Mechanical analysis of piezoelectric vibration energy harvesting devices

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Abstract - A modal model of the dynamics behavior of a bimorph vibration energy scavenger aimed at wireless system networks is proposed and validated via FEM simulations. Experimental set-ups are developed with the aim of assessing the validity of the model. The set-ups are tested by measuring the performances of commercially available energy scavengers. Repetitive bending tests allow determining the equivalent bending stiffness of the scavengers. Subsequent dynamics tests allow then obtaining results in terms of power vs. applied resistive loads. The validity of all the tools needed for the optimization of the performances of custom made scavengers in terms of their power outputs, while considering constraints such as dimensions and strength limits, is thus proven. They will be used to upgrade the design and the performances of vibration energy scavengers for wireless sensor networks.

I. INTRODUCTION

Systems for harvesting environmental energy have attracted a lot of attention in the development of pervasive wireless sensor networks [1]. Most commonly used energy harvesting concepts include thermoelectric conversion, photovoltaics, RF and electromagnetic radiation power sources and (random or sinusoidal) kinetic excitation (including that of human motion) [2-7]. Of the latter, the most common is vibration energy scavenging which is already being used, especially in the field of wireless monitoring of machine tools or to power cell phones, and can be divided into piezoelectric, electromagnetic and capacitive. In this frame, harvesting of kinetic energy of vibrations via piezoelectric bimorph cantilevers so as to generate electric power is particularly advantageous due to design simplicity, miniaturization potential as well as the inherent linearity of the mechanical behavior and of the electromechanical coupling [8].

In order to be able to properly design vibration energy harvesters, their behavior in terms of dynamics response, electromechanical coupling and charge distribution, has to be accurately studied [9-11].

In this work a modal model of the dynamics behavior of a bimorph scavenger is proposed. The study is based on the classical Euler-Bernoulli beam model. The obtained results are validated with those obtained via a Finite Element Model (FEM).

Some vibration energy scavengers are already commercially available [5, 6, 12-16], but their usage can be limited due to lack of relevant data on the values of their electromechanical characteristics. This makes the behavior of commercial scavengers hard to model.

Experimental set-ups are thus developed with the aim of validating further the modal model of the behavior of the scavengers, but also to verify the performances of commercially available piezoelectric vibration energy scavengers.

Repetitive bending tests on a tensile machine allow determining the equivalent bending stiffness of the used scavengers. Dynamics tests make then possible obtaining results in terms of power vs. applied resistive loads and thus to correlate the behavior of the studied devices to the modal model, especially when it will be coupled to electromechanical performances' models proposed recently in literature [9, 17].

The respective experimental set-ups are described and their suitability to be used for the foreseen applications is proven. This will make possible, in the following phase of the work, to develop custom vibration energy harvesting devices with optimized performances in terms of power outputs, taking concurrently into consideration not only dynamics but also dimensional and material strength constraints.

II. MODAL MODEL OF A CANTILEVER WITH A TIP MASS

In order to maximize energy conversion, the response of the piezoelectric vibration energy harvester (Fig. 1.) has to be tuned to the excitation source present in its surrounding. To analyze the dynamics response of the considered device, the respective modal model has therefore to be considered. Indicating then with E the Young's modulus of the material, with ρ its density, with I_y the second moment of inertia of the cross section of area A and with $u_x(z, t)$ the beam deflection in x direction, the force and moment equilibrium of the Euler-Bernoulli beam is considered. The bending stiffness $k(z) = E(z)I_y(z)$ and the mass distribution along the cantilever $m(z) = \rho(z)A(z)$ are supposed to be constant, whereas the shear forces are neglected. The following relation is hence obtained [18]:

$$m\frac{d^2u_x}{dt^2} = -\frac{\partial^2}{\partial z^2}\left[k\frac{\partial^2 u_x}{\partial z^2}\right] \tag{1}$$

Following the standard modal expansion method, the variables are separated in the time and space domains

$$u_x(z,t) = q(z)\cdot\eta(t) \tag{2}$$

where $q(z)$ is the tip deflection, while $\eta(t)$ is a harmonic function equal to $\eta(t) = \sin(\lambda t + \phi)$. The obtained solution is then:

$$\frac{EI_y}{\lambda^2\rho A}\frac{d^4q(z)}{dz^4} = -q(z) \tag{3}$$

in which λ denotes the eigenfrequency of the cantilever.

