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Sound Insulation by Laminated Viscoelastic Plates

Plates are basic components in industrial structural design whose acoustical properties are often very important to reduce noise, e.g., in a car or a building. The sound insulation by plates is influenced by various effects, e.g., by the plate's mass but also by their damping capability. Hence, for a realistic analysis, damping should be taken into account which can approximatively be modelled by the use of a viscoelastic material law.
Moreover, in automotive industry and structural engineering, laminated plates are very common to improve stiffness and stability, where plates with a viscoelastic core and rigid faces show additionally an excellent acoustical behaviour. Those laminated panes with a viscoelastic core are used in the design of windscreens, and, to improve sound insulation, in the design of building windows.
Here, the mechanical behaviour of such a laminated plate is described by using a homogenisation procedure. The damping property of the plate is influenced by the stiffness of the viscoelastic core, it's dissipation factor, and by the thickness of the dissipating layer. These properties of a three-layer-plate are mapped on homogenised material parameters of a simple plate via the Ansatz of Ross, Kerwin and Ungar (RKU-Ansatz).
Finally, the sound insulation effectiveness of plates is analysed numerically.

1. Numerical Calculation of Transmission Loss

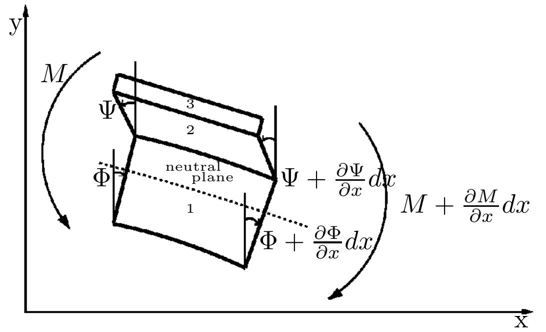
The Transmission Loss of plates is numerically determined by applying a coupled Finite-Element-/Boundary Element Method. Vibrating plates and sound field in interior domains are treated with Finite Elements Method, while the sound radiation into the outer unbounded fluid domains is modelled by the Boundary Element Method. The interactions of the structural and fluid parts of the system are represented via a strong coupling of system parts and numerical methods on the basis of the principle of virtual works [1].

2. Modeling of laminated plates

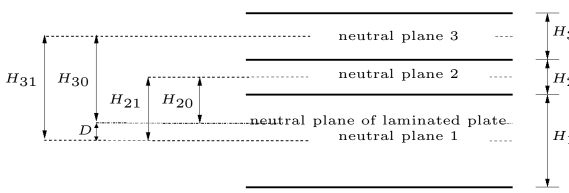
The damping of flexural vibrations should be described with the equations of a three-layer plate with a visco-elastic intermediate layer to determine the sound insulation of laminated viscoelastic plates. There are two different ways to analyze the vibrations of laminated plates: One can either use the classic theory of laminates to put down to a finite element formulation or one can maintain the finite element formulation for the one-layer plate and use homogenized material parameter to treat the problem of vibrating laminated panes. This way is sufficent if only the radiated sound of the attenuated, vibrating plate is of interest and so it is used in the following.

Here, the approach of Ross, Kerwin and Ungar [2] is applied to derive homogenized material parameters of a three-layer plate. They assume that bending vibrations of the plate are shared by all the three layers, and, that the visco-elastic middle layer of the plate is sheared due to the sticked connection with the two other layers.

From equilibrium condition at a differential element (fig. (a)) and considering geometry data of the three layers (fig. (b)), the bending stiffness of the laminated plate is calculated with this so-called RKU-approach. The



