes the most widely used technique because the measurements result is fundamentally unbiased at the natural frequencies of individual, well separated modes and for frequency band averaged results in the presence of many modes.

Bies and Hamid (1980) compared the measurements of the loss factor through PIM and by the reverberant decay method and concluded that they lead to different results. They also concluded that loss and coupling loss factor may be determined by inversion of the power balance equations and, provided that the equations are well conditioned for inversion, good results may be expected.

Clarkson e Pope (1981) pointed out the methodology to determine by and indirect way the loss factor and the modal densities of flat plates and cylinders through decay method. They emphasize the importance of the impedance punctual correction and also used the results got by PIM to confirm those got by the decay method.

Brown and Norton (1985) provided some comments on the experimental determination of modal densities and loss factor based on his works on lightly damped shells and plates. One of their conclusions is that the steady state power flow technique for loss factor estimation is critically dependent on the estimation of input power. Other important conclusion is the added mass that appears between the measurement transducer and the structure. It has to be accounted for when using the point mobility technique for modal density estimates. For a flat plate these errors can be significant at much lower frequencies.

De Langhe (1996) proposed a methodology to determine the parameters of SEA based on PIM and consider the decay method as complement of PIM. He also emphasize that PIM is the most suitable method to determine the loss factor and the coupling loss factor for a system.

Whenever it is necessary to determine the loss factor of a complex structure, some questions arise like the place where the structure must be excited in order to get a better distribution of the input energy, the enough amounts of response points which represent the system total kinetic energy and the limitations of the PIM. The objective of this paper is to answer these questions.

2. THE POWER INPUT METHOD (PIM)

The power injection method is a powerful method for obtaining frequency-averaged loss factor of structures under steady state vibration. It is based on the basic SEA power balance, Eq. (1).

For structural system under steady state conditions, the damping loss factor in the frequency band $\boldsymbol{\eta(\omega)}$ can be defined by (Bloss and Rao, 2005):

$$\eta(\omega) = \frac{E_{in}}{E_{SE}} \tag{2}$$

where E_{in} is the input energy and E_{SE} is the strain energy.

This equation considers that, in steady state conditions, the total input energy is dissipated by the damping of the system in the heat form.

Based in the power spectral density of the input force G_{ff}, the input energy can be defined by (Bloss and Rao, 2005):

$$E_{in} = \frac{1}{2\omega}\mathrm{Re}[Y_p(\omega)]G_{ff}(\omega) \tag{3}$$

where $Y_p(\omega)$ is the driving point mobility function.

Some assumptions were done to estimate E_{SE}. The first is that the strain energy is replaced by the kinetic energy E_{KE}, because it is impossible to measure the force and the velocity for each response point. Replacing strain energy with kinetic energy is valid assumption at natural frequencies or when average across many modal resonances within a frequency band (Bies and Hamid, 1980). The total kinetic energy of a system can be evaluated by:

$$E_{KE} = \frac{1}{2}\int_v \rho . G_{ii}(\omega).dv \tag{4}$$

where $\boldsymbol{\rho}$ is the system density , $\boldsymbol{v}$ is the volume of the system, and $\boldsymbol{G_{ii}}$ is the power spectral density function of the velocity vector. For experimental application, it is possible to approximate a continuous system by a discrete system composed of small parts. On this form, the kinetic energy of a discrete system can be evaluated replacing the volume integral in Eq. (4) by a summation:

$$E_{KE} = \frac{1}{2}\sum_{i=1}^{N} m_i G_{ii}(\omega) \tag{5}$$

where $\boldsymbol{N}$ is the number of measurement locations, $\boldsymbol{m_i}$ is the mass of the discrete portion of the structure, and $\boldsymbol{G_{ii}}$ now is the power spectral density of the velocity response of each response point. In order to reduce errors in the estimation of the loss factor, it is necessary to consider a considerable amount of response points.

For systems with linear behavior, the transfer mobility function can be related to the power spectral density functions through the following equation:

$$\left|Y_t(\omega)\right|^2 = \frac{G_{ii}(\omega)}{G_{ff}(\omega)} \tag{6}$$

where $\boldsymbol{Y_t}$ is the transfer mobility function. This equation is written under the assumption that the measurements are taken with minimal noise content (Bloss and Rao, 2005).

