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Measurement and SEA modelling of sound transmission of ribbed-stiffened panels

Carlos Henrique Gomes[1], Samir Nagi Yousri Gerges[1], and Roberto Jordan[1]

[1] Laboratory of Acoustics and Vibration, Mechanical Engineering Department, Federal University of Santa Catarina, 88040-900, Florianópolis-SC, Brazil, E-mail: *chgomes@yahoo.com* or *samir@emc.ufsc.br*

Abstract: The acoustic fields generated around an aircraft in flight act in the mid and high frequency regions, where the high modal density of the structure hinders dynamics analysis through deterministic methods. In this context, the search for effective methodologies to model vibro-acoustic characteristics at mid and high frequencies has been the motivation for several scientific studies. As an alternative approach, Statistical Energy Analysis (SEA) allows the study of energy diffusion in vibro-acoustic systems in mid and high frequency regions. SEA provides a procedure for calculating the flow and storage of dynamical energy in a complex system. This present study aims to describe the vibro-acoustic characterization of a structure similar to an aircraft fuselage. Several SEA models were considered to compare the analytical formulations found in the literature with measurement data. The measurements were carried out in two adjacent reverberation rooms which permitted an evaluation of the sound transmission loss and the vibration level of the panels: simple and ribbed-stiffened. The results show that SEA modelling provides a good prediction of the vibro-acoustic performance of simple and ribbed-stiffened panels. The importance of an accurate evaluation of resonant and non-resonant SEA parameters is thus highlighted. In this regard, the revised model for computing the coupling loss factors was evaluated and the results gave a much better agreement with measured data than the results from the SEA commercial software. The main contributions of this study are the detailed analysis of the hypotheses adopted during the definition of SEA subsystems and an accurate prediction of the vibro-acoustic performance through SEA models in simple and ribbed-stiffened panels in the mid and high frequencies regions.

Keywords: Ribbed-stiffened panels, Statistical Energy Analysis – SEA, Measurements, Sound Transmission Loss, and Aircraft fuselage.

INTRODUCTION

In a world where high noise levels are present, acoustic comfort is an important consideration in the design and operation of airplanes. The noise levels generated by the propulsion system and the turbulent boundary layer can be intense enough to result in an unacceptable interior noise environment causing passenger discomfort, interference with communications, and malfunction of electronic equipment. In order to solve problems in structures such as aircrafts it is necessary to understand the vibro-acoustic properties and characteristics of fuselage panels.

Interior noise prediction is concerned with the sound and vibration wave fields that can exist in the structure and interior spaces (cavities) of the aircraft. At low frequencies, the vibro-acoustic response is dominated by long-wavelength standing wave fields. Deterministic approaches such as finite element analysis attempt to describe each mode explicitly and resolve the associated variations in response over space and time. However, aircraft noise problems typically involve a very large number of modes over a broad frequency range. At high frequencies, where the short wavelengths wave fields are present, these modes become both expensive to compute and highly sensitive to uncertain physical details of the system.

In this context, the search for effective methodologies to model vibro-acoustic characteristics at mid and high frequencies has been the motivation for several scientific studies. As an alternative approach, Statistical Energy Analysis (SEA) addresses these challenges by taking a statistical approach and allows the study of energy diffusion in vibro-acoustic systems in mid and high frequency regions, Lyon (1995) and Craik (1996). The system is represented by an ensemble: a population of similar systems with statistically described variations in properties.

The basic SEA concept is to consider coupled structures and acoustic spaces as systems of resonant modes whose modal resonance frequencies lie within a narrow frequency band. The SEA model is composed of subsystems which are groups of similar modes.

The SEA parameters for response predictions are modal density (number of resonant modes of an element in a frequency band), damping loss factors (DLFs, rate of dissipative losses of energy of a subsystem), and coupling loss factors (CLFs, rate of power flow from one subsystem to another). Loads and other inputs are quantified according to the amount of time averaged, band-limited power they inject into a subsystem. Thus, the average statistical characteristics, such as mean-squared structural velocity or acoustic pressure, can be determined through a linear system of algebraic power balance equations for the subsystems.

The energy stored in a structural element or acoustic enclosure is generally dominated by the resonant modes. The group of modes that resonate outside the frequency band under consideration, which are therefore called non-resonant, may play a role, in transmitting energy from one element to another. This non-resonant transmission path is significant in the problems related to sound-structure interaction, such as the transmission of sound through metal panels, Gomes (2005), Gomes (2006a), Leppington et al (1987), and Szechenyi (1971).

Since the SEA modeling and evaluation of parameters are based on assumptions and approximations, the importance of an accurate assessment of the resonant and non-resonant SEA parameters is highlighted in this study, Gomes (2006b).

