Integrated Vibro-acoustic Calculation Procedure of Composite Structures using Coupled Finite and Boundary Element Method

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Abstract
Lightweight composite structures for high-technology applications have to fulfill high demands on low constructive weight combined with an adequate stiffness. In general, the low structural weight leads to high vibration amplitudes due to low forces of inertia and causes an undesired sound radiation. This effect can be compensated by exploiting the high vibro-acoustic potential of fibre-reinforced composites. For this purpose, an integrated calculation algorithm for the transmission loss, considering the eigenfrequencies (modal analysis) and the estimation of modal damping values, is presented. Therefore, an ANSYS loop is combined with an especially developed VA One batch file. The developed model enables the calculation engineer to study the influence e.g. of the fibre orientation on the radiated sound power of a fibre-reinforced structure with a complex geometry.

1 Introduction

Lightweight structures for high technology applications often have to balance high demands of contrary targets. So the low weight leads to higher vibration amplitudes and mostly to higher sound radiation levels, as well. A possibility to solve such conflicts is given by the adjustable behaviour of fibre-reinforced composites. For basic structures, like beams or plates, semi-analytic solutions to calculate the resulting sound radiation or transmission loss exist. Currently, the high potential of complex fibre-reinforced structures, though, can't be predesigned in an efficient way using commercial tools. So in this paper a new calculation algorithm considering the modal damping of complex fibre-reinforced structures as well as an optimization loop for these structures is shown. The main part of the algorithm generated uses the ANSYS Programmable Design Language (APDL). In a first step, the modal damping of the desired structure is calculated using an adapted model of the complex mode shapes and the Modal Assurance Criterion (MAC) [1, 2, 5]. Furthermore, the parametric geometry is meshed for Finite Element (FE) and vibro-acoustic calculations. Therefore, the calculated modal basis, using ANSYS, is transferred via a configuration file to the vibro-acoustic software VA One. Subsequently, ANSYS calls VA One and the resulting transmission loss of the structure is calculated. After finishing the calculation, VA One exports the results in a comma separated text-file for further investigations. Afterwards, a "completed" signal is send to ANSYS and a further calculation loop could be started. By defining optimization parameters within ANSYS, a powerful automatic working optimization tool for complex fibre-reinforced composite structures is obtained.

2 Fundamentals

To control the vibro-acoustic behaviour of complex structures, several possibilities are available. In this paper, a model to influence the transmission loss of complex fibre-reinforced structures is shown. At the beginning, a short overview of relevant acoustical effects and design parameters is given.

The transmission loss TL_{mass} of a single-leaf wall can be estimated by the mass law, with *m'* as mass per unit area, *f* as frequency, ρ as density of air and *c* as speed of sound in air [10]. That empirical formulation indicates that the transmission loss increases by 6 dB for every doubling of the mass per unit area.

$$TL_{mass} = 20\log\left(\frac{\pi m' f}{\rho c}\right) \tag{1}$$

Furthermore, (1) shows that a doubling of the frequency also leads to an increase of the transmission loss by 6 dB. Because of the usual use cases with broadband excitation, the frequency dependence is of minor interest, while the interaction of mass and transmission loss is essential.

In addition to this mass law, several other components influence the resulting transmission loss of a structure. Having a look at the transmission loss spectra of a simple wall (Figure 1), three other significant regions could be seen besides the mass controlled area: a stiffness controlled area in the low frequency range, a resonance controlled area between the stiffness and the mass controlled area and a coincidence controlled region within the mass controlled area.

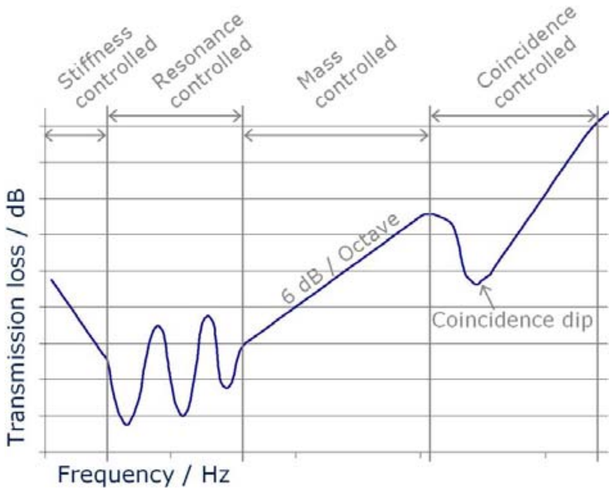

Figure 1: Main parts of a transmisison loss spectrum for a single wall

The decreasing characteristic in the low frequency range is caused be stiffness effects. This happens in an equivalent way to the damping behaviour of a spring-mass system with a continuous external excitation. By the stiffness law (2), an amendment of the mass law, the decrease is described by a recession of 6 dB per doubling of the mass unit area or doubling of the frequency [10].

