ICSV14

Cairns • Australia

9-12 July, 2007

14th International Congress on Sound & Vibration

STIFFENERS EFFECT ON SOUND TRANSMISSION THROUGH A THIN PLATE EXCITED BY A TURBULENT BOUNDARY LAYER (CORCOS MODEL) – COMPARISON WITH A DIFFUSE SOUND FIELD

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Abstract

In the context of understanding and predicting airborne sound transmission through aircraft structures, the main objectives of this work are to study the stiffeners effect on a thin plate under a turbulent boundary layer excitation (Corcos model) and to compare this effect to the one on a plate excited by a diffuse field.

A numerical simulation using FEM software ACTRAN© has been carried out on a thin plate stiffened by frames and stringers, having its critical frequency equal to 7000 Hz.

Both excitation fields are modelled as distributed weakly stationary random process. For the diffuse sound field, the classical spatial correlation function sin(kr)/kr is used, and the Transmission Loss TL is computed. For the turbulent boundary layer excitation, the Corcos model is used, and an equivalent Insertion Loss IL is determined.

For both excitation fields, for the stiffened and unstiffened plate, TL and IL were obtained and compared in the 100 Hz – 2200 Hz range. For the turbulent boundary layer excitation the aerodynamic coincidence frequency is equal to 1700 Hz.

The main results are the following:

- for both excitation fields, the stiffened plate TL and IL are lower compared to the unstiffened ones in the 200 Hz – 1500 Hz range;
- the differences between the stiffened and unstiffened structures depend on the excitation field both in levels and trends, thus the transmission mechanisms differ.

Further investigation is compulsory to determine the most sensitive parameters for sound transmission through stiffened structures under turbulent boundary layer excitation.

1. INTRODUCTION

The turbulent boundary layer excitation of aircraft structures is one of the major interior noise sources over a large frequency range. In the context of understanding and predicting the airborne sound transmission typically a diffuse sound field is used. It is therefore all the more important to determine to which extent this acoustic field is representative of the real excitation for interior noise assessment.

Although several authors studied the stiffeners effect on plates (or cylinders) under diffuse sound field excitation [1] [2] [3] [4], or more recently under turbulent boundary layer excitation [5], none of them compared this effect for both excitation fields. Moreover no comprehensive study over a large frequency range could be found, particularly for the turbulent boundary layer case.

The main objective of the present paper is therefore to analyse and compare the stiffeners effect on a rectangular thin plate either excited by an acoustic field or a turbulent boundary layer excitation.

2. SIMULATION DESCRIPTION

2.1 Structures description

Figure 1 gives the main characteristics of the rectangular plates on which we carried out simulation. The stringers are equally spaced. The stiffeners and the plate are made of the same homogeneous isotropic material.

The critical frequency of the unstiffened plate, which correspond to the maximum radiation efficiency is given by equation (1), and is equal to 7000 Hz.

$$f_c = \frac{1}{2\pi} c^2 \sqrt{\frac{M}{D}} \qquad (1)$$

c is the sound celerity, M is the surface density of the plate and D its bending stiffness.