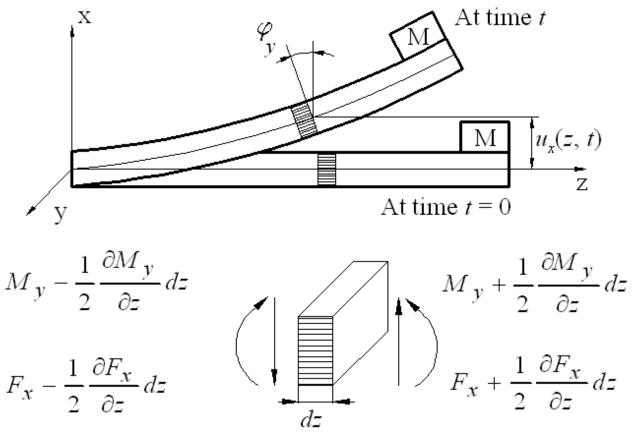

Fig. 1. Model of the cantilever with a tip mass.

Introducing the substitution

$$a = \sqrt{\lambda} \cdot \sqrt[4]{\frac{\rho A}{EI_y}} \tag{4}$$

the assumed solution of equation (3) is:

$$\begin{aligned} q(z) = A\sin(az) + B\cos(az) + \\ + C\sinh(az) + D\cosh(az) \end{aligned} \tag{5}$$

The relevant boundary conditions for the cantilever are considered next:

fixed end ($z = 0$)

$$q(0) = 0, \quad q'(0) = 0 \tag{6}$$

free end ($z = L$):

$$q''(L) = 0 \tag{7}$$

Introducing in the conventional model the tip mass M and considering eq. (4), the second boundary condition at the free end becomes [19]:

$$q'''(L) = -\frac{\lambda^2}{EI_y} Mq(L) = -\frac{M}{m} a^4 q(L) \tag{8}$$

A transcendental equation is hence derived:

$$\begin{aligned} \frac{M}{ml} \cdot al \cdot (\cosh al \cdot \sin al - \sinh al \cdot \cos al) = \\ = 1 + \cosh al \cdot \cos al \end{aligned} \tag{9}$$

This equation has to be solved numerically to obtain the coefficients al and thus, via Eq. (4), the eigenfrequencies λ.

The respective modal model, implemented in MATLAB, was tested using a reference rectangular cantilever. The obtained eigenfrequencies λ are shown in Table 1., while the respective mode shapes are depicted in Fig. 2.

TABLE I
CALCULATED EIGENFREQUENCIES

λ_1 (Hz)	λ_2 (Hz)	λ_3 (Hz)	λ_4 (Hz)	λ_5 (Hz)
90.7	1242	3950	8201	13998

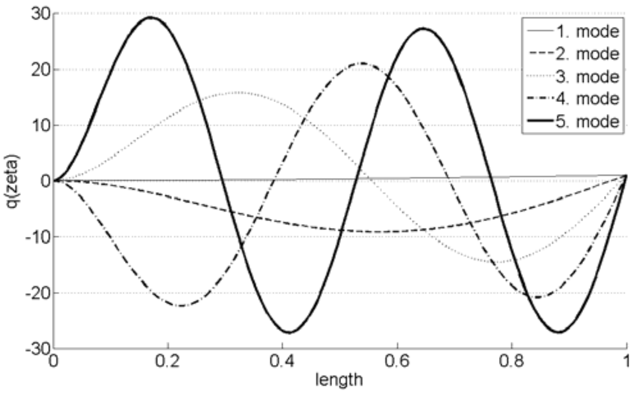

Fig. 2. Mode shapes of a cantilever beam carrying a tip mass.