(a) Differential Element under bending deformation



(b) Geometry of a laminated plate

approach leads to the following expression for the homogenized bending stiffness B

$$\begin{aligned} B &= K_1\frac{H_1^2}{12}+K_2\frac{H_2^2}{12}+K_3\frac{H_3^2}{12}-K_2\frac{H_2^2}{12}\frac{\partial\psi}{\partial\phi}+K_1D^2+K_2(H_{21}-D)^2 \\ &+ K_3(H_{31}-D)^2-H_2\frac{\partial\psi}{\partial\phi}\cdot\left(\frac{K_2}{2}(H_{21}-D)+K_3(H_{31}-D)\right), \end{aligned} \tag{1}$$

with Young's Modulus $E\,[N/m^2]$, a distance $D\,[m]$ (see fig. (a), shear modulus $G\,[N/m^2]$, a distance $H\,[m]$ (see fig. (a)), stiffness $K = E\cdot H[N/m]$, wave number $\kappa = \omega/c\,[1/m]$, flexural angle $\phi\,[rad]$ and the shear angle of layer 2 $\psi\,[rad]$. The indices of the material and geometry data indicate the number of the plate's layer (see fig. (b)).
In equation (1), the relation between the shear angle and the flexural angle $\frac{\partial\psi}{\partial\phi}$ can be replaced by

$$H_2\frac{\partial\psi}{\partial\phi} = H_2\frac{(\partial^2\psi/\partial x^2)}{(\partial^2\phi/\partial x^2)}=\frac{H_{31}-D}{1+(G_2/K_3H_2\kappa^2)}, \tag{2}$$

where the distance D between the neutral plane of the first layer and the neutral plane of the laminated is given by

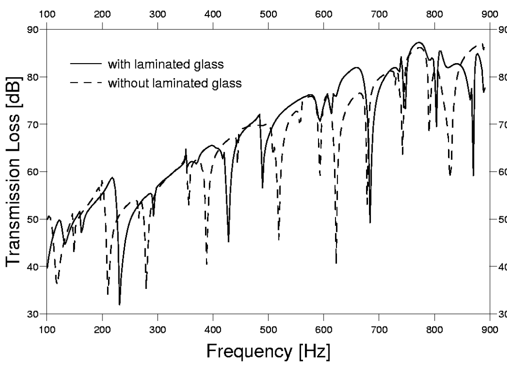
$$D = \frac{K_2\cdot\big(H_{21}-(H_{31}/2)\big)+(K_2\cdot H_{21}+K_3\cdot H_{31})\cdot g}{K_1+(K_2/2)+g\cdot(K_1+K_2+K_3)}, \tag{3}$$

with the shearing parameter $g = (G_2)/(K_3\cdot H_2\cdot\kappa^2)$.

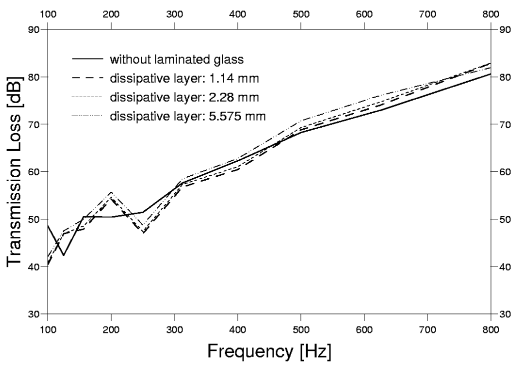
3. Numerical examples

The influence of laminated glass on the sound insulation is studied with the following example. Figures (c) and (d) shows a comparison between a conventional insulation window with two simple panes and a window whose outer pane is replaced by a laminated glass. The visco-elastic material of laminated pane's middle layer consists of polyvinyl butyral with Young's Modulus of $1.5\cdot 10^{+7}\,[Pa]$, density of $1100\,[kg/m^3]$, and poisson's ratio of 0.4. The material data for the considered glass are a Young's Modulus of $6.32\cdot 10^{+9}\,[Pa]$, density of $2300\,[kg/m^3]$, and a poisson's ratio of 0.24.

In figure (c), one can see that the peaks in the Transmission Loss of a window with laminated glass are damped compared to a conventional window, because of the energy loss in the visco-elastic middle layer of the laminated glass. A look on the third-averaged Transmission Losses shows (d) that the insulation of a window with laminated glass is better than that of a window with two simple panes, and that this positive effect on sound insulation increases with thickness of the visco-elastic layer and with frequency.



(c) Not third-averaged



(d) Third-averaged

4. References

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