Frequency Self-tuning Scheme for Broadband Vibration Energy Harvesting

MICKAËL LALLART,* STEVEN R. ANTON AND DANIEL J. INMAN

Department of Mechanical Engineering, Center for Intelligent Material Systems and Structures, Virginia Tech, Blacksburg, Virginia 24061, USA

ABSTRACT: The recent proliferation of microscale devices has raised the issue of energy harvesting for replacing batteries that present maintenance and recycling problems. Particularly, piezoelectric seismic microgenerators offer the advantages of easy maintenance and high power output, but are very sensitive to frequency drifts that can dramatically decrease their performance. The purpose of the present article is to expose a technique to ensure that the harvester resonance frequency matches the base motion frequency, without any external intervention. The principles of the proposed method rely on ultralow-cost frequency sensing combined with an energy-efficient stiffness tuning, through the use of an additional actuator. Experimental results carried out to validate the model show that such an approach permits increasing the effective bandwidth of the structure by a factor of 4 in terms of mechanical vibrations and having a 100% frequency band gain in terms of total power output of the device (i.e., taking into account the energy spent by the actuation). The total energy produced by the harvesting device, taking into account the actuation cost, is discussed as well.

Key Words: control, energy harvesting, piezoelectric, actuator.

INTRODUCTION

THE recent development of low-power devices combined with the increasing interest in terms of self-powered sensors and sensor networks has created a demand for supplying low-power electronic devices (Lallart et al., 2008a). In order to overcome the drawbacks of batteries that present limited lifespan, maintenance issues and recycling problems, using the ambient environment as an energy source has been proposed (Krikke, 2005). Various sources can be considered for energy harvesting, for instance solar (Hamakawa, 2003), magnetic (Shearwood and Yates, 1997), wind (Priya, 2005), but a particular emphasis has been placed on vibration energy harvesting for supplying energy to small-scale devices (Roundy et al., 2003). In this domain, piezoelectric elements are commonly used thanks to their relatively high conversion abilities and integration potentials (Glynne-Jones et al., 2001; Sodano et al., 2004; Badel et al., 2005; Guyomar et al., 2005; Lefeuvre et al., 2006; Anton and Sodano, 2007; Lallart and Guyomar, 2008; Lallart et al., 2008b).

When dealing with piezoelectric energy harvesting, several configurations are possible. First, the piezoelectric element can be used either in a static or a dynamic fashion, as exposed by Keawboonchuay and Engel (2003), the latter case being the most favorable as it provides much higher power outputs. Hence, much work has been focused on driving a microgenerator near one of its resonance frequencies in order to optimize the energy extraction. Two types of configurations can be considered in such an arrangement. Either the active material is directly bonded on the host structure near high stress locations, directly converting the strain, or an additional structure (e.g., cantilever beam) can be used to host the active material. This latter configuration, sensitive to the base acceleration, allows reduced maintenance, but requires a fine tuning of the resonance frequency to the base motion. In particular, the base motion can see its frequency contents changing due to external conditions such as temperature or pressure effects as well as aging. Hence, new approaches have to be considered to ensure the robustness of the harvester in order to guarantee that its resonance frequency matches with the host structure motion (Roundy and Zhang, 2005; Leland and Wright, 2006; Challa et al., 2008). Nevertheless, only a few researchers have described a self-tuneable system, combining both frequency detection and self-actuation (most studies on frequency

*Author to whom correspondence should be addressed.
E-mail: mickael.lallart@insa-lyon.fr
Figures 3 and 8–11 appear in color online: http://jim.sagepub.com

1045-389X/10/09 0897–10 $10.00/0 DOI: 10.1177/1045389X10369716

tuning consider passive tuning, which requires an external operator).