A detailed analysis of the hypotheses adopted during the definition of SEA subsystems has been previously carried out and an accurate prediction of the vibro-acoustic performance through SEA models was achieved for simple and ribbed-stiffened panels in the mid and high frequencies regions, Gomes (2006c).

SOUND TRANSMISSION LOSS USING SEA MODELS

The relationship between the Transmission Loss (*TL*) and the Statistical Energy Analysis (SEA) is based on the formulation used in the experimental determination of the transmission loss with the aid of two reverberant chambers. The SEA model is composed by the subsystems: emission chamber (1), panel (2), and reception chamber (3), see Fig. 1.

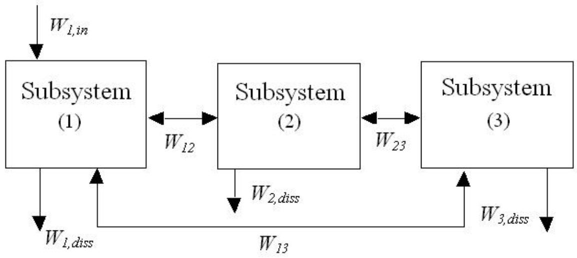

Figure 1 – SEA Model representing the set-up adopted for the experimental evaluation of the Transmission Loss.

The expression for the evaluation of the Transmission Loss from the SEA energy levels proposed by Price (1970) is given by:

$$TL_{SEA} = 10\log\left(\frac{E_1}{E_3}\right) - 10\log\left(\frac{V_1}{V_3}\right) + 10\log\left[\frac{S_2}{0{,}161V_3}\left(\frac{2{,}2}{f\eta_3}\right)\right] \tag{1}$$

where S_2 is the panel surface area, V_i is the volume of the subsystem *i*, η_i is the damping loss factor of the subsystem *i*, E_i is the total energy of the subsystem *i* and f is the central frequency of the band (usually 1/3 - octave).

ANALYTICAL SEA PARAMETERS

In this section, the main concepts behind the analytical SEA parameters will be shown and discussed in relation to sound transmission loss applications.

Simple panel

The coupling loss factor (CLF) of the resonant transmission path, panel–reverberant chambers, are proportional to the radiation efficiency of the panel, Craik (1996), and is given by:

$$\eta_{21} = \frac{\rho_0 c_0 S_2}{\omega M_2}\sigma_{rad} \tag{2}$$

where: ρ_0 is the air density, c_0 is the speed of sound, ω is the angular frequency ($2\pi f$), M_2 is the mass of panel e σ_{rad} is the average radiation efficiency in frequency.

A first evaluation of the average radiation efficiency in frequency bands was proposed by Maidanik (1962), who studied the interaction between the panel vibrational field and the acoustic resonant field. Simplified expressions were proposed for the average radiation efficiency of the panel for each frequency range. In the high frequency range, frequencies above to the critical frequency, the influence of the finite panel dimensions on the resonant radiation phenomenon can be neglected. Thus, the radiation efficiency is identical to that of the infinite panel efficiency. However, in the low and mid frequency ranges, the amplitude of the radiation efficiency was proportional to the perimeter of the panel.

An accurate evaluation of the average radiation efficiency was proposed by Leppington et al (1982) for the mid and high frequency ranges. In their study, the contribution of the resonant modes to each area of the two-dimensional space of the bending wavenumber was revalued and asymptotic formulations of radiation efficiency were proposed for each resonant mode of the panel. Leppington showed that the singularity points of the Rayleigh integral could be classified into two different groups. The first group, *primary stationary points*, has a greater contribution to the radiation efficiency; its order of magnitude is around $(k)^{-1}$, where k is the acoustic wavenumber. The second group of singularities is denominated *secondary stationary points* and its contribution has an order of magnitude around $(k)^{-3/2}$. In order to evaluate the average radiation efficiency in the frequency range, the angular average was carried out in the first quadrant of the dimensionless space of the bending wavenumber.

With the aim of making a comparison between the literature and the commercial software formulations for the CLF, the AutoSEA2 formulation for CLF was investigated. The AutoSEA2 CLF is based on Leppington et al (1982). However, the software formulation neglects the contribution of the secondary stationary points in the evaluation of the radiation efficiency of each resonant mode of the panel. According to Vibro-Acoustic Sciences (1994), the presence of such a contribution does not result a significant change in the value during the integration process of the average radiation efficiency in the frequency range.

The evaluation of the non-resonant CLF is usually based on the transmission coefficient (τ). The several formulations found in the literature for the non-resonant CLF are discussed below.

One of the first proposed formulations, Price (1970), is based on the “Mass Law” transmission coefficient which neglects the finite dimensions and the elastic characteristics of the panel. In a general way, the “Mass Law” theory offers a good estimate of the non-resonant transmission, because the panel behaves only as a linking element, that is, the panel has no resonant characteristics.

