Electrostatic tuning of the bending stiffness of simple, slender multi-layer composite structures.

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ABSTRACT

Vibration control and suppression in structures plays a central role in the extension of their service life and improvement of their reliability. While in many cases the solution of this problem implies the introduction of external damping devices, it is also conceivable to adaptively modify their vibratory properties, so that the occurrence of severe vibrations due to resonance phenomena can be curbed at its origin.

The modification of the shear stress transfer at the interface between the core and the faces of a sandwich beam has been shown to have a remarkable effect on the bending stiffness of the structure. Such modification can be obtained by applying a normal stress between the core and the un-bonded, electrically insulated faces of the sandwich by means of a strong electrical field.

An intermediate behavior between fully bonded and un-bonded layers in terms of inter-laminar shear stress can be achieved by temporary electrostatic bonding of the components. The outlined approach to the reduction of transversal vibrations in thin multi-layer beams is promising and can in principle be applied to multi-layer plates.

Keywords: Vibration control, adaptive system, sandwich structure, bending stiffness

1. BENDING STIFFNESS OF SIMPLE AND SANDWICH CANTILEVER BEAMS

1.1. Static behavior

The ability to modify the vibratory properties of a structure, such as its natural frequencies represents a useful addition to the tools available for effective vibration suppression. In order to achieve the ability to modify the vibratory properties of a structure, such as a simple cantilever beam, an influence onto its stiffness D in (1) has to be exerted.

$$D = EI_z = E \iint y^2 dS \quad (1)$$

In the case of structural elements made of one bulk material, such as a simple cantilever beam, a modification of the behavior of the system can be achieved either by a modification of the elastic properties of the constituting material or by a change of the geometry, typically the cross-section, of the beam. In the first case an effect on E in (1) will be achieved, in the second case I_z will be affected.

At a fundamental level, the elastic properties of solids are governed by the electric attraction and repulsion forces between atoms of a material. The shape of the potentials that determine the elastic properties of the material is mainly determined by the nature of the bond (metallic, covalent, ionic etc.), the geometric distribution and the electronic properties of the atoms. [1] For most engineering materials, the options for a substantial modification of their elastic properties are limited.

Special materials such as shape memory alloys (SMA) are an example of a case where it is possible to modify the behavior of a material. The use of SMA has already been proposed for the implementation of adaptive devices, such as an adaptive tuned mass damper. [2] [3] [4] NiTi SMAs exhibit a change in their (pseudo)elastic properties up to threefold in connection with a body centered cubic (*bcc*) to monoclinic phase transformation. The phase transformation takes place upon heating or cooling of the material and transfer of the corresponding phase transformation energy. [5]

A change in bending stiffness due to a modification of the cross-section of the beam (e.g. from a circular cross-section to an elliptical cross-section with the same cross-sectional area, as shown in Figure 1) is theoretically possible but presents practical difficulties in its realization, especially in the case of engineering materials presenting elastic moduli of the order of tens of GPa.

Figure 1: Effect of deformation of a circular cross-section to an elliptical cross-section on the second moment of area I_z .

In sandwich structures, thin, stiff faces (typically made of CFRP or another high strength, high modulus material) are glued on a lightweight, low modulus thick core to obtain a composite structure with a high bending stiffness. The transfer of shear stresses at the interfaces between faces and core plays a very important role in the stiffening of the sandwich structure. For common engineering applications it is a requirement that the stiff faces be perfectly bonded to the core, that means that a complete transfer of stresses between layers of the sandwich structure takes places. The behavior of such structures has been extensively investigated. [6] Some of the most important relations describing the properties of such structures are summarized in the following section.

Figure 2: Schematic representation of a sandwich cantilever beam (length l , width b , face thickness t_f , core thickness t_c) subject to a force P at a distance l from origin.

The flexural rigidity of a sandwich structure is given by:

$$D_{sandwich} = \int E z^2 dz = 2 \cdot \frac{E_f b_f t_f^3}{12} + \frac{E_f b_f t_f d^2}{2} + \frac{E_c b_c t_c^3}{12} = 2D_f + D_0 + D_c \quad (2)$$

Where E_f and E_c are the Young's moduli of the faces and core, respectively, t_f and t_c are the thicknesses of the faces and core (Figure 2), respectively and b_f and b_c the corresponding widths, and $d = t_f + t_c$.