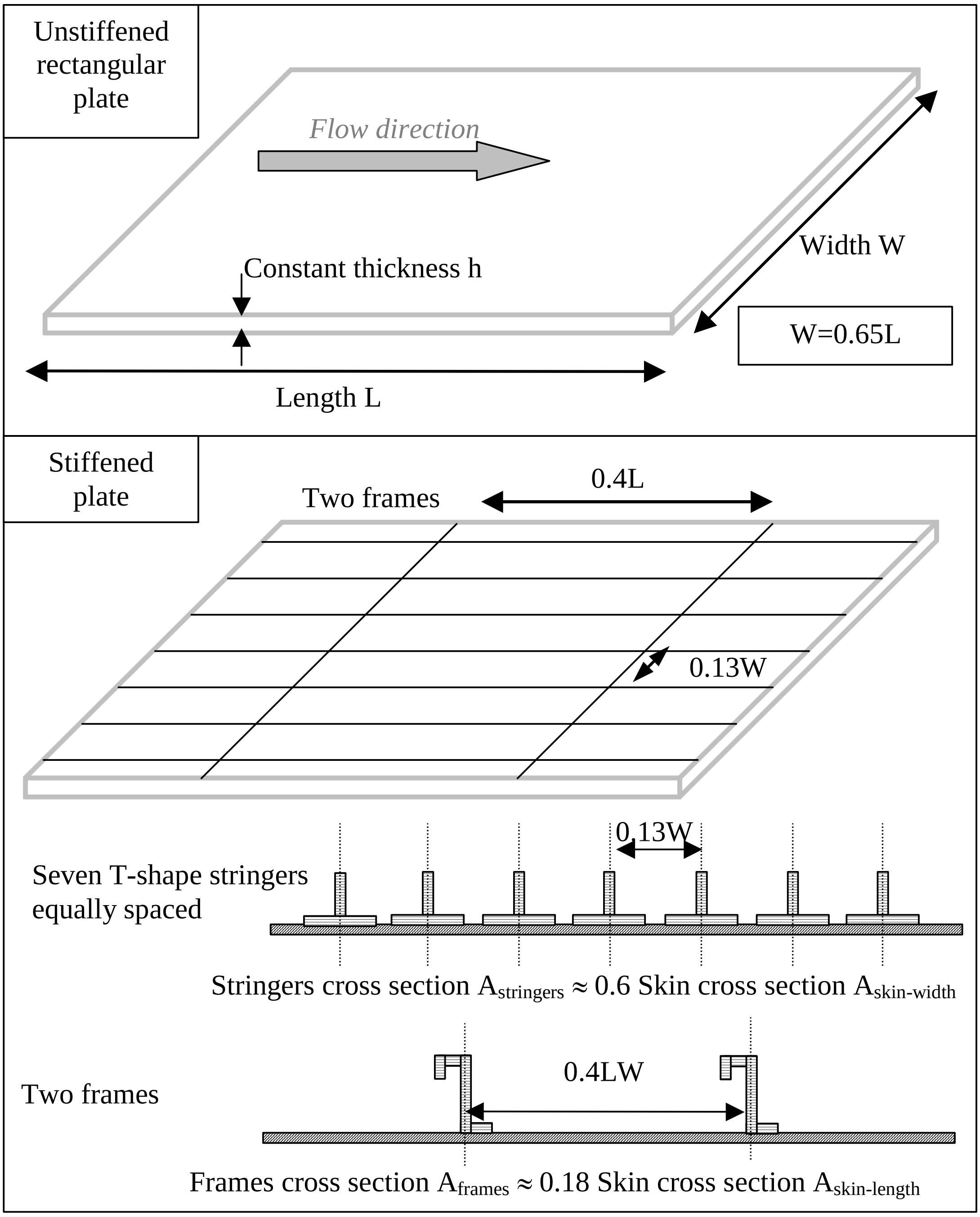

Figure 1. Unstiffened and stiffened plate characteristics – Frames, stringers and skin are all made of the same isotropic material.

2.2 Simulation description / Main assumptions

We studied the vibroacoustic behaviour of the unstiffened and stiffened plates with FEM software ACTRAN© in the [200 Hz ; 2200 Hz] range. The plates were clamped along their edges.

2.2.1 Random excitations

Random excitations can be considered as stationary random distributed processes, characterized by a reference power spectrum $S_{pp}(\omega)$ and a spatial correlation function $f_{cross}(P,P')$, taking into account the cross effects. ω is the circular frequency, P and P' are points along the loading surface. Two different random excitations were considered:

- an acoustic diffuse field, $f_{cross}(P,P')$ being defined by equation 2; k being the acoustic wavenumber:

$$f_{cross}(P,P') = \frac{\sin\left(k\left|P-P'\right|\right)}{k\left|P-P'\right|} \tag{2}$$

- a turbulent boundary layer excitation (Corcos model [9]) defined by equation 3, y and x being respectively the stream wise and span wise directions, α_x, α_y being the constants of the Corcos model, and U_c the convection velocity:

$$f_{cross}(P,P') = e^{-\alpha_x\omega\left|x-x'\right|/U_c}e^{-\alpha_y\omega\left|y-y'\right|/U_c}\cos\left(\frac{\omega\left|y-y'\right|}{U_c}\right) \tag{3}$$

2.2.2 Stiffeners modelling

The stiffeners were modelled using ACTRAN© 2D elements, which means that traction, normal bending and shear are taken into account, as well as translation and rotary inertia. We considered that torsion, non-normal bending, as well as stiffeners radiation can be neglected at least for a first approach and for the frequency range on which computation has been performed.

2.2.3 Meshing

The meshing criteria was defined in order to match the bending wavelength of the unstiffened plate as well as the spatial correlation to the excitation field. This lead to a model of 36800 DOFs at 2200 Hz.

