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WAVE-RESOLVING AIRCRAFT FUSELAGE MODEL FOR CABIN NOISE PREDICTIONS UNDER DISTRIBUTED FLUID LOADINGS

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Abstract. *For the development of future passenger aircraft comprising novel technologies and designs, a reduction of cabin noise is one major aim. An analysis of the aircraft cabin sound pressure levels can be achieved by mechanical models. In opposite to energy-based methods, wave-resolving models allow a detailed investigation of distributed fluid loadings and the wave propagation within the fuselage. A turbulent boundary layer loading on the outer skin of the aircraft is investigated exemplary. As the loading bring along a specific footprint on the aircraft skin and the model resolves the bending waves, the load input can be applied directly. Besides the outer skin, the aircraft fuselage model considers the stiffeners, the insulation, the inner cabin lining and the cabin fluid of the entire cabin. Each part is validated separately and included in the aircraft fuselage model. The structural and fluid domains are fully coupled and discretised with finite elements using 2D shells and 3D continua resulting in millions of degrees of freedom. The simulation shall not model a specific aircraft, but rather demonstrate on the example of a generic research aircraft the crucial modelling and solving aspects of typical aircraft parts. Within this contribution, a section of the fuselage is presented having a specific focus on the cabin fluid modelling. The damping characteristics of the seats are considered by impedance boundary conditions within the 3D fluid domain and compared to a homogenising approach using a frequency-dependent damping loss factor. The necessary input parameters are measured in a reverberation chamber having available an aircraft seat bench. The numerical comparison of the modelling approaches in the fuselage model shows that local differences can be expected but in general the SPL is decreasing similarly with increasing frequency.*

1 INTRODUCTION

Facing an increasing demand for mobility, the number of flights is expected to raise significantly in future. Many passengers are exposed to cabin noise during flights. Under high noise levels, health and comfort issues may occur. A simulation of cabin sound pressure levels in early design stages can help to avoid a high exposure by integrating sound reduction measures if necessary.

Based on preliminary aircraft design data as provided in the Collaborative Research Center 880 [1], a wave-resolving fuselage model can be derived. The modelling of structure-borne sound waves in the primary and secondary structure and the fluid waves in the insulation and the cabin itself allows a detailed analysis of the sound transfer paths. As waves in all domains are considered, sound reduction measures like, e.g., acoustic black holes and their placement can be studied directly without any assumptions on their efficiency. Discretising and solving the model delivers the sound pressure level (SPL) distribution in the cabin under certain loads. Fluid loads like jet noise or the turbulent boundary layer (TBL) bring along a characteristic footprint on the outer skin. Only a correct consideration of phase differences and amplitudes of such loads ensure a precise prediction of cabin noise. Especially for novel aircraft concepts, resulting sound sources are not known and may be assessable through complex simulations. In [2], e.g., the cabin noise due to jet noise is investigated for a comparison of an UHBR engine with a conventional engine. The model is extended by a TBL load in [3] suggesting that this load might be one of the dominating loads in aircraft with UHBR engines.

However, appropriate modelling assumptions for all fuselage parts are required in order to receive reasonable responses in the cabin. The cabin fluid itself including seats and passengers is expected to play a major role in the entire model. Hence, a comparison of models of different complexity is aimed for in this contribution. Simple approaches considering a homogenised cabin fluid offer a simple and fast modelling procedure. A detailed consideration of seats is more complex but may offer crucial quality differences in the model. A boundary element solution of an aircraft passenger cabin is applied in [4] to the cabin fluid of an aircraft segment. The numerical model is compared to experiments – differences are mainly attributed to the neglected influence of the surrounding structure. In the model applied here, all surrounding structures are considered. Qualitatively, in [4], the numerical sound pressure distribution near the seats matched experimental results under a consideration of impedance boundary conditions for the seat surfaces. Hence, this shall be the local approach applied here within the fuselage model described in the next section.