When it is considered that the mass by unit area is equal and homogenous in all the panel, or when all measurement points uniformly spaced throughout the system, the damping loss factor in the frequency band ***η(ω)*** can be evaluated by introducing Eqs. (3), (5) and (6) into Eq. (2):

$$\eta(\omega) = \frac{\langle \mathrm{Re}[Y_p(\omega)] \rangle}{\omega . M . \langle \overline{|Y_t(\omega)|}^2 \rangle} \quad (7)$$

The input power can be calculated with a simultaneous measurement of the force and the velocity at the driving point using an impedance transducer.

For the measurement of the average damping of a group of modes resonating in a frequency band, one must be sure that all the desired modes are responding approximately equally. The measurement should be repeated and averaged for multiples excitation and response points in order to spatially average over different mode shapes. The excitation points should be located near high response regions of the subsystem, where the distribution of modal responses tends to be more uniform (Lyon and DeJong, 1995).

3. EXPERIMENTAL SETUP AND PROCEDURE

The structure under study is a curved ribbed-stiffened rectangular panel with an area of 2,06 m^2. For the estimation of the loss factor, the panel was excited using shaker. A random excitation signal was employed to excite uniformly from 0 Hz to 6400 Hz.

During the measurements the panel and the shaker were suspended with nylon strings. The shaker was attached to the panel through a stinger-impedance head assembly.

Forty response points were uniformly distributed over the structure. These points were chosen knowing that the points on the reinforcement beams display minor vibration levels (greater rigidity) and the midpoints on skin and the free edge of the panel display high vibrations levels.

Eight driving points were chosen. Five of these points (E1 to E5) were localized in flexible regions (on the skin) and the other three driving points (A1 to A3) in stiff regions (on the beams).

Low mass accelerometers were used to reduce the effects of mass loading to structure.

The setup of this experiment is shown in Fig. 1.

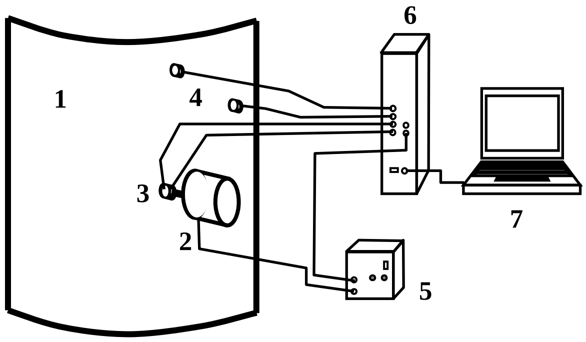

Figure 1. The measurement setup: 1) Panel, 2) Shaker, 3) Impedance head, 4) Accelerometers, 5) Power amplifier, 6) Analyzer, 7) Computer.

All the measurements were corrected in order to cancel the effects of the existing mass between the piezoelectric element of the impedance head and the structure (Fig. 2). This mass affects seriously the FRF, especially at high frequencies. The corrected mobility functions can be calculated by (Baldanzini and Pierini, 2002):

$$Y_{pc} = \frac{Y_p}{1 - i.\omega.m_c.Y_p} \quad (8)$$

and

$$Y_{tc} = \frac{Y_t}{1 - i.\omega.m_c.Y_p} \quad (9)$$

where Y_{tc} is the corrected transfer mobility and Y_{pc} is the corrected driving point mobility and m_c is the correction mass.

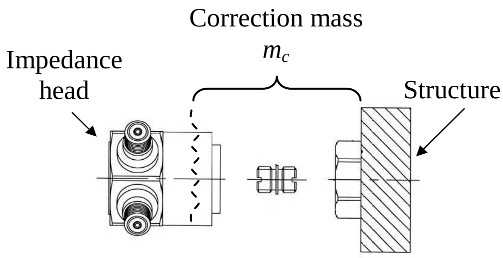

Figure 2. Correction mass for attached between impedance head and the structure (PCB).