$$TL_{stiffness} = 20\log\left(\frac{\pi m' f}{\rho c}\cdot\left(1-\frac{\Omega^2}{f^2}\right)\right) \tag{2}$$

The stiffness law allows an initial estimation of the transmission loss for the whole frequency range. The change between the stiffness and the mass controlled areas is defined by the first relevant mode shape Ω of the structure.

For more detailed information about the resonance controlled area, the mode shapes of the investigated structure have to be characterized. The frequency range of the resonance controlled area is defined by the eigenfrequencies themselves. The lowest mode shapes showing a “global” vibration pattern, have a significant influence on the transmission loss, because the resulting vibrations are able to emit acoustical waves (Figure 2). Higher mode shapes with a shorter bending wave length aren't able to emit such relevant waves and the mass law prevails the resonances in that higher frequency range and dominates the resulting transmission loss.

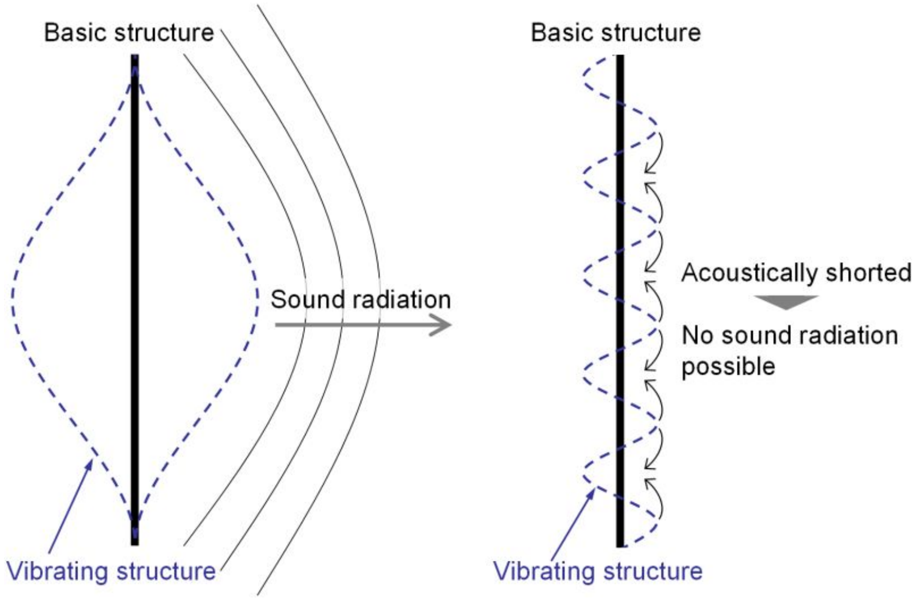

Figure 2: Sound radiation of lower and higher mode shapes (shematic)

The last significant region of the transmission loss spectrum is coincidence controlled. Coincidence occurs, when the projected wavelength of the incident sound waves equals the wave length of the bending waves of the structure (Figure 3).

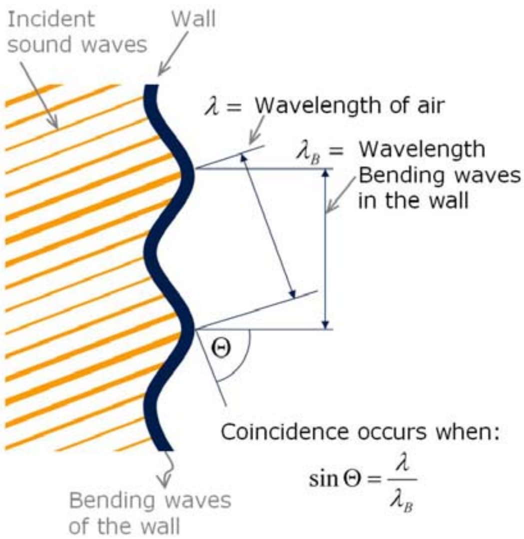

Figure 3: Fundamental parameters of the coincidence effect

The lowest frequency, where this effect appears is called the critical frequency. It depends on the speed of sound *c* in the environmental medium, the bending stiffness *B* and the mass per unit area *m'* of the structure.