2 FUSELAGE MODEL

The preliminary design data by the design tool PrADO [1, 5] within the CRC 880 provides a dimensioning of the entire primary structure of a virtual research aircraft for 100 passengers. For the purpose of cabin noise simulations, these information on materials, thickness distribution and cross-sections are automatically transferred into a finite element shell model of the outer skin, the stiffeners and the floor section. In Fig. 1, the FE model of the fuselage is shown. Basically, the model underlies the assumptions of linearity and symmetry. For simplicity, a small model with three seat-rows is applied in this study. Curved sandwich plates fixed to the stiffeners are considered as typical inner sidewall and ceiling panels with a constant thickness of 0.01 m. An aramid honeycomb core is homogenised to a 3D continuum meshed by 27-node quadratic hexahedrons while a glass-fibre-reinforced plastic layer is modelled at each side by shell elements perfectly fixed to the honeycomb core.

The double wall gap between outer skin and inner cabin linings is filled by glass wool. The fibres material is considered as homogenised 3D equivalent fluid volume meshed by hexahedrons as well. The passenger cabin itself is basically filled by air with typical density, temperature and humidity values of the cabin air in cruise. Different modelling approaches for the cabin fluid volume including damping, reflection and refraction of the aircraft seats shall be in focus of this contribution. As visible in Fig. 1, seat domains are already separated in groups in order to apply different modelling approaches keeping the finite element mesh unchanged.

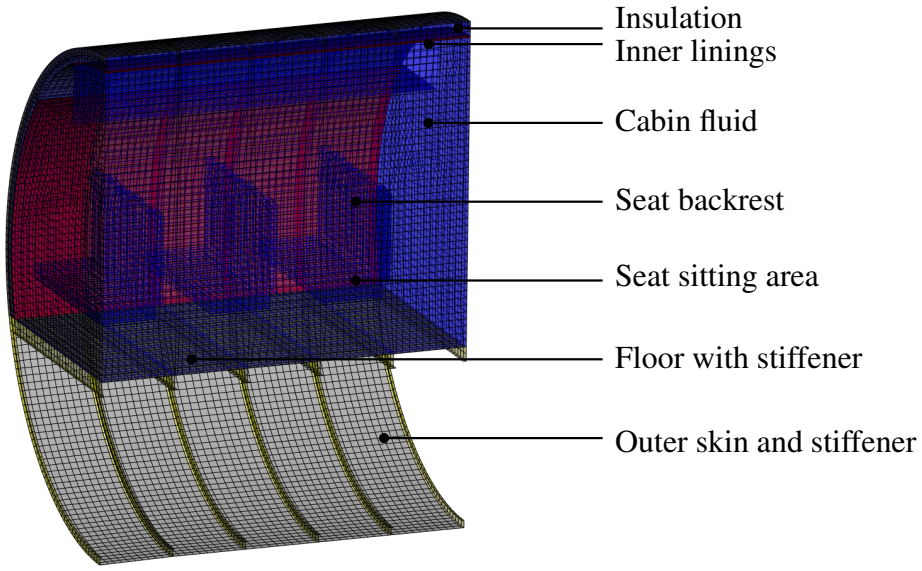


Figure 1: Meshed fuselage section model including structural and fluid domains; Symmetry boundary conditions are applied.

Fluid and structural domains are strongly coupled resulting in a total computational cost of 7.3 min per frequency step[1]. As exemplary load for a comparison of the cabin models, an artificially generated TBL load is considered as described in detail in [3].

3 MODELLING ASPECTS IN THE PASSENGER CABIN

In previous studies of the aircraft model from Sec. 2 which can be found in [2, 6, 3], a homogeneous constantly damped Helmholtz fluid domain is considered according to Eq. 1. A constant complex speed of sound $\underline{c}$ is applied for the entire fluid in order to consider the damping properties of seats and passengers. $\underline{p}$ is the sound pressure, ω is the angular frequency and $\mathbf{x}$ is the location in the aircraft cabin.

