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A new simulation and optimization tool for calculating the attenuation of airborne and structure-borne sound of maritime silencers

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ABSTRACT

The acoustical impact of silencers is restricted by the influence of structure-borne sound. Thereby, excitation and radiation of structure-borne sound play an important role, as well as its transmission along the silencer system. In the design process of a silencer the influence of structure-borne sound on the damping effect is mostly considered by empirical estimations, which can cause an oversized construction. For calculating structure-borne sound in maritime silencers with their typical large dimensions, the application of statistical methods is a possible problem-solving approach. In this contribution, a new simulation and optimization tool is presented, considering the decreased damping effect of silencers due to structure-borne sound. The tool is based on Statistical Energy Analysis (SEA). Results of analytical and experimental SEA are compared. These results were obtained on downscaled silencers and have been transferred to real maritime silencers. Especially the SEA-parameters loss factor and coupling loss factor, airborne and structure-borne sound energies and the number of relevant modes are considered. Based on the measured damping of a silencer compared with numerical and statistical calculations, the capability for optimizing the design process of large silencers is shown.

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1 INTRODUCTION

Silencers are used in diverse applications for reducing the emission of airborne sound. With increasing requirements on their weight, dimension and emission of pollutants, silencers must ensure especially the compliance with noise limit values. However, the acoustical impact of silencers is restricted by the influence of structure-borne sound. Thereby, excitation and radiation of structure-borne sound play an important role, as well as its transmission along the silencer system consisting of various ducts and mufflers.

An example of a silencer system is shown on the left side in Figure 1. The source, e.g. an engine or a fan, emits primarily airborne sound but also structure-borne sound because of its vibration. Furthermore, an indirect excitation of structure-borne sound due to coupling of airborne sound occurs. The structure-borne sound energy is transmitted along the duct and the enclosure of the splitter silencer. After that, a portion of this energy is radiated in the form of airborne sound behind the silencer. This effect reduces the acoustical impact of the splitter silencer. In practice this propagation path has to be interrupted, e.g. by use of vibration isolators or sound absorbing material around the enclosure.

Calculating the influence of structure-borne sound on the damping effect represents a major challenge when dimensioning a silencer. In many cases it is considered by empirical estimations or assumptions made by manufacturers with long-term experience. On the right side of Figure 1, an empirical estimation of the normalized transmission loss of silencers lined with absorbing material is shown for different lining ratios. The curves are based on Piening’s equations to calculate the damping of silencers and plotted against the dimensionless frequency parameter η [1]. At low frequencies ($\eta << 1$) the transmission loss increases with frequency squared. In the middle frequency range ($\eta \approx 1$) the transmission loss is at a maximum but limited at 1.5 dB independent of the lining ratio. This limitation is made preventively because of the well-known presence of structure-borne sound. The lining ratio only affects the frequency range of maximum transmission loss. At high frequencies ($\eta > 1$) the transmission loss decreases with frequency squared.

Piening’s equations can be sufficiently accurate in the case of simply designed absorption silencers. Often there is a need for a more accurately calculation for dimensioning a wide variety of silencers with regard to their overall acoustical impact. Especially in the case of large dimensions like in the maritime sector, statistical methods can be applied. These methods remain comparatively easy to handle at large dimensions and high frequencies, in contrast to numeric simulation methods like the Finite Element Method or the Boundary Element Method. In this paper, the development of a statistical approach for calculating the attenuation of airborne and structure-borne sound of maritime silencers based on Statistical Energy Analysis (SEA) is presented. This approach has been tested on a large maritime silencer under laboratory conditions and the related results are shown.

The research is part of a joint project involving Gesellschaft für Akustikforschung Dresden mbH, Fraunhofer IWU and LUHE-STAHL GmbH.