The developed model is validated using the ANSYS finite element modeling (FEM) software by employing 2D BEAM3 elements as well as by using SOLID45 3D elements. A MASS21 point mass is placed at the free end of the cantilever while all degrees of freedom (DOFs) at the clamp are constrained (Fig. 3). Both FEM models validate the results obtained via the proposed analytical model with the relative differences being always within 2% (Table 2).

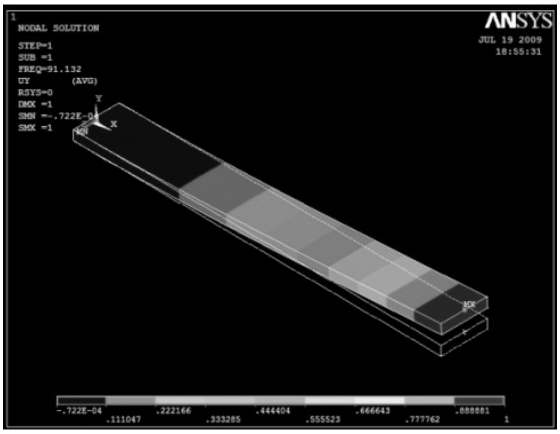

Fig. 3. First fundamental mode shape obtained via FEM analysis.

TABLE II
COMPARISON OF EIGENFREQUENCIES OBTAINED WITH THE VARIOUS APPROACHES

Eigenfreq. (Hz])	Proposed model	FEM 2D	FEM 3D	Model vs. 3D
λ_1	90,7	91,1	90,6	0,99
λ_2	1242	1237,8	1241	0,99
λ_3	3950,2	3740,8	3944	0,99
λ_4	8201,1	8291,6	8177	0,99
λ_5	13998	13781	13931	0,99

III. EXPERIMENTAL SET-UPS FOR THE ASSESSMENT OF THE PERFORMANCES OF ENERGY SCAVENGERS

In order to validate experimentally the developed models, but also to assess the behavior of commercially available vibration energy scavengers, suitable experimental set-ups have to be developed.

In fact, the available data sheets on the electro-mechanical characteristics of MIDE scavengers [15] allow evidencing a very intricate structure composed by nine alternating layers of: woven fiberglass reinforced epoxy laminate sheets, epoxy adhesive films, layers of piezoelectric materials and copper clad polyimide laminates. This structure was seen also by using an Olympus type SZX16 stereomicroscope (Fig. 4) [20] as well as via an analysis performed on an Oxford instruments INCA-based energy dispersive X-ray spectroscopy apparatus where the X-rays are emitted by the analyzed sample in response to being hit by electrons emitted by a scanning electron microscope (SEM) [21].

A. Determination of Bending Stiffness

Based on the above shown structure of commercially available scavengers, it is obvious that assessing their performances will be possible only if an equivalent elastic modulus, and thus the respective bending stiffness, is experimentally evaluated.

For this purpose a campaign of repetitive bending measurements on a VEB Thüringer Industriewerk Rauenstein tensile machine has been set-up (Fig. 5). Tests were performed on V21b and V25w type MIDE vibration energy scavengers (indicated in Fig. 5 with 2) simply supported on a suitable holder (indicated with 1). Load F was applied via a suitable loading system mounted on the tensile machine and indicated in Fig. 5 with 3, while the applied load was measured by using a Z6FD1 HBM load cell [22]. The deflections w of the scavenger was measured via a HBM inductive displacement transducer of the type W1T3 [22], indicated in the figure with 4, which has a measurement range of ±1 mm and a sensitivity of 10 V/mm. In the considered limited range of displacements, the measured load vs. deflection data showed a perfectly linear behavior, while the repeatability of the measurements is always within ±2%.

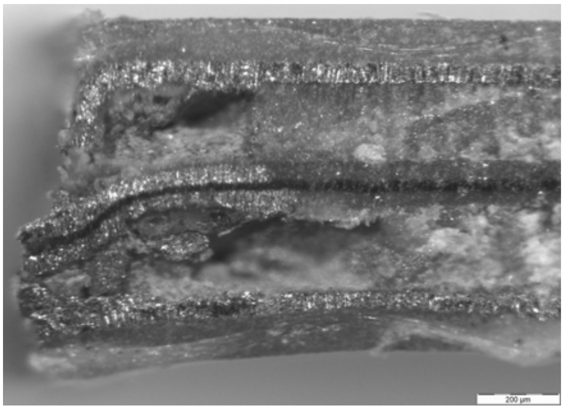

Fig. 4. Cross-section of a MIDE energy scavenger seen through a stereomicroscope.

