Contribution of stiffeners to the sound transmission of aircraft skin panels

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Introduction

Traditionally, the sound radiation of aircraft skin panels is calculated using a structural model with stiffners and an acoustical model where the stiffners are omitted. We present a numerical study on the impact of this omission. The numerical model consists of three parts. Firstly, the pressures in a diffuse sound field are modeled. Secondly, a Finite Element (FE) model describes the structural response caused by the sound field. Thirdly, a Boundary Element (BE) model yields the sound intensity.

The impact of stiffeners is determined by comparing the results of a BE model with and without stiffeners. The emphasis lies on the results of this model. Several stiffener configurations and frequencies are compared and a number of new analysis techniques are introduced to pinpoint the cause of these differences. Differences of up to 35% have been found for the geometries that have been tested.

Section 2 introduces the numerical model that has been used. Sections 3 and 4 present an overview of the results and conclusions are drawn in section 5.

Model

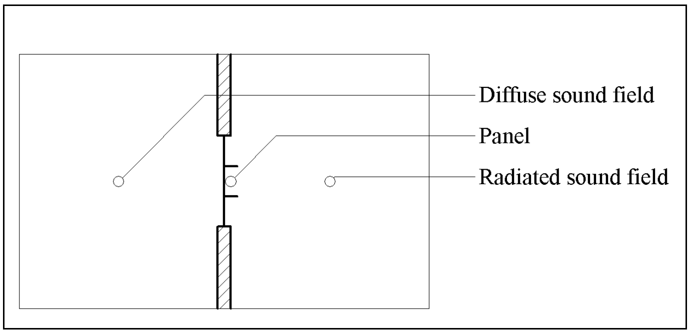

Figure 1: Schematic overview of the model

The model represents the experimental setup used in [2]. A diffuse sound field excites a panel, causing structural vibrations. In turn, these vibrations cause sound to be radiated (see figure 1). The radiated sound power is calculated for two cases. In the case which is commonly used in the industry, the model of the radiated sound does not contain stiffeners although the structural model does. In the second case, stiffeners are included in both the acoustical and the structural model.

A diffuse sound field consists of incoming plane waves from all sides, which are modelled to be mutually uncorrelated. This means that the sound field has been modeled statistically. The cross-correlation between the pressure $p(x,\omega)$ at point x and frequency ω and $p(y,\omega)$ at point y is known to be [3]

$$\mathbf{C}_{xy} = E\left(p(x,\omega)\overline{p(y,\omega)}\right) \tag{1}$$

$$\mathbf{C}_{xy} = \mathbf{C}_0 \frac{\sin(k\,|\,x-y\,|)}{k\,|\,x-y\,|} \tag{2}$$

where E, k and C_0 are the is the expected value, the wave number and the auto-correlation of the pressure, which is independent of the location and the frequency. Due to this random excitation, the response of the structure and the radiated sound field are random as well.

A FE model is used to model the vibration of the panel. This approach describes the dynamics of the panel in the frequency domain as follows.

$$\left(\mathbf{K} + i\omega\mathbf{D} - \omega^2\mathbf{M}\right)\mathbf{u} = \mathbf{f} \tag{3}$$

$$\mathbf{K}_D\mathbf{u} = \mathbf{f} \tag{4}$$

Where **u** and **f** are the displacement vector and the force vector. **K, D** and **M** are the stiffness, damping and mass matrices respectively. Modal reduction is performed to reduce the computation times, but the equations required to achieve these efficient calculations are not discussed here. In terms of cross-spectra, equation 4 becomes

$$\mathbf{C}_{ff} = E\left(\mathbf{f}\mathbf{f}^H\right)$$

$$\mathbf{C}_{ff} = E(\mathbf{K}_D\mathbf{u}\mathbf{u}^H\mathbf{K}_D{}^H)$$

$$\mathbf{C}_{ff} = \mathbf{K}_D E\left(\mathbf{u}\mathbf{u}^H\right)\mathbf{K}_D{}^H$$

$$\mathbf{K}_D\mathbf{C}_{uu}\mathbf{K}_D{}^H = \mathbf{C}_{ff} \tag{5}$$

Where $\mathbf{C}_{uu}$ and $\mathbf{C}_{ff}$ are the cross-correlation matrices of the displacements and the forces respectively and $(\cdot)^H$ denotes the Hermitian transpose. The matrix $\mathbf{C}_{ff}$ can be derived from the cross-correlation matrix of the pressures (equation 2) such that equation 5 provides cross-correlation of the structural displacements $\mathbf{C}_{uu}$.

