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Experimental and Numerical Studies of Thermoelastic Damping

Christin Zacharias[1]* and Carsten Könke[1]

Micro Abstract
The use of correct damping parameters is a decisive aspect in the numerical simulation of dynamical problems and indispensable to predict and reduce reliably vibration amplitudes. In this contribution, experimental and numerical studies to identify damping coefficients of simple geometries are presented. In the experiments the focus was set to measure the pure material damping excluding all disturbing environmental influences. In the numerical investigation, the thermoelastic approach was used.

[1]Institute of Structural Mechanics, Bauhaus-University Weimar, Weimar, Germany
***Corresponding author**: christin.zacharias@uni-weimar.de

Introduction

Damping is defined as the irreversible transition of mechanical energy into other forms, mostly thermal energy [7]. In all dynamic processes damping has a considerable effect on the amplitude, the time history or even the existence of vibrations. Hence determing the sources and the intensity of dissipation is important for a wide variety of applications.
To classify damping mechanisms the different physical causes should be analysed. It is often distinguished between internal and external damping [7]. Basically external damping includes all effects outside the system boundaries, e.g. air damping, acoustic radiation or friction at the bearings. The term „Internal damping“ refers to all dissipating effects within the system boundaries, e.g. the material damping or contact-surface friction between parts of the system. In this work we focus on the measurement and calculation of material damping, therefore the other damping phenomena are eliminated as well as possible.
Material damping is caused by anelastic material behaviour and several underlying physical effects. Lazan gives a good review on this topic [3]. Probably the most significant cause of material damping is the thermoelastic effect. Zener shows already in his work of 1937 the meaning of this relation and describes an approach which is used till today [8]. The theory was often adapted and developed for a wide field of applications, e.g. Micro- and Nanomechanical resonators. Advanced theories can be found for example in [4] or [2]. The base of the thermoelastic effect is the deformation caused by flexure. During the bending of a component a strain gradient occurs over the thickness. This leads to an adiabatic change in temperature (Thomson's Principle):

$$\Delta T = \sigma\alpha\frac{T}{\rho C} \tag{1}$$

where ΔT is the temperature change, σ is the flexural stress, α is the coefficient of thermal expansion, ρ is the density of the material and C_p is the heat capacity. Since stress and strain have different signs there is a temperature gradient along the thickness of the specimen. The compensating heat flow causes further thermal strain, which mainly characterizes the resulting heat flux [1]. The loss factor Q^{-1} is defined as the quotient of the damping energy and the total strain energy [3]. This results in an expression for the maximal energy loss under the given flexure:

$$Q^{-1} = \frac{\alpha^2 ET}{\rho C} \tag{2}$$

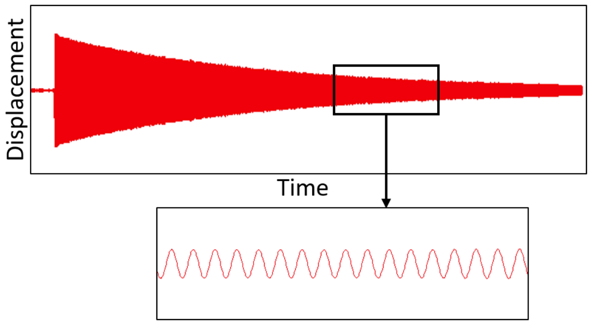

Figure 1. Viscous decay curve

where E is the isothermal Young's modulus. However the real energy dissipation is always less than the one described in equation 2, because it depends strongly on the frequency. In the equation developed by Zener this dependency is considered by the calculation of a peak frequency f_0 which leads to a maximum in energy dissipation. A detailed derivation of these expressions can be found in [8].

$$Q^{-1} = \frac{\alpha^2 ET}{\rho C} \cdot \frac{f f_0}{f^2 + f_0^2} \tag{3}$$

$$f_0 = \frac{\pi \lambda}{2d^2 \rho C} \tag{4}$$

where f is the considered frequency, λ is the thermal conductivity and d is the thickness of the specimen.
To prepare damping analyses on complex components, simple geometries are ivestigated experimentally and numerically. In a first step a series of thin aluminium beams is designed. They show essentially one-dimensional behaviour and they are therefore suitable to study the damping in pure beam flexure. The specimens have an increasing thickness from 1 mm to 9 mm and they are dimensioned to have the same range of eigenfrequencies in each beam to get comparable results concerning the geometry. Furthermore the experiments and simulations are extended to a two-dimensional structure. Therefore, a thin plate with a thickness of 3 mm was produced to investigate the plate modes. In this work the results of the beams and the plate are presented.