However, to challenge this postulation, several studies have been carried out: Leppington (1987), Gurovich (1978), and Szechenyi (1971). In these investigations the simplified assumptions associated with the “Mass Law” are questioned and new formulations are presented for the quantification of the non-resonant transmission path contribution.

Leppington revalued the transmission coefficient, presenting two main advances regarding the “Mass Law” theory. The first advance was the hypothesis of the thick plate theory, instead of the thin plate theory used by “Mass Law” formulation. The second advance is related to the contributions of the grazing incident angles.

A new proposal for the evaluation of the transmission coefficient was presented by Gurovich (1997) who evaluated the non-resonant CLF based on an exact formulation in terms of the orthogonal natural modes of the bending vibrations for the simple panel and also the Huygens integral for the sound pressure of radiated waves. Gurovich’s main contribution was to take into account the effect of finite dimensions of the panel in the calculation of CLF. Therefore, an asymptotic formulation of the diffuse sound transmission coefficient, for low frequencies, was obtained by integrating the incident power and the transmitted power through the panel.

The AutoSEA2 formulation is similar to definition of the non-resonant CLF proposed by Lyon (1995). In this formulation, the sound transmission coefficient is based on the impedance of subsystems, Szechenyi (1971).

Ribbed-stiffened panel

In general, the vibrational behavior of a ribbed-stiffened panel is strongly influenced by the presence of the beams, which increases inertia effects and stiffness of the structure. In this regard, Langley (1999) classified the vibrational modes into two different groups, according to the dynamic behavior of a structure attached by beams. The first group is denominated the global modes, which have large wavelengths and strongly related phases. The second group is denominated the local modes, which have small wavelengths and weakly correlated phases.

In order to evaluate the natural frequencies of a ribbed-stiffened panel, the model of Mikulas (1965) was used in this study. In this model, the influence of beam placement on the amplitude of natural frequencies was analyzed for several configurations of ribbed cylinders. The equivalent orthotropic plate approach was adopted. The beams are assumed to be identical and are equally spaced for each one of the directions. The in-plane forces are included, although the inertia effects are neglected. Under the assumption that the displacements of the plate and beams are the same as those of the fixation points, the deformation energy of the system is evaluated. Although this formulation is related to ribbed cylinders, this analytical proposal can be extended to the evaluation of the natural frequencies of ribbed planes, with the infinite radius assumption.

For the evaluation of the resonant CLF of a ribbed panel, two approaches to the evaluation of the average radiation efficiency were compared. In the first approach, Maidanik (1962), the frequency average of radiation efficiency is proportional to the perimeter of the panel in the frequency range below the critical frequency. For a ribbed panel, the original perimeter is substituted by an “*equivalent*” perimeter which is composed of the original perimeter of the simple panel plus twice the total length of the reinforcement beams.

The second approach relates the radiation efficiency to the parameter joint acceptance. The physical concept of the joint acceptance is to quantify the strength of coupling between two waves or wave fields where they are joined along a line or over an area, taking into account the relative amounts of phase reinforcement and cancellation, that is, the constructive or destructive overlap, over the space of the junction. In other words, the joint acceptance describes how well the vibration modes of the panel are harmonized with the characteristics of the external pressure field.

The relationship between the average radiation efficiency (σ_{rad}) and the joint acceptance (j_n) was investigated initially by Maidanik (1962) and later formalized by White (1966) and is given by:

$$\sigma_{rad} = \frac{2kS_2}{\pi}\left\langle j_n{}^2 \right\rangle \tag{3}$$

where k is the acoustic wavenumber and $\left\langle j_n{}^2 \right\rangle$ is average joint acceptance in the frequency band.

The joint acceptance definition, relating to the generalized force corresponding to a homogeneous and perfectly random pressure field, was detailed by Mead (1968). For a diffuse pressure field acting on a panel with (finite) dimensions a and b, the parameter joint acceptance in rectangular coordinates, is given by:

$$j_n{}^2 = \frac{1}{a^2b^2}\int_0^b\int_0^a \phi_n(x_2, y_2)\int_0^b\int_0^a \rho(\overline{x_1}, \overline{x_2}, \omega)\phi_n(x_1, y_1)\, dx_1 dy_1 dx_2 dy_2 \tag{4}$$

where ω is the angular frequency, $\overline{x_i}$ or (x_i, y_i) is the rectangular coordinate of a point *i* and ϕ_n is the mode shape of the *n*-th vibrational mode shape of the panel.