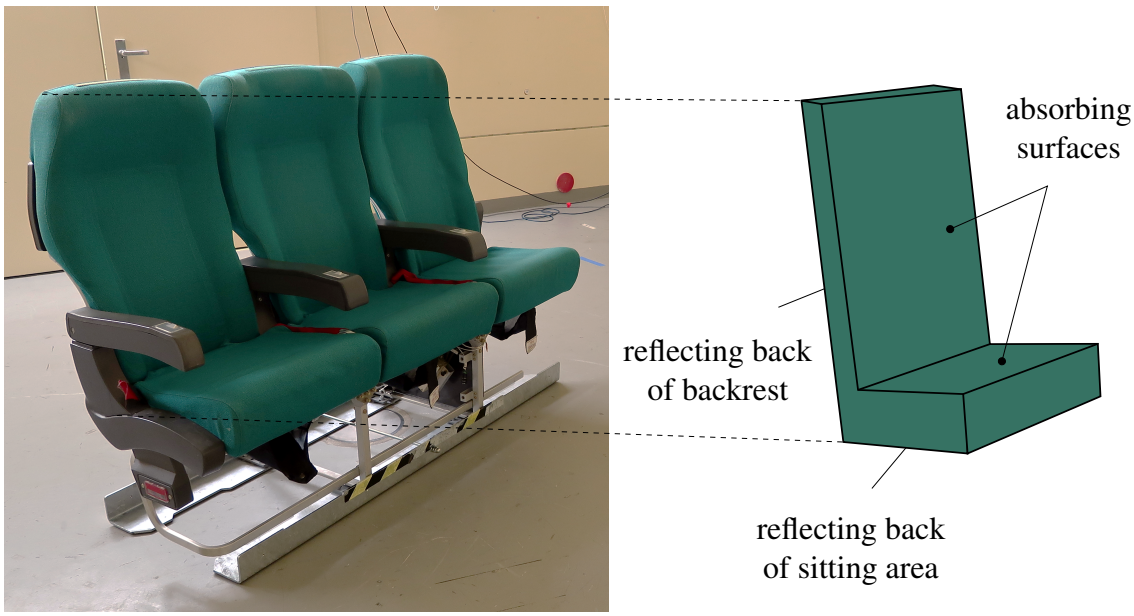
$$\nabla^2 \underline{p}(\mathbf{x}, \omega) + \frac{\omega^2}{\underline{c}^2} \underline{p}(\mathbf{x}, \omega) = 0 \tag{1}$$

$\underline{c}$ is calculated according to Eq. (2) by use of the fluid loss factor η_f and the imaginary unit i [7].

$$\underline{c}^2 = c^2 \left(1 + i\eta_f\right) \tag{2}$$

[1]Using the direct MUMPS solver and the institute's in-house code elPaSo at 20 cores on the phoenix cluster of TU Braunschweig

In order to assess the consideration of $\underline{c} = const$, the model complexity is increased in this contribution. As the surrounding structures of the cabin (sidewall panels, floor) are considered with material damping properties in the model, the seats and the passengers are mainly expected to influence the cabin fluid domain. For the purpose of experimental studies on the damping characteristics of seats, three aircraft seats are available which are shown in Fig. 2 (a). As sketched in Fig. 2 (b), each seat is transformed to simplified cornered domains and considered as separate block in the FE model. Two approaches are investigated: A frequency-dependent loss factor and an introduction of surface impedances for the seat rows.



a) b)

Figure 2: (a) Aircraft seats in reverberation chamber and (b) sketch of simplified seat assembly.

A first extension of the previous model is a consideration of a **frequency-dependent loss factor** $\eta_f(f)$ leading to a frequency-dependent speed of sound $\underline{c}(f)$ (Eq. (2)). The frequency-dependent parameter is yielded by measurements. The reverberation time $T_r(f)$ is measured in a reverberation chamber resulting in the equivalent absorption area $A_t(f)$ according to DIN ISO 354 [8] for the three seats shown in Fig. 2. The equivalent absorption area of three seats is linearly scaled to 100 seats which neglects any influences between seat rows. The effect of passengers has been studied as well, but the change of $A_t(f)$ is not significant. For the equivalent absorption area with 100 seats $A_{t,100}(f)$, the reverberation time $T_{r,\text{cabin}}$ is calculated according to Eq. 3 [8] inserting the properties of the passenger cabin.

$$T_{r,\text{cabin}}(f) = \frac{55.3 V_{\text{cabin}}}{\left(A_{t,100}(f) + 4 V_c m_{\text{cabin}}(f)\right) c} \tag{3}$$