To calculate the radiated sound power, the pressures at the surface must be calculated. A boundary element (BE) model is used for this porpose. This model yields the following frequency domain description.

$$\mathbf{p} = \mathbf{Z}\mathbf{v} = i\omega\mathbf{Z}\mathbf{u} \tag{6}$$

Where **Z** and **p** are the impedance matrix and the pressure vector respectively. The radiated sound power follows from the cross-spectral matrix between pressure and velocity. It is given by

$$\mathbf{C}_{pv} = i\omega\mathbf{Z}\mathbf{C}_{uu} \tag{7}$$

The radiated sound power is

$$W = \frac{1}{2}\Re\left(\int_\Gamma \mathbf{C}_{p(x)v(x)}dx\right) \tag{8}$$

Where $\Re$, Γ and $\mathbf{C}_{p(x)v(x)}$ denote the real part, the surface of the radiating side of the panel and the cross-spectrum between the pressure and velocity at point x respectively. This equation may be expressed in terms of the BE results as follows

$$W = \Re\left(tr\left(\mathbf{BC}_{pv}\right)\right) \quad (9)$$

Where $\Re$ and tr denote the real part and trace respectively

$$\mathbf{B}_{ij} = \frac{1}{2}\int_{\Gamma} N_i(x) N_j(x) dx \quad (10)$$

Where Γ is the surface of the radiating side of the panel and $N_i(x)$ is the shape function of node i.

The boundary element model and its corresonding impedance matrix are calculated for the case where the stiffeners are omitted as wel as the case where stiffeners are present. The relative difference between these cases calculated as

$$\gamma = \frac{W_s - W_{ns}}{W_s} \quad (11)$$

Where W_s and W_{ns} denote the case where the stiffeners are present and the stiffeners are omitted respectively.

Equations 1-11 model one-way coupling from a sound field to structural vibrations to the radiated sound field, which means that the acoustical pressures caused by the vibration of the structure have no impact on the vibration of the structure. Numerical experiments with full coupling show that the errors are negligible.

Simulations

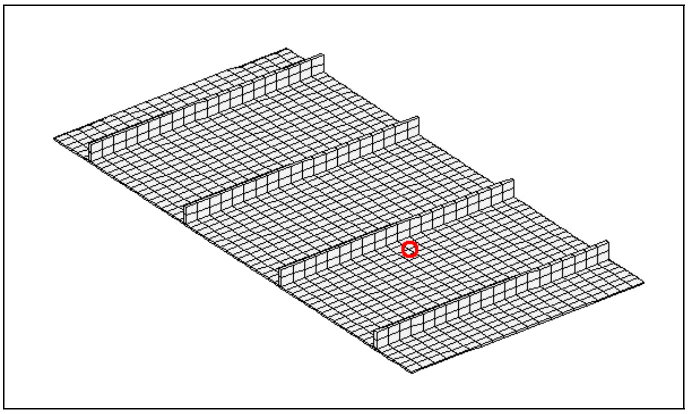

Figure 2: Boundary element model of a panel with 4 stiffeners. (○) node β, which will be discussed in the next section.

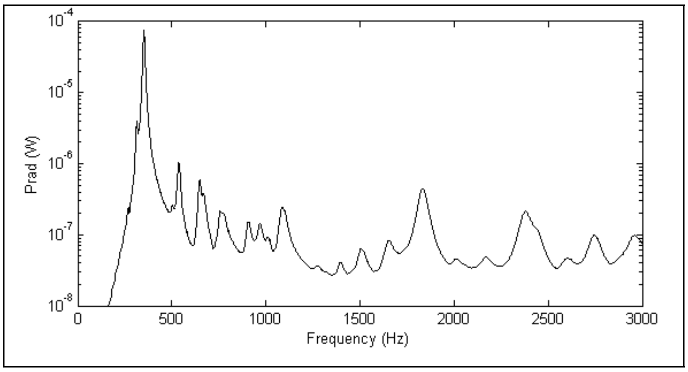

Figure 3: Radiated power versus frequency

A plate of 0.3m×0.6m with 4 stiffeners is modeled (see figure 2). The mechanical properties are modeled after a composite material, using E=32.5 and ρ=2000kg/m^3. For a more detailed description, see [1]. Modal damping is applied with a damping coefficient ζ=0.03.