Additionally, if the core of the sandwich is isotropic, a shear contribution to the transversal deformation has to be considered. With a shear stiffness S

$$S = \frac{G_c b_c d^2}{t_c} \quad (3)$$

the total deformation can be written as a sum of bending and shear deformation. In the case of a discrete transversal force P acting on the beam at a distance l from the origin, the total deformation can be written as:

$$w_{sandwich} = w_{bending} + w_{shear} = \frac{Pl^3}{6D_{sandwich}} \left[3 \left(\frac{x}{l} \right)^2 - \left(\frac{x}{l} \right)^3 \right] + \frac{Px}{S} \quad (4)$$

The shear stress between faces and core is found to be

$$\tau_i = \frac{T_x}{b_f D_{sandwich}} \cdot \frac{E_f b_f t_f d}{2} \quad (5)$$

with $T_x = dM_x/dx$, where M_x is the bending moment acting on the structure.

$$\tau_i = \frac{T_x}{b_f D_{sandwich}} \cdot \frac{E_f b_f t_f d}{2} = \frac{T_x E_f b_f t_f d}{b_f \left(2 \cdot \frac{E_f b_f t_f^3}{12} + \frac{E_f b_f t_f d^2}{2} + \frac{E_c b_c t_c^3}{12} \right)} \approx \frac{P}{b_f d} \quad (6)$$

In most cases, the first and the third term in $D_{sandwich}$ are very small, compared to the middle term and can therefore be neglected, yielding the approximation on the right hand side of (6).

The extreme case of de-bonding (i.e. no adhesion between core and faces on the whole contact area between them) is hardly of any relevance for practical purposes. In this case, the shear stress transferred at the face-core interface is none. Simple considerations and inclusion of the boundary conditions show that for a loose bundle composed of the same faces and core as considered in (2) the stiffness is

$$D_{bundle} = \frac{2E_f b_f t_f^3}{12} + \frac{E_c b_c t_c^3}{12} \quad (7)$$

Again, for a core with a sufficiently low bending stiffness, the second term can be neglected in a first approximation, so that the ratio between the stiffness of a sandwich and the stiffness of a loose bundle can be calculated as:

$$\frac{D_{sandwich}}{D_{bundle}} = \frac{2 \cdot \frac{E_f b_f t_f^3}{12} + \frac{E_f b_f t_f d^2}{2} + \frac{E_c b_c t_c^3}{12}}{\frac{2E_f b_f t_f^3}{12} + \frac{E_c b_c t_c^3}{12}} \approx 3 \left(\frac{t_c}{t_f} + 1 \right)^2 \quad (8)$$

For an accurate comparison of the deformation behavior of a sandwich with bonded vs. un-bonded faces the shear deformation of the soft core sandwich should also be taken into consideration.

1.2. Free vibrations

The natural frequency of the free vibration of a structure depends on its elastic properties. The vibratory behavior of simple structures has been extensively investigated and described [7]. In general terms it is found that the natural frequency of an oscillator is proportional to the square root of the stiffness k of the elastic element in the oscillator and inversely proportional to the square root of the mass W of the oscillator.

Figure 3: A one mass oscillator with mass W and spring stiffness k

For the simple oscillator shown in Figure 3, the frequency of the free oscillation is:

$$f = \frac{1}{2\pi} \sqrt{\frac{kg}{W}} \quad (9)$$

Where g is the acceleration due to gravity. While this simple relationship applies to an idealized system and more detailed models are needed for real systems, the general character of the dependence of vibration frequency on the stiffness of the system:

$$f \propto \sqrt{k} \quad (10)$$

remains unchanged. In particular for beams in which no shear deformation has to be taken into account, the free vibration frequencies f_B for pure bending vibrations in the x-y plane can be written as:

$$f_B = \frac{1}{2\pi} \sqrt{\frac{D}{\rho} \left(\frac{\mu_{Bi}}{L} \right)^2} \quad (11)$$

