AIRCRAFT SIDE WALL WITH IMPROVED LOW-FREQUENCY SOUND INSULATION USING A LINING PANEL WITH HELMHOLTZ RESONATORS

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The continuous development of aircraft engines aims to reduce fuel consumption by increasing the bypass ratio. However, an increased bypass ratio changes the sound emission of the engine, such that the sound spectrum inside the cabin can be dominated by low frequency tones with high sound pressure levels, penetrating more efficiently the current cabin wall than sound of higher frequencies. Thus, the comfort level of the passengers could be adversely affected.

This contribution presents a new side wall panel design that is capable to significantly increase the transmission loss of low frequency tones. The rear side of the lining is fully covered by a resonator system of a cantilever integrated into a Helmholtz resonator. With this approach, the lining with resonators can be tuned with respect to two resonance frequencies to improve the sound transmission loss of the cabin wall. A transfer matrix for the Helmholtz resonator panel is derived and the transmission loss of the proposed side wall design is determined. The analytically calculated transmission loss is compared to numerical and experimental results obtained under diffuse field conditions.

Keywords: Helmholtz resonators, vibration resonators, transmission loss, double wall, bandwidth

1. Introduction

For more efficient aircraft engines, the development tends to larger bypass ratios and gear driven fans. As a consequence, the noise emission of the engines will change, such that in addition to broadband sound discrete frequencies with high sound pressure in the low frequency range will arise. The present aircraft wall can be acoustically seen as a double wall, consisting of the outer fuselage skin and the inner cabin lining, with embedded insulation. This is a sufficient sound barrier in the high frequency region, i.e. above 1000 Hz, but in the low frequency region, where a number of tones of future aircraft engines are expected, the sound insulation is insufficient. Because of lightweight aeronautical design principles, a better sound insulation cannot be achieved by increasing the double wall mass or spacing. Therefore, a new concept using a resonator system with cantilevers integrated into Helmholtz resonators inside the double wall will be introduced. Two methods for integrating Helmholtz resonators inside the aircraft side wall can be found in the literature. On the one hand, they could be embedded into porous materials to
enhance the typically weak absorption behavior of - compared to the acoustic wave length - thin absorber blankets \(^1\). On the other hand, Helmholtz resonators could be incorporated in one or both walls of the aircraft double wall, which was investigated by Kuntz et al. \(^2\).

Helmholtz resonators are only acoustically effective in a small frequency region around their resonance frequency. To overcome this limitation a lot of effort was made to design Helmholtz resonators with two or more resonance frequencies. One possible design is a double chamber Helmholtz resonator \(^3\), which consists of two cavities in series. The continuation of this design is a stack of multiple resonators behind each other \(^4\). In Ref. \(^5\) a build-up of two Helmholtz resonators whose cavities are connected with a separating membrane is presented. A flexible plate or membrane can also be installed inside the Helmholtz resonator cavity \(^6\) or as an end plate \(^7\). These solutions require either a higher space consumption, a higher mass or a more complex manufacturing process compared to the single frequency Helmholtz resonator.

The present resonator system improves the described state of the art solutions through an integration of a cantilever into a Helmholtz resonator. Mass and volume remain unchanged compared to a single Helmholtz resonator. Figures 1(a) and (b) show one resonator as a unit cell of the resonator array on the lining, which is depicted schematically on Fig. 1(c).

A u-shaped slit is incorporated on one side of the resonator, such that a cantilever (CL) arises. The slit defines the area of the Helmholtz resonator neck, which is connected to the resonator cavity. The thickness of the resonator cover plate defines both, the length of the neck and the thickness of the cantilever. In this paper, the coupled resonator system is called hereafter Cantilever-Helmholtz resonator. It exhibits two tunable resonance frequencies, which can be designed closely together to broaden the overall bandwidth of a conventional Helmholtz resonator.

A transfer matrix model for the Cantilever-Helmholtz resonator panel is derived and will be used to calculate the transmission loss of the panel as well as the aircraft side wall. The analytically calculated transmission loss is compared to numerical and experimental results under diffuse field conditions.