The term $\rho(\overline{x_1}, \overline{x_2}, \omega)$ has a form similar to the space-correlation coefficient, which is a function of the frequency , Mead (1968). For the diffuse pressure field, Clarkson (1985) showed that this is given by:

$$\rho(\overline{x_1}, \overline{x_2}, \omega) = \frac{\sin[k(\overline{x_1} - \overline{x_2})]}{k(\overline{x_1} - \overline{x_2})} \tag{5}$$

In the AutoSEA2 software, the second approach is adopted by the "*Ribbed Panel*" tool, which evaluates the average radiation efficiency of a ribbed panel. However, the calculation of the space-correlation coefficient using Eq. (5) requires a large computational effort. Thus, a separable form approach is used by AutoSEA2 software, Vibro-Acoustic Sciences (1994).

In non-resonant transmission, the procedure adopted for the evaluation of the CLF is similar to that used for the simple panel, but the equivalent mechanical properties are used, Vibro-Acoustic Sciences (2001).

EXPERIMENTAL SEA PARAMETERS AND TRANSMISSION LOSS MEASUREMENTS

Experimental evaluations of SEA parameters have become an excellent tool in the building of hybrid models. In this regard, since an analytical evaluation of the damping loss factors is not possible, some experimental procedures were carried out in this study: Fahy (1997), Clarkson (1981), Clarkson (1983), and ISO 3745 (1977).

In order to make a good experimental evaluation of the transmission loss of the metal panels, the experimental procedures employed in this study were based on ISO 140 – 1 (1990) and ISO 140 – 3 (1995). The Laboratory of Vibrations and Acoustics (LVA) has two adjacent reverberant chambers, whose dimensions are: emission chamber (7.49 x 7.49 x 2.63 m), and reception chamber (7.90 x 5.60 x 4.50 m), and they have a test opening of approximately 10 m^2 (2.10 x 5.0 m).

In this study, two test samples were investigated. The first sample is a simple aluminum panel whose dimensions are: 1.80 m x 1.13 m with 2 mm of thickness. The second sample is a ribbed-stiffened panel, that is, a simple panel reinforced with beams in both directions. The cross-section dimensions of the beams, as well as the spaces between them, were based on the typical structure of an aeronautical fuselage. The longitudinal beams have an L-shaped section (25 x 14 mm) with a spacing of 0.20 m and the transverse beams have a U-shaped section (14 x 48 x 14 mm) with a spacing of 0.40 m. Both groups of beams, 5 longitudinal and 4 transverse beams, have a uniform thickness of 2 mm.

NUMERIC RESULTS

This section will show the numeric results for both metal panels: simple and ribbed-stiffened. In the first case, the simple panel was represented by a single SEA subsystem. The SEA models of the ribbed-stiffened panel were built using two modeling approaches. In the first approach, the ribbed panel is represented by an equivalent single subsystem; this SEA model is referred as to the equivalent SEA model. In the second approach, the model is referred to as the explicit SEA model, which is composed of several SEA subsystems corresponding to each structural component.

Simple panel: single SEA subsystem

Usually in the SEA, the model for transmission loss is composed of three subsystems: source chamber (1), panel (2), and reception chamber (3). Fig. 2 shows this SEA model whose subsystems are uncoupled to provide a better visualization of the SEA subsystems.

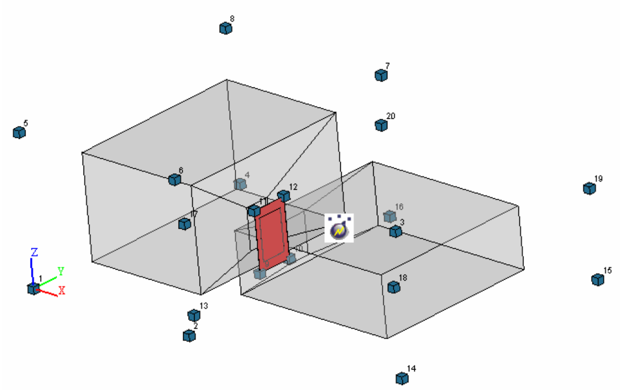

Figure 2 – AutoSEA2 model for a simple panel.

Since the damping loss factors (DLFs) and the input power are related to experimental procedures, a detailed analysis of the coupling loss factors (CLFs) was made. Firstly, the resonant CLFs which are associated with the connections were assessed: panel - source chamber and panel - reception chamber. These CLFs were considered to be identical. For others resonant CLFs, a reciprocity relation between adjacent subsystems was considered, Lyon (1995). Previous analyses of the several formulations of the resonant CLFs have suggested that the formation amplitudes are practically equivalent, except for small discrepancies observed in the proximity of the critical frequency, $f_c =$ 6019 Hz.

The frequency range analyzed in this study was 100 Hz to 10 kHz, in one-third octave bands. An analysis of the validity of the SEA model showed that the region for reliable results lies at bands above 250 Hz. In this region, the basic results for the energy flow between two vibration resonators can be extended to the two groups of coupled modes and also to several groups of coupled modes, Gerges (2000).