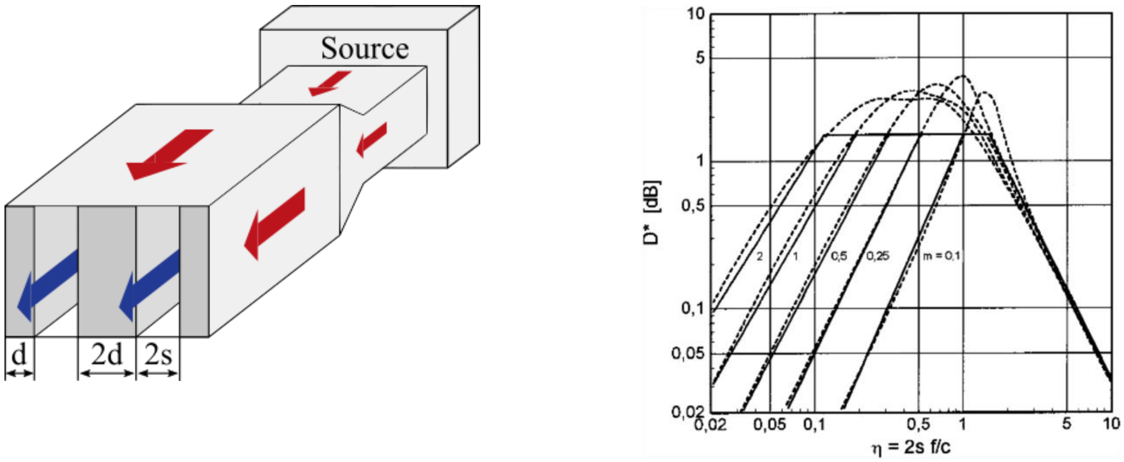

Fig. 1 – Left: Example of a lining with geometric parameters d and s in a splitter silencer with propagation of airborne sound (blue) and structure-borne sound (red) – Right: Normalized transmission loss of homogenous fibrous or porous silencers as a function of the dimensionless frequency parameter η for different lining ratios m = d/s [1]

2 STATISTICAL ENERGY ANALYSIS OF COUPLED SUBSYSTEMS

In SEA, energy flows between subsystems and energy conversions are examined. Commonly, a subsystem is a physical component of the technical structure of interest, but can also be ambient air or a specific kind of structure-borne sound waves. The energies are temporal and local mean values within frequency bands, usually third-octave or octave bands. Thus, SEA does not calculate the vibration behavior of a subsystem at a defined time, at a certain place or at a specific frequency. The basics, applications and restrictions of SEA are described in the references [2] and [3]. At this point, the principle is shown by means of two subsystems in Figure 2. Therein, both subsystems are excited to oscillate by the input energies $E_{IN,1}$ or $E_{IN,2}$ respectively. A part of the resulting vibrational or acoustical energies of the subsystems, E_1 and E_2, is transferred between the subsystems, characterized by the coupling loss factors η_{12} and η_{21}. The other part is transferred into another form of energy, e.g. thermal energy, characterized by the loss factors η_{11} and η_{22}.

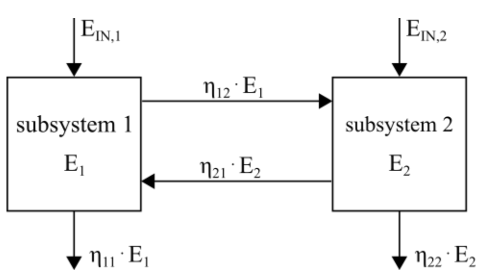

Fig. 2 – Energy flow in a SEA model consisting of two subsystems

Similar to the node laws known from electrical engineering, energy balance equations can be set up for both subsystems. When subsystem 1 is excited with energy $E_{IN,1}$ the following equations represents the energy balance for both subsystems:

$$E_{\mathrm{IN},1} = \eta_{11} \cdot E_{11} + \eta_{12} \cdot E_{11} - \eta_{21} \cdot E_{21},$$
$$0 = \eta_{22} \cdot E_{21} + \eta_{21} \cdot E_{21} - \eta_{12} \cdot E_{11}. \tag{1a}$$

When subsystem 2 is excited to oscillation because of input energy $E_{IN,2}$ the following equations for the subsystems can be written:

$$0 = \eta_{11} \cdot E_{12} + \eta_{12} \cdot E_{12} - \eta_{21} \cdot E_{22},$$
$$E_{\mathrm{IN},2} = \eta_{22} \cdot E_{22} + \eta_{21} \cdot E_{22} - \eta_{12} \cdot E_{12}. \tag{1b}$$

The use of a matrix notation is suitable especially in the case of a huge number of subsystems. For two subsystems, the following equation system based on Equations (1a) and (1b) is set up:

$$\begin{bmatrix} E_{\mathrm{IN},1} & 0 \\ 0 & E_{\mathrm{IN},2} \end{bmatrix} = \begin{bmatrix} \eta_{11} + \eta_{12} & -\eta_{21} \\ -\eta_{12} & \eta_{22} + \eta_{21} \end{bmatrix} \cdot \begin{bmatrix} E_{11} & E_{12} \\ E_{21} & E_{22} \end{bmatrix}. \tag{2}$$