![Figure 1: Unit cell of one Helmholtz resonator with cantilever resonator and resonator panel combined with the aircraft lining.](image)

### 2. Derivation of the Transfer Matrix

The schematical build-up of a laterally infinite resonator panel with density \(\rho_s\) and thickness \(h_s\) is shown in Fig. 2. The Cantilever-Helmholtz resonator presents an oscillator with two degrees of freedom. One degree of freedom is the movement of the fluid volume through the slit with the area \(S_{HR}\) and the second is the movement of the cantilever beam with area \(S_{CL}\). Both movements are coupled by the air in the resonator cavity \(V_0\), acting as a spring. The mean displacement \(\bar{u}_{HR}\) of the fluid in the slit as well as the mean displacement \(\bar{u}_{CL}\) of the cantilever beam cause a volume change in the cavity given by

\[
\Delta V = \Delta V_{HR} + \Delta V_{CL} = -S_{HR}\bar{u}_{HR} - S_{CL}\bar{u}_{CL}.
\]
This volume change generates a pressure difference $\Delta p$ inside the resonator. In the considered low-frequency region, the amplitude of $\Delta p$ is assumed to be uniform inside the resonator cavity at every frequency, because all dimensions of the resonator are assumed to be smaller than the acoustic wavelength. With the bulk modulus of the fluid $\rho_0 c_0^2$, $\rho_0$ and $c_0$ being the density and speed of sound of the fluid, respectively, the pressure difference is calculated as [8]

$$\Delta p = \rho_0 c_0^2 \frac{\Delta V}{V_0}. \quad (2)$$

The pressure difference expressed in terms of the particle velocity of the fluid in the throat $v_{HR} = i\omega \bar{u}_{HR}$, the vibration velocity of the cantilever $v_{CL} = i\omega \bar{u}_{CL}$ and the vibration velocity of the panel $v_s$ results in

$$\Delta p = \rho_0 c_0^2 \frac{\Delta V}{V_0} = -\frac{\rho_0 c_0^2}{V_0} S_{HR}(v_{HR} - v_s) + S_{CL}(v_{CL} - v_s) = -\frac{k_1}{i\omega} (\sigma_{HR}(v_{HR} - v_s) + \sigma_{CL}(v_{CL} - v_s)), \quad (3)$$

where $S_0$ is the area of the unit cell, $\sigma_{HR} = S_{HR}/S_0$, $\sigma_{CL} = S_{CL}/S_0$, $\sigma = \sigma_{HR} + \sigma_{CL}$, $k_1 = \rho_0 c_0^2/h_0$, and $h_0 = V_0/S_0$. Detailed explanations on the physical behavior and the calculation of resonance frequencies of the coupled system are listed in [9].

A transfer matrix $\mathbf{T}$ is derived, which links pressure $p_L$ and velocity $v_L$ on the one side to the pressure $p_R$ and velocity $v_R$ on the other side of a laterally infinite structure via

$$\begin{pmatrix} p_L \\ v_L \end{pmatrix} \mathbf{T} = \begin{pmatrix} p_R \\ v_R \end{pmatrix}. \quad (4)$$

The equations of motion for the panel, the air volume in the slit and the cantilever beam are

$$Z_s v_s = p_L - \sigma \Delta p - (1 - \sigma) p_R, \quad (5)$$
$$Z_{HR} v_{HR} = \Delta p - p_R \quad \text{and} \quad (6)$$
$$Z_{CL} v_{CL} = \Delta p - p_R, \quad (7)$$

where $Z_s$, $Z_{HR}$ and $Z_{CL}$ are the effective mechanical impedances of the panel, the air in the slit and the cantilever, respectively. With $i = \sqrt{-1}$, the angular frequency $\omega = 2\pi f$, the mass per unit area