In which the stiffness D and the density ρ of the beam are found again in a similar form as in (9). L in (11) stands for the length of the beam and μ_{Bi} is a parameter in the solution of the differential equation of motion that depends on the boundary conditions of the problem. If shear deformation is also taken in consideration, it is found that the frequency of the vibration can be approximated as follows [8]:

$$\frac{1}{f^2} \approx \frac{1}{f_B^2} + \frac{1}{f_S^2} \quad (12)$$

Where f is the vibration frequency of the beam, f_B has the same meaning as in (11) and f_S is the vibration frequency for the pure shear vibration. Similarly as in (11) f_S can be written as:

$$f_S = \frac{1}{2\pi} \sqrt{\frac{S}{\rho} \left(\frac{\mu_{Si}}{L} \right)^2} \quad (13)$$

As μ_{Bi} in (11), μ_{Si} is a parameter in the solution of the differential equation of motion that depends on the boundary conditions of the problem. In the case of a beam built in on one side, $\mu_{Bi} = (i - 0.5) \cdot \pi$ and $\mu_{Si} \approx (i - 0.5) \cdot \pi$

An in depth discussion of vibration problems transcends the scope of this work. Based on the relationships outlined in the previous sections one can assume that the vibration frequency of a beam depends both on its bending and the shear stiffness in a non-linear fashion.

2. WORKING PRINCIPLE OF ELECTROSTATICALLY TUNABLE SANDWICH BEAMS

As outlined in the previous section, the transfer of shear between faces and core makes a remarkable difference in the overall stiffness as well as in the vibratory properties of the system. In common engineering applications the bond between the components is obtained by the application of an adhesive. This generally creates a covalent or an otherwise strong bond between components and adhesive layer. The use of adhesives (Figure 4, left) presents the advantage of being permanent and comparably inexpensive. Under normal conditions the bond cannot be modified in its strength, which makes the stiffness of the sandwich structure invariable, for practical purposes.

Figure 4: Schematic representation of adhesive (left) and electrostatic (right) bonding between faces and core of a sandwich structure.

Hence, one can deduce that, if the transfer of shear stress could be influenced, the bending stiffness of the system would be also affected.

As an alternative to the use of adhesives for the transfer of shear across the face-core interfaces, stresses can be transferred by means of friction, if a sufficiently large normal stress at the contact surfaces is given. It is possible to apply a homogeneously distributed normal stress between faces and core of a sandwich, across an insulating layer with appropriately high dielectric number, if the faces and core are electrically conductive and insulated from one another, as outlined in Figure 4. The application of a strong electrical potential (in the order of a few kV) across a thin insulating layer generates fairly large normal stresses at the interface, as shown in Figure 5. The electrostatic stress σ_i at the contact surface between two planar electrically loaded bodies is given by (14):

$$\sigma_i = \frac{\epsilon_r \epsilon_0 U_{hv}^2}{2\delta^2} \quad (14)$$

Where ϵ_r is the dielectric number of the dielectric layer material, ϵ_0 is the dielectric constant of vacuum, U_{hv} is the electric potential between the electrodes and δ is the thickness of the dielectric layer material.

For example, assuming an ideal interface between faces and core and the absence of air gaps between components, normal stresses well over 100kPa can be obtained by the application of a 3500V potential across a 0.05mm poly(vinylidene fluoride) (PVDF) dielectric layer (with $\epsilon_r = 8$). Ceramic materials such as different forms of Titanium oxide present dielectric numbers exceeding 100.

Figure 5: Normal stress at the interface between two electrically conductive bodies separated by a dielectric layer as a function of layer thickness d and applied potential U_{hv} . The dielectric number ϵ_r of the dielectric layer is assumed to be $\epsilon_r = 8$ (approximately the dielectric number of poly(vinylidene fluoride), PVDF).

Based on a simple model for solid-solid friction (15):

$$\tau_i = \mu \cdot \sigma_i \quad (15)$$

the shear stress τ_i at the interface between two bodies subjected to a normal stress σ_i is assumed to be linearly proportional to the normal stresses. [9] μ varies as a function of the materials involved in the friction processes and can generally lie between 0.01 and 0.5.

The effect of normal stresses at the core-face interfaces on the stiffness of a sandwich beam consisting of two carbon fiber reinforced plastic (CFRP) faces and a silicone-elastomer core separated by a PVDF dielectric layer have been reported in [10]. The presented results showed that with increasing potential between layers, the overall stiffness of the system increased, due to the ability to transfer higher shear stresses at the face-core interface.

The goal measurements presented in the following sections is to confirm that the increased stiffness of the system leads to an increase in the first natural frequency of the beam in bending mode.