$$f_{Coincidence} = \frac{c^2}{2\pi\sqrt{\frac{B}{m'}}} \tag{3}$$

The bending stiffness of a symmetric sandwich structure could be calculated by (4), where the index *FS* means face sheet parameter and index *C* core parameter. The parameter *h* describes the thickness of the layer.

$$B_{SW} = \frac{E_{FS} h_C^2 h_{FS}}{2(1-\nu^2)} \left(1 + \frac{h_{FS}}{h_C}\right)^2 \tag{4}$$

In summary, there are several possibilities to influence the transmission loss spectrum of a single-leaf structure. With the currently used pre-designing algorithms, it is not possible to consider all of these effects. So within the scope of this work, an integrated simulation process considering a damping and radiation adapted model for complex fibre-reinforced structures will be shown

3 Integrated transmission loss model

To optimize the transmission loss of complex fibre-reinforced structures, a closed loop calculation of the structural behaviour in all frequency ranges is required. Furthermore, the panel stiffness and damping, which depend on fibre angle, also have to be considered for all frequencies. For this complex calculation algorithm, a combination of a commercial finite element (FE) tool (ANSYS) and a vibro-acoustic tool (VA One) is chosen. Both tools are required, because ANSYS doesn't provide the possibility to simulate infinite fluids, especially if they have to be divided into two semi-infinite fluids by an infinite rigid wall. VA One on the other hand, doesn't include a built-in optimization tool and relies on existing FE solvers for computing the modal basis of complex materials. By combining these two tools, the full vibro-acoustic potential of VA One could be used within an ANSYS-driven optimization loop. Figure 4 shows the developed algorithm using both of these tools.

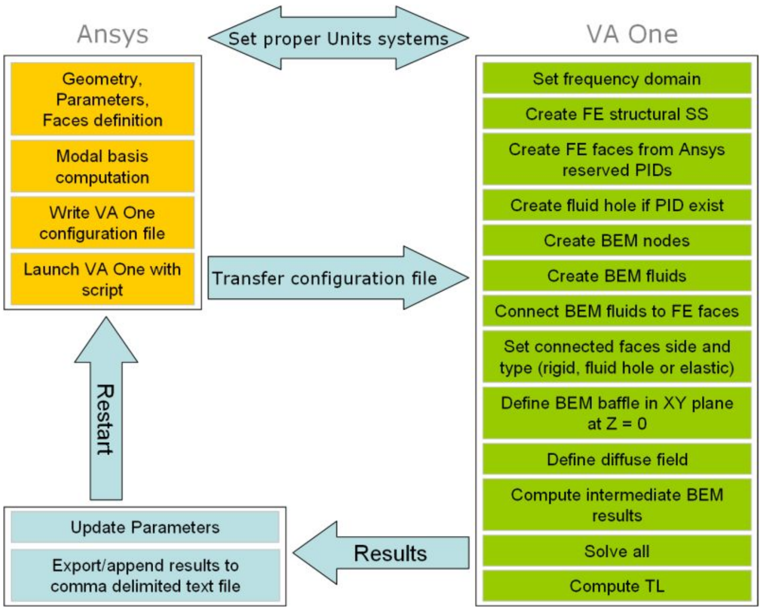

Figure 4: Flowchart of the integrated calculation algorithm using ANSYS and VA One

The geometry has to be created by an external CAD tool and imported into ANSYS or could be created in ANSYS itself. Afterwards, the system parameters, like frequency, element edge length and faces, have to be defined. Two FE-meshes are created in ANSYS, one for the structure itself and another, coarser one for the boundary element (BE) calculation in VA One. After calculation of the modal basis in ANSYS, the results and the meshes are exported into a VA One configuration file. By launching a predefined batch-file, VA One imports all data from the configuration file and creates the parameter-defined BE-model. After solving the BE-model, VA One exports the calculated transmission loss results into a comma separated text file and sends a "completed" signal to ANSYS. Finally, ANSYS updates the desired model parameters and starts the calculation loop again.

