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Vibro-Acoustic Characterisation of Thermoplastic Fibre Reinforced Composites

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ABSTRACT

Improving efficient products and processes, endless fibre-reinforced composites offer a high potential in adjustability of material parameters and aptitude for mass production at the same time. Due to high material costs and complex manufacturing techniques, a widespread application has not started yet. Dynamic properties, however, are essential in requirement specifications for newly developed devices caused by an increasing dynamic and acoustic sensitivity of lightweight structures. The shortage of natural resources demands for efficient and low-cost materials with excellent mechanical properties, both, for dynamic and for static load cases. In this paper, static and dynamic material properties of thermoplastic composites with glass- and carbon-fibre reinforcements are determined. In detail, the investigations are based on the experimental modal analysis of plate specimens to examine the elastic engineering constants and characterise the damping behaviour of the composites. Furthermore, the acoustic analysis, tests of free oscillation of thin beams and Finite-Element-Models of experimental setups are for the validation of the material parameters. The results of all investigations are compared to state-of-the-art metal-based composites with plastic mid-surfaces being typically used for reduction of structure born noise and vibration.

INTRODUCTION

The upcoming complex requirements for future component parts, which are joining the claimed efficiency in material and energy, will lead to composite materials and structures with adjustable characteristics and functions. Fibre reinforced plastics (frp) offer a lot of efficient and cost-saving technologies to produce even large plastic products, such as bulkheads, roofs, engine oil-pans and further more. The combination of continuous fibre reinforcements and thermoplastic matrices leads to high stiffness structures being most suitable in automotive applications, such as body panels and pans, which tend to be easily induced to vibrations and conduct solid-borne sound. According to their high damping properties polymer systems are used in a wide range to enhance the acoustic properties of dynamically loaded devices. The system can be adapted to the individual load-case achieving maximum lightweight potential, due to anisotropic material properties. On the contrary, stiff metal sheets with thin plastic mid surfaces are available. Due to the high costs the applications are restricted to special applications.

PREVIOUS WORK

A lot of effort has been spent to develop acoustic optimised metal sheets using thin polymer films as interlayer for sound decoupling. The synthesis of aligned fibre-reinforced plastics with metal sheets to hybrid multi-layer composites offers high freedom of design to vary the lightweight-properties. Especially material damping and acoustic emissions as well as failure tolerance can be enhanced, compared to the single components. The reduction of sound radiation has also been the focus of interests for the first large-volume production of metal composites with unfilled polymers [1-3]. Commercial available hybrid composites under the trade name Bondal® consisting of steel and visco-elastic acrylic layers are mainly used to reduce actively solid-borne sound. While vibration face-sheets are sliding slightly on each other and vibrational energy is dissipated due to shear in the polymer layer fraction. The vibro-acoustic coupling causes solid-borne sound level and can be reduced up to 20 dB compared to conventional steel [2]. The same goals were focused for the development of aluminium-polymer composites (e.g. Alucobond®, Dibond®) [6]. Suchlike sandwich structures offer increased flexural rigidity and flexural strength, whereas the mechanical in-plane properties like specific strength and stiffness are clearly decreased.

As a further enhancement of the lightweight-properties and failure tolerance multi-layer composites with aligned fibre-reinforced thermosets have been developed, counting frp-metal-composites Glare (Glass Fibre Reinforced Aluminium), Carall (Carbon Fibre Reinforced Aluminium) and TiGr (Titanium Graphite Laminate). Based on numerous research studies the manufacturing and joining processes were optimised [7-9] and the specification of elastic and strength characteristics were carried out [7, 10-16].

Though, using frp with thermoplastic matrix offers a wide range of advantages in manufacturing such as low cycle times and the ability of mass production. Moreover endless fibre reinforcements lead to high stiffness and low density at the same time.

Nevertheless an increasing lightweight potential always leads to a higher sensitivity for structure borne sound emission. Thus, the prediction of the dynamic behaviour using the Finite-Element-Method is essential for the application in all fields of engineering, automotive or aircraft industries.