$V_{\text{cabin}} = 118\,\text{m}^2$ is the fluid volume in the cabin and $m_{\text{cabin}}(f)$ is the air absorption coefficient. Finally, $\eta_{\text{cabin}}(f)$ is related to $T_{r,\text{cabin}}(f)$ as given in Eq. (4) [9] with frequency f.

$$\eta_{\text{cabin}}(f) = \frac{2.2}{fT_{r,\text{cabin}}(f)} \tag{4}$$

In Fig. 3, the resulting loss factor is shown. In the low frequency range ($< 250\,\text{Hz}$), a homogenised loss factor of $3\,\%$ can be expected by the seat cushions. With increasing frequency, the loss factor decreases to $1\,\%$ damping at $1000\,\text{Hz}$. As most passenger cabins are similarly designed, a transferability of these results is expected to a certain extent.

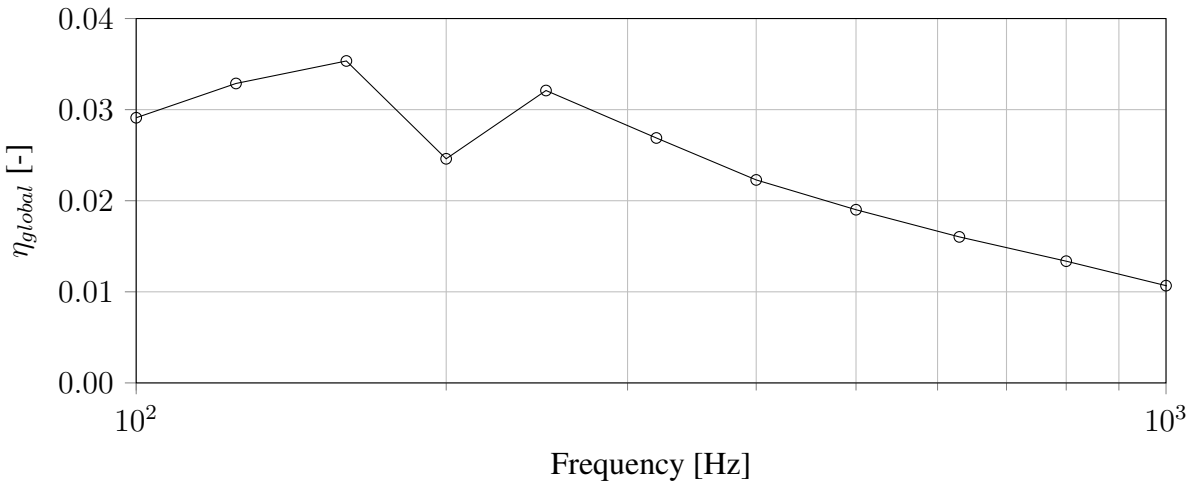


Figure 3: Homogenised loss factor in the cabin with volume $V_c = 118\,\text{m}^2$ for 100 passengers based on measurement of the reverberation time of three seats.

The second approach is similar to the above described approach by [4] – an acoustic **surface impedance** $\underline{Z}_a$ according to Eq. 5 [10] is placed in the fluid domain for the seat's backrest and the sitting area. The seat positions are given by the preliminary design data.

$$\underline{Z}_a(f) = \frac{\underline{p}(f)}{\underline{v}_n(f)} \tag{5}$$

The impedances are experimentally determined in the reverberation chamber as well. By placing an intensity probe near the surface, $\underline{p}$ and $\underline{v}$ are calculated on the basis of the two pressure signals and the cross-correlation of these. This way, the impedances of the surfaces (backrest and sitting area) are approximately given. The results for the $\underline{Z}_a$ are depicted in Fig. 4. In the low frequency range ($< 250\,\text{Hz}$), the values show high variations which is consistent with findings in [11]. In these frequency ranges, rather a smoother curve is expected which can be shown by comparing with porous material models or impedance tube measurements. With increasing frequency, the real part of the impedance converges to the impedance of air (blue curve) which is physically reasonable as the absorption performance of the seat cushion is expected to increase with reduced wave lengths.