3.1 Geometry and parameter definition

The developed algorithm is programmed using the ANSYS Programmable Design Language (APDL). Herewith, it is possible to import a predefined geometry from a CAD program or create a parameter-based structure. To demonstrate the performance of the developed calculation algorithm, a fibre-reinforced sandwich tray was chosen. The geometric parameters are shown in Figure 5.

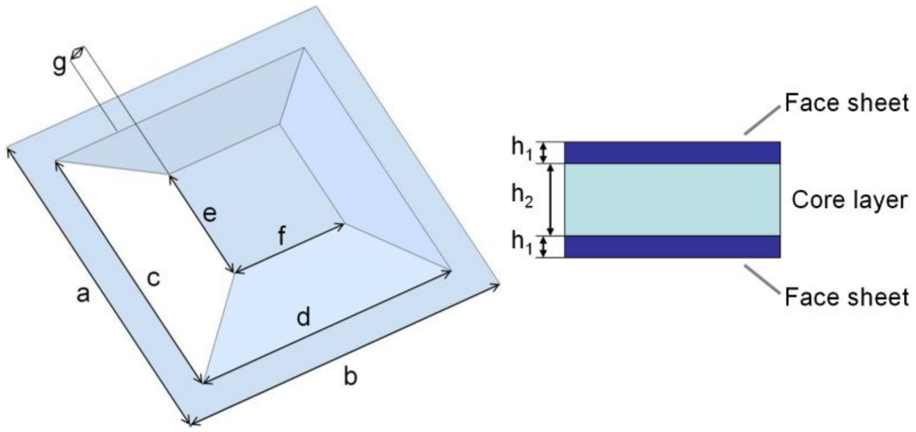

Figure 5: Dimensions of the selected test structure

The usage of a complex fibre-reinforced sandwich construction allows variations of additional parameters. So a lay-up of unidirectionally reinforced fiberglass face sheets and a structural foam core layer was chosen. Because of the anisotropic material characteristics (Figure 6), several mode shapes of the structure could be influenced without a significant changing of the overall stiffness by varying the face sheet fibre angle [4, 7, 9].

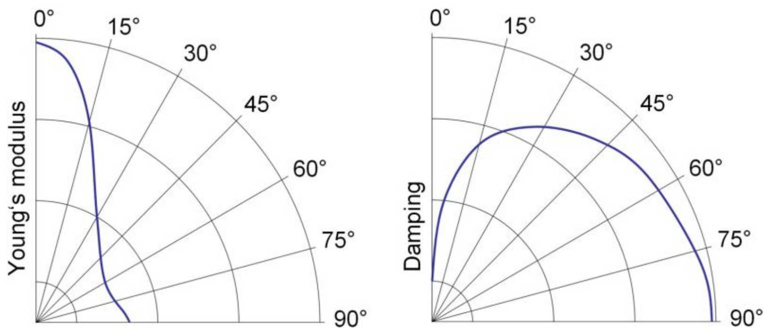

Figure 6: Young's modulus and damping value for an unididectional fibre-reinforced composite

Furthermore, variations of the thickness ratio between face sheets and core layer allows variations of the dynamic behaviour without changing the overall mass. These parameters, namely fibre angles of face sheets and thickness ratio, offer a high potential for lightweight structures with an adjusted vibro-acoustic behaviour. In the model, the thickness ratio is defined by (5)

$$r = \frac{h_1}{h_2} \tag{5}$$

To assure a constant overall mass m, the thicknesses h_1 and h_2 are calculated from (6)

$$h_1 = \frac{r}{2r\rho_1 + \rho_2} \cdot \frac{m}{A} \qquad h_2 = \frac{1}{2r\rho_1 + \rho_2} \cdot \frac{m}{A} \tag{6}$$

Figure 7 shows the thickness values for different thickness ratios from 0.01 to 100.

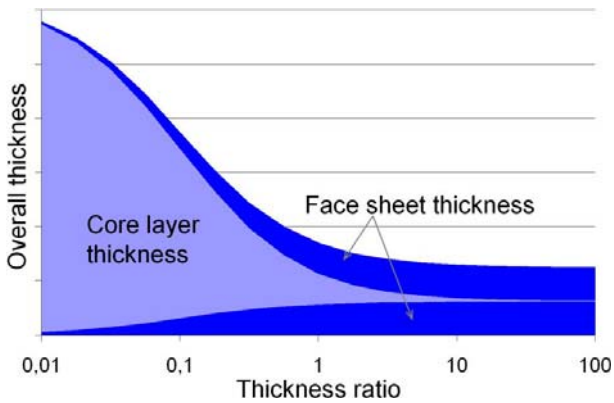

Figure 7: Overall thickness for different thickness ratios between face sheets and core layer

The huge overall thickness for a small thickness ratio is caused by the different material-densities of the face sheets (high density) and the core layer (low density). To ensure a constant overall mass, the overall thickness has to be reduced for thicker face sheets.

