ICSV13 - Vienna
The Thirteenth International Congress on Sound and Vibration
Vienna, Austria, July 2-6, 2006

MEASUREMENT AND SEA MODELLING OF SOUND TRANSMISSION OF RIBBED-STIFFENED PANELS

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Abstract

Nowadays, acoustic comfort is an important consideration in the design and operation of airplanes. 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, an alternative approach, Statistical Energy Analysis (SEA) allows the study of energy diffusion in vibro-acoustic systems in mid and high frequency regions. 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. Two classes of the panels were investigated: simple and ribbed-stiffened. 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.

INTRODUCTION

In a world where high noise levels are present, acoustic comfort is an important consideration in the design and operation of airplanes. The acoustic fields generated around an aircraft in flight act on the mid and high frequency regions, where the high modal density of the structure hinders dynamics analysis through deterministic methods (FEM and BEM).

In this context, an alternative approach, Statistical Energy Analysis (SEA) allows the study of energy diffusion in vibro-acoustic systems in mid and high frequency regions [1]. 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.

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. 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 single and ribbed-stiffened panels in the mid and high frequencies regions, Gomes, C. H. [2].

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 expression for the evaluation of the Transmission Loss from the SEA energy levels proposed by Price & Crocker [3] 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

Analytical SEA parameters: single panel

The coupling loss factor (CLF) of the resonant transmission, panel–reverberant chambers, are proportional to the radiation efficiency of the panel [4]. A first evaluation of the average radiation efficiency in frequency bands was proposed by Maidanik [5]. Simplified expressions were proposed for the average radiation efficiency of the panel for each frequency range. An accurate evaluation of the average radiation efficiency was proposed by Leppington *et al*. [6] 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 Raleigh integral could be classified into two different groups: primary and secondary stationary points. 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.

The evaluation of the non-resonant CLF is based on the transmission coefficient (τ). The several formulations found in the literature for the non-resonant CLF are discussed. One of the first proposed formulations, Price & Crocker [3], is based on the "Mass Law" transmission coefficient which neglects the finite dimensions and the elastic characteristics of the panel.

However, to challenge this postulation, several studies have been carried out, [7], [8], and [9]. 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 contribution. Leppington revalued the transmission coefficient, presenting two main advances. The first advance was the consideration of the thick plate theory, instead of the thin plate theory used by "Mass Law" theory. 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 [9] who evaluated the non-resonant CLF based on an exact formulation in terms of the orthogonal natural modes of the bending vibrations for the 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.

Analytical SEA parameters: 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 order to evaluate the natural frequencies of a ribbed-stiffened panel, Mikulas & McElman's model was used [10]. The equivalent orthotropic plate approach was adopted. The beams are assumed to be identical and are equally spaced for each one of the directions.

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 [5], 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 single panel plus twice the total length of the beams. The second approach relates the radiation efficiency to the parameter "joint acceptance". The relationship between the average radiation efficiency (σ_{rad}) and the joint acceptance (j_n) was investigated initially by Maidanik [5] and later formalized by White & Powell [11] and is given by:

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

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

In non-resonant transmission, the procedure adopted for the evaluation of the CFLs is similar to that used for the single panel, but the equivalent mechanical properties are used, AutoSEA2 Application note nº 28 [12].

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, [1], [13], [14], and [15].

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 [16]. 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).

The first sample is a single aluminium panel whose dimensions are: 1.85 m x 1.18 m with 2 mm of thickness. The second sample is a ribbed-stiffened panel, that is, a single panel reinforced with beams in both directions. 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 have a uniform thickness of 2 mm.

RESULTS

This section will show the numeric results for both metal panels: single and ribbed-stiffened. In the first case, the single 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.

Single panel

Usually in the SEA, the model for transmission loss is composed of three subsystems: source chamber (1), panel (2), and reception chamber (3); see Figure 1.

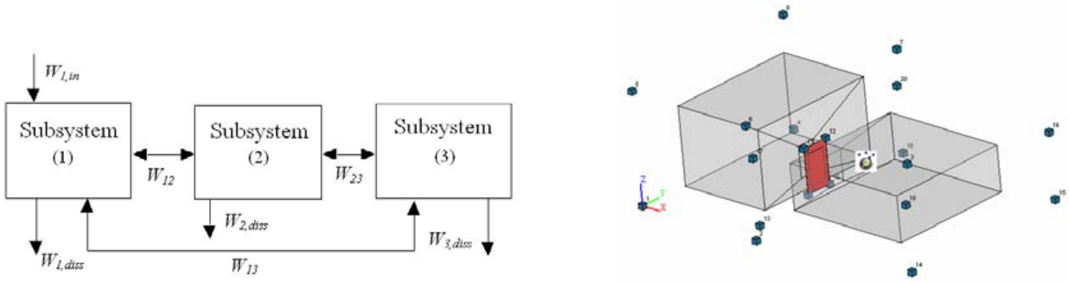

Figure 1 – SEA Model for a single panel and AutoSEA2 model.

Since the damping loss factors (DLF) and the input power relate to experimental procedures, a detailed analysis of the coupling loss factors (CLF) was made. Firstly, the resonant CLFs which are associated with the connections were assessed: panel - source chamber and panel - reception chamber. These CLF were considered to be identical. For others resonant CLFs, a reciprocity relation between adjacent subsystems was considered [1].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 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, f_c = 6019 Hz. 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, Figure 2. The results suggest that the non-resonant CLF proposed by Gurovivh represents the dynamic behavior at low frequencies in a more satisfactory way [9]. 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.