In order to evaluate the contribution of resonant and non-resonant transmission paths, distinct SEA models were built considering the presence of each one of the transmission paths in isolated and joint form. Although they are omitted here, the results suggest that each one of the transmission paths is associated with a different predominant range in the frequency domain. Thus, the predominance of the non-resonant transmission path occurs below the critical frequency. On the other hand, the predominant resonant transmission path became more relevant above the critical frequency.

Similarly to the case of the resonant transmission path, a quantitative analysis of the non-resonant transmission path was carried out. For this, the resonant CLFs were considered constant and several non-resonant CLF formulations were then compared with each other, Fig. 3.

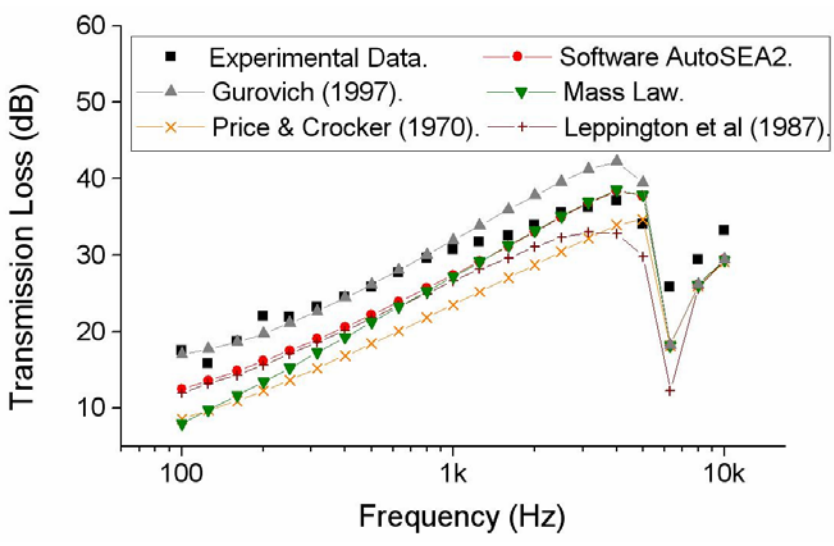

Figure 3 – Transmission Loss: the non-resonant path of a simple panel.

The results suggest that the non-resonant CLF proposed by Gurovich (1997) represents the dynamic behavior at low frequencies in a more satisfactory way. On the other hand, for the mid frequencies in the proximity of the critical frequency, the CLF based on the "Mass Law" transmission coefficient provides a good agreement with the experimental data, similar to case of the AutoSEA2 formulation.

According to Fig. 3, a new formulation was proposed for the non-resonant CLF of the proposed model, Gomes (2005). For the low frequencies, Gurovich's formulation was used. On the other hand, for the region which is near to the critical frequency, the formulation with the "Mass Law" transmission coefficient was considered.

Additionally, the influence proportioned by the boundary condition is to take into account during the evaluation of the resonant CLF, in order to improve the results for the frequency range in the proximity of the critical frequency, as it is shown in detail by Leppington et al (1984).

The proposed resonant CLF is given by:

$$\sigma_{rad} = \frac{Uc_0}{2\pi^2 f^{1/2} f_c^{1/2} S_2 \left(\gamma^2 - 1\right)^{1/2}} \left[\ln\left(\frac{\gamma+1}{\gamma-1}\right) + \frac{2\gamma}{\gamma^2-1} \right] \; if \;\; f < f_c , \tag{6}$$

$$\sigma_{rad} = \left(\frac{2\pi f}{c_0}\right)^{1/2} a\left(0{,}5 - 0{,}15\frac{a}{b}\right) \; if \;\; f = f_c , \tag{7}$$

$$\sigma_{rad} = \left(1 - \frac{f_c}{f}\right)^{-1/2} \; if \;\; f > f_c . \tag{8}$$

where: a, b are the dimensions of panel ($a \le b$), $U = 2(a+b)$ is the perimeter of panel, and $\gamma = \left(f_c / f\right)^{1/2}$ is the ratio of frequencies.

The proposed non-resonant CLF is given by:

$$\eta_{13} = \frac{\rho_0{}^2 c_0{}^3 S_2^3}{\omega M_2{}^2 V_1} \left\{ \begin{array}{l} \left\{ \dfrac{\pi^9 c_0{}^2}{2^{11} S_2} \left[1 - \left(\dfrac{10\omega}{2\pi f_c} \right) \right] + \omega^2 \right\}^{-1} , if \;\; f_{11} < f < \dfrac{f_c}{10} \\ \dfrac{\pi}{\omega^2} , if \;\; \omega \ge \dfrac{f_c}{10} \end{array} \right\} . \tag{9, 10}$$

The results from the proposed SEA model provided a much better agreement with the experimental data than the results from the SEA commercial software, see Figure 4.