The tests were carried out for undamped and damped conditions. For the damped condition, a constrained layer viscoelastic material (viscoelastic material + metal cover sheet) was used. The covered area was about 51% of the total area. Both panel conditions are shown in Fig. 3.

The rigid points are those over the surfaces of the reinforcing beams. The flexible points are localized in the skin. The parts of the panel structure are shown in Fig. 3.

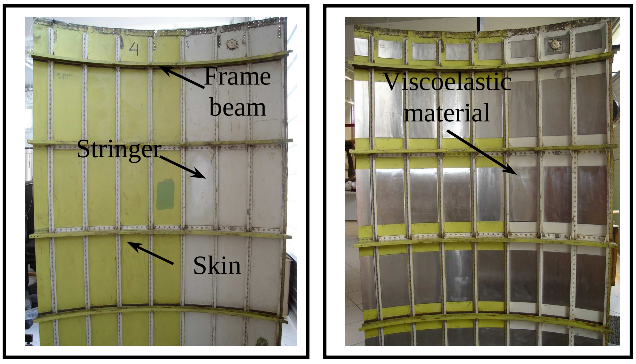

Figure 3. Fuselage panel without and with viscoelastic treatment.

During the measurements, the real part of the driving point mobility has to be verified. It should be positive over the frequency range of interest. To obtain accurate loss factors estimations, it is essential to have highly accurate measurement of the driving point FRF, otherwise large errors can be introduced (Bloss and Rao, 2005)

Therefore it is necessary to pay much attention to each test result when the accelerometers are moving between different measurement locations.

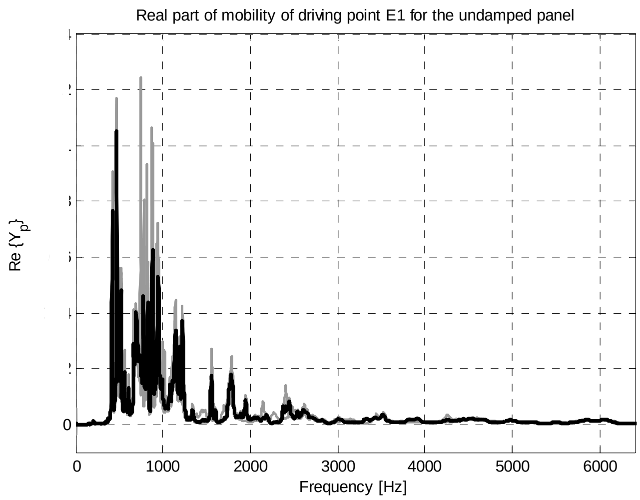

Figure 4. Example of the monitoring of the mobility real part; measurement made on the undamped panel.

In order to measure the sound radiation of a plate, a final experiment was carried out, with and without damping materials. The applied viscoelastic treatment is the same that was used in previous tests (see Fig. 3).

The fuselage panels were suspended by wires inside a semi-anechoic chamber (with 110 m^3 free volume) and were driven by a shaker at point A1 with a broad band random signal, with frequency range from 0 to 6.4 kHz. A microphone was positioned in front the panel, at a distance of 2.4 m (Fig. 5).

Figure 5. Driven panel inside the semi-anechoic chamber.

4. PRESENTATION AND DISCUSSION OF RESULTS

The tests were carried out for undamped and damped panels and the calculations performed using a computer routine. The input power is evaluated using the real part of the mobility, shown in the Fig. 6 and Fig. 7 for both conditions. High levels of energy are obtained when the driving points are localized in flexible regions (skin). For the damped condition (Fig. 7), the mobilities of the flexible points present high attenuation. It could be related to the fact that these points are located in areas covered by viscoelastic material. As a consequence, part of the input vibratory energy is dissipated before it can be transmitted to the rest of the structure (Fig. 8).

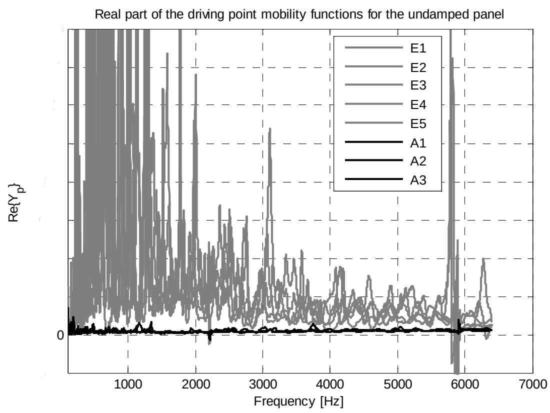

Figure 6. Real part of the driving point mobilities for the undamped panel.