One of the most important problems when considering frequency identification in resonant structures lies in detecting whether the frequency or the magnitude is changing. Additionally, a drop or a rise in the magnitude can be caused by both an increasing or a decreasing frequency when the device is tuned, or an excitation magnitude change. Hence, frequency tuning based only on the deflection magnitude cannot be properly achieved. A possible method for frequency detection relies on zero crossing, but in the case of seismic microgenerators, both base and harvester frequencies need to be detected, leading to a complex and costly method. In Zhu et al. (2008), the resonance frequency matching is achieved using a magnet that applies a supplementary spring force on the harvester, while the sensing principles rely on the voltage maximization. This method, therefore, suffers from the previously exposed drawback, that is, the frequency change direction cannot be detected efficiently. The use of magnets also induces a supplementary damping effect that reduces the harvester power output as well. A particularly interesting property for a fine self-tuning at the resonance frequency with respect to the base motion consists of using the phase information between the two signals (i.e., driving force – or acceleration – and deflection), as the phase shift equals $\pi/2$ when the piezogenerator is tuned. In this domain, Peters et al. (2009) have proposed a very simple and efficient way for tuning the resonance frequency of a harvester based on the signum of the displacement sensors' signals, while the actuation is done by applying a pre-stress on a piezoactuator, hence altering the stiffness. Unfortunately, neither a theoretical model of the behavior of the complete system nor the total energy produced, taking into account the actuation energy, are presented in this work.

The purpose of this article lies in proposing a low-cost self-tuning technique (i.e., both automatic frequency detection and actuation) based on the properties of systems driven at their resonance frequency and on a non-linear, low-cost stiffness tuning scheme described by Guyomar et al. (2008) for the actuation. In addition, a theoretical model of the proposed self-tuning technique is presented. The proposed detection method consists of using the averaged product of the beam deflection and base acceleration, directly giving the phase between the two signals, and in particular yielding 0 when the phase equals $\pi/2$ (i.e., matched resonance frequency). Then from this information the frequency can be adjusted accordingly by properly tuning the value of a switching voltage source. This article also provides the derivation of the total energy of the system, giving the output power of the microgenerator taking into account the cost of actuation.

The article is organized as follows: the section 'Principles' explains the basics of the proposed self-tuning method, whose theoretical development is presented in the section 'Modeling', as well as the expected power output of the microgenerator (taking into account the total energy flows, that is, harvested energy and required energy for actuation). The section 'Experimental Validation' aims at validating the concept and predicted results through experimental measurements using a structure featuring three piezoelectric layers: one for energy harvesting, one for frequency tuning, and one for deflection sensing.[1] Finally, the last section briefly concludes the article.

PRINCIPLES

The purpose of this section consists of exposing the general principles of the resonance frequency self-tuning scheme. For the sake of simplicity, the harvesting circuitry considered here will be made only of a bridge rectifier connected to a smoothing capacitance C_S and a load R_L (Figure 1), although more efficient energy interfaces can be considered (Badel et al., 2005; Guyomar et al., 2005, 2009; Lefeuvre et al., 2006; Lallart and Guyomar, 2008; Lallart et al., 2008b). The time constant $C_S R_L$ is assumed far greater than the vibration time period, so that the rectified voltage V_{DC} can be considered as constant. The operations of this circuit are as follows. When the absolute value of the piezoelectric voltage reaches the rectified voltage, the piezoelectric element is connected to the harvesting stage made of the capacitor C_S and the load R_L through the rectifier. Hence, a current flow appears from the active device to the smoothing capacitor and load, thus extracting energy. This energy harvesting process stops when the current becomes null, which actually occurs when the deflection reaches either a maximum or a minimum value. Then the piezoelectric element is left in open circuit (null current), so that its voltage varies with the deflection, until it reaches the rectified voltage value.

Frequency Sensing Principles

An efficient frequency sensing technique for resonant systems has to fulfill two major requirements: independence from a magnitude change caused by an excitation magnitude decrease (or increase) and differentiation of frequencies lower or higher than the resonance frequency. In this article, such a method is proposed, based on the simple observation that the product of two sine

[1]Although it may be thought that the additional transducer used as actuator for frequency tuning would rather be used as harvester, it can be shown that for highly coupled, weakly damped structures, the addition of an active layer does not increase the power output due to damping effect (Guyomar et al., 2005). In addition, the third layer (sensor) can be small as no significant energy has to be converted (only the signal is important).

