Acceleration-based strain estimation in a beam-like structure

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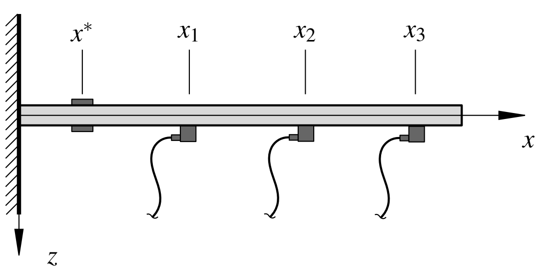
A robust design of the mechanical components of wind turbines with respect to fatigue behaviour is essential in the dimensioning process. The design load assumptions according to the certification guidelines by DNV GL [1] are conservative compared to the actual loads acting on a physical wind turbine, implying a discrepancy between the actual and anticipated fatigue progression. Thus, in order to assess the individual life time of a physical wind turbine, reliable tracking of the actually endured fatigue loads at critical spots is of great interest. Direct measurement of strains in operating wind turbines imposes high requirements on the measurement equipment and is in any case only possible at accessible measurement positions. An indirect derivation of fatigue indicators by motion quantities, e.g. accelerations, is more reasonable in this context. The aim of this contribution is to provide a basis for the fatigue estimation based on a limited number of acceleration measurements applying the modal decomposition and expansion (MDE) approach [2]. First practical tests were conducted on a steel cantilever beam, according to Fig. 1, which can be regarded as a simplified model of a wind turbine tower. Further concepts will be investigated on a small scale wind turbine [3] as well as on a full scale prototype turbine [4].

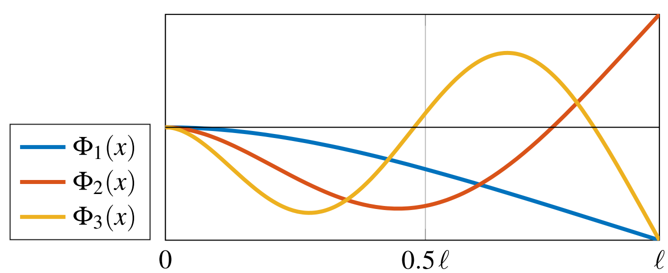
The dynamic displacement $w(x,t)$ of continuous systems can be represented by a linear combination of their mode shapes $\Phi_j(x)$ and modal coordinates $q_j(x)$ with $j = 1,\dots,\infty$. In practice, depending on the frequency range of excitation, usually only a small set of n lower modes provide a significant contribution to the dynamic displacement. Thus, the omission of higher modes (modal truncation) allows for the approximation of $w(x,t)$ by a finite sum,

$$w(x,t) = \sum_{j=1}^{\infty} \Phi_j(x)\, q_j(t) \approx \sum_{j=1}^{n} \Phi_j(x)\, q_j(t)\,. \tag{1}$$

a



b



c

Cantilever beam properties			Position	x/ℓ	Sensor type
Young's modulus	E	210 GPa	x^*	0.15	Vishay strain gauges
Density	ρ	$7850\,\mathrm{kg/m^3}$	x_1	0.38	PCB accelerometer
Length	ℓ	665 mm	x_2	0.65	PCB accelerometer
Cross section	A	$45 \times 8\,\mathrm{mm}$	x_3	0.90	PCB accelerometer

Fig. 1: **a** Sensor locations on the cantilever beam with accelerometers at positions x_k, $k = 1,2,3$ and strain gauges at position x^*. **b** The first three uni-axial bending modes. **c** Experimental setup parameters.