In this paper different structure and air borne sound measurements are used to determine the dynamic properties of thermoplastic frp and are compared to Finite Element Simulations.

FREE VIBRATIONS OF VISCOUS UNDERDAMPED BEAMS

Basically, the material properties have been determined measuring free vibrations of cantilever beams. It enables an easy validation due to detailed theoretical descriptions and a dominant first natural frequency.

Experimental setup

All cantilever beam specimens are of the same length and width. The vibration direction has been chosen horizontal to avoid the influence of gravity and a zero drift. The velocity is measured by a laser-scanning vibrometer to assure a high accuracy and eliminate effects of additional masses likewise using accelerometers. The whole measuring equipment consists of (see):

- specimen holder (1) with force sensor (2) and specimen (3)
- a Laser-Doppler-vibrometer (out-of-plane) with a single-point sensor head Polytech® OFV-503 (4)
- a modular vibrometer controller Polytech® OFV-5000 using the digital broadband velocity decoder VD-09 and the broadband digital displacement decoder DD-900
- A-D-converter National Instruments® PXI 5922
- PC for data acquisition and interpretation within National Instruments® LabVIEW

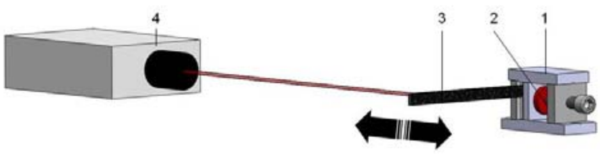

Figure 1. Measuring setup: beam in specimen holder and Laser-Doppler-Vibrometer

Free vibrations of cantilever beams

Within the experiments bending stiffness and the damping behaviour have been determined measuring the free vibrations of a beam. Concerning [17,18,19], the natural frequency of a beam is

$$\omega_j = \lambda_j^2 \sqrt{\frac{E \cdot I}{\rho \cdot A \cdot l^4}} \tag{1}$$

whereas the dimensions length l and height h of the beam are essential. Using the constraints of a cantilever beam leads to the frequency equation of λ

$$1 + \cos\lambda \cdot \cosh\lambda = 0 \tag{2}$$

with the solutions λ_j shown in **Table 1**.

Table 1. Solutions of the frequency equation for cantilever beams

j	1	2	3	> 3
λ_j	1.8751	4.6941	7.8548	$(2j-1)\frac{\pi}{2}$

In this case, the first natural angular frequency has been used to determine the material parameters.

$$\omega_1 = 2\pi \cdot f_1 \tag{3}$$

Using homogenous specimens of rectangular cross section geometry, the beam oscillating in height direction leads to an independence of the YOUNGs modulus from its width:

$$E = \frac{48\pi^2}{\lambda_1^4} \frac{f_1^2 \cdot \rho \cdot l^4}{h^2} \tag{4}$$

Moreover, damping has been reasonably assumed to be viscous, so the natural angular frequency is

$$\omega = \omega_1 \sqrt{1 - \vartheta^2} \tag{5}$$

with the damping ratio

$$\vartheta = \frac{\delta}{\omega_1} \tag{6}$$

or the decay constant δ describing the exponential abatement in the equation of motion.

$$q(t) = C \cdot e^{-\delta t} \cdot \cos(\omega t - \alpha) \tag{7}$$

Last, the logarithmic decrement can be used for the characterisation of the viscous damping, too.

$$\Lambda = \frac{2\pi\vartheta}{\sqrt{1 - \vartheta^2}} \tag{8}$$

Experimental results

The experiments have been done using three to six specimens per material. All specimens have been excited ten times with a small displacement. In general, the reproducibility of the measurements was very good. Within the results of one single specimen, the error was less than 0.5% for the determination of the natural frequency and 0.05% for the damping ratio. Comparing different specimens, the reproducibility of the natural frequency still is less than 5% whereas the decay constant varies up to 13%. It has to be mentioned that there is a significant influence of the amplitude and way of excitation examined. Thus, the excitation has been kept constant for all the experiments.