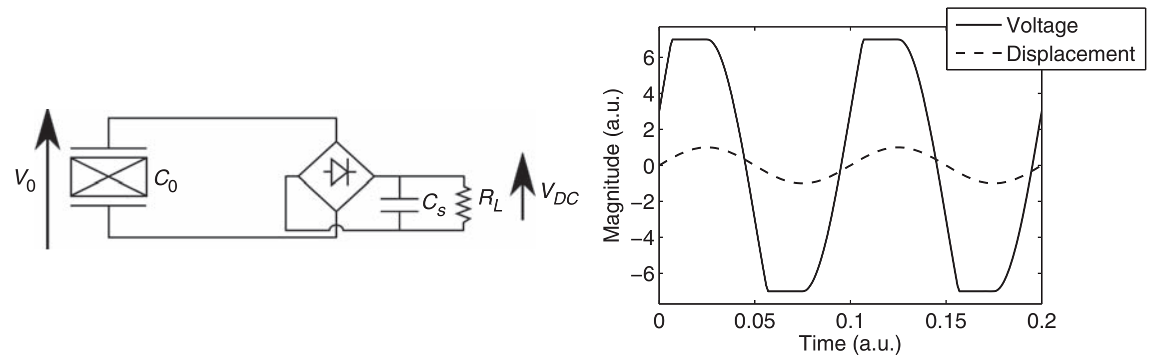

Figure 1. *Standard energy harvesting schematics and waveforms.*

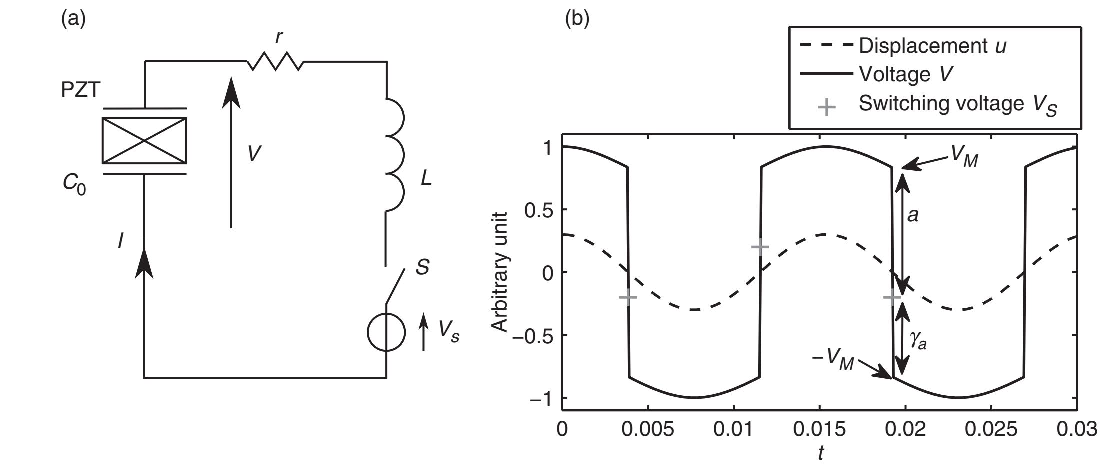

Figure 2. *Frequency tuning circuitry and typical waveforms.*

functions of the same frequency ω gives information about their respective phase ψ:

$$\sin(\omega t)\sin(\omega t+\psi)=\frac{1}{2}\Big(\cos(\psi)-\cos(2\omega t+\psi)\Big). \quad (1)$$

As the resonance frequency is obtained for a phase of $\pi/2$ between the base motion and active structure deflection, the first term of the previous expression (obtained from Equation (1) using a low-pass filter) would be 0 when the structure is tuned and positive (respectively negative) if the resonance frequency is too low (respectively high) compared to the excitation frequency. Then from this information, it is possible to finely tune the following actuation technique for an efficient self-sensing microgenerator.

Actuation principles

When designing systems aiming at magnifying the performance of a harvester, it is crucial to keep in mind that the available energy is limited. Indeed, using an actuation scheme that requires more power than the harvester abilities would be useless. In this article, a simple, low cost but efficient approach for stiffness control described in Guyomar et al. (2008) is used. This technique consists of switching the piezoelectric element on an electrical network for a brief time period each time the deflection crosses a zero value (Figure 2).