Based on Equation (2) a system of equations can be set up for n subsystems in the form of Equation (3). Therein η_{ii} is the loss factor of subsystem i, η_{ij} is the coupling loss factor from subsystem i to subsystem j and E_{ij} is the energy of subsystem i when subsystem j is excited. The elements of the loss factor matrix can either be determined by measurement of input and subsystem energies and subsequent transposing the equation system or by analytical calculations. Depending on the approach it is called experimental or analytical SEA:

$$\begin{bmatrix} E_{\mathrm{IN},1} & 0 & \cdots & 0 \\ 0 & E_{\mathrm{IN},2} & \cdots & \vdots \\ \vdots & \vdots & \ddots & \vdots \\ 0 & \cdots & \cdots & E_{\mathrm{IN},n} \end{bmatrix} = \begin{bmatrix} \sum_{k=1}^{n} \eta_{1k} & -\eta_{21} & \cdots & -\eta_{n1} \\ -\eta_{12} & \sum_{k=1}^{n} \eta_{2k} & \cdots & \vdots \\ \vdots & \vdots & \ddots & \vdots \\ -\eta_{1n} & \cdots & \cdots & \sum_{k=1}^{n} \eta_{nk} \end{bmatrix} \cdot \begin{bmatrix} E_{11} & E_{12} & \cdots & E_{1n} \\ E_{21} & E_{22} & \cdots & \vdots \\ \vdots & \vdots & \ddots & \vdots \\ E_{n1} & E_{n2} & \cdots & E_{nn} \end{bmatrix}. \tag{3}$$

3 STATISTICAL ENERGY ANALYSIS MODEL OF SILENCERS

3.1 Entire SEA model

Figure 3 shows the developed SEA model to describe the propagation of airborne and structure-borne sound in an exemplary duct system. It consists of a source, a silencer, one duct on the input side and one duct on the output side of the silencer. Within the model, subsystems are defined representing airborne and structure-borne sound inside both ducts and the silencer. This distinction enables a separate treatment of coupling and propagation of structure-borne sound. According to Figure 3, the subsystems 1 and 4 – airborne and structure-borne sound of the duct on the input side – are directly excited by the corresponding input energies $E_{IN,1}$ and $E_{IN,4}$, which are emitted by the source. The remaining subsystems of the model are indirectly excited due to coupling to other subsystems. It is assumed that there is no coupling between subsystems 2 and 5 – airborne and structure-borne sound within the silencer – and therefore no coupling loss factor. This assumption is made because maritime silencers have mostly a lot of absorption material on the inner and outer side of their enclosures. The excitation of subsystem 4 combined with structure-borne sound propagation along the silencer and following radiation in the form of airborne sound is assumed as the main reason for the reduction of the silencer's acoustical impact.

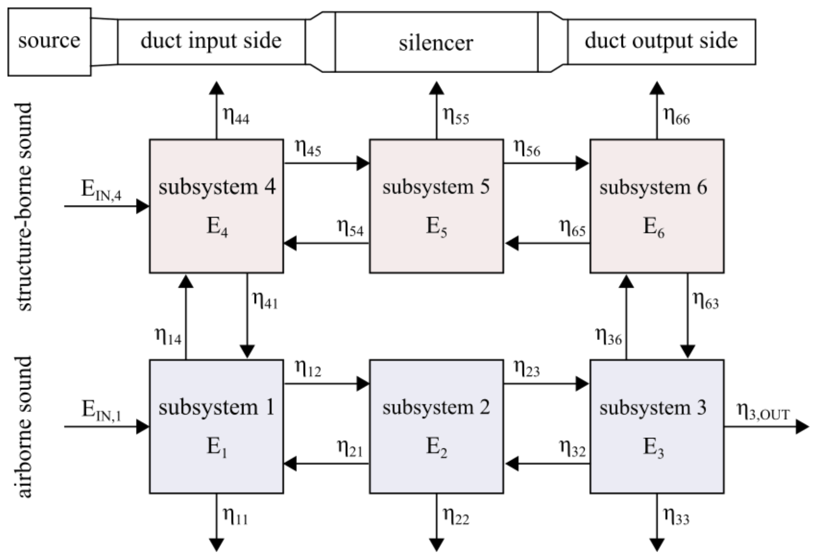

Fig. 3 – Entire SEA model for calculating excitation and propagation of structure-borne sound and airborne sound in a silencer

3.2 Modeling of airborne sound

The transmission loss D_{ts} is defined as the ratio between induced airborne sound energy into the silencer and passed through sound energy [4]. Transferring this definition into the SEA model in Figure 3, it corresponds with the ratio of energies induced in and passed through subsystem 2. If only airborne sound propagation is considered, the SEA model is reduced to the three coupled subsystems in Figure 4.

