OPTIMISATION OF VISCOTHERMAL DAMPING OF DOUBLE WALL PANELS

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1. INTRODUCTION

Viscosity and thermal conductivity can cause dissipation of acoustic energy in thin air layers [1,2,3]. Recently a new acoustic finite element was developed for a thin air layer trapped between two flexible panels [4,5]. This element also includes the thermal and viscous effects in the air layer and the mutual interaction between the pressure distribution in the air and the vibrating panels. The finite element was validated experimentally and is used in the presented study for optimising the viscothermal damping of double wall panels.

2. PROBLEM DEFINITION

In Figure 1 two chambers are shown which are separated by a double wall panel. In chamber A sound is generated. Part of the sound is transmitted to chamber B. The sound transmission through the double wall panel is studied by numerical simulations with the finite element program B2000 [6]. It will be shown that the dimensions and the properties of the plates and the air layer can be adjusted in such a way that the air layer dissipates a maximum amount of energy in a certain frequency range.

Important parameters for creating damping are the thickness ratio of the plates, their natural frequencies and the width of the gap.

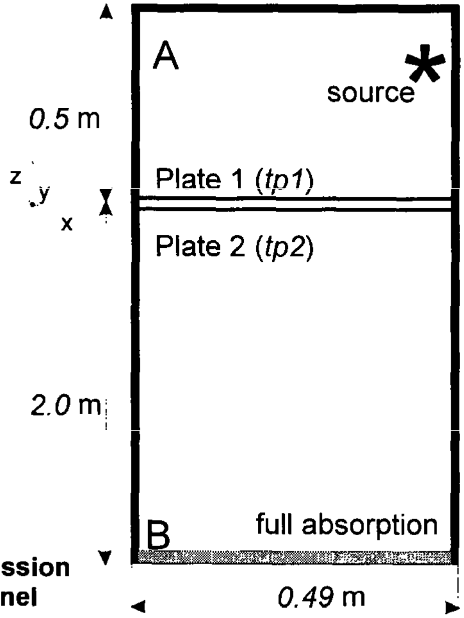

Figure 1. Sound Transmission through a double wall panel

3. ENERGY LOSS CALCULATIONS

To reduce the calculation time a 2-dimensional problem is considered. By choosing the appropriate boundary conditions the variations in the y-direction equal zero. At the wall facing plate 2 ($z=-2.0$) the specific impedance is $Z=1.0$, which represents full absorption.

Chambers A and B are modelled by standard acoustic elements. In chamber A a pressure perturbation is introduced in the upper right corner. Plate elements are used to model the two aluminium plates. The new viscothermal acoustic elements are used to model the air in the gap between these plates. By demanding continuity of velocity, a fully coupled acousto-elastic finite element formulation results:

$$-\omega^2\begin{bmatrix} M_s & 0 \\ M_c & M_a(\omega) \end{bmatrix}\begin{bmatrix} \{u\} \\ \{p\} \end{bmatrix}+\begin{bmatrix} K_s & -K_c \\ 0 & K_a \end{bmatrix}\begin{bmatrix} \{u\} \\ \{p\} \end{bmatrix}=\begin{bmatrix} \{0\} \\ \{P^*\} \end{bmatrix} , \qquad (1)$$

with ω the angular frequency, $\{u\}$ the vector with structural degrees of freedom and $\{p\}$ the vector with acoustic pressures. M_S and K_S are the structural mass and stiffness matrices, $M_a(\omega)$ and K_a are the acoustic mass and stiffness matrices. M_C and K_C are the coupling matrices. The right hand side of the equation represents the excitation.

Due to the influence of viscosity and thermal conductivity on the effective speed of sound, which appears in the viscothermal elements, the acoustic mass matrix is complex and frequency dependent. Due to the coupling the system of equations is asymmetric.

Frequency response calculations are performed in the frequency range from 0 to 300 Hz. The energy loss (EL) is calculated from the time averaged energy input in plate 1 ($\overline{W}_1$) and the time averaged energy output from plate 2 ($\overline{W}_2$):

$$EL = 10\log\left(\frac{\overline{W}_1}{\overline{W}_2}\right) . \qquad (2)$$

$\overline{W}_1$ and $\overline{W}_2$ are calculated by [7]:

$$\overline{W} = \frac{1}{2}\int \mathrm{Re}\{pv^*\}dA , \qquad (3)$$

where p is the acoustic pressure and v the acoustic velocity, which equals the structural velocity at the interface. When the viscous and thermal effects are discarded $\overline{W}_1 = \overline{W}_2$. So $EL=0$ and no energy is dissipated in the gap.

4. PARAMETER ANALYSIS

The most important parameters, the ratio of the plate thicknesses and the layer thickness are varied. The thickness ratio *tp1* : *tp2* is varied in such a way that the total thickness remains 3 mm to ensure the same amount of mass. The gap width is varied for a constant thickness ratio (*tp1* =1 mm, *tp2* = 2 mm).

The results for some typical configurations are given in Figures 2 and 3. In Figure 2 the effect of the thickness ratio of the plates forming the double wall panel is shown, while Figure 3 depicts the effect of the gap width.