Experimental Studies

The most important aspect in the experimental setup is to avoid all unintented influences to damping and to measure the pure energy dissipation within the material. To eliminate damping sources from the surrounding air, all experiments are performed in a vacuum chamber. The limited space leads to special requirements concerning the support and excitation of the specimen as well as the measurement of the vibrations. A decisive factor for the damping of a system is the friction especially at joints and bearings. To avoid this influence, the mode shapes are determined in a modal analysis and the suspension is set to the nodes of vibration of the relevant eigenmode. This implies that a new experimental setup is necessary for every mode shape. To realize the suspension, small bore holes were drilled in the nodes of the respective eigenmode ($d \approx$ 2 mm). This kind of support pretends free vibration modes and prevents rigid body movement. The excitation of the specimen is achieved by an automatic impulse hammer, which is installed inside the vacuum chamber and controlled from the outside. This method enables a minimum of contact. To excitate the desired mode shape at the best, the point of excitation should have a high deflection in the eigenform. The measurement of the vibration is contactless from the outside and realized by a laser vibrometer. The velocity of the vibration is measured at one charcteristic point of the eigenmode and a decay curve is recorded. In the case of sole material damping there is a viscous decay, i.e. an exponential envelope curve can be calculated:

$$D(t) = C \cdot e^{-\zeta\omega t} \tag{5}$$

where ω is the angular frequency, C is the initial displacement and ζ specifies the damping coefficient that we use as a measure to describe and compare the damping. The damping coefficient corresponds to half the loss factor Q^{-1}. Every experiment was conducted at least five times and the diagrams show the mean values of the results.
Figure 2 shows the results of the experiments on the aluminium beams with a thickness of 1 mm to 9 mm. For each specimen the first three eigenfrequencies were considered. The diagram includes the theoretical approach according to Zener [8] as well. It points up that the experiments match the theory very well. There is a dependency of the damping coefficient on the frequency. In higher frequency ranges the material damping becomes lower. Furthermore the dependency on the dimensions becomes apparent. The thinner beams show higher damping values, however, the differences become smaller within the thicker specimen.
The experimental results of the plate are shown in figure 3 together with the results of the numerical studies. It can be seen that there is no clear relation between frequency and damping coefficient like in the one-dimensional case. However, the diagram shows a correlation of the mode shape and the damping ratio. Especially by considering mode 1 to mode 4 it becomes clear that the bending mode shapes that deform similar to the one-dimensional beams, show higher damping values than mainly torsional mode shapes.

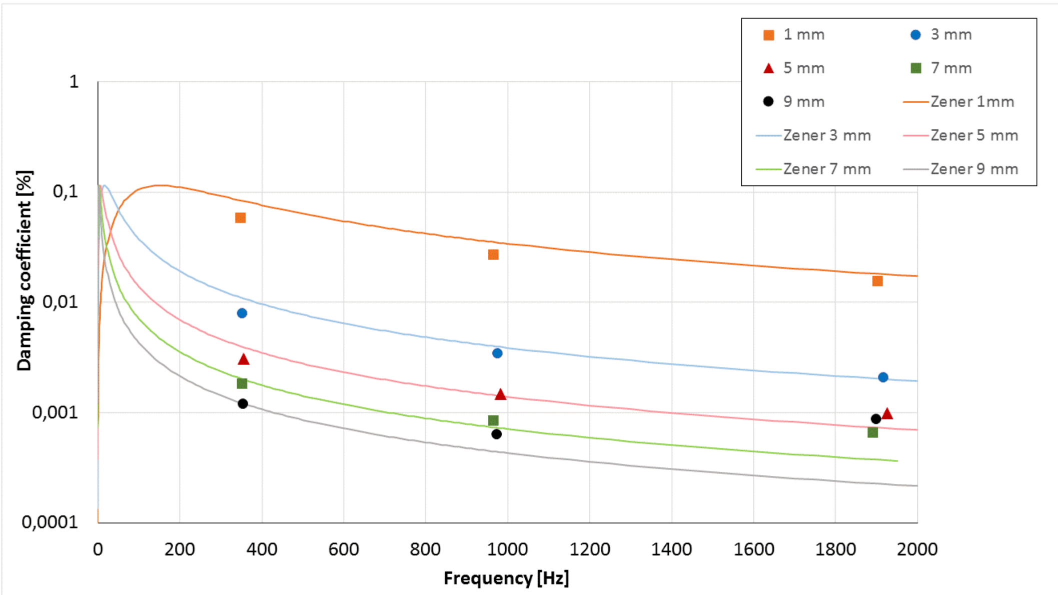

Figure 2. Experimental damping coefficient of thin aluminium beams (thickness 1 mm to 9 mm) and comparison to the theoretical damping according to ZENER [8].