The electrical circuit is made of an inductance L (whose losses are modeled by a resistor r) in series with a tuneable voltage source V_S and a digital switch S. Hence, when the piezoelectric element is connected to this circuit, it shapes a resonant electrical network and the piezoelectric voltage starts oscillating with respect to V_S. Particularly, if the switching time period is chosen to be equal to half an electrical oscillation period t_e given as[2]:

$$t_e=2\pi\sqrt{LC_0}, \quad (2)$$

this leads to a voltage inversion around V_S. However, due to internal losses in the switching circuit, this

[2]The inversion can actually be stopped automatically by the use of diodes.

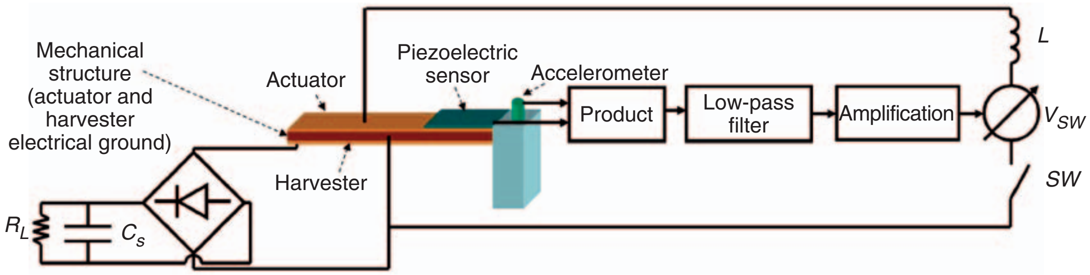

Figure 3. Self-tuned energy harvester schematic.

inversion is not perfect and characterized by the inversion coefficient γ ($0 \leq \gamma \leq 1$).

Such a treatment, therefore, shapes a piezoelectric voltage that can be decomposed into two voltages: one voltage proportional to the deflection, and one piecewise constant voltage that is in phase with the deflection. This last voltage, whose amplitude can be tuned through the voltage source V_S (which varies with time), applies a pre-stress on the piezoelectric element which, therefore, allows the control of the stiffness over a range bypassing the short and open-circuit stiffnesses, for a typical power requirement 10 times less than a classical proportional control (Guyomar et al., 2008).

Overview of the Self-tuning System

Hence, the basic concept of the self-tuning system, depicted in Figure 3, relies on the low-pass filtered product of the base acceleration with the cantilever deflection signal of the piezoelectric sensor, that gives the phase between the two signals (Equation (1)). Then, from this information, the voltage source for frequency tuning can be chosen in order to ensure that the excitation frequency matches the resonance frequency using the previously exposed scheme.

MODELING

This section aims at investigating the power output that can be expected from the self-tuned piezoelectric generator and comparing the performance of the proposed system with the classical implementation.

For simplicity, the electromechanical structure will be modeled by a rough but effective electromechanically coupled single degree of freedom (SDOF) model such as a spring-mass-damper system (Badel et al., 2007; Erturk and Inman, 2008):

$$\begin{cases} M\ddot{u} + C\dot{u} + K_E u = -\mu_1 M a - \alpha V \\ I = \alpha\dot{u} - C_0\dot{V} \end{cases} \tag{3}$$

where u, a, V, and I refer to the beam deflection, base acceleration, piezoelectric voltage and current, respectively. M, C, and K_E denote the dynamic mass, structural damping coefficient and short-circuit stiffness, respectively, while α and C_0 are defined as the force factor and piezoelectric clamped capacitance. μ_1 is the excitation correction factor due to the mass of the beam without tip mass.

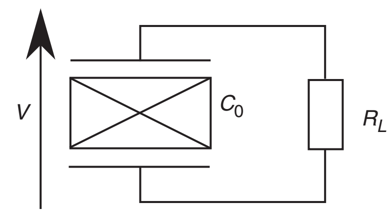

Figure 4. AC standard energy harvesting schematics.