Comparing Figure 2 and Figure 3 the time domain signals show the typical viscous decay characteristic. Thus, this experiments have been used to validate the YOUNGS modulus of the specimens and to determine the decay. Furthermore, the dominant first natural frequency could be observed in the frequency domain leading to the damping ratio and the logarithmic decrement respectively constants (Table 2).

The intrinsic lines presented in Figure 2 and Figure 3 show a good coherence. So the predicted mechanical model is valid for data processing of the measured results.

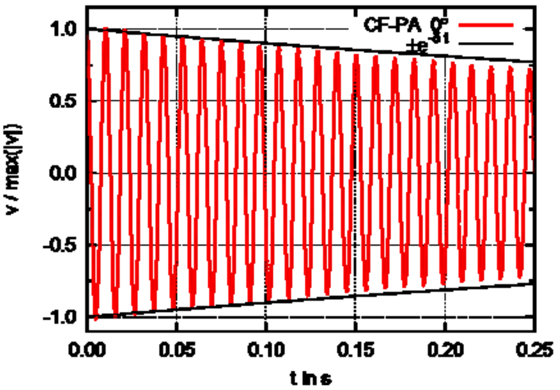

Figure 2. Decay curve of CF-PA-0° beam and exponential intrinsic line

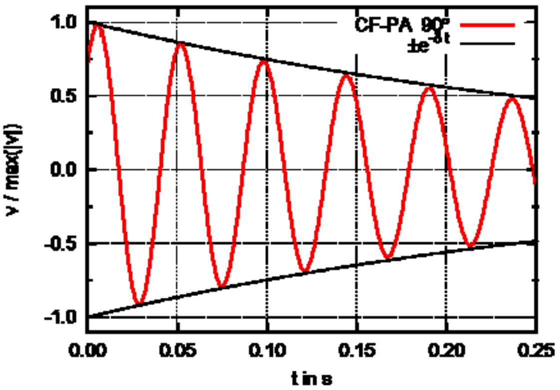

Figure 3. Decay curve of CF-PA-90° beam and exponential intrinsic line

Due to the anisotropic material behaviour, the YOUNGS modulus, natural frequency, decay constant and damping ratio show a significant dependency on the fibre angle (compare Figure 4 and Table 2).

Table 2. Measurement results of free cantilever beam vibrations

	CF-PA 0°		CF-PA 45°		CF-PA 90°	
	MV	STD	MV	STD	MV	STD
f in Hz	87,7	4,4	38,9	1,6	21,8	0,2
δ in s^{-1}	1,02	0,01	4,76	0,15	2,91	0,04
∂	0,002	2,8E-04	0,020	7,0E-04	0,021	3,9E-04
E in N/mm^2	98235	5501	16858	1347	5293	89

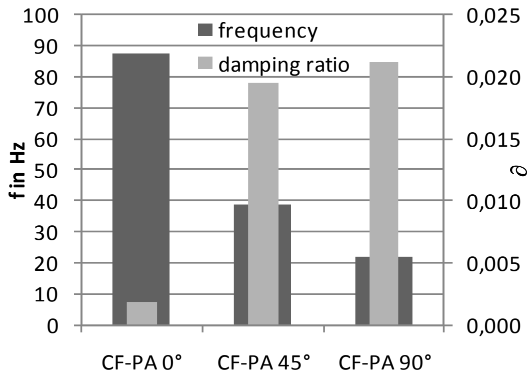

Figure 4. Frequency and damping ratio of CF-PA depending on the fibre angle

FREE VIBRATIONS OF THIN PLATES

To examine the coupling of structural vibrations and sound pressure, free vibrations of thin plates have been measured. Therefore the accelerations and the sound pressure close to the specimen have been determined and compared.