2.3 Transmission loss (TL) and Insertion Loss (IL) calculation

We computed the vibroacoustic indicators defined by [8]. For the diffuse sound field excitation, the classical Transmission Loss TL defined by equation (4) was determined, Π^{inc} being the incident acoustic power, Π^{rad} the acoustic radiated power. S_p (Pa^2/Hz) is the power spectral density of the acoustic pressure in far field, c the sound celerity, ρ_0 (kg/m^3) the fluid density and S the plate surface S (m^2).

$$TL(dB) = 10\log_{10}\left(\frac{\Pi^{inc}}{\Pi^{rad}}\right) \text{ with } \Pi^{inc} = \frac{S_p}{4\rho_0 c}S \tag{4}$$

For a turbulent boundary layer excitation, there is no incident acoustic pressure in far field S_p but only a power spectral density S_{pp} on the plate surface. Therefore we defined an equivalent incident power Π^{equ}, equal to the one of a diffuse sound field with a power spectral density equal to S_{pp} on the plate surface. The equivalent incident power and Insertion Loss is given by equation (5).

$$\mathrm{IL(dB)} = 10\log_{10}\left(\frac{\Pi^{equ}}{\Pi^{rad}}\right) \text{ with } \Pi^{equ} = \frac{S_{pp}}{8\rho_0 c} S \quad (5)$$

3. STIFFENERS EFFECT ON SOUND TRANSMISSION

3.1 Diffuse sound field excitation

We first determined as a baseline the stiffeners effect on the Transmission Loss curves of the plate (defined by figure 1) excited by an acoustic diffuse field, cf. figure 2.

The TL of the unstiffened plate is very classical and increases with a 6 dB/octave slope. This is the mass law behaviour below the critical frequency, equal to our case to 7000 Hz.

The TL of the stiffened plate is much lower than the one of the unstiffened in the mass law area, the difference remaining almost the same over the frequency range, and equal to 5 to 10 dB. This result is classical and was obtained experimentally by [4] for instance. The decrease of the TL compared to the unstiffened case is mainly due to spatial coincidence between the stiffened plate structural modes and the acoustic excitation field.

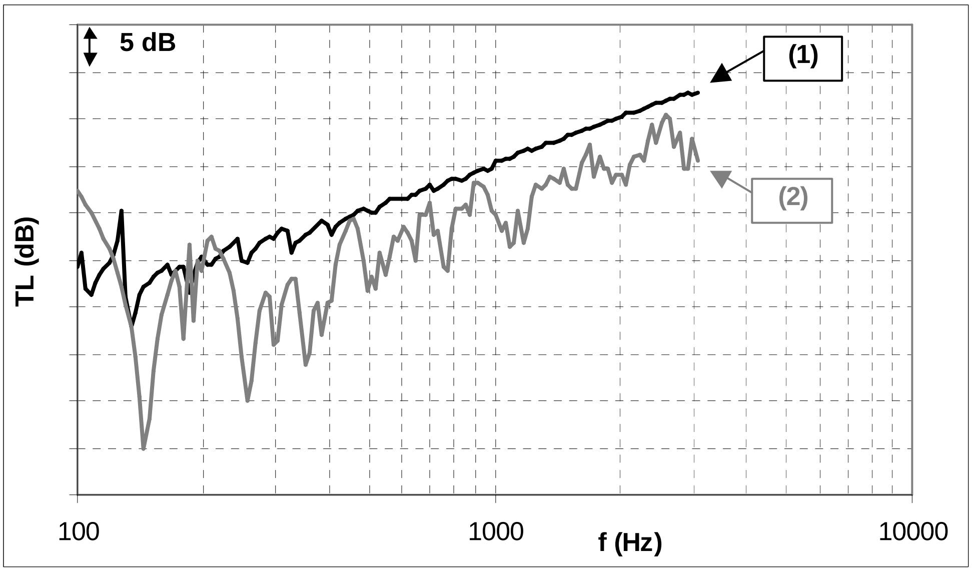

Figure 2. Stiffeners effect on the rectangular thin plate for a diffuse sound field excitation. (1) Unstiffened plate. (2) Stiffened plate.