The commercial FE package ANSYS has been used to calculate the matrices of the structural calcuation and to perform modal reduction. An in-house direct boundary element code has been used to calculate the matrices of the acoustical calculation [4]. Although an indirect boundary element code is better suited to these thin-walled structures, the thickness of 4mm proves to be a good compromise between a good description of the geometry and small numerical errors due to sufficiently thick walls [1]. MATLAB has been used to implement equations 2-11 in order to calculate the transmitted sound power based on FE and BE results.

Figure 3 depicts the radiated sound power of the panel with stiffeners. The modal behavior can be seen clearly and especially the lower modes cause a large radiated sound power. Figure 4 depicts the relative difference between the case with and without stiffeners. The difference is well within 10% in the frequency range of 0-2750Hz. The negative peak around 648Hz is -4.3% and the positive peak around 1450Hz is 6.8%. At 2900Hz, the difference is 36%, which can be a considerable error in practice. The next section will analyse the differences at 2900Hz and 650Hz respectively.

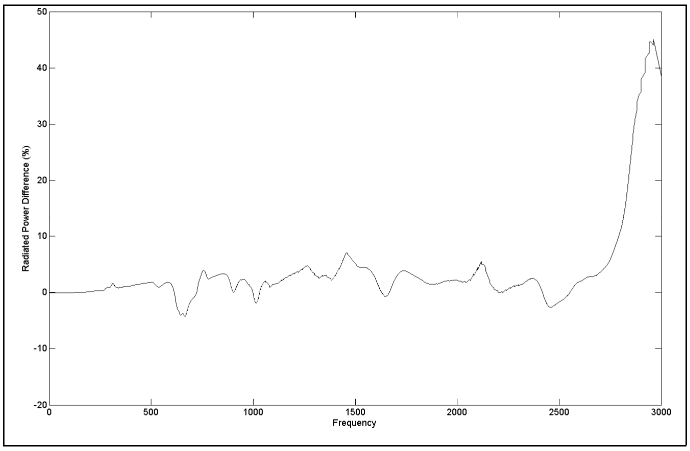

Figure 4: γ versus frequency

Analysis of the results

To understand the difference in radiated sound power, it is convenient to start by plotting the operational deflection shape. Since the model is statistical, the are multiple incoherent shapes of vibration. Figure 5 depicts the deflection shape, correlated to an arbitrary node.

At 2900Hz, the difference is mainly caused by vibration of the stiffeners: if the stiffeners are modeled to stand still, the difference decreases from 36% to 7%. Simulations of T-shaped stiffeners demonstrate simmilar differences around 1250Hz, which shows that this difference is not caused by numerical inaccuracies due to the high frequency.

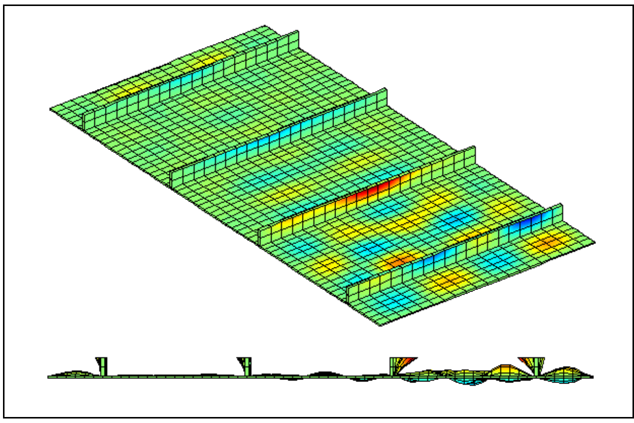

Figure 5: Operational deflection shape at 2900Hz, correlated to an arbitrary node