3.EXPERIMENTAL

To demonstrate the functionality of an electrostatically tunable composite cantilever beam, a sandwich beam consisting of:

- two carbon fiber reinforced plastic (CFRP) layers, each 200 mm long, 12 mm wide, 0.15 mm thick, coated on both sides with a ~0.08 mm thick poly(vinylidene fluoride) (PVDF) film, $\epsilon_r=8$
- one carbon-filled rubber core, 200 mm long, 10 mm wide, 2.3 mm thick, coated on both sides with a ~0.05 mm thick Poly(tetrafluoroethylene) (PTFE) film, $\epsilon_r=2$

was chosen as a testing object. With this configuration an effective dielectric number of $\epsilon_r=3.7$ and an effective dielectric layer thickness $\delta=0.13\text{mm}$ (sum of the thicknesses of the two layers) are obtained. The main mechanical properties of the sandwich components are listed in Table 1.

Property	Faces	Core
Material	Carbon fiber reinforced plastic	Particle filled siloxane elastomer
Thickness t [mm]	0.15	2.3
Width b [mm]	12	12
Young's modulus E [GPa]	120	~0.004
Length l [mm]	140	140

Table 1: Key properties of the components of a CFRP-elastomer sandwich beam

The effective dielectric number ϵ_r is obtained by calculating the total capacity C_{tot} of the PVDF and PTFE capacitances in series

$$\frac{1}{C_{tot}} = \sum_i \frac{1}{C_i} \quad (16)$$

and calculating back ϵ_r with:

$$C = \frac{\epsilon_r \epsilon_0 A}{\delta} \quad (17)$$

Where ϵ_0 is the dielectric constant of vacuum, ϵ_r the dielectric number of the medium in the capacitance, A the surface of the electrodes and δ the distance between electrodes.

Figure 6 shows schematically the composite beam and the set-up used for the experiments. The use of the PTFE film was necessary to avoid excessive sticking of the rubber core to the PVDF film. The carbon filled rubber core was chosen because of its comparably good electrical conductivity.

The beam was positioned vertically and fixed 140 mm from the bottom end, where two small FeNdB magnets were positioned. Their function was to make it possible to apply a transversal force $P(t)$ by means of a variable magnetic field $B(t)$, generated by a current $I(t)$ that circulated in the coils.

Figure 6: Schematic representation of the test set-up for the static cantilever bending and frequency sweep measurements.

Two series of measurements were performed:

In order to determine the static properties of the beam, as described in [10], a series of static cantilever bending tests was performed. The transversal force $P(t)$ exerted on the beam increased stepwise from $P(0)=0$ to $P(t_1)=P_{max}$ by increasing the intensity of the current circulating in the coil, then it was reduced to $P(t_2)=0$ and then further to $P(t_3)=-P_{max}$, the load was then increased back to $P(t_4)=0$ and finally to $P(t_5)=P_{max}$, thus describing a complete cycle from P_{max} to $-P_{max}$.

In order to determine the vibratory properties of the system, specifically the first natural frequency of the transversal vibration in the x - y plane frequency sweep measurements were performed. The excitation current $I(t)$ circulated through the coil was defined as

$$I(t) = \sin(f(t) \cdot t) \quad (18)$$

Where:

$$f(t) = f_o + \frac{(f_e - f_o)}{t_{sweep}} \cdot t \quad (19)$$

Where f_o is the start frequency, f_e the end frequency and t_{sweep} the duration of a sweep. The parameters chosen for the frequency sweep were $f_o=2\text{Hz}$, $f_e=30\text{Hz}$ and $t_{sweep}=500\text{s}$.

For all tests, the current $I(t)$ was supplied by a Kepco bipolar operational power amplifier BOP 20-20M. For the static tests, the amplifier was driven by the analog output tension of a National Instruments 6036 E DAQ board. For the

frequency sweep measurements the operational amplifier was driven by an Agilent 33120A Function/Arbitrary Waveform Generator.

The transversal displacement of the composite beam was measured at two by means of a Micro-Epsilon optoNCDT laser displacement sensors (marked as LDT in Figure 6).