Numerical Simulation

To verify the results numerically we used a finite-element analysis developed by Serra and Bonaldi [6]. The underlying approach is the coupling of the mechanical behaviour with the thermal properties. The shape functions of the elements do not only consider the nodal displacement but also the temperature shift. This approach is already implemented in ANSYS with 20-node brick elements. To calculate the damping coefficient in dependency on the frequency, a frequency domain analysis was performed with a harmonic excitation in the range from 0 Hz to 2000 Hz.

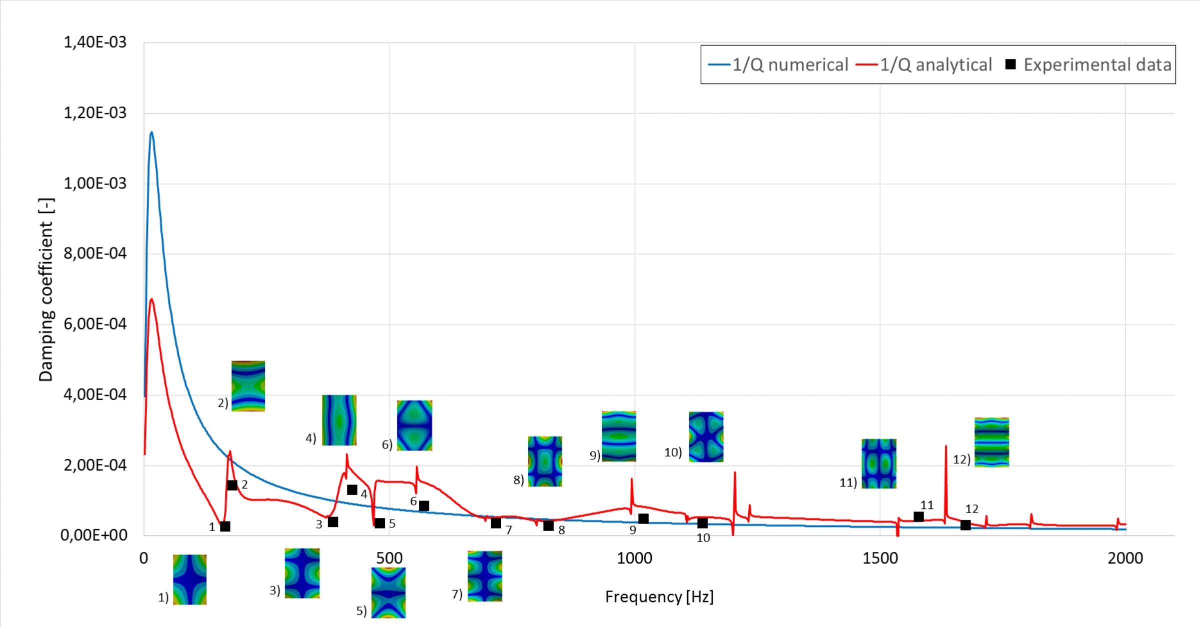

Figure 3. Numerical and experimental results for a thin aluminium plate (thickness 3 mm). For comparison, the analytical solution according to Zener for a specimen thickness of 3 mm is shown.

Figure 3 shows the results for the aluminium plate with a thickness of 3 mm. The loss factor Q^{-1} is calculated by dividing the imaginary solution by the real solution in every substep integrated over the whole model. Therefore poles appear at the resonant frequencies. For comparison the diagram also includes the analytical solution for a beam with the same thickness according to Zener [8]. This graph does of course not map the two-dimensional plate modes. First of all it can be noted that the numerical solution matches the experimental results very well. As mentioned earlier, there is a strong correlation between the damping coefficient and the mode shape. In the flexure dominated mode shapes, e.g. eigenmode 2 and 4, the damping coefficient is significantly higher. As a result the damping ratio varies strongly in the lower frequency range. In the higher frequency range the mode shapes show more and more mixed deformation and the damping coefficient only varies a little. Since the finite element formulation including the thermoelastic coupled shape functions is very time and CPU-consuming we are looking for a faster and simpler solution to consider the thermoelastic effect. A promising approach could be the separation of different mode shape parts.

Conclusions

In the experimental damping determination all influencing damping sources from the environment have to be eliminated to get the pure material dissipation. This includes the air resistance, acoustic emissions into the surrounding or friction in bearings and joints as well as a possible influence of measuring instruments. In a simplified approach we assume viscous dissipation behaviour, i.e. the decay curve can be fitted by an exponential function. The material damping can be calculated theoretically very well with the thermoelastic approach. In case of one-dimensional bending of beams even the analytical formula developed by Zener leads to good results. To simulate more complex plate modes there exist finite elements with thermoelastically coupled shape function. They show good results in comparison to the experimental data, but the simulation is very time-consuming.

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