Figure 2: Schematic representation of an infinite Cantilever-Helmholtz resonator panel. The blue filled area represents the fluid volume in the u-shaped slit of the resonator.
of the panel \( m'' \) and neglecting the bending stiffness for frequencies below the critical frequency, the mechanical impedance of the panel is \( Z = \imath \omega m'' \). The mechanical impedance of the air in the slit \( Z_{HR} = R_{HR} + \imath \omega m''_{HR} \) is generally composed of a resistive part \( R_{HR} \), accounting for the viscous damping in the resonator slit, and a reactive part \( \omega m''_{HR} \), accounting for the vibration of the fluid mass, and is calculated for a micro-slit with the analytical formula given in Ref \[10\]. The mechanical impedance of the cantilever is \( Z_{CL} = C_{CL}/\imath \omega + \imath \omega m''_{CL} \) \[11\]. In the low frequency region, it can be assumed that the vibration of the cantilever is dominated by the first eigenmode with the associated eigenfrequency \( \omega_{CL,1} \). The stiffness of the cantilever \( C_{CL} \) can then be replaced by \( C_{CL} = m''_{CL} \omega_{CL,1}^{-2} \). \( \omega_{CL,1} \) is calculated with consideration of the stiffness of the support using the analytical model given in Ref. \[12\]. The continuity constraints for the particle velocity on each side of the panel are given by

\[
\nu_L = \nu_s \quad \text{and} \quad \nu_R = (1 - \sigma) \nu_s + \sigma_{HR} \nu_{HR} + \sigma_{CL} \nu_{CL}.
\]

Substituting \( \nu_s \) by \( \nu_L \), Eqs. \( (3) \) to \( (9) \) can be transformed to the following system of equations for determining the values of \( p_R \) and \( v_R \):

\[
\begin{bmatrix}
0 & \frac{k_{s}}{\imath \omega} \sigma & -\frac{k_{s}}{\imath \omega} \sigma_{HR} - \frac{k_{s}}{\imath \omega} \sigma_{CL} & 1 \\
-1 & Z_s & 0 & 0 & \sigma \\
0 & 0 & Z_{HR} & 0 & -1 \\
0 & 0 & 0 & Z_{CL} & -1 \\
0 & 1 - \sigma & \sigma_{HR} & \sigma_{CL} & 0
\end{bmatrix}
\begin{bmatrix}
p_L \\
v_L \\
p_{HR} \\
v_{HR} \\
v_{CL} + \Delta \rho
\end{bmatrix}
= \begin{bmatrix}
0 & 0 & 0 & 0 & 0 \\
0 & \sigma - 1 & 0 & 0 & 0 \\
0 & 0 & -1 & 0 & 0 \\
0 & 0 & -1 & 0 & 0 \\
0 & 0 & 0 & 0 & 1
\end{bmatrix}
\begin{bmatrix}
p_R \\
v_R
\end{bmatrix}.
\]

Introducing the matrix \( C \), which extracts the values of \( p_L \) and \( v_L \) from the unknown vector \( x \) via

\[
\begin{bmatrix}
p_L \\
v_L
\end{bmatrix} = \begin{bmatrix}
1 & 0 & 0 & 0 & 0 \\
0 & 1 & 0 & 0 & 0
\end{bmatrix} x,
\]

leads to the transfer matrix \( T = CA^{-1}B \) of the infinite Cantilever-Helmholtz resonator panel. The transfer matrix of the double wall is calculated by multiplying the transfer matrices of each layer. Insulation layers are modelled as an equivalent fluid and walls are modelled as a limp wall using the mechanical impedance with a mass per unit area and neglecting the bending stiffness. The infinite transmission coefficient \( \tau \) at a specific plane wave incidence angle \( \theta_i \) can be calculated from the matrix elements according to \[13\]. To account for a finite sized panel, the transmission coefficient \( \tau \) is corrected according to \[14\] with \( \tau = \tau \sigma_R \cos \theta_i \), where \( \sigma_R \) is the radiation efficiency of the finite panel. Furthermore, the diffuse field transmission coefficient \( \tau_{f,d} \) is obtained by integrating over the incidence angles \( \theta_i \) \[13\]. Using the finite diffuse field transmission coefficient, the transmission loss is finally calculated with \( TL = -10 \log \tau_{f,d} \).

### 3. Experimental and Numerical Validation of the Transfer Matrix Model

A resonator test panel has been built to validate the analytically calculated transmission loss for a single resonator panel and a double wall build-up. The resonator panel consists of a resonator array with 252 unit cells. The sizes of the cantilevers, the cavities inside the resonators, the unit cells and the panel are listed in Table \[1\]. The panel is made from of a lightweight closed-cell polymethacrylimide foam (tradename Rohacell), with the density \( \rho_R = 32 \text{ kg/m}^3 \), Young’s modulus \( E_R = 36 \text{ MPa} \), and Poisson’s ratio \( \nu = 0.38 \). The resonator cavities as well as the 1.8 mm wide slits were milled in the foam and a
Table 1: Geometry of cantilever, cavity, unit cell and resonator panel; all dimensions are in mm.