Experimental setup

Two thin plates of CF-PA and of Bondal have been suspended on one single string. The plate dimensions are 250 x 98 x 1.25 mm with fibres placed lengthwise for the CF-PA plate and 300 x 300 x 1 mm for the Bondal plate. An accelerometer (acc.) has been attached on the bottom of the plate as shown in Figure 5. Moreover, two microphones (mic. 1 and mic. 2) have been placed close to the specimen surface. The distance of about 10 cm is necessary to avoid any contact caused by pendular vibrations of the plate after excitation.

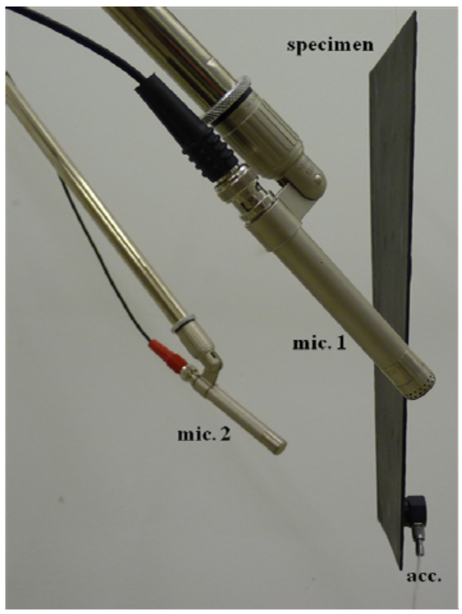

Figure 5. Experimental setup: modal analysis of structural vibrations and sound emission of thin plates

The plate is excited using a modal hammer with a steel cap using this signal as trigger. The whole measuring consists of:

- two ICP measuring microphones MICROTECH GEFELL MM210 (Nr. 1583, 1575)
- 4-chanal frequency analysis SINUS Harmonie BNC
- Software SINUS Samurai v. 1.7.16
- amplifier 600 W SOLTON SPA600X
- single axis accelerometer KISTLER
- Kalibrator LARSON DAVIS CAL2000, Nr. 6990

To assure measuring accuracy all measurements have been done in a vibro-acoustic lab with low background noise.

Experimental results

Every measurement has been repeated five times, recording the time domain signals of the two microphones and the accelerometer time synchronic. All sensors have been sampled with 44.1 kHz. Comparing microphones and accelerometer, the three signals show a similar behaviour in time and frequency domain (Figure 6 and Figure 7) for the CF-PA plate.

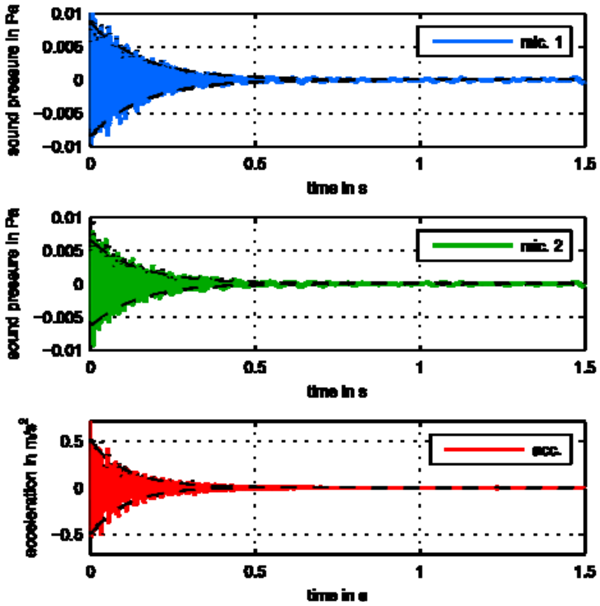

Figure 6. Time domain signals of free vibrations of the CF-PA plate: comparison of microphones and accelerometer

In detail, the time domain signals differ slightly in the decay behaviour. As before, the decay constant has been determined but shows significant standard deviations compared to the natural frequencies.