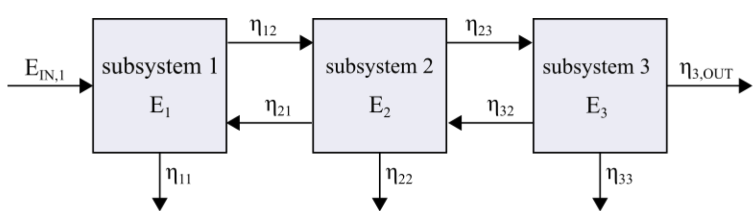

Fig. 4 – SEA model considering only airborne sound in a silencer

By the definition of transmission loss and the reduced SEA model in Figure 4, the following equation can be set up for calculating the transmission loss considering the net effective energy flow through the silencer:

$$D_{\mathrm{ts}} = 10 \cdot \log_{10}\left(\frac{\eta_{12} \cdot E_1 - \eta_{21} \cdot E_2}{\eta_{23} \cdot E_2 - \eta_{32} \cdot E_3}\right) \text{ dB}. \tag{4}$$

The acoustical impact of the silencer within the SEA model is represented by the energy loss factor η_{22} of subsystem 2. Thus, in a next step, the connection of transmission loss and loss factor η_{22} needs to be formed. Depending on the silencer type and the size of the entire SEA model, the mathematical connection is more or less extensive. In the scope of this research the SEA model is validated by acoustical measurements at a maritime absorption silencer. For this reason, the SEA parameters within the model are adapted according to Equation (5) and Figure 5, which show the made definitions of loss factors and coupling loss factors considering airborne sound propagation in the case of an absorption silencer:

$$\begin{gathered} \eta_{11} = \eta_{33} = \eta_{21} = \eta_{32} = 0, \\ \eta_{12} = \eta_{23} = \eta_{3,\mathrm{OUT}} = 1. \end{gathered} \tag{5}$$

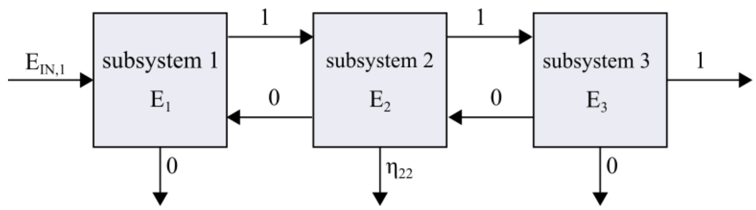

Fig. 5 – Simplified SEA model considering airborne sound in an absorption silencer

Therein only wave propagation in direction from the source to the outlet opening is considered, assuming that there is no relevant reflection or another source within the silencer system. After setting up the energy balance equations for the three subsystems in Figure (5), based on Equations (4) and (5) a new defining equation for the transmission loss of an absorption silencer as a function of its loss factor η_{22} can be written according to Equation (6). At this point, the propagation of structure-borne sound has not been considered yet:

$$D_{\mathrm{ts}} = 10 \cdot \log_{10}(\eta_{22} + 1)\ \mathrm{dB}. \tag{6}$$

3.4 Modeling of structure-borne sound

Figure 6 shows the excitation and propagation path of structure-borne sound. In the later simulation, this path can be regarded separately from airborne sound. Independent from the silencer type, the coupling loss factors between the pictured subsystems have to be calculated for every direction. When the path is coupled with the airborne sound path, the overall transmission loss of the silencer can be calculated considering the previous assumptions for airborne sound:

$$D_{\mathrm{ts}} = 10 \cdot \log_{10}\left(\frac{E_1}{E_3}\right)\ \mathrm{dB}. \tag{7}$$

As mentioned above, specific kinds of structure-borne sound waves can be represented by unique subsystems. This distinction should be made when the total vibrational energy is substantially affected by the energies of several kinds of waves. Because of mostly thin-walled ducts within a silencer system, the total vibrational energy is assumed to be equal to the energy of bending waves. Therefore, subsystems 4, 5 and 6 represent bending waves within the specific structure-borne sound subsystem.