Energy Harvesting Interface

Although a realistic energy harvesting interface for powering electronic devices relies on AC/DC conversion (Figure 1), it is proposed here, for the sake of simplicity, to consider that such an interface can be linearized so that the rectifying stage can be considered as a pure resistive load as depicted in Figure 4.

In this case, the electrical equation of Equation (3) turns to:

$$\frac{V}{R_L} = \alpha\dot{u} - C_0\dot{V}, \tag{4}$$

yielding the harvested power P as a function of the angular vibration frequency ω:

$$P = \frac{V.V^*}{2} = \frac{1}{2}\frac{R_L(\alpha\omega^2)}{1 + (R_L C_0 \omega)^2} {u_M}^2, \tag{5}$$

where * denotes the complex conjugate and u_M refers to the displacement magnitude.

However, harvesting electrical energy also modifies the mechanical behavior of the system. According to the equation of motion of Equation 3, the transfer function giving the displacement magnitude yields:

$$u_M = \frac{\mu_1 M a_M}{-M\omega^2 + j\omega\left(C + \frac{R_L\alpha^2}{1+(R_L C_0\omega)^2}\right) + \left(K_E + \frac{R_L{}^2\alpha^2 C_0\omega^2}{1+(R_L C_0\omega)^2}\right)}, \tag{6}$$

with a_M being the acceleration magnitude. For weakly damped systems, this expression can be approximated around the resonance frequency by:

$$u_M = (u_M)_{res}\sin(\varphi), \tag{7}$$

where $(u_M)_{res}$ denotes the resonance frequency displacement magnitude:

$$(u_M)_{res} = \frac{\mu_1 M a_M}{\omega\left(C + \frac{R_L\alpha^2}{1+(R_L C_0\omega)^2}\right)}, \tag{8}$$

and φ is the phase angle between the base acceleration and beam deflection, which not only depends on the mechanical parameters of the structure, but also on the connected load (when the load corresponds to the optimal load so that the harvested power is maximal, or for low coupled piezoelectric materials, however, the impact of the load on the phase angle φ is weak however).

Self-tuned Resonator

From a given frequency and load, it can be seen from Equation (7) that the phase angle that maximizes the deflection magnitude and therefore the maximum power output is equal to π/2, meaning that the system is excited at its resonance frequency. In this subsection, the analysis of the self-tuning mechanism that ensures the excitation at this frequency is proposed and developed.

For simplicity reasons, only the mechanical aspect of the self-tuned resonator will be investigated in this subsection (i.e., the energy harvesting effect will not be taken into account). In this case, the electromechanical equation of motion is still given by Equation (3), but the electrical and electromechanical quantities (i.e., V, I, α, and C_0) refer to the actuator parameters. Using the previously described system, the actuation voltage source V_S is given by:

$$V_S = \pm\beta\frac{\alpha}{C_0}\langle V_{\text{base}} \,.\, V_{\text{cant}}\rangle, \tag{9}$$

where $\langle V_{\text{base}} \,.\, V_{\text{cant}}\rangle$ is the mean value of the product of the base accelerometer and beam deflection sensor signals and β is a user-defined tuning factor. It can be noted that β needs to be negative for a proper tuning, as for a frequency lesser (respectively higher) than the vibration frequency, $\sin(\varphi)$ is positive (respectively negative), while the stiffness has to be decreased (respectively increased), meaning that the switching voltage V_S has to be negative (respectively positive). The ± sign is negative (respectively positive) for a falling (respectively rising) switching process. Considering that the sensitivities of the accelerometer and beam deflection sensor are respectively given by S_{acc} and S_{defl} and defining β_0 such as:

$$\beta_0 = -S_{\text{acc}} S_{\text{defl}} \frac{\beta}{a_M}, \quad \beta \le 0, \tag{10}$$

the voltage source expression turns to:

$$V_S = \mp\frac{1}{2}\beta_0\frac{\alpha}{C_0}u_M\cos\varphi. \tag{11}$$

Hence, assuming that the deflection remains sinusoidal (i.e., only the first harmonic of the actuator voltage has an effect on the structure), the equation of motion turns to (Guyomar et al., 2008):