As shown in Fig. 5 (b), the meshed cabin fluid volume assigned to the seat row is substituted by the surface impedance. This means, the finite elements in these regions are removed introducing a reflecting surface at the back of each seat cushion besides the impedance at the front of each seat cushion (backrest and sitting area). For the homogenised approach (Fig. 5 (a)), the mesh is the same including the filled gap. Here, no reflecting boundaries are introduced.

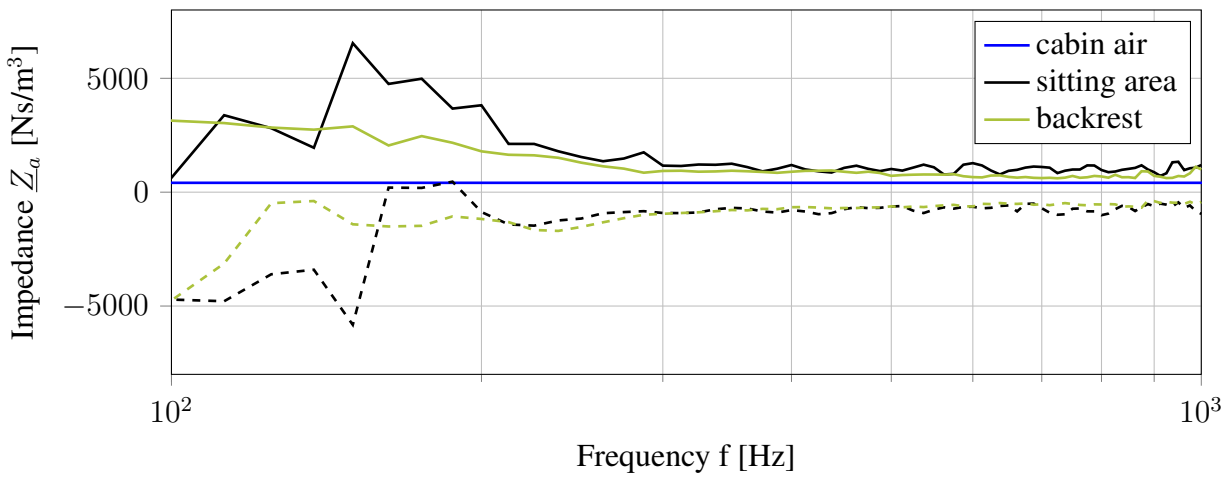
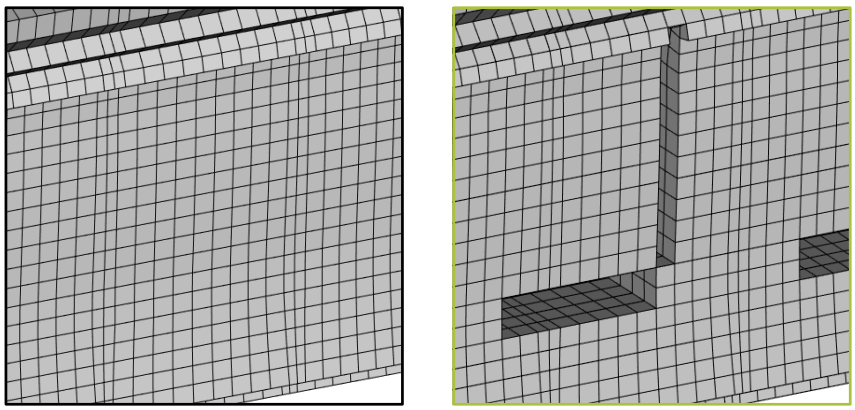


Figure 4: Acoustic impedances measured near the seat surfaces; The dotted lines are the imaginary parts.



a) b)

Figure 5: (a) Continuous mesh for homogenised approach; (b) Removed elements in seat region.

By removing the elements in the cabin fluid domain for the seat model, the computational costs are reduced by 17 % in comparison to the homogenised approach. Of course, the modelling effort is higher as the seat volumes must be considered in the model. In the vicinity of the sidewall panels, the meshing procedure is slightly more complex.