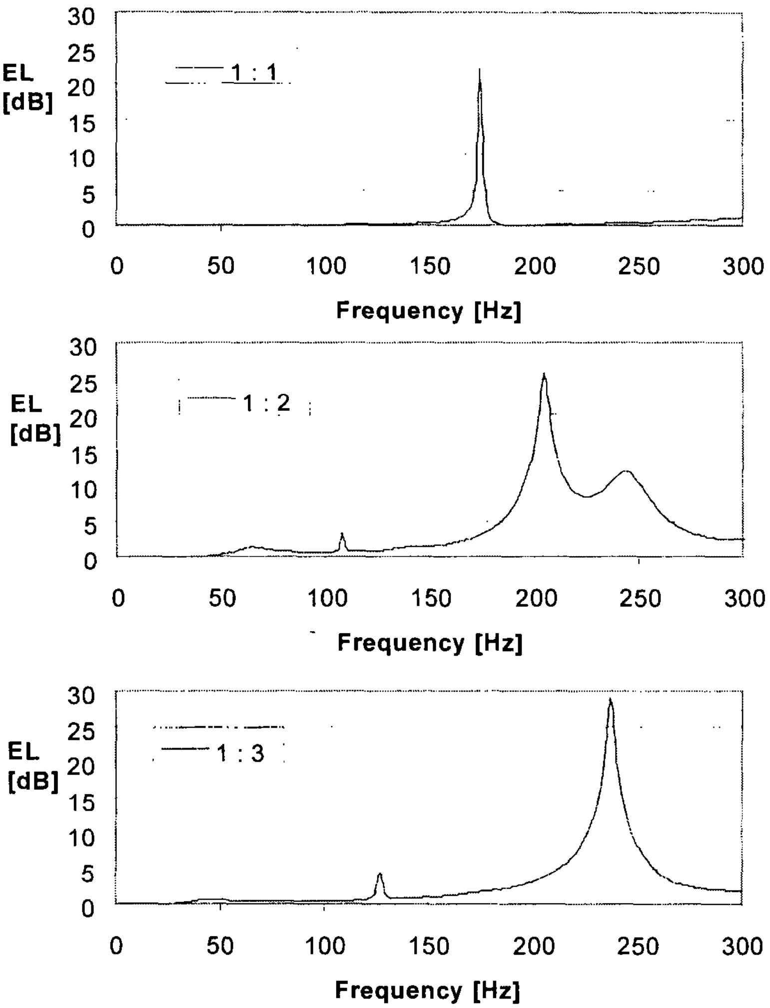

Figure 2. Energy loss for three different thickness ratios: 1 : 1, 1 : 2, 1 : 3

As could be expected, energy dissipation occurs in small bandwidths around structural eigenfrequencies. A significant amount of energy is dissipated in the layer by means of viscous shear, so especially modes with a pumping mechanism cause dissipation of acoustic energy.

When the mean energy loss in the frequency range from 175 to 275 Hz is taken as the objective function, in this case the configuration with a thickness ratio of 1 : 2 and a gapwidth of 1.0 mm is found as the optimum.

5. CONCLUSIONS

It has been demonstrated that the new viscothermal finite elements are well suited for optimising the damping of double wall panels. The effect of viscothermal damping is limited to bandwidths around structural eigenfrequencies of the double wall panel.

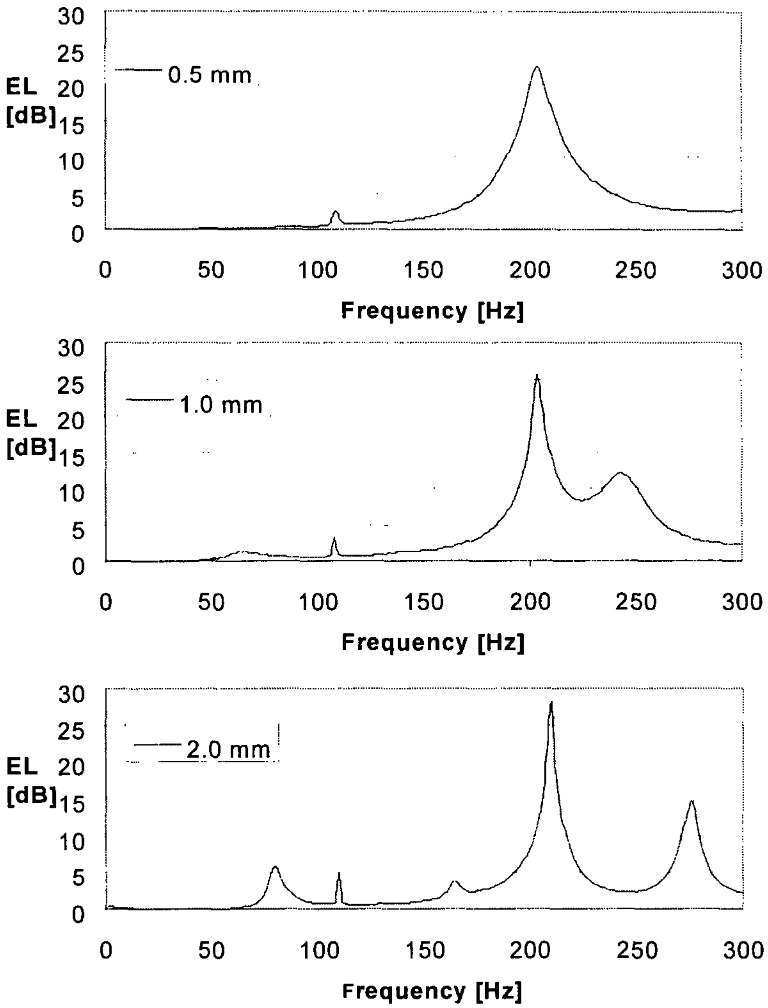

Figure 3. Energy loss for three different gap widths: 0.5 , 1.0 , 2.0 mm

6. ACKNOWLEDGEMENTS

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7. REFERENCES

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