The layout of the measurement apparatus is shown on the photo given in Fig. 6.

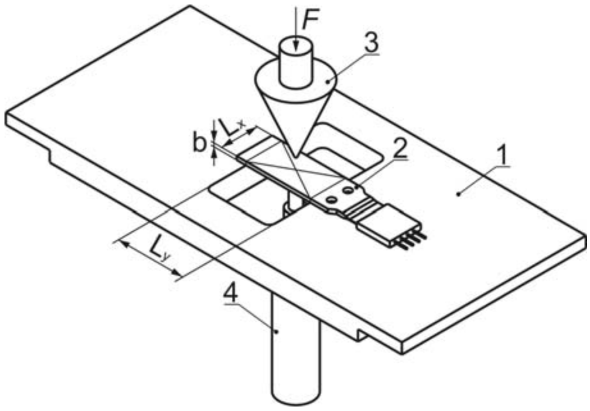

Fig. 5. Experimental set-up for the determination of the equivalent bending stiffness.

Plate theory, i.e. the expression which correlates the modulus of elasticity E of a simply supported plate to its dimensions (thickness b and width L_x) as well as with the deflection w for a given centered point load F, where k_w is a geometrical coefficient which depends on the L_y/L_x ratio, [23] is then used to obtain the value of Young's modulus:

$$E = \frac{12FL_x^2}{k_w b^3 w} \tag{10}$$

The obtained values of the equivalent modulus of elasticity of the scavengers are then, respectively, 3.24 GPa for the V25w and 4.22 GPa for the V21b scavenger (the difference being probably attributable to the different thicknesses of the various layers that make up the device).

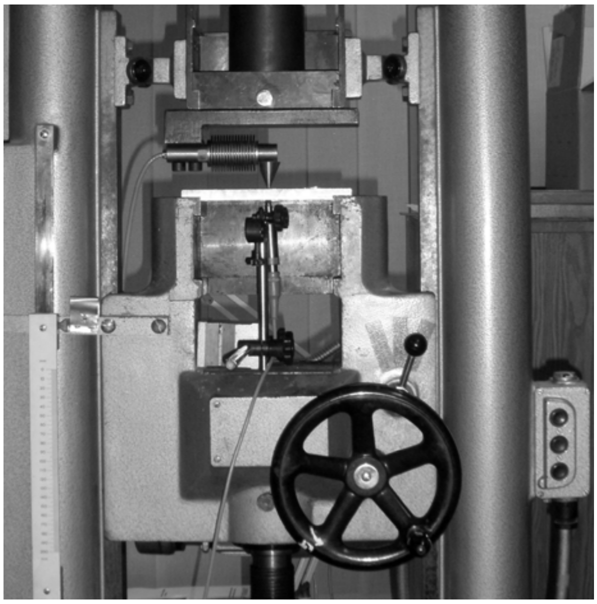

Fig. 6. Detail of the measurement set-up for the determination of the bending stiffness.

B. Set-up for Dynamics Analysis

A set-up for the dynamics analysis of the performances of the available vibration energy harvesters was developed within the Laboratory for Precision Engineering of the Department of Mechanical Engineering Design of the Faculty of Engineering of the University of Rijeka [24].

The set-up is shown in Fig. 7 and is based on a Schenk AG electrodynamics shaker of the type Vibroexciter 41 (indicated in the figure with 1) with the corresponding signal generator and power amplifier of the type Vibropower 41. The coils in the exciter generate a dynamics excitation controllable in force units in a selectable frequency range (that can be swept) of up to 1 kHz. The excitation is transmitted to a suitable holder (indicated in Fig. 7 with 2) and thus to the scavenger (indicated with 3).