According to Figure 2, a “hybrid” formulation was proposed for the non-resonant CLF of the revised model. 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. For the intermediate range, an average between the latter two formulations was proposed [2].

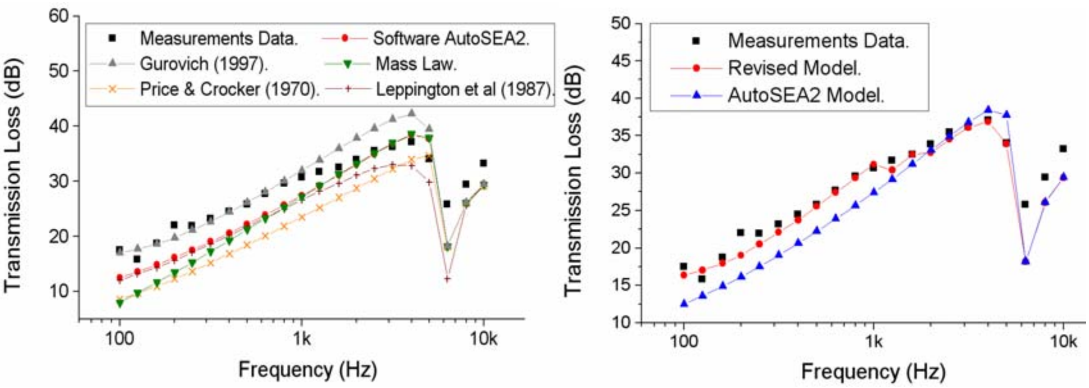

Figure 2: Part (a) – Transmission loss: the non-resonant path of a simple panel Part (b) – Results from the revised SEA model: the transmission loss of a single panel.

The results from the revised SEA provided a much better agreement with the measured data than the results from the SEA commercial software, see Figure 2. 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 [9]. Secondly, in the coincidence frequency range, a weaker dependence on the finite panel dimensions was observed. However, it should be noted that the discrepancies between the SEA results and the measurement 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 for the evaluation of the CLFs for both resonant and non-resonant transmission paths. 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 [2]. On the other hand, the non-resonant CLFs of ribbed-stiffened showed small differences compared to the CLF of the single 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 AutoSEA2 results were from the "*Ribbed Panel*" tool. Therefore, the AutoSEA2 results, as well as the results from the revised SEA model, were compared with the measurement data, see Figure 3.

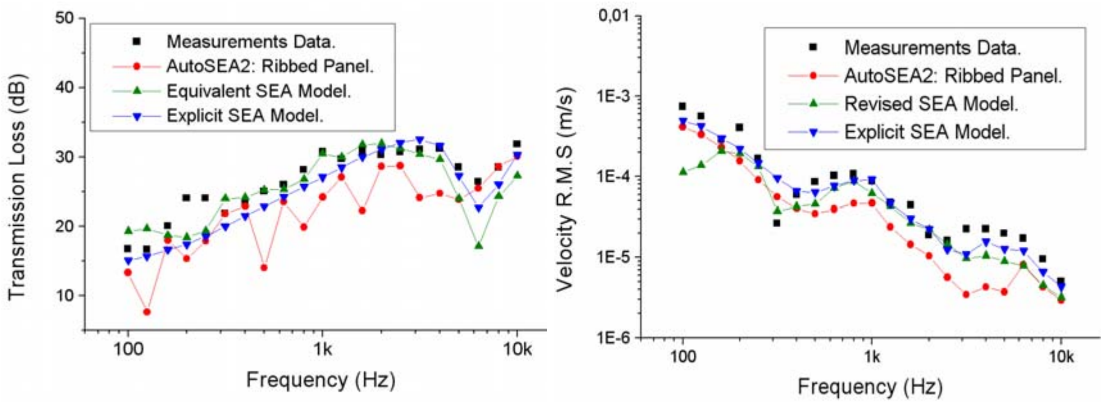

Figure 3 – The transmission loss and the velocity R.M.S: the measured data and the SEA models results.

The revised SEA model provided a much better agreement with the measured data than the results from the SEA commercial software, 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. 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 constructed 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 most relevant aspect of an explicit model is the model validity because, in general, the modal subsystem densities show low magnitudes. Therefore, the explicit model results are reliable only in the high frequency range, above 4 kHz. The SEA transmission loss from the explicit model and from the equivalent model were compared with the transmission loss measurements and the structural velocity of the panel, Figure 3.

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.

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 analytical formulations found in literature with experimental measurements.

The results showed that SEA models provide a good prediction of the vibro-acoustic performance of single 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 revised model for computing the coupling loss factors was evaluated and the results gave a better agreement with measured data.

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 single and ribbed-stiffened panels in the mid and high frequencies regions.

ACKNOWLEDGEMENTS

The authors wish to acknowledge the financial support of the Conselho Nacional de Desenvolvimento Científico e Tecnológico (CNPq) of Brazil and the Coordenação de Aperfeiçoamento de Pessoal de Nível Superior (CAPES) of Brazil.

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