VIBRO-ACOUSTIC DESIGN OF PASSIVE AND ACTIVE TEXTILE COMPOSITES WITH HIGH DAMPING PROPERTIES

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SUMMARY: Anisotropic textile composites offer a directional dependent mechanical property profile, which flexibly can be designed to the external loads. Especially, a specific improvement of the modal damping and sound emission characteristics is achievable by this new group of textile composites using a material-adapted combination of piezo-ceramic actors (active control) and material damping effects (passive control). An essential precondition for such a vibro-acoustic design of passive and active textile-reinforced composites is a detailed knowledge of the complicated coherences between the structural-dynamic and acoustic behavior. For this purpose, combined analytical and numerical methods were developed at the ILK allowing to include ("passive") anisotropic parameter functions like directional dependent stiffness, damping and sound radiation as well as ("active") piezo-element parameters like piezo-behavior, -size, -position and -voltage into the design process.

KEYWORDS: Textiles, design, anisotropic material damping, sound radiation, coupled FE/BE analysis, optimization

INTRODUCTION

Lightweight composite structures for high-technology applications increasingly have to fulfill high demands not only on low constructive weight and adequate stiffness but also on reduced vibrations and sound radiation. These good vibro-acoustic qualities are needed to meet the rising comfort and environmental requirements. Here, especially textile-reinforced composites offer adjustable vibro-acoustic material (passive control) properties and provide the opportunity for a material-adapted integration of piezoelectric elements for active damping and active sound reduction measures (active control). The specific combination of different textile architectures and the large variety of passive and active design variables give the possibility to synergetically achieve the challenging demands on coupled dynamic and acoustic properties for optimal design of lightweight structures. An essential precondition for such holistic design concept is a detailed knowledge of the structural-dynamic and acoustic coherences of the different behavior of textile anisotropic composites. Thus, the specific anisotropic vibration and damping characteristics of passive and active textile composite structures have to be coupled with the acoustic behavior by physically based simulation models.

For this purpose advanced analytical and numerical methods have been developed at the ILK in order to include not only stiffness and strength but also the passive and active controlled sound radiation in the design process. Here, especially material-adapted numerical methods such as hybrid finite element and boundary element methods are predestinated for the modeling of the coupled field problems. As practice-oriented sub-model structures plane load bearing composite structures are chosen (e. g. Fig. 1).

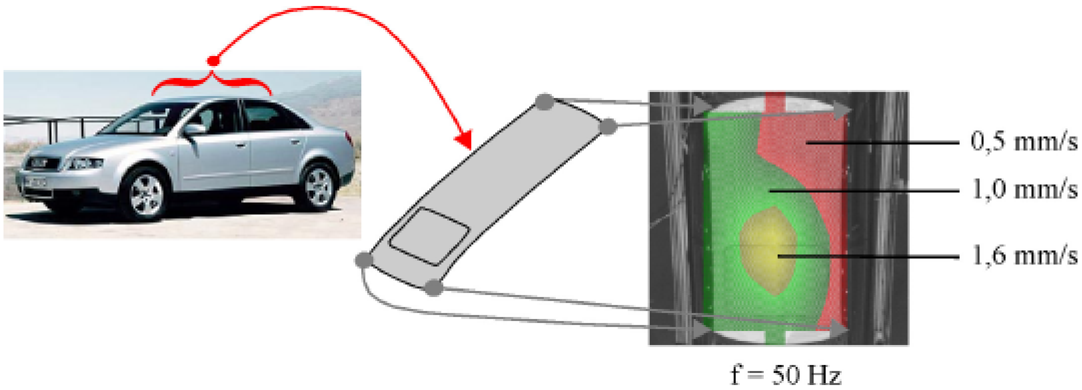

Fig. 1: Distinct flexural vibration mode of a car roof (AUDI AG)

The performed parameter studies reveal the physically based coupling of the manifold dynamic and acoustic design variables. The knowledge of these coupled dependencies allows to specifically develop vibro-acoustically tailored textile composite structures for high-technology applications.