3.2 Calculation of the modal basis

Especially the high material damping of composites leads to the necessity of including the modal damping into the vibro-acoustic calculation process.

Commercially available FE-software tools like ANSYS just consider dynamic damping values via proportionality factors α and β [1, 2, 6]. That approach allows good correlations for isotropic calculations but no practical results for anisotropic materials like fibre-reinforced composites, with directional material damping.

A possibility to calculate the modal damping values of complex fibre-reinforced composite structures is given by the method of complex eigen-values [5]. Usually, this method leads to fully occupied complex-valued matrices, which can not be solved by commercial FE tools. Thus, the method was adapted to be used with these-solvers [1]. Figure 8 shows the developed approach.

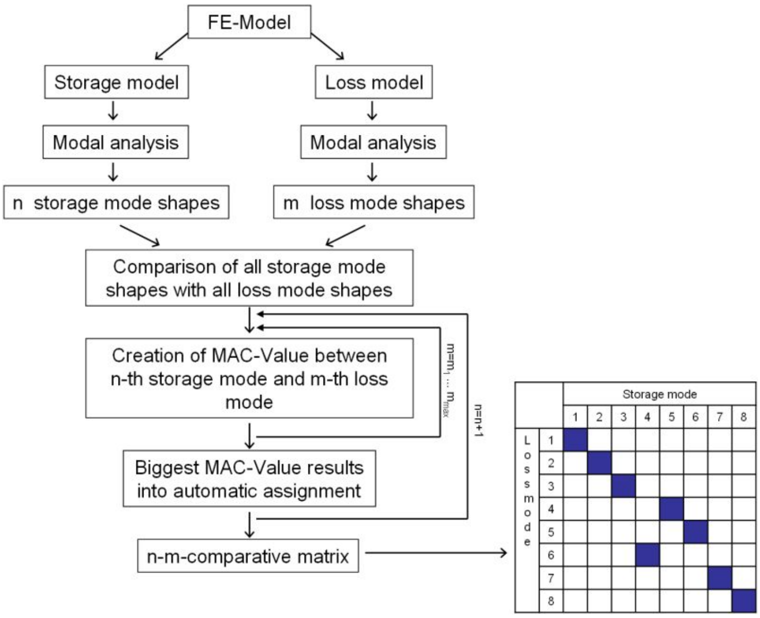

Figure 8: Flowchart of the adapted method of complex eigen-values

Beginning with the equation of motion for damped vibrations in complex notation

$$\underline{\underline{M}}\,\underline{\ddot{u}}(t) + \underline{\underline{K}}^{*}\underline{u}(t) = \underline{0} \tag{7}$$

a complex eigen-value problem can be deduced using the complex approach $\underline{u}(t) = \underline{\eta}^{*} \cdot e^{i\omega t}$ [5].

$$\begin{gathered}\left[(\underline{\underline{K}}' + i\underline{\underline{K}}'') - (\omega' + i\omega'')^2 \underline{\underline{M}}\right]\left(\underline{\eta}' + i\underline{\eta}''\right) = \underline{0} \\ \left[\underline{\underline{K}}^{*} - \omega^{*2}\underline{\underline{M}}\right]\underline{\eta}^{*} = \underline{0}\end{gathered} \tag{8}$$

Equating coefficients of (8) leads to the following relations

$$\begin{gathered}\left[\underline{\underline{K}}' - (\omega'^2 - \omega''^2)\underline{\underline{M}}\right]\underline{\eta}' = \underline{0} \\ \left[\underline{\underline{K}}'' - 2\omega'\omega''\underline{\underline{M}}\right]\underline{\eta}'' = \underline{0}\end{gathered} \tag{9}$$

Using these equations, the eigen-values $\omega'^2_{FE} = \omega'^2 - \omega''^2$ and $\omega''^2_{FE} = 2\omega'\omega''$ can be separately calculated using commercial FE-tools.