4 RESULTS

In Fig. 6, the SPL is plotted for the two described modelling approaches. The position corresponds to the seat in the central row near the sidewall panel. The height is chosen in accordance with a typical passenger ear height. Basically, the two SPL curves are similarly decreasing over frequency and show comparable basic sound levels. The overall damping characteristics seem similar as the dynamics of both curves is comparable as well. For a fast comparison of aircraft configurations (e.g. in third octave bands), the global loss factor may be appropriate in order to save modelling effort and avoid further mistakes during the introduction of impedances and reflecting boundaries. Furthermore, the impedance approach brings along several assumptions

like one impedance value at normal sound incidence which requires a more critical interpretation of results.

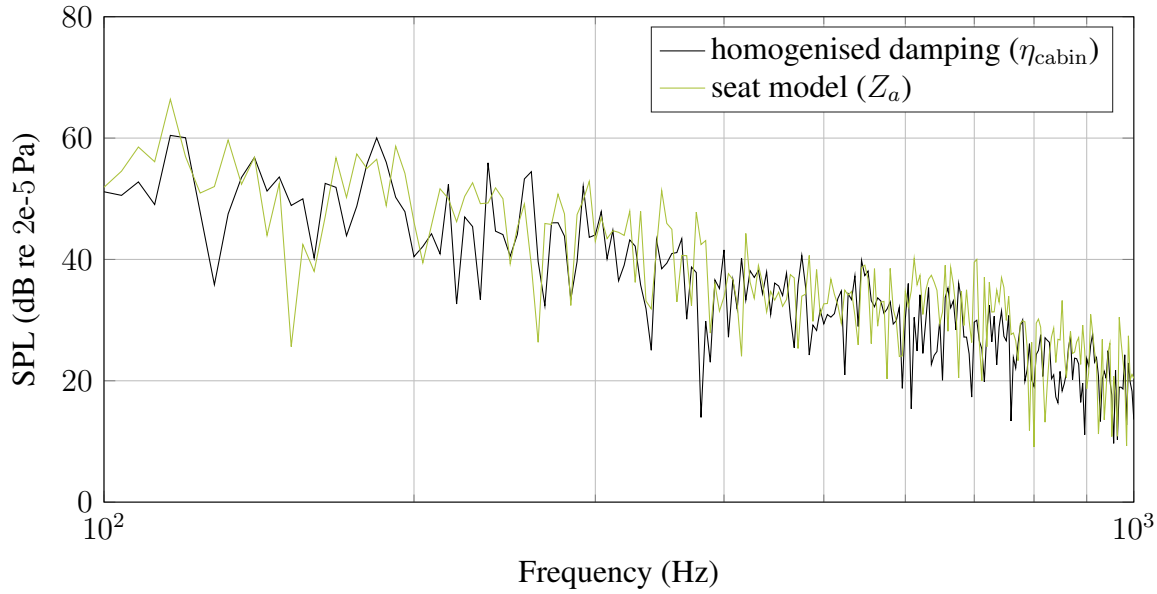


Figure 6: Sound pressure level at central seat position near the sidewall panel under TBL load with two different modelling approaches for the cabin.

Having a closer look, some resonances occur in both approaches (e.g. around 300 Hz and 530 Hz) but in most frequency regions there are local differences up to 10 dB and slightly more resonances in the response of the seat modelling approach. In the case of the local seat model, standing waves between seat rows and acoustic effects near seats can occur besides global resonances of the entire cabin fluid. In Fig. 7, pressure distributions in a cabin slice are shown for several frequencies. For the global pressure distribution at 168 Hz, for instance, only a slight distortion of the global shape is observed. For shapes in longitudinal direction as, e.g. seen at 184 Hz, the backrests of the seats seem to have a major influence. Furthermore, as it can be observed at 232 Hz, local effects (here below the seat sitting area) occur.

Concluding the results, detailed seat models seem necessary for applications or analyses which require a knowledge of local effects. For a rough and global comparison of the sound pressure levels, a global loss factor is appropriate. For active sound reduction measures, like noise cancelling in aircraft cabins, these local information are crucial and a seat model must be considered. Also, for a detailed investigations of tonal components like propeller noise, a seat model should be preferred in order to cover local effects at specific frequencies which might interfere with the tonal noise transferred into the cabin.