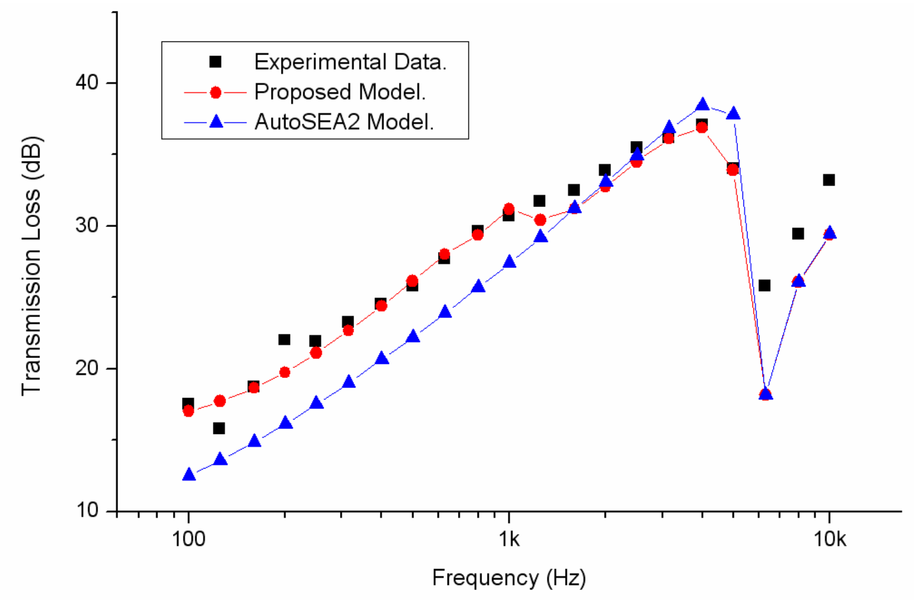

Figure 4 – Results of Transmission Loss from the proposed SEA model for a simple panel.

Firstly, the results suggest that the assessment of the Transmission Loss had a possible dependence on the finite dimensions of a panel in the lowest frequency range, Gomes (2006b). Secondly, in the coincidence frequency range, a weaker dependence on the finite panel dimensions was observed. Hence, the formulation based on the "Mass Law" transmission coefficient described the dynamic behavior in a more satisfactory way.

Another relevant result provided by SEA models is the analysis of energy diffusion in vibro-acoustic systems in mid and high frequency regions. In the Fig. 5, the input powers of the reception chamber subsystem are shown. According this, the non-resonant contribution is associated to energy flow from the emission chamber and its predominant frequency region is low frequency range. On the other hand, the resonant contribution is associated to radiation from simple panel mainly above to critical frequency.

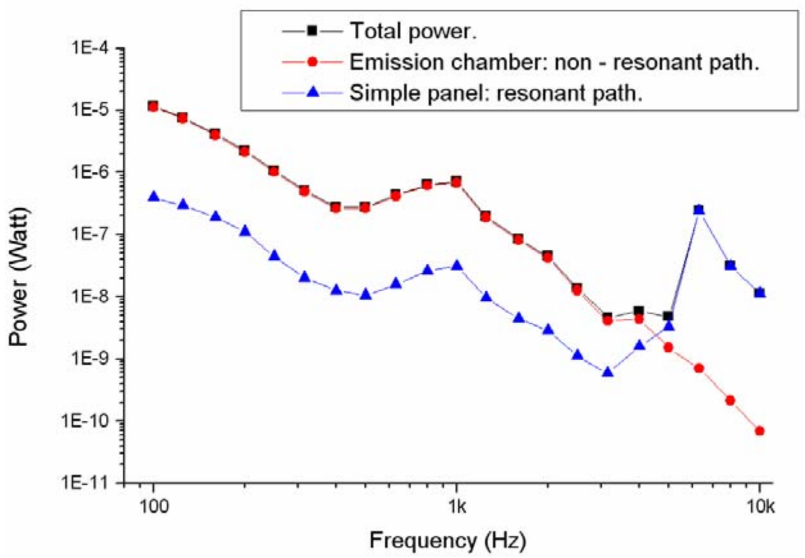

Figure 5 – The input powers of reception chamber

Finally, the accurate analysis of the transmission paths was an essential procedure in the attribution and evaluation of the CLFs in the SEA model. However, it should be noted that the discrepancies between the SEA results and the experimental data could be due to the neglecting of the damping increase due to the presence of rubber strips used in the fixation system of the metal panel, mainly in the coincidence frequency range.