To perform these calculations, two FE models are required, a so called storage model and a loss model. Both models are based on the same geometry, but differ in the material parameters. The storage model uses the real parts of the complex material parameters and the loss model the imaginary parts. By a pairwise assignment of the eigenfrequencies, the modal loss factors of the selected mode shape η_n can be calculated.

$$\eta_n = \frac{f''^2_{n,FE}}{f'^2_{n,FE}} \tag{10}$$

3.3 Classification of mode shapes using modal assurance criterion (MAC)

The calculation of the modal loss factor using the adapted method of complex mode shapes requires a mapping of the calculated mode shapes using the storage and the loss model. Because of the different anisotropic material properties for the two models, the order of the calculated mode shapes may change (Figure 9). Assigning the mode shapes by hand is time-consuming and error-prone. Furthermore, there is no potential for an automatic process which will be a fundamental for an automatic optimization loop.

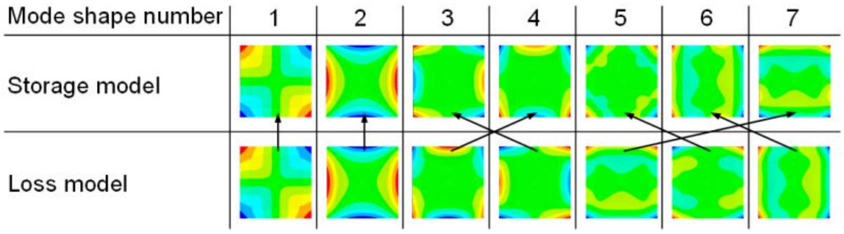

Figure 9: Example for manually mapped storage and loss mode shapes

The short overview of the first seven mode shapes of a simple tray structure shows the complexity of the manual classification.

To reduce the error-proneness and create an automatic process, the Modal Assurance Criterion (MAC), which is usually used for the mapping of measurement and simulation results, is adapted to compare and allocate the mode shapes. Therefor, the eigen-vectors of both vibration modes are compared.

$$MAC_{ij} = \frac{\left| \underline{n}_j^T \underline{n}_i \right|^2}{\underline{n}_j^T \underline{n}_j \cdot \underline{n}_i^T \underline{n}_i} \tag{11}$$

The magnitude of the resulting MAC value defines the correlation between the two mode shapes. A value of 1 means a perfect correlation of the compared mode shapes.

3.4 Preparation of the transmission loss calculation

For the calculation of the transmission loss using VA One, ANSYS exports the structural geometry, the BE mesh data and the modal result file and calls VA One in batch mode. VA One reads a configuration file written by ANSYS which describes the transmission loss calculation. The definition file contains filenames of the structure, the BE mesh and the modal basis to be used. VA One automatically creates the structural mesh, the fluid faces, the fluid domains, the infinite rigid plane between the two rooms and a transmission hole where the structure is installed. This is done through the use of reserved identification numbers (PID) in the definition file. VA One automatically creates the BE fluids and all necessary connections. Finally, VA One defines a diffuse field excitation, sets structural modal damping following a predefined frequency range and launches the Transmission Loss (TL) computation. Figure 10 shows the resulting BE-model.

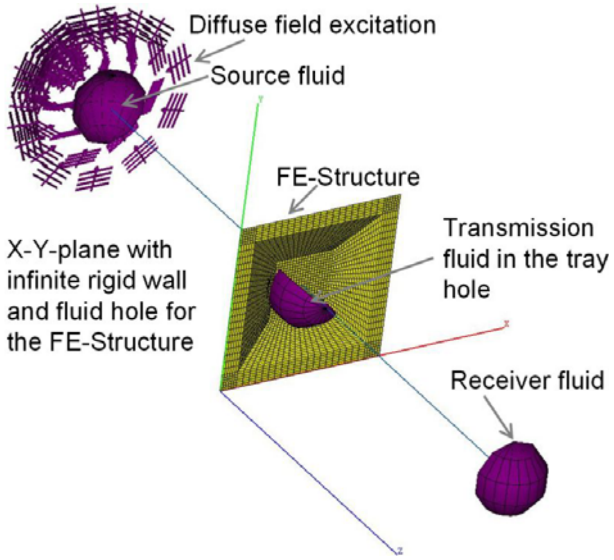

Figure 10: Layout of the generated BE-model

The results of the transmission loss calculation are written in a comma separated text file for further analysis. Table 1 exemplarily shows the results of the first frequencies for the defined structure.