STRUCTURAL-DYNAMIC ANALYSIS OF TEXTILE COMPOSITE PLATES

Textile-reinforced lightweight-components are sensitive for dynamic excitation due their low mass and the resulting low forces of inertia. This sensitivity leads to higher vibration-amplitudes and consequently to a higher sound-emission. Thus, material-adapted design methods for dynamically loaded textile composite structures have to be developed, which realistically describe their complex structural-dynamic behavior [1-3]. For most dynamic problems flexural vibrations are the typical form of structural vibrations causing a high sound-radiation because of the strong structure-fluid-coupling. Here, many associated problems can be analyzed using plane load bearing structures such as composite plates. The structural-dynamic analysis of anisotropic composite plates requires problem-specific idealizations of the real geometry and bearing-conditions. Furthermore, appropriate calculation-methods like the First Order Shear Deformation Theory (FSDT) have to be used to characterize the special deformation behavior of the different composite classes like unidirectional reinforced (UD) single layer composites or multilayered composites.

The FSDT considers the influence of shear deformation effects on the basis of the REISSNER-MINDLIN approaches analogue to the TIMOSHENKO beam theory [4, 5]. Each normal will stay a straight line in the deformed state. In contrast to the Classical Lamination Theory (CLT) the normals to the mid-plane are not normal to the deformed mid-plane (see Fig. 2). Thus, additional shear-angles $\hat{y}_x$ and $\hat{y}_y$ have to be considered, which are not equal to the first derivative of the deflection $\hat{w}$ [1].

[1] The roof symbols denote the time dependence of the physical quantities $\hat{y}_x, \hat{y}_y, \hat{w}$.

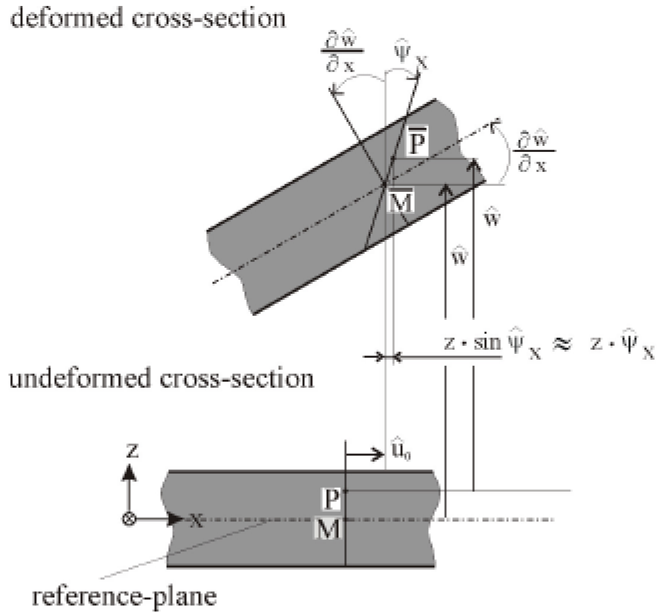

Fig. 2: Cross-section deformation according to the FSDT

The linear strain-displacement relations yield:

$$
\begin{aligned}
\hat{e}_x &= \frac{\partial \hat{u}_0}{\partial x} + z\frac{\partial \hat{y}_x}{\partial x}, & \hat{g}_{yz} &= k_1\left(\hat{y}_y + \frac{\partial \hat{w}}{\partial y}\right), \\
\hat{e}_y &= \frac{\partial \hat{v}_0}{\partial y} + z\frac{\partial \hat{y}_y}{\partial y}, \quad \textit{and} & \hat{g}_{xz} &= k_2\left(\hat{y}_x + \frac{\partial \hat{w}}{\partial x}\right), \\
\hat{e}_z &= 0, & \hat{g}_{xy} &= \left(\frac{\partial \hat{u}_0}{\partial y} + \frac{\partial \hat{v}_0}{\partial x}\right) + z\left(\frac{\partial \hat{y}_x}{\partial y} + \frac{\partial \hat{y}_y}{\partial x}\right).
\end{aligned}
\tag{1}
$$

The functions $\hat{u}_0$ and $\hat{v}_0$ are the x and y displacements of the reference-plane, respectively. The necessary shear corrections factors k_1 and k_2 are here chosen analogue to REISSNER and MINDLIN (see [4, 5]).