The displacement was measured with a sampling frequency of 64 Hz for the static test and 256 Hz for the 2Hz...30Hz frequency sweep. The components of the sandwich beam were connected to a high voltage power supply (Stanford Research Systems PS 350), so that the necessary electrical potential U_{hv} could be applied between faces and core.

The complete test setup was controlled from a LabView interface on a PC via an IEEE 488 bus.

4. RESULTS AND DISCUSSION

4.1. Static tests

Figure 7: Force-displacement diagrams of the cantilever bending tests performed on the electrostatically tunable sandwich beam described in section 3. The value of S corresponds to the estimated average slope of the force-displacement curve dP/dw .

The diagrams in Figure 7 show the hysteretic behavior of the system, especially for low values of the potential U_{hv} . The noticeable hysteresis can be explained by the presence of friction processes at the contact surface between faces and core. For U_{hv} values starting at 1500 V, the area of the hysteresis curves decreases markedly. Also, a nearly bilinear behavior of the system, displaying a marked softening at increasing loads is evident, for measurements at low potentials.

The overall stiffness of the system increases at high potentials, where the non-linearity of the system is less pronounced than at low potential levels ($U_{hv}=0\ldots1000\text{V}$). This makes a simple quantitative description of the evolution of the stiffness over the whole span of U_{hv} values difficult. To this purpose, an average slope of the force displacement diagrams (in N/m) was estimated for all U_{hv} values as an indicator of overall stiffness.

Figure 8: Average slope of the force displacement diagram as a function of the potential U_{hv} .

Figure 8 displays the change in stiffness with increasing potential U_{hv} . The initial potential step $0\text{V}\rightarrow500\text{V}$ does not affect the stiffness in a remarkable way, probably because at 0V potential a small air gap is present. The gap is likely closed at the first step. In the following voltage steps, the stiffness increases in an approximately linear way.

4.2. Frequency sweeps

As expected based on the static behavior of the beam, a remarkable change of the first natural frequency f of the electrostatically tunable sandwich beam can be measured as a function of the increasing potential U_{hv} (Figure 9).

The frequency f as a function of S_l is displayed in Figure 10.

Figure 9: Transfer function $|G|$ measured during a $2\text{Hz} \ldots 30\text{Hz}$ frequency sweep for different U_{hv} values ($0\text{V} \ldots 3000\text{V}$).

The frequency sweep measurements confirm the observations made in the static experiments. The first voltage step does not have a remarkable effect on the behavior of the beam. The following steps have a more marked effect on the peak frequency as well as on the amplitude of the recorded vibration.

Figure 10: First natural frequency of the transversal vibration of the sandwich beam as a function of the average slope SI .

While it is not possible to recognize with confidence the expected relationship (10) between stiffness and natural frequency f from the data displayed in Figure 10, a clear lack of linearity in the frequency increase with increasing stiffness can be observed and a clear convexity from above can be observed, as for a root function.

The measurements performed show that a marked shift in the vibration frequency of a sandwich beam can be obtained by modulating the shear stress transfer at the face-core interface. In the case presented, an increase of the vibration frequency by 400% could be achieved while the average slope of the static force-displacement diagrams as measured in the cantilever bending tests showed a tenfold increase.

While the approach used for the estimation of the average stiffness of the sandwich beam as an effect of increased shear stress transfer between faces and core is certainly not sufficiently accurate to model the behavior of the system, it gives a first insight in the effect of the modification of the shear stress transfer via the application of electrostatic forces between faces and core of a sandwich structure. More refined models will be used in the future, taking the softening behavior of the beam in to account. Also, displacement measurements along the main axis of the beam will allow for the calculation of the shear contribution to the lateral displacement of the beam, thus making it possible to estimate D and S for the calculation of f_B and f_s in (11) and (13).

The combination of materials used for the beam investigated in this work was chosen to make the observation of the presented stiffening effect fairly easy. Especially the selection of the core material, in the first phase of the project was limited by the requirements in terms of electrical conductivity. For practical applications, the deposition of a thin metallic layer on the outer surfaces of the core would overcome this limitation giving a greater freedom in the selection of the materials.

The goals of future work will be selection and further development of appropriate models to describe the dynamic behavior of the system as a function of the electrostatic potential as well as the optimization of the sandwich materials. In particular stiffer cores will be tested and more effective dielectric materials, such as Titanium oxides and Lead Zirconate Titanate (PZT) dielectric layers will be used.

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