	Fibre angle	0	5	10	15	20	25
Frequency	5	78,1	77,5	76	73,7	67,5	64,7
	10	69,3	68,7	67,2	64,8	58,6	55,9
	15	64,0	63,4	61,9	59,5	53,3	50,5
	20	60,1	59,5	58,0	55,6	49,4	46,6
	25	57,1	56,6	55,0	52,6	46,4	43,6
	30	54,6	54,0	52,5	50,0	43,8	41,0
	35	52,2	51,6	50,0	47,6	41,4	38,5

Table 1: First results of transmission loss calculation for different fibre angles and a thickness ratio of 0.1

After creating the output file, VA One is closed by the batch file and ANSYS resumes the loop. In ANSYS, the desired parameters will be changed and the next calculation loop using VA One will be started.

4 Results for a CFRP-RHC-sandwich tray

For the first investigations, a quadratic sandwich tray with the dimension a=b=560 mm, c=d=450 mm, e=f=200 mm and g=50 mm was chosen. In the model, all outside edges where clamped. To optimize the transmission loss of the selected tray structure, the fibre orientation of the face sheets and the thickness ratio between face sheets and core layer were varied. The selected frequency range is given by the standard for sound insulation in buildings from 50 to 5000 Hz [3].

By the variation of the fibre angle of the face sheets, the stiffness and the damping ratio of the structure is changed. For the first investigations, the fibre angles for both face sheets were varied in the same way. Because of the interaction between the anisotropic material characteristics and the geometry, the highest stiffness values were reached for an orientation of 0 °. Furthermore, the reinforcement results in doubled mode shapes which usually are almost identical but diverge slightly for fibre angles around 30 °. The resulting transmission loss considering the mentioned effects is shown in Figure 11 for a thickness ratio of 0.01. Because of the quadratic geometry, the results are symmetric to a fibre angle of 45 °.

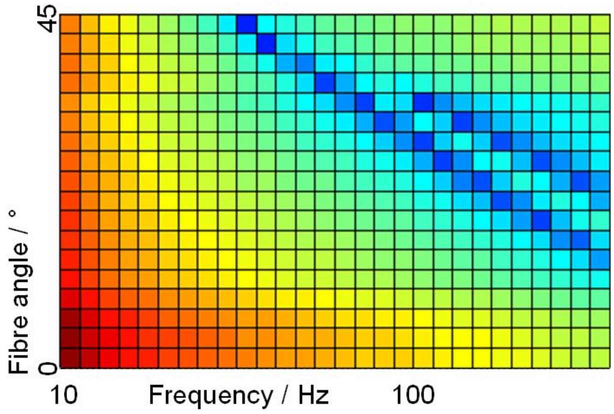

Figure 11: Results of the transmission loss calculation for a thickness ratio of 0.01

For the chosen thickness ratio, the stiffness-dominated region of the transmission loss spectrum is much wider for small fibre angles than for greater ones (till 45 °). That's a result of the stiffening effect of the reinforcement in interaction with the chosen geometry. The blue lines indicate the crossover to the resonance dominated region. So the fall-offs are corresponding to the first mode shapes of the structure. In comparison with other thickness ratios, it's mentioned that a value of 1 leads to the smallest stiffness controlled region and also to the lowest eigenfrequencies, which results in a low transmission loss spectrum (Figure 12).

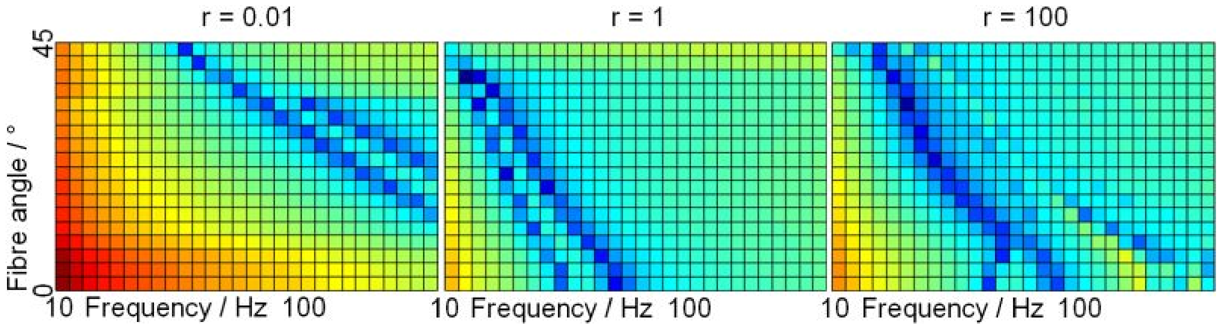

Figure 12: Comparison of the transmission loss results for different thickness ratios