5 CONCLUSION

In this contribution, the SPL in an aircraft cabin is simulated using a finite element model of a typical fuselage section under an artificially generated TBL load. A comparison of two different modelling approaches is conducted – a homogenised damping loss factor for the entire cabin fluid is compared to a local modelling of seats introducing local impedances for the seat cushions. Both input parameters are frequency-dependent and received by measurements of a real seat bench. The resulting SPL at one passenger ear delivers comparable results regarding

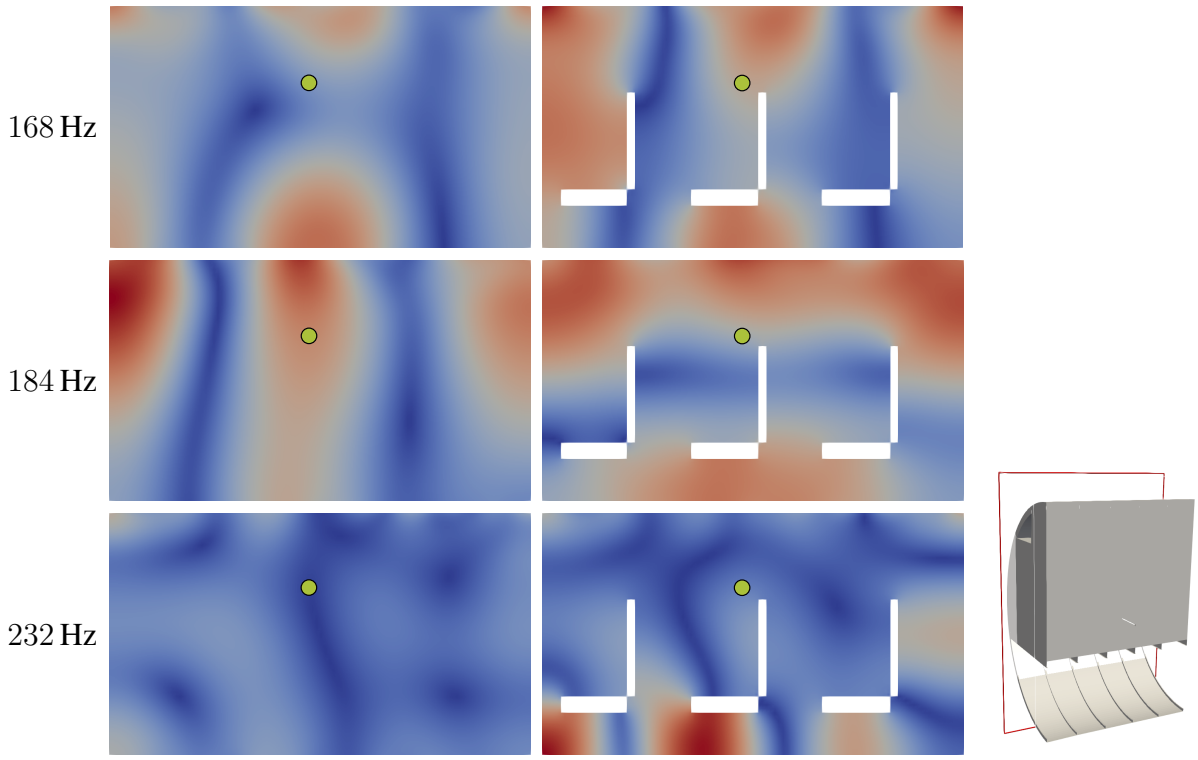


Figure 7: Qualitative pressure distribution in slices of the passenger cabin fluid; Each row refers to one example frequency with identical contour map ranges per row; The evaluation position for Fig. 6 is indicated by the green dots.

basic sound levels and damping performances over frequency. A detailed comparison of the pressure distribution in the cabin shows local effects at and between seat rows which lead to differences up to 10 dB in the model response.

For studies in future, a consideration of few tria elements in the volume mesh and non-coincident meshes in general may reduce the modelling effort. In addition, a consideration of porous material domains for the seat cushions might be a third approach. Furthermore, an experimental setup without surrounding aircraft parts seems reasonable in order to validate the local approach using impedances and reflecting backs of the seats. Finally, a more precise measurement of the seat impedances, especially for lower frequencies, is aimed for in future.

6 ACKNOWLEDGEMENT

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