The FFT shows less noise in the frequency share. Nevertheless the natural frequencies can be detected within the signal of the accelerometer and as major peaks in the airborne sound. In addition, both microphones show noise peaks up to 100 Hz and hide one important peak above 400 Hz.

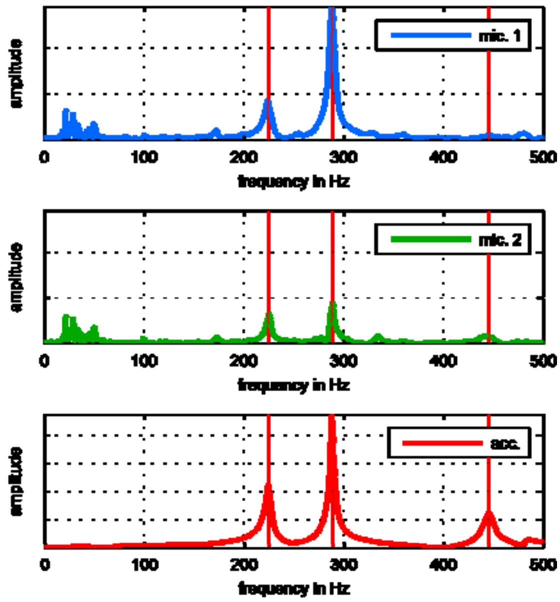

Figure 7. Frequency domain signals of free vibrations of the CF-PA plate: comparison of microphones and accelerometer

Even though all measurements have been excited with slightly different amplitudes, the natural frequencies are determined with a high accuracy using the accelerometer (Figure 8) or the microphones (Figure 9). In contrary the time domain decay behaviour is sensitive to the excitation and measurement system.

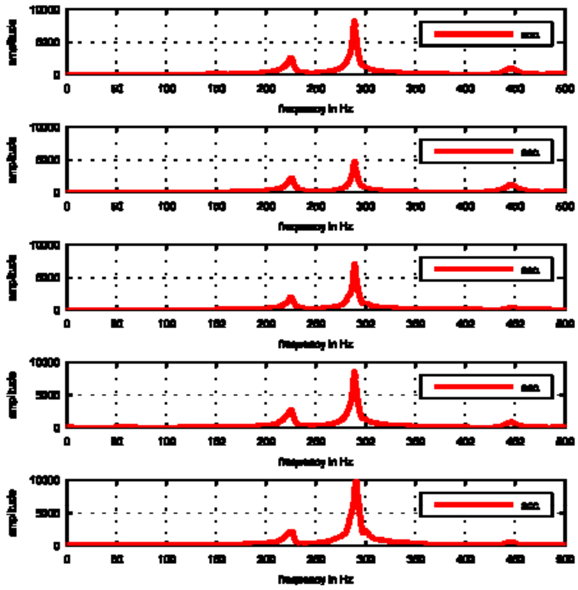

Figure 8. Frequency domain signals of the accelerometer: reproducability of measuring results

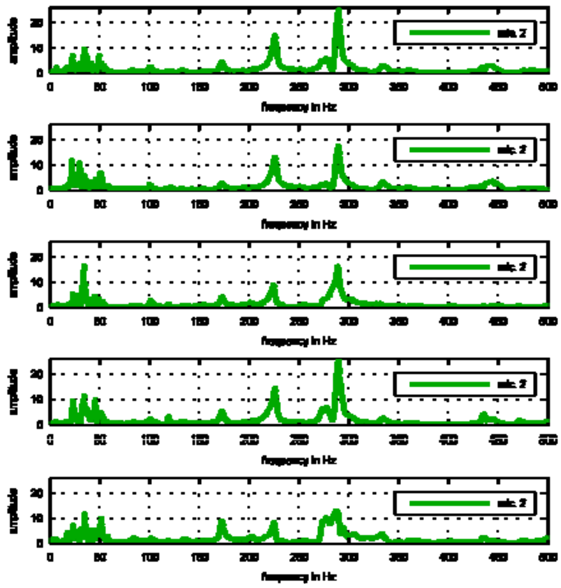

Figure 9. Frequency domain signals of one microphone: reproducibility of measuring results

According to that, sound pressure is a possibility for a simple determination of the frequency behaviour of free vibrations of thin plates with roughly good accuracy, as long as the major natural frequencies are above the low-frequency noise.