Having a look at Figure 12, a raising thickness ratio seems to lead into lower transmission loss values. But the plotted charts aren't useful to identify the parameter set with the best overall transmission loss. Therefore, the standard for sound insulation of buildings suggests a single value for the transmission loss. This single value allows a better comparison of different transmission loss spectra. Figure 13 shows the resulting single values for selected combinations of fibre orientation and thickness ratio.

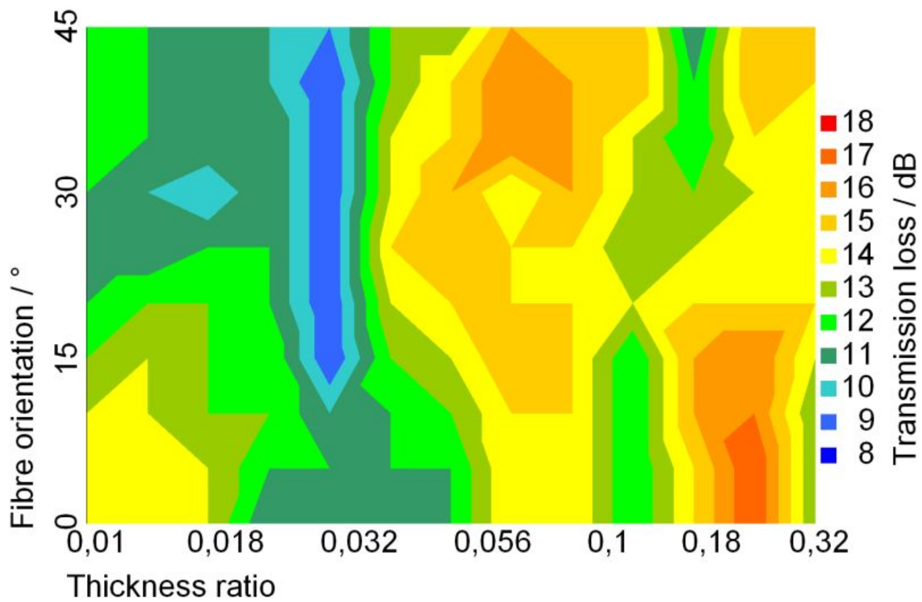

Figure 13: Single value transmisison loss for selected combinations

Because of the complex interaction between the sandwich lay-up, the geometry and the anisotropic material behaviour, the best overall transmission loss is reached for a fibre orientation of nearly 0 ° and a thickness ratio of 0.2. Particularly, the reachable difference of nearly 10 dB between the best and the worst construction without changing the overall mass indicates the high vibro-acoustic potential of fibre-reinforced sandwich constructions.

5 Conclusion

Beginning with a new model to calculate the modal damping values for complex fibre-reinforced composite structures, the investigations lead to a complex optimization tool for the transmission loss of sandwich structures. Within the scope of this work, it was shown, how common FE software will be able to calculate modal damping values. Therefor, the method of complex mode shapes was adapted and two temporary models were created. The storage model calculates the real parts of the mode shapes and the loss model the imaginary parts. By an automatic mapping procedure using the modal assurance criterion, the resulting damping values can be calculated. The FE software ANSYS is used for the creation of the geometry, to calculate the modal parameters and control the whole loop. VA One as a powerful vibro-acoustic software is started by ANSYS and performs the transmission loss calculation. The required parameters were given by a configuration file, created and exported by ANSYS, and the model creation was done by VA One. The results of the transmission loss calculation are saved in a comma separated text file and can be evaluated in further steps. After the calculation of the transmission loss for the active loop, VA One sends a “completed” signal to ANSYS and the next loop with predefined changing of parameters can start.

As a result of first investigations using that tool, the influence of different fibre angles and especially different thickness ratio of a sandwich tray using the same overall mass could be shown. Further evaluations of the resulting spectra are currently in work.

In conclusion, the developed optimization tool, using ANSYS and VA One, is a powerful assistance for vibro-acoustic driven design processes especially for complex lightweight structures

Acknowledgements

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