3.2 Turbulent boundary layer excitation

We then determined the stiffeners effect for the same plate studied in §3.1 under turbulent boundary layer excitation (Corcos model). We indicated on the IL curves the value of the aerodynamic coincidence frequency given by equation 6 and equal in our case to 1700 Hz. This frequency corresponds to the equality between the flexural wave velocity of the plate and the flow convection velocity U_c.

$$f_{caero-Plate} = \frac{1}{2\pi} U_c^2 \sqrt{\frac{M}{D}} \quad (6)$$

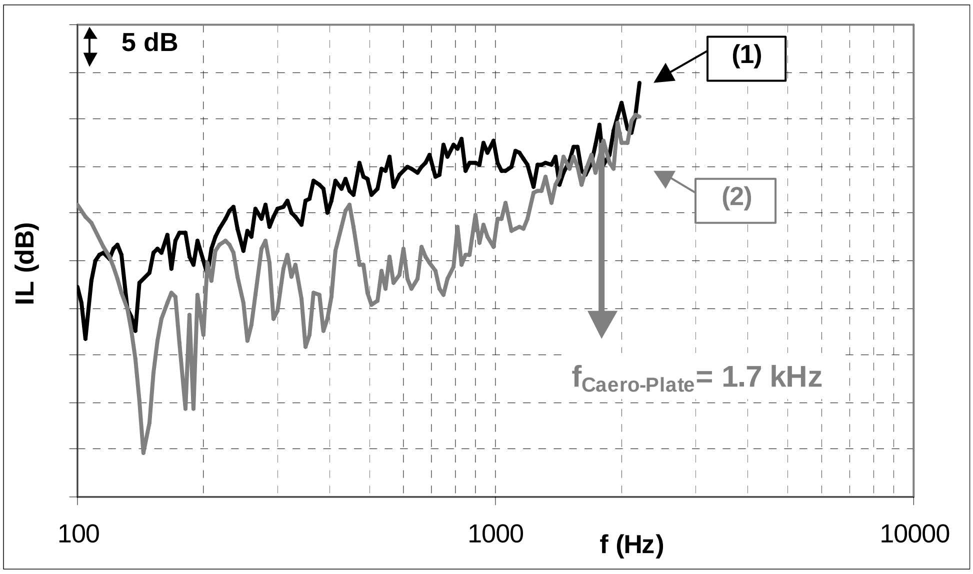

Figure 3. Stiffeners effect on the rectangular thin plate for a turbulent boundary layer excitation (Corcos model). (1) Unstiffened plate. (2) Stiffened plate.

The IL of the unstiffened plate is similar to the one obtained by [8], that is, an increase of the IL with a minimum around the aerodynamic coincidence area. The stiffeners effects are the following:

- a decrease (up to 15 dB) of the IL over a large frequency range, below $f_{caero\text{-}Plate}$, this decrease becomes lower while the frequency increases;
- almost no effect around the aerodynamic frequency area; consequently no deep minimum at $f_{caero\text{-}Plate}$ for the stiffened plate IL.

The stiffeners effect under turbulent boundary layer excitation is therefore different in trend and level to the one under acoustic diffuse field excitation. Those are new results.

Note that Liu [5] noticed that the radiated power under turbulent boundary layer excitation of a stiffened plate was higher to the one of an unstiffened plate below 1 kHz. Nevertheless his study presented no analysis with respect to the aerodynamic coincidence area, and no results for acoustic excitation.

3.3 Conclusion

Our main results are the following:

- for both excitation fields, the stiffeners decrease the Transmission Loss or the Insertion Loss compared to unstiffened structures in the [200 Hz ; 1500 Hz] frequency range; these trends were noticed by previous authors;
- the stiffeners effect is different both in levels and trends depending on the excitation field, thus the transmission mechanism differ, this is a new result:
    - under a diffuse sound field excitation the stiffened plate TL is lower to the unstiffened one, the differences (5 to 10 dB) remain almost unchanged in the mass law zone;
    - under a turbulent boundary layer excitation, the stiffened plate IL is lower to the unstiffened one below $f_{caero\text{-}Plate}$; in the $f_{caero\text{-}Plate}$ area, there is almost no difference between the unstiffened and stiffened curves.

4. CONCLUSION / PERSPECTIVES

One can conclude that using the real excitation is compulsory to make precise assessment of interior noise due to turbulent boundary layer. These results show that further studies and analysis are required to analyze precisely the coincidence phenomena between the plates modes and the excitation field. We also intend to cover a higher frequency range up to the structural critical frequency.
Another perspective of the present paper is to extend this analysis to more complex structures such as curved plates and cylinders, with double wall configurations (blankets and trim panel).

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