Relation (1) is exploited for the principle of MDE. With a given set of acceleration measurements $\ddot{\boldsymbol{w}}(t) \in \mathbb{R}^{m\times 1}$ at m discrete positions x_k, $k = 1,\ldots,m$, the structural dynamics can be approximately described by

$$\ddot{\boldsymbol{w}}(t) \approx \boldsymbol{\Phi}\,\ddot{\boldsymbol{q}}(t) \quad\Longrightarrow\quad \ddot{\boldsymbol{q}}(t) \approx \boldsymbol{\Phi}^{-1}\,\ddot{\boldsymbol{w}}(t)\,, \tag{2}$$

with $\boldsymbol{\Phi} \in \mathbb{R}^{m\times n}$ including the n mode shapes vectors at the m measurement locations and $\ddot{\boldsymbol{q}}(t) \in \mathbb{R}^{n\times 1}$ being the modal accelerations. The mode shape vectors $\boldsymbol{\Phi}$ are to be derived either by experimental modal analysis, finite element (FE), multibody [5] or idealised analytical models. This assumes that the experimental mode shapes correspond sufficiently to their numerical counterparts. In the context of this contribution analytical mode shapes are chosen as a first approximation. If n matches m, i.e. the number of measurement location equals the number of modes of interest, Eq. (2) can be directly solved for $\ddot{\boldsymbol{q}}(t)$. With identified modal accelerations $\ddot{\boldsymbol{q}}(t)$ and given analytical curvature mode shapes $\Phi_j''(x),\ j = 1,\ldots,m$, the maximal axial strains $\varepsilon_x(x^*,t)$ at any location x^* can be approximated according to the EULER-BERNOULLI theory,

$$\varepsilon_x(x^*,t) = -w''(x^*,t)\,z_{\max} \approx -z_{\max}\sum_{j=1}^{n}\Phi_j''(x^*)\,q_j(t)\,, \tag{3}$$

while $z_{\max}$ is the distance between outer fibre and neutral axis. Relation (3) requires a twofold time integration of the modal accelerations $\ddot{\boldsymbol{q}}(t)$. To prevent drift due to biased signals and unknown initial conditions high-pass filtering is applied. The integration process has been validated by simulation models as well as practical measurements.

The outlined procedure was investigated in several experiments. Test specimen was a homogeneous steel cantilever beam with properties listed in Fig. 1**c**. Three PCB accelerometers were used to observe the first three uni-axial bending modes. Optimal sensor locations were determined by a validated genetic optimisation algorithm on the basis of numerical modal analysis [6]. Reference strains at sensor location x^* were measured using Vishay strain gauges in a full bridge circuit. The physical cantilever beam clamping exhibits a certain degree of flexibility. For this reason the curvature mode shapes $\Phi_j''(x),\ j = 1,\ldots,m$ required for Eq. (3) were derived by an analytical cantilever beam model with a torsional spring as clamping boundary condition. A reasonable spring stiffness is determined under the condition, that the second eigenfrequency of the physical beam is to be reproduced by the analytical model. The results for a free vibration test after initial static deflection of the beam are shown in Fig. 2.

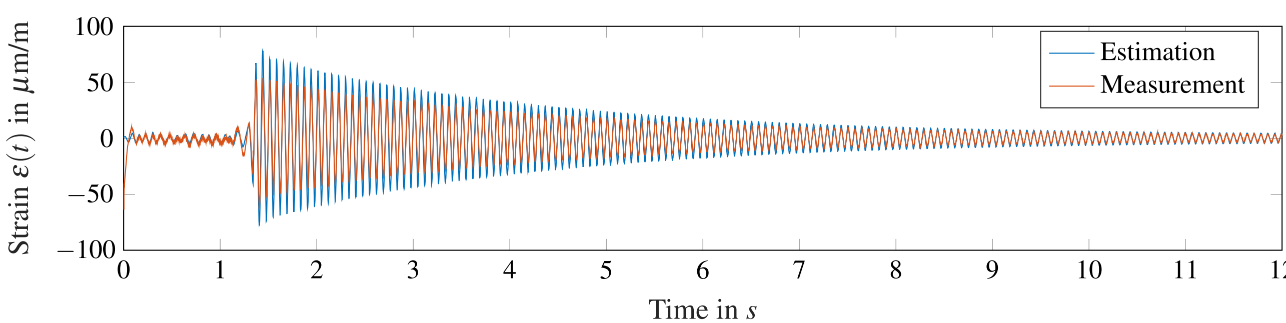


Fig. 2: Results of a free vibration test. Strain measurement and estimation based on analytical mode shapes with flexible clamping boundary condition.

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