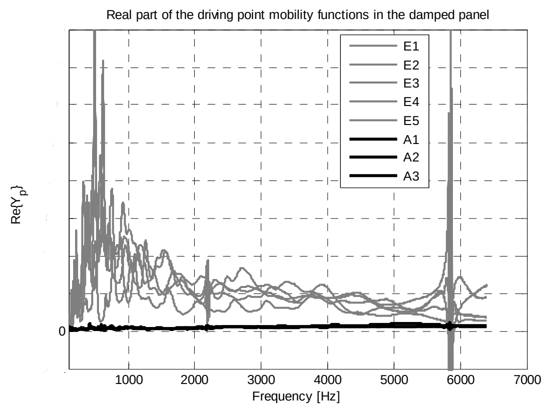

Figure 7. Real part of the driving point mobilities for the damped panel.

The kinetic energy levels can be analyzed through the squared modulus of the transfer mobility. Observing the dispersion of the transfer mobilities, the vibratory energy is distributed more uniformly when the panel is excited in the stiff points (Fig. 9 and Fig. 10).

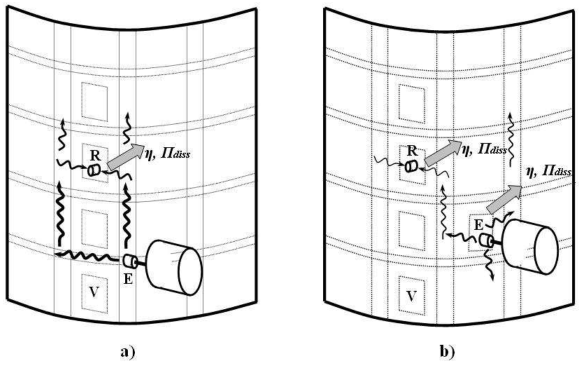

Figure 8. The propagation of the vibration: a) when the panel is excited in a stiff point and b) when the panel is excited in a flexible point. (R response point, E driving point, V viscoelastic material).

High levels of kinetic energy appear at response points located near the driving point (R35 e R40 in the Fig. 9). This effect is more evident for the damped panel (Figs. 10, 11 and 12). The difference between the energy levels can be higher that 20 dB at some frequencies. These differences can be caused by the proximity between the affected response point and the driving point. The driving point generates an outspoken direct field nearby the point of excitation. Because the subsystem vibration energy, which is comprised in the basic SEA equations, constitutes exclusively reverberant energy, one should not include the kinetic energy of the response points affected by direct field. It is essential to have knowledge of the extent of the direct field. De Langhe (1996) presents some equations to determine the extent of the direct field for beams and plates.

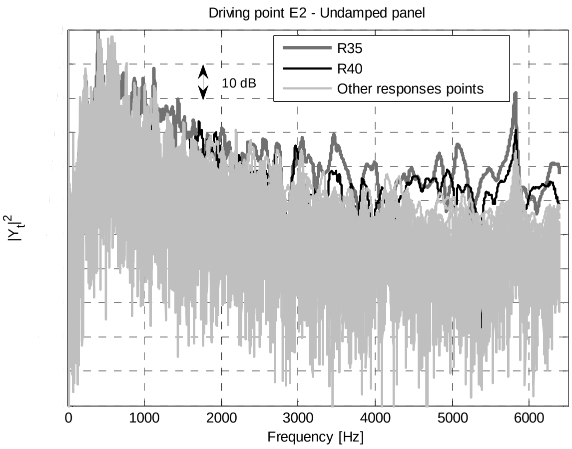

Figure 9. Squared modulus of the transfer mobility for the forty response points measured when the undamped panel is excited at driving point E2.