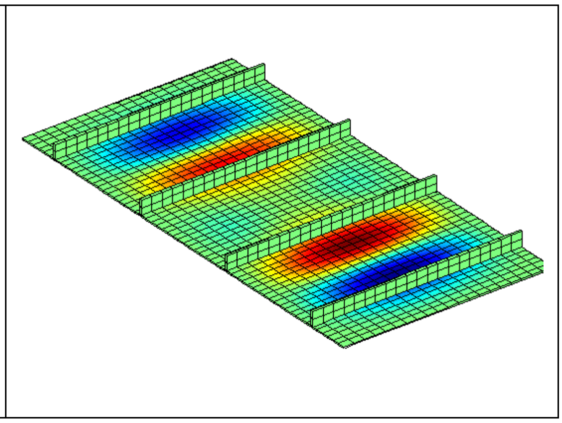

Figure 6: Operational deflection shape at 648Hz

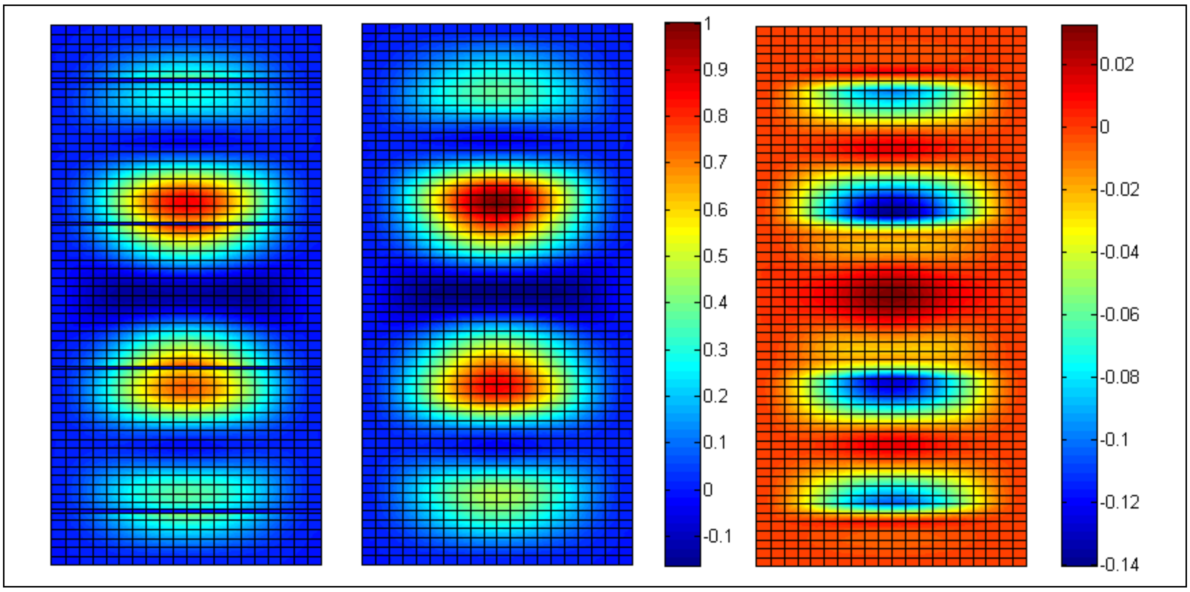

Figure 7: Sound intensity. Left: acoustical model with stiffeners, stiffener vibration set to zero. Middle, acoustical model without stiffeners, right: difference in intensity between the two panels.

At 648Hz, the deflection shape is dominated by the (1,5) mode shape, which has an eigenfrequency of 648Hz (see figure 6). The negative peak is caused by the fact that the sound can reflect on the stiffeners: if the vibration of the stiffeners is set to zero, then the difference in radiated soundpower increases from -4.3% to -14.1%. In figure 7, the sound intensity of a panel with stiffeners which stand still is compared to the sound intensity of a panel without stiffeners.

This case demonstrates that the behavior can be difficult to explain even if the results are known. The fact that a flat panel radiates more sound than a panel with stiffeners is contrary to the intuition that stiffeners allow pressure to build up, causing a larger radiated sound power. Although this intuition is correct if the deflection shape between the stiffeners has a large volumetric displacement, it is incorect for this case, where the increase of pressure on one side is compensated by a larger negative pressure on the other side. In this case, the effect of the negative pressure outweighs the effect of the positive pressure, causing a negative γ.

Conclusion

This article studied the impact of stiffeners on the radiated sound power of aircraft skin panels by means of a simulation study. Based on the simulations presented here, as well simulations of two different geometries, it is concluded that the difference tends to be within 10% if the vibration of the stiffeners itself is neglected. The vibration of the stiffeners has caused a difference of 35% in this case and a difference of around 50% in the case of T-shaped stiffeners, at frequencies where the vibration of the stiffeners is large.

References

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