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A framework for backbone experimental tracking : Piezoelectric actuators, stop-sine signal and Kalman filtering

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ABSTRACT

This paper deals with the use of piezoelectric patches for nonlinear dynamic identification. The patches are glued on the structure to identify amplitude-dependent damping and natural frequency; their positions are defined in order to perform the excitation concentrated on the first bending mode. Their locations on the structure allow to perform "stop sines" tests, as, unlike electrodynamic shakers, piezos are embedded on structures and do not modify the studied structure after the excitation signal is switched off. Although, despite the piezo and the *stop-sine*, the signal is still modulated by other frequency components or polluted by random signals, a post processing with the extended Kalman Filter allows a very good determination of the modal damping and the natural frequency, especially when they depends on the free vibration amplitude.

1. Introduction

In order to compute the vibration levels of built-up structures, the prediction of damping remains a great challenge. Simulation tools, Computer Aided Design and Finite Element Method are used to predict both inertia and stiffness with a pretty good accuracy, but damping is often badly estimated. As a consequence, the vibration levels are also wrongly predicted. The damping might be induced through several common ways such as viscoelastic materials, pressure loss in fluids or solid friction. The latter remains badly modeled, whereas the joints, such as welded points, bolted joints or rivets, are widely used to link the parts of the mechanisms and the structures.

1.1. State of the art

Among all the studies that focus on friction-damping, it is commonplace to separate the works that highlight energy dissipations coming from macro-sliding, and micro-sliding. In the first category, the damping is due to localized friction points, see Berthillier et al. [1] or Poudou et al. [2] for instance. In this case, simulations and tests are easy to perform because the contact region is generally localized and the slipping occurs all over the contact area. In the second case, the damping comes from partial sliding between the parts. Thus the sliding region is generally badly known and the motion is

Acronyms: EKF, Extended Kalman Filter; EMCC, Electro-Mechanical Coupling Coefficient; EL, Energy Losses; FRF, Frequency Response Functions; BC, Boundary Conditions; SC, Short Circuit; OC, Open Circuit; FE, Finite Element

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Nomenclature

Roman Symbols

T_{ij}	Mechanical stress tensor (order 2)
u_i	Displacement field
S_{ij}	Mechanical strain tensor (order 2)
s_{ijkl}	Material constitutive tensor (order 4)
$,j$	Denotes the partial derivation in all the equations, according to the coordinate e_j
jj	In all the equations, denotes the discrete sum according to the coordinate e_j
P	Polarization of the piezos
K_i	Electro-mechanical modal coupling coefficient
D_i	Electric displacement field
E_i	Electric field
d_{ijk}	Electromechanical coupling tensor (order 3)
$s_{ijkl}E$	Piezo material constitutive tensor (order 4)
φ	Electric potential
ω_{0i}, ξ_i	Natural frequency and modal damping of the ith mode
f_i	Modal electromechanical force for the ith mode
ω_i^{OC}, ω_i^{SC}	Natural angular frequency in open circuit and short circuit of the ith mode
ω	Angular frequency of the harmonic excitation
$x(t)$, x_n	Continuous-time and discrete-time analytic signal of displacement
$x_{1,n}$, $x_{2,n}$	Real and imaginary parts of the analytic signal.
$x_{3,n}$	Instantaneous phase of the signal
$a(t)$, a_n	Continuous-time amplitude, discrete-time amplitude
$\phi(t)$	Instantaneous phase
f_n	Instantaneous frequency
W_n	Process noise
V_n	Noise observation random process
Kg_n	Kalman gain

governed by stick-slip waves between the parts. It was shown experimentally by many authors (Goodman et al. [3], Beards et al. [4], Pian [5], Ungar [6]) that, in turn, the damping is strongly dependent to the vibration amplitude. This is due to the pressure and the shear stress variations. This leads to difficulties in modelling this case, see Festjens et al. [7] and Caignot et al. [8]. In simulations, geometrical defects and loading trajectories have to be carefully taken into account. To avoid this difficulty, it is also possible to measure and identify meta-models, see Festjens et al. [9], on specifically designed test-benches. Several testing devices have been designed in order to achieve this goal Fig. 1. The advantages of most of the

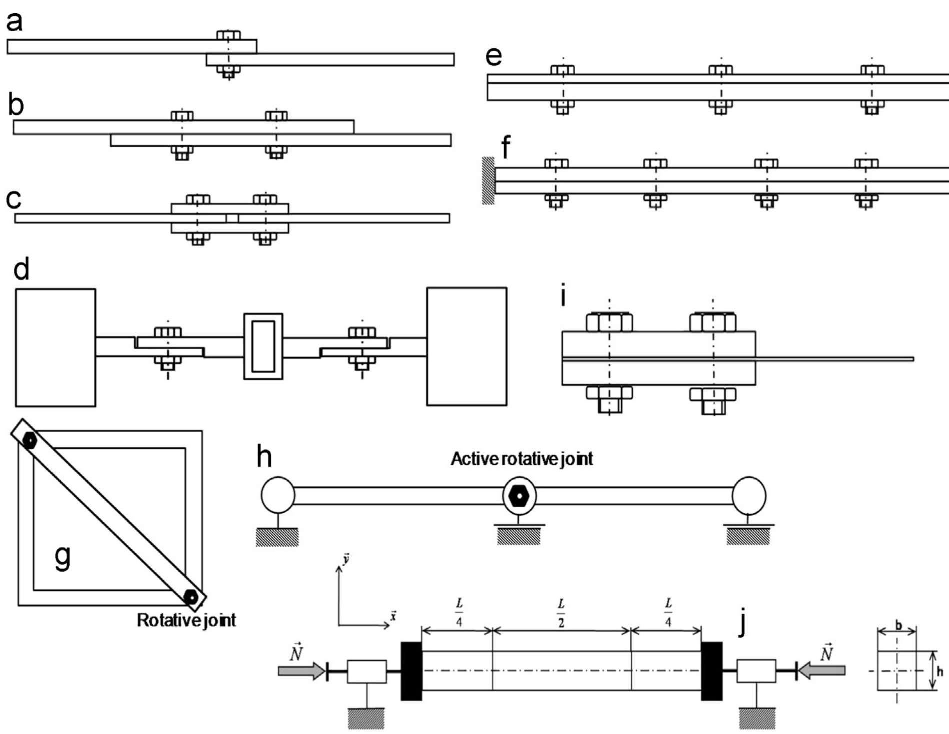

Fig. 1. Examples of testing devices developed to highlight friction induced damping in joints. (a) Beam assembly with a single bolted joint from Ahmadian et al. [11]. (b) Beam assembly with two bolted joints from Metherell et al. [12] or Esteban et al. [13]. (c) Beam assembly symetrically screwed with two bolted joints from Song et al. [14]. (d) Structure with two blocks, one spring and beam assembly with two bolted joint and with special lap joint geometry from Goyder et al. [15]. (e) Beam assembly with distributed bolted joints in free conditions from Heller et al. [16,17]. (f) Beam assembly with distributed bolted joints in clamped-free conditions from Goodman et al. [3], Nanda et al. [18,19]. (g) Polyarticulated structure with rotative friction joints from Beards et al. [4]. (h) Beam assembly with an active rotative joint from Gaul et al. [20]. (i) Cantilever quartz beam dedicated to the study of microsliding in the clamp from Nouira et al. [21]. (j) Clamp clamp cutted beam with pure microsliding in the interfaces from Peyret et al. [22].