Ribbed-stiffened panel: equivalent SEA model

For the ribbed-stiffened panel, different SEA models were also built in order to evaluate the performances of the resonant and non-resonant CLFs. Two SEA parameters are required for building the model: the average radiation efficiency, which is associated with the resonant CLFs; and the modal density which describes the subsystem capacity of the energy storage.

Although they are not shown in this paper, large discrepancies were found between the amplitudes of the different formulations for the resonant CLFs under study, Gomes (2005). On the other hand, the non-resonant CLFs of ribbed-stiffened showed small differences compared to the CLF of the simple panel.

An evaluation of the SEA model validity was carried out and showed that reliable results were found for bands above 500 Hz. The analysis of the transmission path suggested that the non-resonant transmission path was more significant for the low frequencies, while the resonant transmission path was predominant for the mid and high frequency ranges.

The SEA model which gave the best results is discussed here. The SEA parameters considered were: Maidanik's formulation for the resonant CLF and Gurovich's formulation for the non-resonant CLF. The modal density was evaluated using the mode counting method, Bremner (1994). The AutoSEA2 results were from the "*Ribbed Panel*" tool. Therefore, the AutoSEA2 results, as well as the results from the equivalent SEA model, were compared with the experimental data, see Fig. 6 and 7.

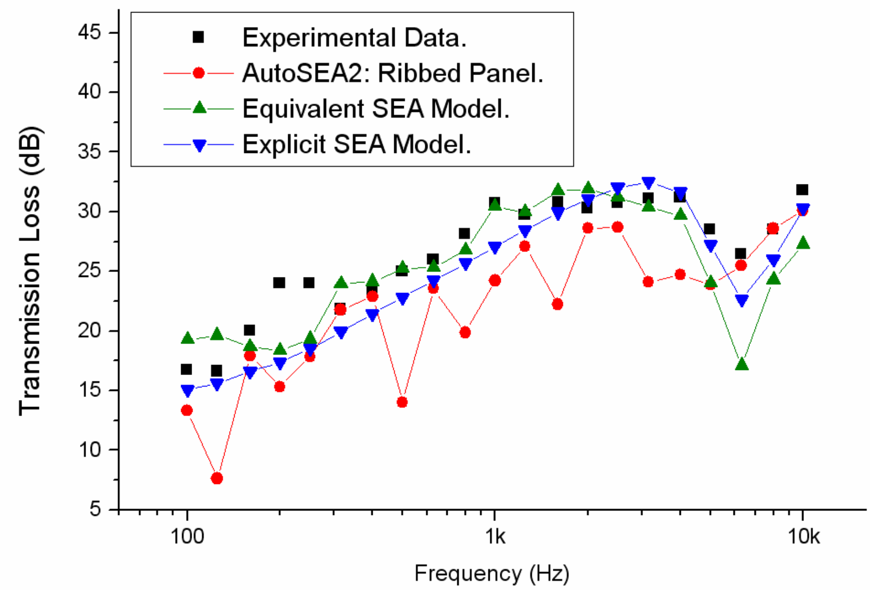

Figure 6 – The Transmission Loss: the experimental data and the SEA models results.

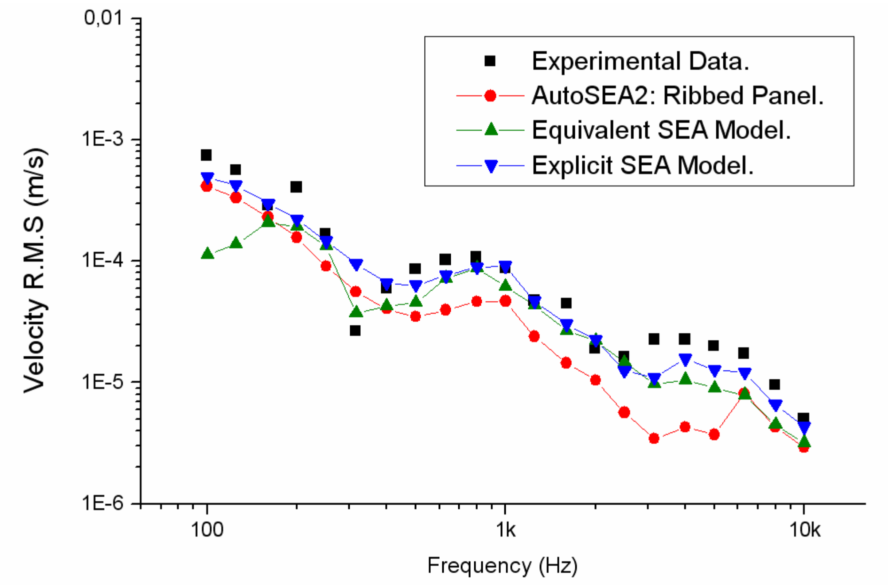

Figure 7 – The velocity R.M.S: the experimental data and the SEA models results.