The material law of the investigated textile composites can be formulated on the basis of the model of hyper-elasticity with linear material behavior

$$
\begin{pmatrix} \hat{s}_x^{(k)} \\ \hat{s}_y^{(k)} \\ \hat{t}_{yz}^{(k)} \\ \hat{t}_{xz}^{(k)} \\ \hat{t}_{xy}^{(k)} \end{pmatrix} = \begin{pmatrix} \tilde{Q}_{11}^{(k)} & \tilde{Q}_{12}^{(k)} & 0 & 0 & \tilde{Q}_{16}^{(k)} \\ \tilde{Q}_{12}^{(k)} & \tilde{Q}_{22}^{(k)} & 0 & 0 & \tilde{Q}_{26}^{(k)} \\ 0 & 0 & \tilde{Q}_{44}^{(k)} & \tilde{Q}_{45}^{(k)} & 0 \\ 0 & 0 & \tilde{Q}_{45}^{(k)} & \tilde{Q}_{55}^{(k)} & 0 \\ \tilde{Q}_{16}^{(k)} & \tilde{Q}_{26}^{(k)} & 0 & 0 & \tilde{Q}_{66}^{(k)} \end{pmatrix} \begin{pmatrix} \hat{e}_x \\ \hat{e}_y \\ \hat{g}_{yz} \\ \hat{g}_{xz} \\ \hat{g}_{xy} \end{pmatrix} \quad \textit{or} \quad \underline{\hat{s}}^{(k)} = \underline{\underline{\tilde{Q}}}^{(k)}_{5x5}\, \underline{\hat{e}} \,. \tag{2}
$$

Here, the elements $\tilde{Q}_{ij}^{(k)}$ denote the reduced stiffnesses of the k^{th} single layer.

For the description of the structural behavior of the anisotropic textile composites the well known structural law is used (see Fig. 3)

$$
\begin{pmatrix} \underline{\hat{N}} \\ \underline{\hat{M}} \end{pmatrix} = \begin{pmatrix} \underline{\underline{A}} & \underline{\underline{B}} \\ \underline{\underline{B}} & \underline{\underline{D}} \end{pmatrix} \begin{pmatrix} \underline{\hat{e}}^0 \\ \underline{\hat{k}} \end{pmatrix} \tag{3}
$$

with A_{ij}, B_{ij} and D_{ij} being extensional, coupling and bending stiffnesses analogue to the CLT. The additional shear stiffnesses of the FSDT are determined to

$$A_{ij} = \int_{-h/2}^{h/2} \tilde{Q}_{ij}^{(k)} dz = \sum_{k=1}^{N} \tilde{Q}_{ij}^{(k)} \left(z_k - z_{k-1} \right) \quad \left(i, j = 4,5 \right), \tag{4}$$

wherein N is the maximum number of layers and z_{k-1} is the lower limit of the k^{th} single layer (see Fig. 3).

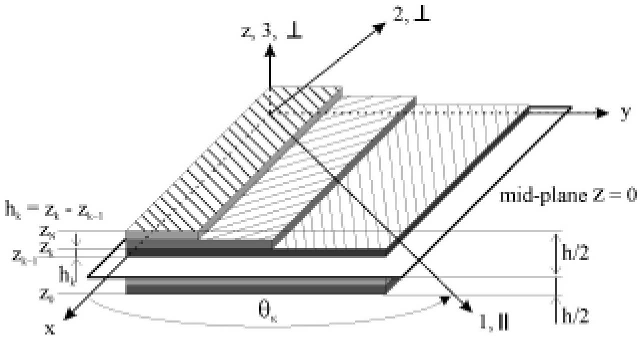

Fig. 3: Multilayered composite element and coordinate systems

The determination of free standing waves of plain load bearing structures is of specific importance for the structural dynamic analysis of anisotropic textile composites. Free standing plate waves show large stationary resonance amplitudes and consequently produce an increased sound radiation. Therefore, the analysis of the eigenfrequencies and eigenforms is essential for the development of vibro-acoustic models. The calculation of free standing plate waves is done here especially considering the specific deformation behavior of textile reinforced multilayered composites using the FSDT. The mathematical formulation of this complex problem is based on the well-known HAMILTONian principle for conservative elastic systems. Here, the starting point is the LAGRANGE function

$$\hat{L}(\hat{u}(x,y,z,t)) = \hat{T}(\underline{\hat{u}}(x,y,z,t)) - \hat{\Pi}(\underline{\hat{u}}(x,y,z,t)) \tag{5}$$

with

$$\underline{\hat{u}} = \left[\hat{u}_z, \hat{u}_x, \hat{u}_y \right]^T = \left[\hat{w}, \hat{u}, \hat{v} \right]^T \tag{6}$$

as the time- and spatial dependent deformation vector, $\hat{T}$ as the kinetic energy and $\hat{\Pi}$ as the full elastic potential of the system, which is in the case of free vibrations identical to the strain energy $\hat{U}$.