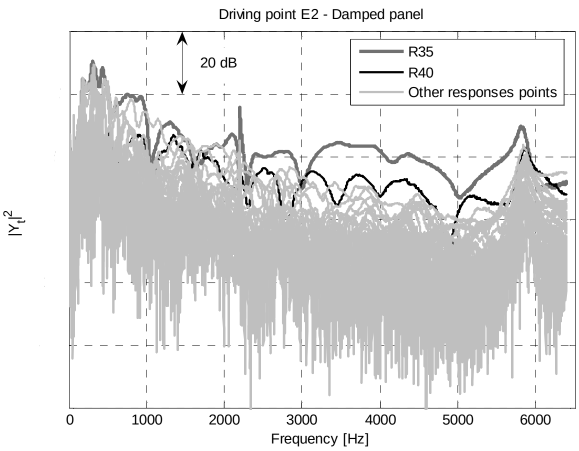

Figure 10. Squared modulus of the transfer mobility for the forty response points measured when the damped panel is excited at driving point E2.

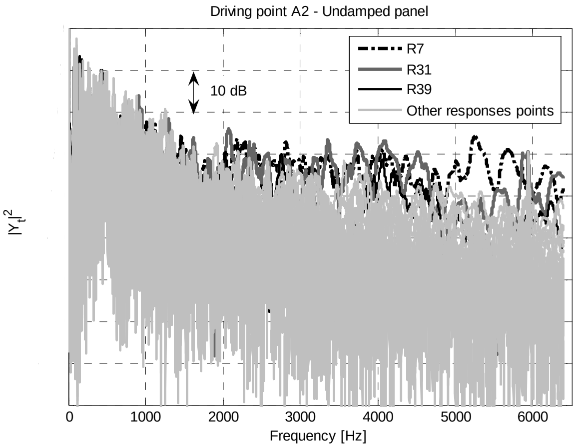

Figure 11. Squared modulus of the transfer mobility for the forty response points measured when the undamped panel is excited at driving point A2.

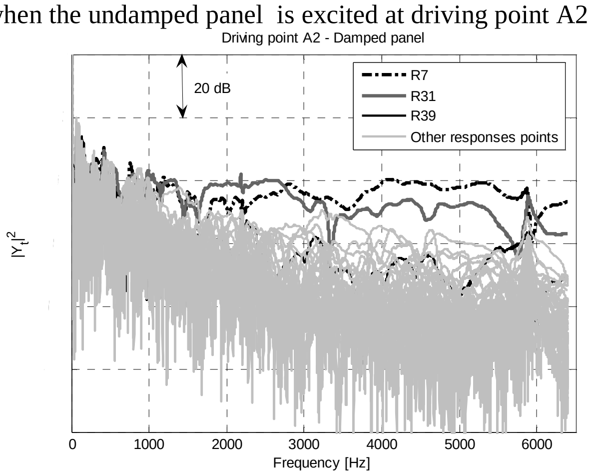

Figure 12. Squared modulus of the transfer mobility for the forty response points measured when the damped panel are excited in the driving point A2. R7, R31 and R39 are the mobilities of the response points near the driving point.

The mean values of the damping loss factor are highly affected by the response points located near the driving point. These effects are shown in Figs. 13 and 14. The damped panel is more sensible to this effect.

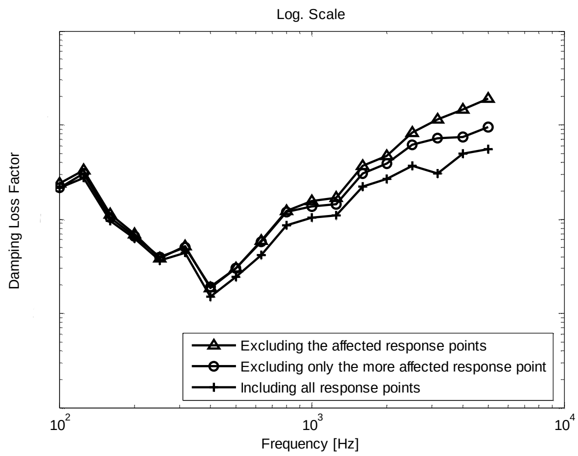

Figure 13. Damping loss factor (DLF) in one-third octave centre band frequency. The effects of the direct vibratory field. Eight driving points (E1 to E5 and A1 to A3) were used to calculate this value. Undamped panel.