<table>
<thead>
<tr>
<th></th>
<th>Cantilever</th>
<th>Cavity</th>
<th>Unit Cell</th>
<th>Panel</th>
</tr>
</thead>
<tbody>
<tr>
<td>Length</td>
<td>36</td>
<td>60</td>
<td>65</td>
<td>1200</td>
</tr>
<tr>
<td>Width</td>
<td>5</td>
<td>65</td>
<td>70</td>
<td>1000</td>
</tr>
<tr>
<td>Thickness</td>
<td>4.5</td>
<td>31</td>
<td>35.5</td>
<td>35.5</td>
</tr>
</tbody>
</table>

Table 2: Masses per unit area of different wall configurations; all dimensions are in kg/m².

<table>
<thead>
<tr>
<th></th>
<th>$m''$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Resonator panel w/o back plate</td>
<td>0.37</td>
</tr>
<tr>
<td>Resonator panel w/ back plate</td>
<td>4.74</td>
</tr>
<tr>
<td>Double wall w/ resonator panel</td>
<td>6</td>
</tr>
</tbody>
</table>

glass fiber reinforced plastic panel with a thickness of 2 mm, density $\rho_s = 2000$ kg/m³, Young’s modulus $E_s = 22$ GPa and Poisson’s ratio $\nu = 0.18$ was applied on the back side of the resonator panel. For the resonator panel with and without the back plate the masses per unit area are listed in Table 2. The double wall build-up and the single resonator panel, respectively, were mounted in a stiff frame into a test window between a reverberation room and a hemi-anechoic chamber. In the reverberation room a diffuse sound field was excited by a dodecahedron-loudspeaker with a total sound pressure level of around 90 dB, which was measured by a 1/2 inch diffuse field microphone on a rotating beam. In the receiving room a narrow-band sound intensity spectrum was measured with a hand-held intensity probe. The Finite Element Method was used for the numerical simulations. The diffuse field was modelled with the plane wave model according to Ref. [15]. The aircraft walls, the resonator panel and the insulation layer were modelled as shells with simply supported boundary conditions, as a solid with roller boundary conditions and as an equivalent fluid with sonically hard walls, respectively. The air in the slit was modelled as an equivalent fluid using the low reduced frequency model to account for thermal and viscous losses arising in the boundary layers. Three perfectly matched layers were applied on the receiving side of the panel.

### 3.1 Resonator Panel

The resonator panel mounted in the transmission window inside a stiff frame is shown in Fig. 3(a). The transmission loss improvement of the panel, shown in Fig. 3(b), was determined by comparing the transmission loss of the resonator panel with open and closed resonator necks. The figure shows reasonable agreement between the numerically and experimentally determined difference in transmission loss with...
two clear peaks at around 500 and 650 Hz. In the analytically calculated transmission loss, the resonance frequencies are further apart than in the experiment (480 Hz and 690 Hz). This inaccuracy can possibly be attributed to the simplifications of the transfer matrix model, i.e. taking just the first resonance frequency of the cantilever into account and assuming that the pressure distribution over the cantilever is uniform.

To characterize the sound transmission loss improvement the 5 dB-bandwidth \( \text{BW}_{5\text{dB}} = \frac{(f_u - f_l)}{\sqrt{f_u f_l}} \) is defined, where \( f_u \) is the upper and \( f_l \) is the lower frequency of the interval, where the difference in the sound transmission loss is higher than 5 dB. Figure 3(b) also shows the difference in transmission loss, determined numerically and analytically, for the resonator panel with fixed cantilevers, such that only the Helmholtz resonators are active. Compared to this reference configuration, the 5 dB-bandwidth of the Cantilever-Helmholtz resonator panel is increased by approximately 6%. An optimization potential will be analyzed later on.