Using the HAMILTONian principal, the deformation vector $\underline{\hat{u}}$ has to be found, which gives a stationary value for the time integral of the LAGRANGE function and is according to the chosen boundary conditions. Considering the LAGRANGE variation $d\,\underline{\hat{u}}(x,y,z,t_1) = d\,\underline{\hat{u}}(x,y,z,t_2) = 0$ and assuming a harmonic time dependence for $\underline{\hat{u}}$ the following variational equation can be derived

$$dU - w^2 \int_V r(z) \underline{u}^T d\,\underline{u} \quad dV = 0 \tag{7}$$

with w being the circumferential frequency and V the plate volume.

For the solution of this fundamental variational equation the spatial dependent displacement vector $\underline{u}$ has to be determined using adapted functions, which describe the displacement fields of standing plate waves according to the chosen boundary conditions. Therefor, the RITZ approximation method is used. The variational equation (7) then yields the following eigenvalue equation

$$\left[\underline{\underline{K}} - \Lambda \underline{\underline{M}}\right]\underline{h} = \underline{0}\,. \tag{8}$$

Here $\underline{\underline{K}}$ denotes the symmetric structural stiffness matrix, $\underline{\underline{M}}$ the symmetric and positive definite mass matrix, $\underline{h}$ the eigenvector made of the RITZ coefficients and $\Lambda = w^2$ the eigenvalues.

VIBRO-ACOUSTIC DESIGN

The development of effective lightweight acoustic concepts for the low frequency range is a special challenge within many technical fields, e. g. the design of vehicles. Here, the use of multi-layered components with high stiffness and low constructive weight often leads to a deterioration of the vibro-acoustic property profile. The low coincidence frequencies result in a heavy structure-fluid coupling already within the first eigenfrequencies of the lightweight components and therefore produce high modal sound radiation [3]. However, classical lightweight sound reducing methods like the absorption concept cannot be used in the low frequency range due to the usually limited constructional space. Here, anisotropic textile composites combined with specific vibration control measures allow to synergetically adapt the directional-dependent material damping as well as the modal spectrum to the external dynamic loads. The resulting innovative possibilities for the design of textile composite lightweight structures with optimized vibro-acoustic and damping properties are yet hardly technically realized.

Based on the described structural dynamic calculation methods analytical vibro-acoustic simulation models for the determination of the sound radiation have been developed at the ILK using physically based coupling of the structural and acoustic fields [6]. Here, the radiated sound power and sound intensity of anisotropic textile composite plates are calculated as problem specific acoustic quantities.

The vibro-acoustic models were fully verified for typical glass and carbon fiber-reinforced multilayered composites by numerical methods (coupled Finite Element (FE) and Boundary Element (BE) Method) as well as experimental investigations (combined Laser Scanning Vibration Interferometry and sound intensity measurements) [6, 7].

Especially for anisotropic textile-reinforced composites the analytical solutions allow efficient parameter studies. There, the calculation of eigenfrequencies, eigenforms, modal loss factors as well as radiated sound power in dependence of material and geometry specific parameters – like fiber angle, laminate lay-up, combination of fiber and matrix, dimensions of the plate – are of importance. Due to the manifold combinations of these different composite specific design variables in the following selected problem fields of special technical relevance are chosen to achieve a better understanding of the complicated physical correlations.

Here, the vibro-acoustic parameter studies were performed for symmetric and balanced multilayered fiber-reinforced polymers of the type $\left[+q/-q\right]_S$ as well as for sandwich systems with fiber-reinforced composite top layers and shear-elastic polymer core layers made of polyurethane $[q/PUR/q]$.