Moreover, the experiments have been repeated with the Bondal plate. Figure 10 shows the high damping in the decay curves, especially for the accelerometer signal. In this case the exponential description of the decay behaviour is not precise enough. Furthermore the comparability of structure plate vibrations and air borne sound cannot be taken for granted.

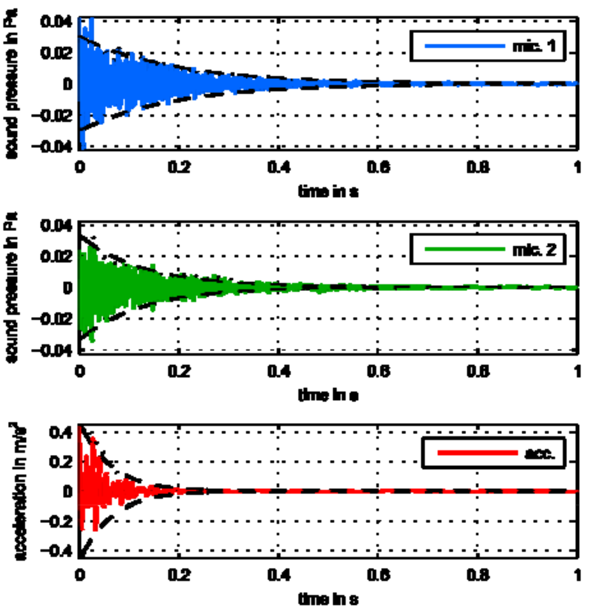

Figure 10. Time domain signals of free vibrations of the Bondal plate: comparison of microphones and accelerometer

The FFT results of microphones and accelerometer only show some peaks similarly. Caused by small sound pressure amplitudes for the highly damped material structural vibrations cannot be determined from the acoustic measurement.

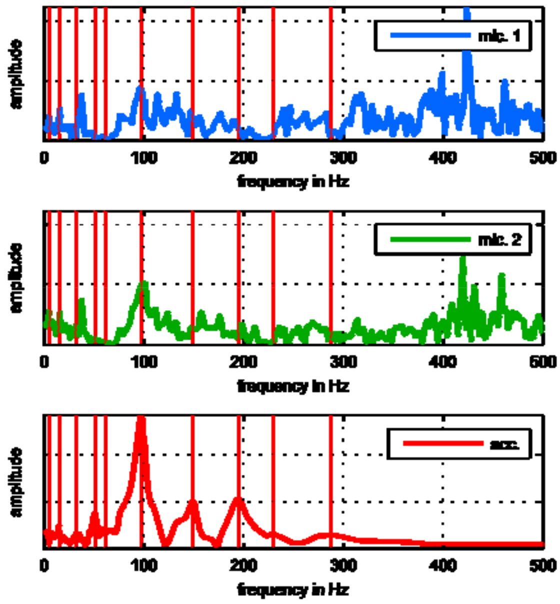

Figure 11. Frequency domain signals of free vibrations of the Bondal plate: comparison of microphones and accelerometer

In contrary, thin steel plates with minor damping than CF-PA showed a very good comparability. Thus, sound measurement is only applicable for materials with small damping ratios.

FINITE ELEMENT SIMULATIONS

Both, beam and plate vibrations have been simulated with Finite-Element-Method using the examined material parameters.