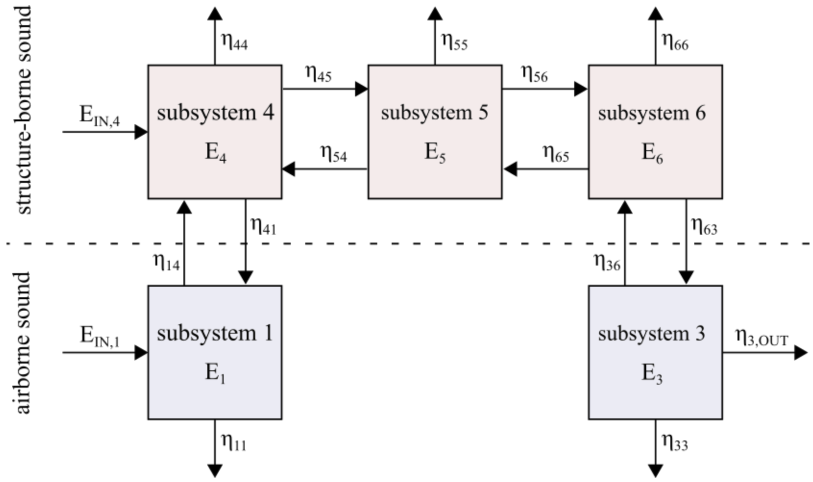

Fig. 6 – SEA model considering structure-borne sound transfer and its coupling to airborne sound in a silencer

4 EXPERIMENTS

The presented entire SEA model in Figure 3 has been developed by means of results from a large number of measurements and theoretical investigations. These first results were obtained from different test objects, especially from down scaled silencers and duct components. Because of the planned purpose in the maritime sector, the SEA model is applied on a real maritime absorption silencer with typically large dimensions. As a basis of comparison, experimental investigations with regard to transmission loss of this silencer have been executed under laboratory conditions. At this point, the measurement set-up and necessary quantities for validating the SEA model are described. Because of spectral averaging in SEA, these quantities have to be seen as spectral averages within frequency bands.

The transmission loss of the silencer was determined by measuring the introduced airborne sound energy and passed through sound energy (E_1 and E_3 in the SEA model). To point out the influence of structure-borne sound on the transmission loss, separate excitations of airborne and structure-borne sound have been realized. Therefor, an electrodynamic loudspeaker or rather a shaker was used. Figure 7 shows some details of the experimental set-up. Microphones were disposed in the middle and at the cylinder barrels of the inner side of both ducts. The silencer itself has been treated as a black box. The airborne sound energies E_1 and E_3 were calculated on the basis of measured time and space averaged sound pressures $\overline{\tilde{p}_{1,3}^2}$ within the inside of both ducts [5]:

$$E_{1,3} = \frac{\overline{\tilde{p}_{1,3}^2} \cdot P_0}{p_0^2 \cdot \omega} \cdot \frac{S_{1,3}}{S_0 \cdot 10^{\frac{K_{1,3}}{10\ \mathrm{dB}}}}. \tag{8}$$

In Equation (8), P_0, p_0 and S_0 are the level references 10^{-12} W, $20 \cdot 10^{-6}$ Pa or 1 m² respectively, ω is the center angular frequency of the spectral band, S and K are the cross sectional areas or rather correction factors regarding transverse modes.

Fig. 7 – Transmission loss measurement of the maritime absorption silencer system under laboratory conditions

In the experiments one-third octave bands were used. Hence, the averaged sound pressure can be written in spectral terms using the single sided spectral density S_{pp} within a frequency band, characterized by the cut-off frequencies ω_1 and ω_2:

$$\overline{\tilde{p}_{1,3}^2} = \frac{1}{N} \cdot \sum_{i=1}^{N} \int_{\omega_1}^{\omega_2} S_{1,3\,pp}(\omega)\,\mathrm{d}\omega. \tag{9}$$

Furthermore, energy E_1 is assumed to be equal to the energy introduced in subsystem 1, hence it is valid that $E_1 = E_{IN,1}$.