3.2 Double wall with Cantilever-Helmholtz resonator panel

The double wall test set up, called laboratory model hereafter, is shown in Fig. 4(a). A homogeneous glass fiber reinforced plastic plate was installed as the second wall with a melamine foam insulation layer (air flow resistance of 12000 Pa/s/m\(^2\), porosity of 0.99) on top of it. Figure 4(b) shows the transmission loss improvement, determined experimentally, numerically and analytically, of a double wall with the Cantilever-Helmholtz resonator panel compared to a mass equivalent double wall without the resonator panel. The mass equivalent double wall was achieved by replacing the resonator panel with a 6 mm thick medium density fiberboard. Two peaks around 500 Hz and 650 Hz are visible, which correspond to the two resonances observed in the measurement results for the Cantilever-Helmholtz resonator panel alone (Fig. 3(b)). The experimentally determined insertion loss has a large 5 dB-bandwidth in the low frequency region for frequencies lower than the first resonance frequency, but the peaks at the resonance frequencies are lower than the peaks of the numerically determined insertion loss. A likely explanation is that the numerical model treats the resonator panel as a solid with ideally smooth surfaces. However, the resonator panel in the experiment was built out of foam, having rough surfaces, which can cause higher viscous losses resulting in lower resonance frequencies and lower transmission losses. Figure 4(b) shows again, that the analytically calculated resonance frequencies are lower for the first and higher for the second resonance peak. However, the experimental and numerical results show an overall good agreement with the analytical results and validate therefore the transfer matrix model.

![Diagram](image1.png)

![Diagram](image2.png)

Figure 4: (a) Schematic representation of the double wall setup. (b) Diffuse field transmission loss improvement for the double wall setup. Comparison of experimental, analytical and numerical results.
4. **Feasibility Study for a Real Aircraft Double Wall with the Resonator Panel**

With the validated analytical transfer matrix model, the potential of a Cantilever-Helmholtz resonator panel integrated into a realistic aircraft cabin wall is estimated. The aircraft double wall set up with realistic dimensions and properties is shown in Fig. 5(a). The transmission loss and the difference in transmissions loss compared to a mass equivalent double wall (named as Ref. DW) were calculated for three different configurations of the resonator panel. The results are plotted in Fig. 5(b). The first configuration is the one of the laboratory model of Table 1. The size of the cavity volume and the length of the cantilever are changed for the second and third configuration, all other dimensions are unchanged. The second configuration has a two times larger cavity volume and a 42 mm long cantilever, the third configuration has a 6 times larger cavity volume and a 54 mm long cantilever. The figure shows clearly that inserting the Cantilever-Helmholtz resonator panel into the aircraft double wall leads to highly increased transmission losses over a large frequency band. The 5 dB-bandwidth is 37%, 86% and 59% for the first, the second and third configuration, respectively. The increase in transmission loss is attributed to a lower double wall resonance frequency with inserted resonator panel even though the double wall spacing and mass per unit area are constant. This phenomenon is subject to ongoing research.

![Figure 5](image_url)  
**Figure 5:** (a) Schematic representation of the aircraft double wall. (b) Diffuse field transmission loss results for three different configurations. — Transmission loss, . . . Difference in transmission loss, - - 5 dB

5. **Conclusion**

In the present contribution, a concept for a new aircraft cabin lining with increased transmission loss in the low frequency region has been investigated. The enhanced sound insulation is realized by covering the rear side of the cabin lining with Helmholtz resonators with integrated vibrating cantilevers. The sound transmission loss of a proof-of-concept design of the panel was calculated analytically with a transfer matrix model and verified and validated by simulations and experiments, respectively. Compared to a panel with deactivated resonators, the experimentally determined transmission loss is improved by 5 to 12 dB at the two resonance frequencies of the coupled resonators. The transmission loss of the double wall is improved by nearly 10 dB (experimental results), compared to a mass equivalent double wall without resonators. Thus, implementing the Cantilever-Helmholtz resonator panel into the aircraft wall increases the sound transmission loss in the low frequency region, while being very lightweight with an additional mass per unit area of 0.37 kg/m², which is 5% of the overall aircraft double wall mass. An
analytical feasibility study of an integrated Cantilever-Helmholtz resonator panel integrated in a realistic aircraft cabin side wall with typical dimensions shows a high improvement potential of the transmission loss in a broad frequency band. An experimental study of the more realistic aircraft cabin side wall design is part of future work.

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