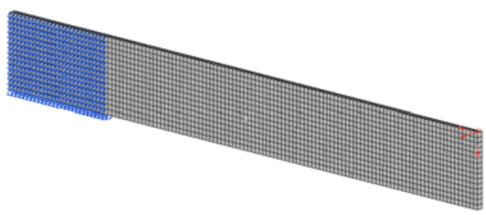

Figure 12. FE-model of beam specimens

Linear hexahedron elements with composite material properties have been used to describe the linear elastic behaviour. Displacement constraints have been used to determine clamping of the cantilever beams whereas the plate has not been constrained at all.

The simulation results correspond to the experiments in predicting the natural frequencies very accurate for the beam specimens. For the CF-PA 0° beam 85 Hz have been determined by FEM compared to 88 Hz from the experiments. Similarly the results for CF-PA 90° specimen were 20 Hz in FEM and 22 Hz in the measurements (Figure 12).

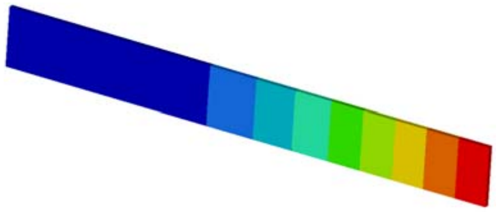

Figure 13. First natural mode of the CF-PA beam

Furthermore, the beam model has been expanded to plates using the same material properties, element types and solver options. Figure 14 shows reasonable results for the natural modes of the CF-PA plate whereat the corresponding frequencies of 91 Hz, 173 Hz, 253 Hz, 258 Hz, 315 Hz and 479 Hz in the observed frequency range differ significantly from the experimental results (compare Figure 7, Figure 8 and Figure 9).

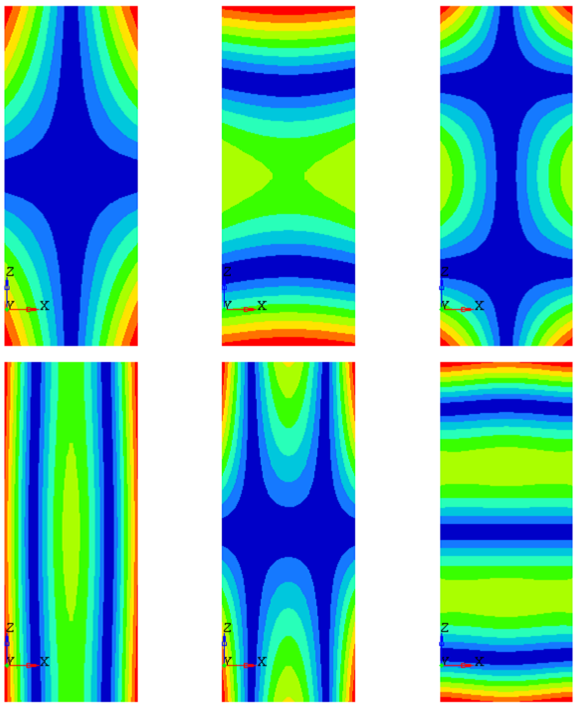

Figure 14. FE-results: Natural modes of the CF-PA plate

SUMMARY AND OUTLOOK

The high stiffness and damping effects of thermoplastic frp have been proofed showing opposed tendencies depending on the fibre angle. In comparison, metal-based composites offer very high damping for bending modes having a significant higher density.

In general, free vibrations of composite beams and plates can be used to determine the elastic and dynamic properties. Nevertheless, high damping ratios limit the possibility of using air borne sound from the excited structures as measuring component. Basic finite element models confirm the validity of the determined natural frequencies for the beams but offer a lot of potential portraying anisotropic plates and the time-depended decay behaviour.

Further investigations using cantilever beams will focus on parameter studies to examine possible non-linearity's, such as clamping forces, excitation amplitudes and frequency dependency using forced vibrations. Moreover vibrations of beams and plates have to be reproduced more detailed using different non-linear Finite-Element models. Last, BEM simulations can help to predict the sound emission of this kind of materials.

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