The time averaged input energy in subsystem 4 – structure-borne sound energy into the duct on the input side – have been determined using an impedance sensor placed between shaker and duct. As shown in Figure 7, the shaker was applied on the cylindrical enclosure. The signals of force and acceleration at the driving point were recorded in phase. Then, the input energy was calculated using the single sided crosspower-spectrum S_{Fx} between force F and displacement x:

$$E_{\mathrm{IN},4} = \mathrm{Re}\left\{\int_{\omega_1}^{\omega_2} S_{\mathrm{Fx}}(\omega)\,\mathrm{d}\omega\right\}. \tag{10}$$

Because of the vibration excitation on the cylindrical enclosure, primarily bending and out-of-plane waves were excited. This corresponds with the taken definition of the structure-borne sound subsystems 4 and 6. The energies of these subsystems were determined using randomly distributed sensors, which measures acceleration normal to the surface. Then the vibrational energies were calculated using the single sided space averaged spectral densities $S_{i,\mathrm{vv}}$ of velocity v, with respect to position *i* of the specific duct:

$$E_{4,6} = m_{4,6\ \mathrm{cyl}} \cdot \frac{1}{N} \cdot \sum_{i=1}^{N} \int_{\omega_1}^{\omega_2} S_{4,6\ i,\mathrm{vv}}(\omega)\ \mathrm{d}\omega. \tag{11}$$

5 RESULTS

At this point the most relevant results from experimental and analytical investigations regarding the transmission loss of the maritime silencer are shown. Figure 8 illustrates the measured transmission loss of a combined airborne and structure-borne sound exication on the one hand and an exclusive excitation of airborne sound on the other hand. Both curves have a characteristic damping path of an absorption silencer consisting of an increasing part, a plateau and a decreasing part (cf. right side of Figure 1). The transmission loss in the case of combined excitation is below the curve of exclusive airborne excitation. It becomes clear that there is an influence of structure-borne sound on the damping effect of the investigated silencer. This issue is present in frequency bands up to 1.6 kHz and the maximum amounts about 5 dB at 250 Hz.

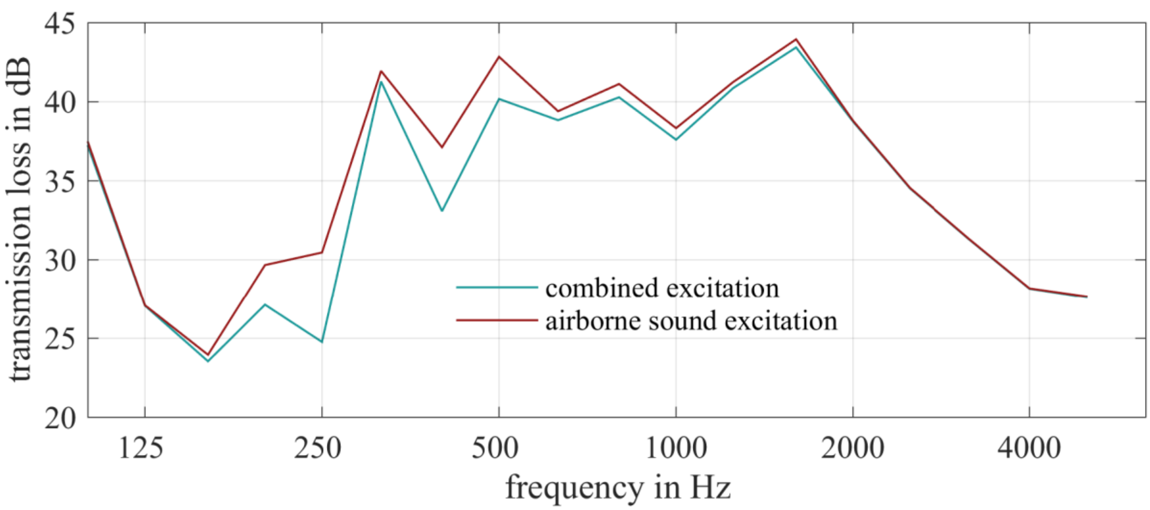

Fig. 8 – Measured transmission loss of the maritime silencer in the cases of combined airborne and structure-borne sound excitation and exclusive excitation of airborne sound

In Figure 9 the measured transmission loss for a combined excitation of airborne and structure-borne sound is compared with two simulation results.

The FEM simulated curve is calculated considering only airborne sound propagation. In practice, those curves are often limited by the manufacturer to a suitable plateau on empirical basis. By means of the FEM simulation the loss factor η_{22} of subsystem 2 – airborne sound within the silencer – has been determined using Equation (6) to perform a SEA simulation.