Excitation acceleration is measured via a Schenk AS-020 piezoelectric accelerometer with a sensitivity of 10.2 mV/(m/s^2) and a measurement range of up to 800 m/s^2 and 15 kHz (indicated in Fig. 7 with 4).

The vibration of the free end of the cantilever is measured by using a MetroLaser VibroMet Model 500V 780 nm wavelength laser Doppler vibrometer (indicated in the figure with 5) [25]. The measurement range of this device goes from 5 μm/s to 800 mm/s.

Variable resistive loads have then been connected to the scavenger, while the whole set-up in interfaced to a LabView v. 8.5 based National Instruments PXI data acquisition system [26].

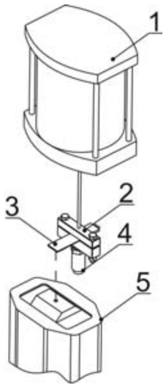

Fig. 7. Experimental set-up for dynamics measurements.

Fig. 8. Lay-out of the experimental set-up for dynamics measurements.

The lay-out of the whole set-up is given in Fig. 8.

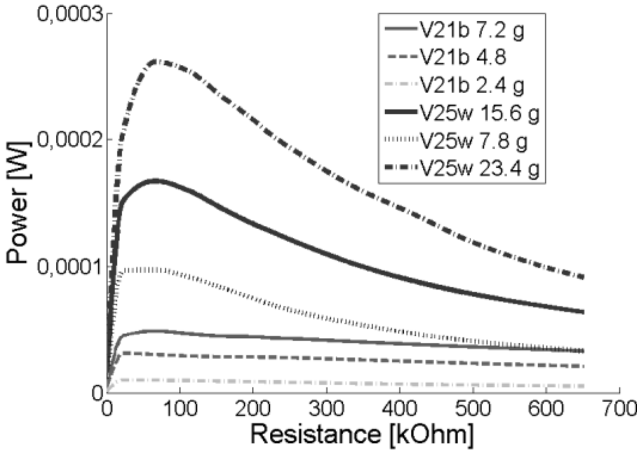

Fig. 9. Preliminary results of the performances of commercial scavengers vs. tip masses and resistive loads.

In a preliminary usage of the set-up, the MIDE vibration energy harvesters V21b and V25w, each with 3 different tip masses and numerous resistive loads, were tested in the dynamics range of ± 3 Hz around the first resonant frequency. The condition of having a pure resistive load connected to the electrodes is not necessarily the most realistic one, since often electric loads consist of rechargeable batteries and other capacitive loads. However, it is simple and useful not only for estimating the resulting power, but also for giving the designer more intuition about the system [17].

The obtained output powers are shown in Fig. 9. The experiments resulted in expected performances. The V25w device with larger piezoelectric material volume outputs more power; with comparable tip masses, the resulting power levels are about twice as large as those obtained on the V21b device. The output power is proportional to the tip mass. Optimal resistances for the employed loading conditions are in the range of 20 - 60 kΩ.

It was hence proven that the developed experimental set-ups are suited to characterize the performances of various scavenger configurations. By employing the above modal model, especially if coupled to the model of the electromechanical behavior of piezoelectric vibration energy scavengers developed recently by Roundy and Wright [17] and Erturk and Inman [9], it will thus be possible not only to correlate the theoretically predicted and experimentally assessed performances of the scavengers, but also to perform the optimization of the performances so as to maximize the output powers for a set of design criteria which may include dimensional constraints and material strength limits.

IV. Conclusion and Outlook

In this work a modal model of the behavior of vibration energy scavengers is developed and complemented with FEM simulations.

The developed experimental set-ups proved to be suitable for assessing the significant parameters and the dynamics performances of the studied devices.

The tools needed to optimize the output power of piezoelectric vibration energy scavengers have hence been successfully developed. In the next phase of the work, they will be used, together with optimization algorithms and electromechanical coupling relations, to upgrade the performances of the studied class of energy scavenger devices for wireless sensor networks. In this framework, the algorithms for the optimization of the geometric parameters of the scavengers have already been developed by using the non linear Sequential Quadratic Programming (SQL) based optimization toolbox of the MATLAB software package [27].

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