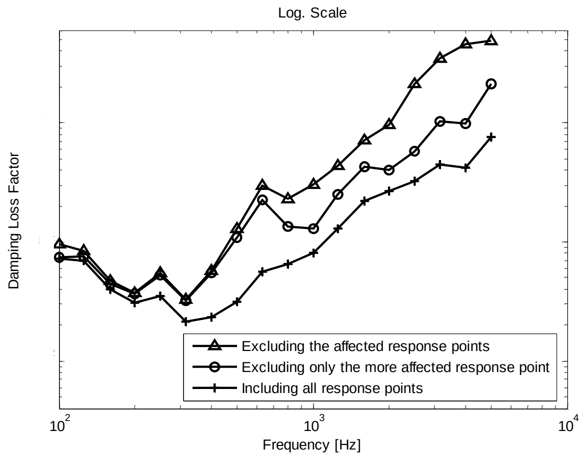

Figure 14. Damping loss factor (DLF) in one-third octave centre band frequency. The effects of the direct vibratory field. Eight driving points (E1 to E5 and A1 to A3) were used for calculate this value. Damped panel.

Another analyzed effect was the selection of the driving point used to obtain the loss factor. For undamped condition, the choice of any type of driving point (flexible or stiff) does not produce great differences (Fig. 15), but when the structure is damped, the loss factor values can be very different (Fig. 16).

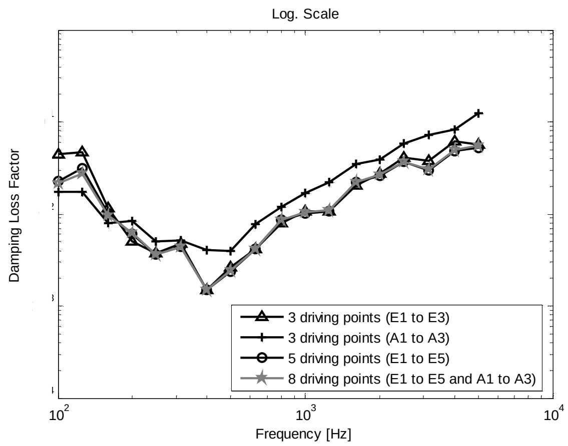

Figure 15. Damping loss factor (DLF) in one-third octave centre band frequency. Effects of the amount and the type of the driving points are presented. The results were obtained considering only the more influenced response points.

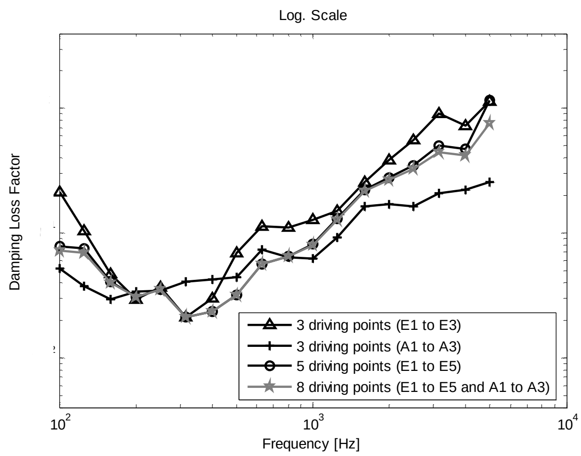

Figure 16. Damping loss factor (DLF) in one-third octave centre band frequency. Effects of the amount and the type of the driving points are presented. The results were obtained considering only the more influenced response points.

The number of responses points used for estimation of the loss factor is analyzed in Figs. 17 and 18. It is clear that 30 or 20 response points are sufficient to obtain good accuracy.