The transmission loss resulting from the – FEM assisted – analytical SEA is also shown in Figure 9. By comparison of this result with the measured curve it can be mentioned, that the investigated silencer is represented sufficiently accurate by use of the SEA model. The maximum difference between both curves reaches a value of 8 dB at 250 Hz. The SEA curve is not calculable below 200 Hz because of low modal densities in airborne sound. For this reason, the SEA result must also be observed critically between 200 Hz and 500 Hz. Over 500 Hz the mode numbers of the airborne sound subsystems are at least four, therefore, a suitable statistical accuracy is assumed. A possible explanation for the good correlation of the SEA results with measured data between 200 Hz and 500 Hz is the sufficient high modal density of the structure-borne sound subsystems.

An empirical estimation based on Piening's equations could not be realized because of the complicated inner structure of the absorption silencer.

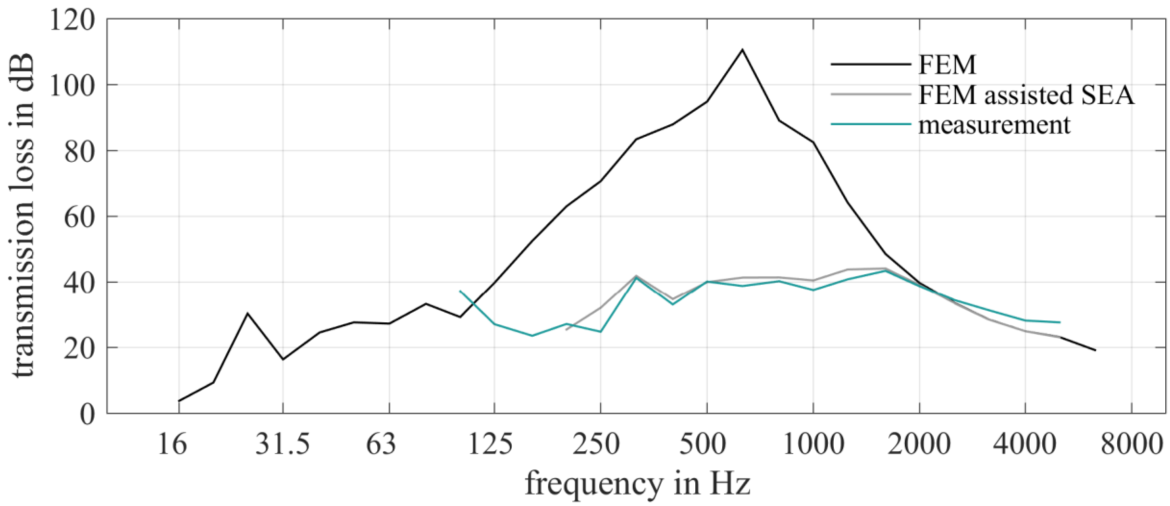

Fig. 9 – Comparison of the transmission loss from a FEM simulation considering only airborne sound with those from a FEM assisted SEA simulation considering both, airborne and structure-borne sound and measurement data for a combined airborne and structure-borne sound excitation of the maritime silencer system

6 CONCLUSIONS

To predict the transmission loss of silencers under the influence of structure-borne sound more accurately than before, a calculation model based on Statistical Energy Analysis has been developed. Within the model, there is a distinction between the propagation of airborne and structure-borne sound, whereby the corresponding subsystems are coupled among each other and exchange energy. As usual in SEA, the energy exchange is characterized by coupling loss factors. In the case of the presented absorption silencer, the coupling loss factors for airborne sound were assumed in a way that just sound propagation from the source to the outlet opening is represented in the model. Thus, the calculation of transmission loss is simplified, because only the loss factor for airborne sound within the silencer forms the connection to the SEA model.

During this research different types of down scaled silencers and duct components were investigated in experimental and analytical approaches. Especially methods for estimation of coupling between subsystems were selected. Until now, the SEA model has been validated by means of a large maritime absorption silencer. The simulation results correspond with the measurements in a good approximation.

The aim is to carry on the model validation by means of more different types of silencers. Furthermore, the simulation of transmission loss will be made with real excitation data for airborne and structure-borne sound. For this, the input energies emitted by a marine diesel engine were measured on a test bench. Finally, the combination of the engine and the silencer within the exhaust gas system will be measured.

7 ACKNOWLEDGEMENTS

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