The specific material parameters of these composites have a complex influence on the sound radiation of the plate structures. An overview of the fiber and matrix influence on the directional dependent radiated sound power of quadratic composite plates made of glass fiber-reinforced epoxy (GF/EP), carbon fiber-reinforced epoxy (CF/EP) and carbon fiber-reinforced polyetheretherketon (CF/PEEK) is given in Fig. 4.

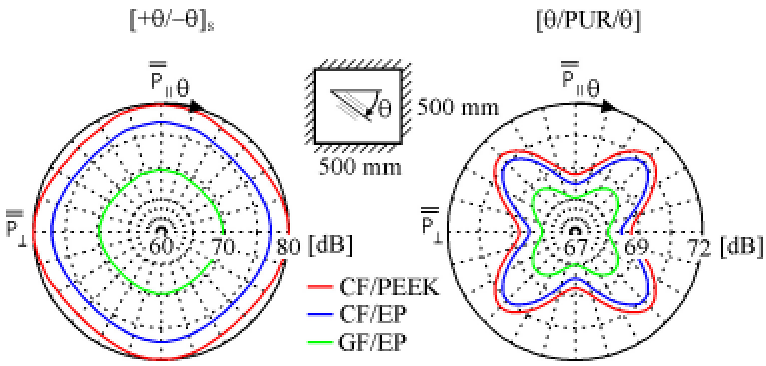

Fig. 4: Polar diagrams of the radiated sound power for different fiber-reinforced composites

The polar diagrams show, that the material damping of the specific fiber-reinforced materials mainly influences the modal sound radiation of multilayered composites. A significant sound reduction can be achieved by adjusting the lay-up to high modal damping values. For fiber-reinforced and sandwich composites a clear reduction of the sound power level is possible, if high damping fiber matrix combinations or shear-elastic damping layers are used. But besides the complex interaction of the material parameters fiber orientation, lay-up and fiber matrix combination a material-adapted vibro-acoustic design of hybrid multilayered composites has also to consider the geometry and bearing conditions. Thus, the vibro-acoustic characteristics have to be specifically designed for each application by the use of the developed analytical simulation models.

STRUCTURAL DYNAMIC ANALYSIS OF ACTIVELY DAMPED TEXTILE COMPOSITES

The effective use of piezo-ceramic elements for affecting the eigenfrequency spectrum of textile-reinforced composites requires a detailed structural dynamic analysis taking into account the coupled mechanical and electrical fields. Therefor, some FE-program systems provide the possibility to perform calculations based on coupled material laws. Furthermore, it is possible to use special programmable design languages in combination with an adapted file structure and MATLAB-routines for efficient parameter studies.

In the following the piezo-size and piezo-voltage are exemplarily chosen from the multiple number of actor-specific design variables for practice-oriented sensitivity analyses. As base-geometry a rectangular CFRP-plate ($1500 \times 1000 \times 4$ mm, fully clamped on all edges) is considered. The length PL of the piezo-element applied to the center of the plate surface was varied between 100 mm and 800 mm for a constant piezo-width PW of 200 mm and 600 mm, respectively (see also Fig. 5).

The used “pure mechanical“ elements Solid 45 (multi-layered composite) and the multi physics elements Solid 5 (piezo) have an universal size of 25 mm. The modal analysis with consideration of the piezo-elements was then performed in a two-stage task as so-called “prestress” analysis.

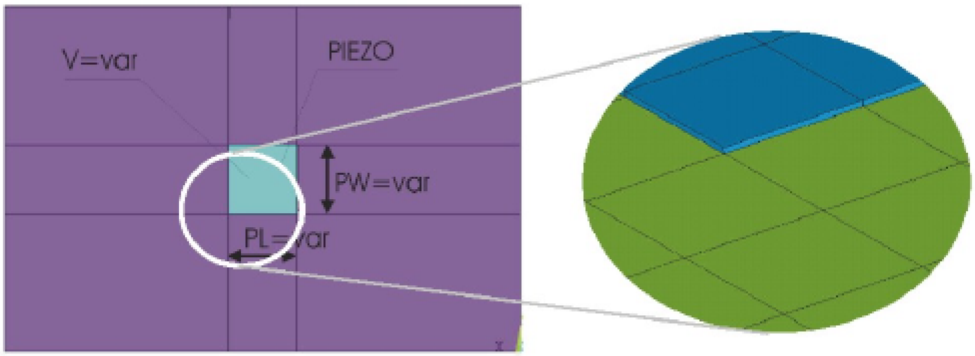

Fig. 5: CFRP composite plate with centered piezo element (turquoise) and design parameter voltage (U), piezo-length (PL) and piezo-width (PW)