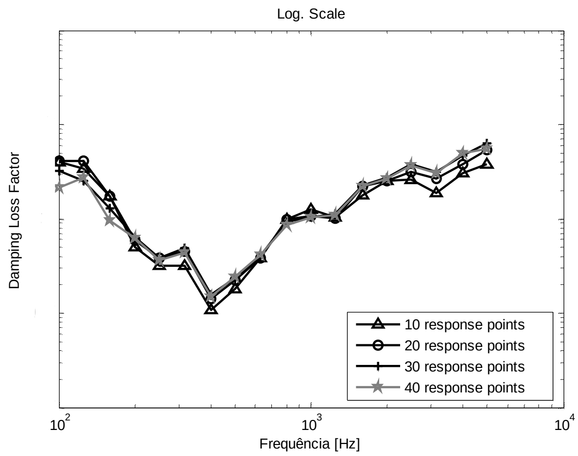

Figure 17. Damping loss factor (DLF) in one-third octave centre band frequency. Effects of the amount of response points are presented. Eight driving points (E1 to E5 and A1 to A3) were used to calculated these values. It was ignored the effect of the direct vibratory field.

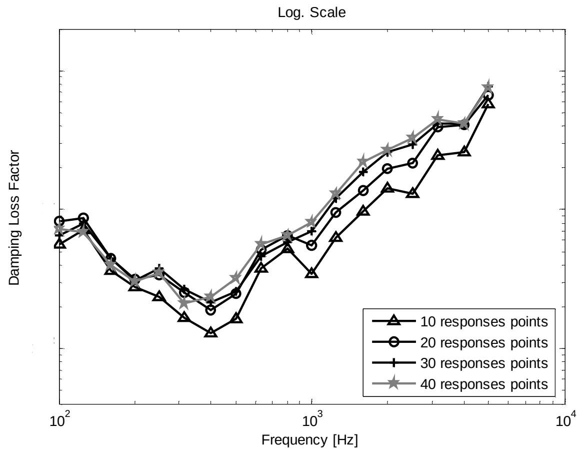

Figure 18. Damping loss factor (DLF) in one-third octave centre band frequency. Effects of the amount of response points are presented. Eight driving points (E1 to E5 and A1 to A3) were used to calculated these values. It was ignored the effect of the direct vibratory field.

Figure 19 shows the sound levels measured inside the semi-anechoic chamber, the background noise and the noise radiated from the damped and undamped panels. Only at high frequencies the background noise approaches the radiated noise.

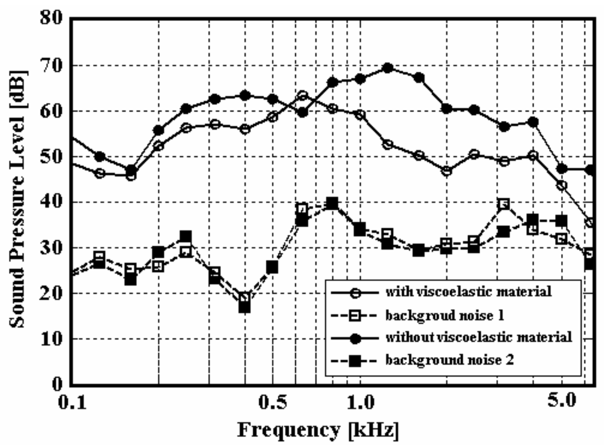

Figure 19: Sound levels produced by radiation from panels.

5. CONCLUSIONS

The fuselage panel was analyzed for undamped and damped conditions. As it is show in Fig. 4, errors are introduced in the estimation of the loss factor when limited number of accelerometers is used at the setup. Variations of the real part of mobility occur when the accelerometers are moving between different measurement locations during the test (theoretically the inject power in the structure must remain equal in order to evaluate the way as each part of structure responds for a same excitation).

High values of the loss factor could be observed in some frequencies in the damped panel when the driving points were located in flexible regions.

Driving points located in stiff regions display better results in the estimation of the loss factor. Three driving points and twenty response points could be sufficient for obtain a good estimation of the loss factor of this structure.

The reduction of the radiated noise when the panels vibrate under shaker excitation is more pronounced above 900 Hz, as can be seen in Fig. 19. The larger noise reductions due to the application of viscoelastic material are perceived between 1 and 2 kHz. At higher frequencies, the effect of the added damping is also reduced.

This paper can be used as a reference for future works which will study the effects of the temperature in the structures damped with viscoelastic materials.

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