The equivalent SEA model provided a much better agreement with the experimental data than the results from the SEA commercial software AutoSEA2, mainly for the mid frequency range. However, there were small discrepancies for the low frequencies due to the non-conformities of the reverberant chambers in relation to the sound field diffusivity.

Although excellent results were obtained using the Maidanik CLF, some observations should be made regarding the geometric and elastic characteristics of the beams, which were neglect, Fahy (1985) The results suggest that the effect of a reinforcement beam on the radiation efficiency of a panel is similar to the effect provided by a simple support condition, and also that the attachment of beams on the surface panel increases the radiation efficiency regardless of the beam cross-sections.

Ribbed-stiffened panel: explicit SEA model

The explicit SEA model was proposed in order to improve the description of the vibro-acoustic characteristics in the coincidence and high frequency ranges. The SEA parameters of the ribbed panel subsystems were determined by the AutoSEA2 software. The damping loss factors (DLF) were considered to be 1% for all the ribbed panel subsystems.

The most relevant aspect of an explicit model is the validity of SEA model because, in general, the modal densities of the subsystems show low magnitudes. Therefore, the explicit model results are reliable only in the high frequency range, above 4 kHz. The SEA predictions for Transmission Loss from the explicit model and from the equivalent model were compared with the Transmission Loss measurements, Figure 6, and the structural velocity of the ribbed panel, Figure 7.

The explicit model results show a good agreement mainly in the critical and high frequency ranges compared with the equivalent model results, for both parameters: the Transmission Loss and the structural velocity of the ribbed-stiffened panel. On the other hand, the time spent on the building of the explicit model and the computational processing time required for its resolution, are greatly superior to the equivalent approach.

Analysis of the effect provided by presence of the reinforcement beams

During this section, a complete analysis about SEA modelling of sound transmission loss of metal panels was carried out. The results from proposed SEA models for both metal panels, simple and ribbed, were satisfactory.

In general, the radiation efficiency of ribbed-stiffened panel is larger than simple panel one with the same dimensions. The effect provided by reinforcement beams is associated to increasing amount of non-canceled areas in comparison to case of simple panel's case. This behavior provided an increase on resonant CLF mainly in low and mid frequency range. In the Fig. 8, the experimental data for transmission loss of both panels were compared. Thus, a reduction of sound transmission loss in the mid frequency range is associated to the increase of resonant CLFs due to the effect provided by the presence of reinforcement beams.

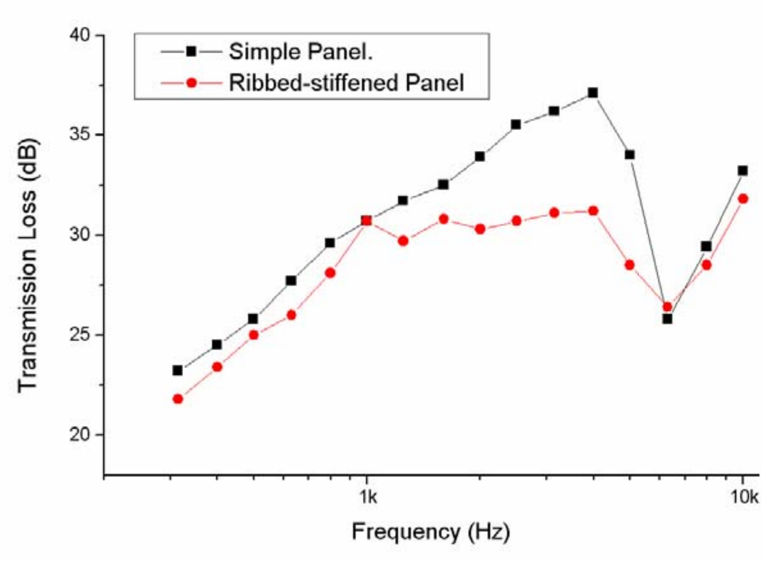

Figure 8: The Experimental data of Transmission Loss: simple and ribbed panels.

SUMMARY

This paper described the characterization of vibro-acoustic phenomena of a structure similar to an aircraft fuselage. Several SEA models were considered to compare the results from the analytical formulations found in literature with the experimental data.

The results showed that SEA models provide a good prediction of the vibro-acoustic performance of simple and ribbed-stiffened panels. Thus, the importance of an accurate evaluation of resonant and non-resonant SEA parameters is emphasized here. In this regard, a proposed model for computing the coupling loss factors was evaluated and the results gave a much better agreement with measured data than the results from the SEA commercial software.

The main contributions of this study are a detailed analysis of the hypotheses adopted during the definition of SEA subsystems and an accurate prediction of the vibro-acoustic phenomena through SEA models in simple and ribbed-stiffened panels in the mid and high frequencies regions.

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