The 4th eigenform of the CFRP plate is chosen here as an acoustically relevant mode shape of higher order, which shows a strong acoustic coupling and therefore a significant sound radiation. The frequencies of this 4th eigenform are clearly influenced by the piezo-voltage and length (see Fig. 6). Within the investigated piezo dimensions the smaller piezo-element with PW = 200 mm shows a maximum of the actively damped eigenfrequencies for a piezo-length PL of approximately 400 mm independent of the applied voltage. For a piezo-width PW = 600 mm such a voltage independent frequency maximum does not exist. Here, the actively influenced eigenfrequency maximum is calculated for the smallest piezo-size and lowest voltage. The minimum eigenfrequency appears for both piezo-widths in case of the largest piezo-element and the highest voltage.

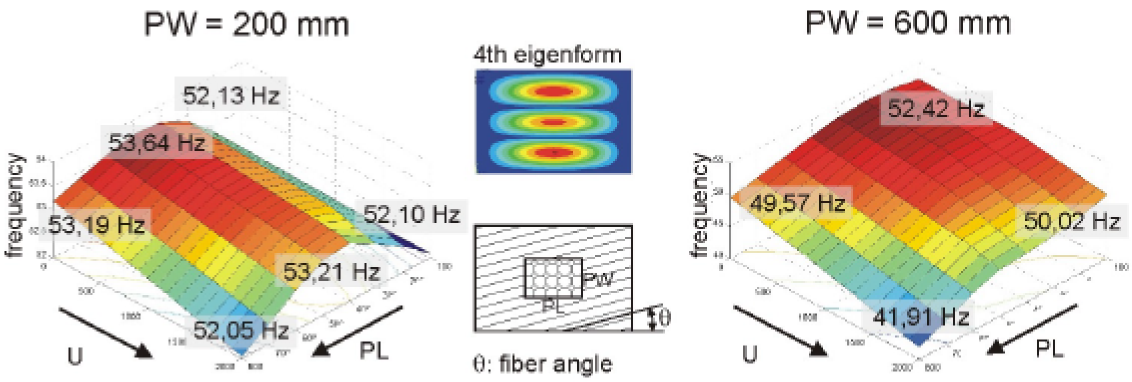

Fig. 6: Influence of piezo-size (PL, PW) and voltage (U) on the actively damped eigen-frequency of a CFRP-plate (q=0°)

The reduction of the eigenfrequencies induced by the piezoelectric field is caused by the “negative” polarization of the applied electric field (negative z-direction). Thus, the piezo-elements show an extension resulting in a lower “active” composite stiffness and therefore lower eigenfrequencies. The formation of a frequency maximum for a piezo-width PW = 200 mm is mainly caused by a higher composite stiffness to mass ratio due to the growing anisotropic piezo-surface. Furthermore, the described effects are strongly dependent on the eigenform. Consequently, a detailed modal analysis considering piezo-electric effects has to be performed for each special application.

CONCLUSIONS

The rapidly progressing development of new function-integrating lightweight structures leads to an increasing use of weight reduced solutions with high performance textile composites. There, the growing customer demands on comfort and safety require not only high specific lightweight characteristics but also reduced vibrations and sound radiation. The optimal use of the anisotropic material characteristics of textile composites with integrated piezo-ceramic actors here offers the possibility to achieve a clearly increased composite damping and improved sound radiation behavior. The main precondition for the development of such active textile composite structures is a holistic design, which includes not only the anisotropic material and damping characteristics but also a physically based coupling of mechanical, electrical and acoustical fields. This vibro-acoustic design strategy is provided by the developed material-adapted simulation methods. Using analytical vibro-acoustic simulation models for the passive composites and material-adapted numerical calculation methods for the actively damped composites a quick determination of the structural-dynamic and acoustic property profiles in dependence of the main composite and actor-specific design parameters is possible. The results of the performed sensitivity analyses make a contribution to a material-adapted development of structural-dynamic and acoustically tailored active textile composites for high technology applications within engine and vehicle construction.

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