$$M\ddot{u} + C\dot{u} + K_E u = -\mu_1 M a - \frac{\alpha^2}{C_0}\left(1 - \beta_0\frac{2}{\pi}\frac{1+\gamma}{1-\gamma}\cos\varphi\right)u, \tag{12}$$

yielding the transfer function:

$$\frac{U(\omega)}{A(\omega)} = \frac{-\mu_1 M}{-M\omega^2 + jC\omega + K_D - \frac{\alpha^2}{C_0}\frac{2}{\pi}\frac{1+\gamma}{1-\gamma}\beta_0\cos(\varphi)}$$
$$\text{with } \varphi = -\arctan\left(\frac{C\omega}{-M\omega^2 + K_D - \frac{\alpha^2}{C_0}\frac{2}{\pi}\frac{1+\gamma}{1-\gamma}\beta_0\cos(\varphi)}\right), \tag{13}$$

where K_D refers to the open-circuit stiffness:

$$K_D = K_E + \frac{\alpha^2}{C_0}. \tag{14}$$

Although the second equation of (13) (giving the value of φ) is not transcendental and can be resolved in the range [−π/2; π/2] using:

$$\cos(\varphi) = \frac{1}{\sqrt{1 + \left(\frac{C\omega}{-M\omega^2 + K_D - \frac{\alpha^2}{C_0}\frac{2}{\pi}\frac{1+\gamma}{1-\gamma}\beta_0\cos(\varphi)}\right)^2}}, \tag{15}$$

the obtained expression of φ remains extremely long, while being relatively straightforward to obtain.

Theoretical Comparison

The previous expressions, therefore, demonstrate the self-tuning process through the closed-loop form of the actuation effect. In this section, a comparative analysis between the deflection obtained with or without the use of the self-tuned actuator is given. In order to make the following charts as independent as possible from the model parameters, the frequency is normalized so that 0 corresponds to the open-circuit resonance frequency and ± 0.5 to the cut-off frequencies (frequencies corresponding to a 3 dB decrease of the deflection) when no control is applied. The axis corresponding to the tuning parameter is referenced to the value of the figure of merits defined as the product of the squared coupling coefficient k^2 by the mechanical quality factor Q_M, multiplied by β_0. The value of the inversion coefficient γ is set to 0.75 (which is a typical value in case of autonomous systems).

The obtained deflection magnitude normalized with respect to its maximal value is depicted in Figure 5. This chart clearly shows the bandwidth enhancement offered by the proposed technique, compared to the case where no tuning is done ($\beta_0 k^2 Q_M = 0$). Figure 6 shows the value of the tuning angle φ, also as a function of the normalized frequency and tuning factor.

From Figures 5 and 6, it can be seen that ideal operation of the device (i.e., flat deflection response) is achieved for a linear angle variation with the frequency. The robustness of the closed-loop control is therefore strongly related to the maximum voltage source value, and hence the tuning coefficient β_0. Although one may think that the value of the tuning coefficient, therefore, has to be set as high as possible, a careful energy analysis should be carried out in order to determine the total energy produced by the harvester.

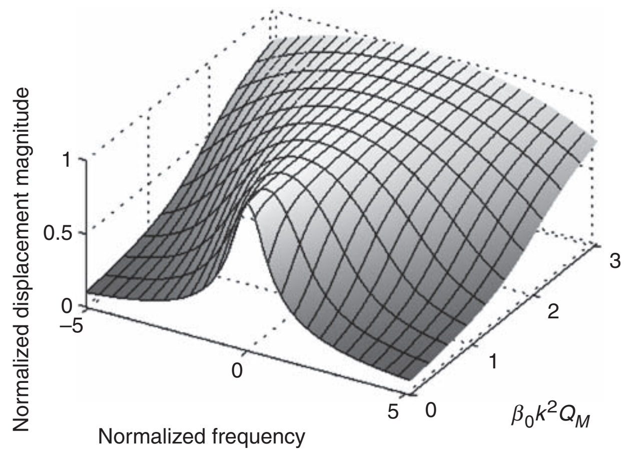

Figure 5. *Theoretical normalized deflection magnitude as a function of the frequency and tuning parameters.*

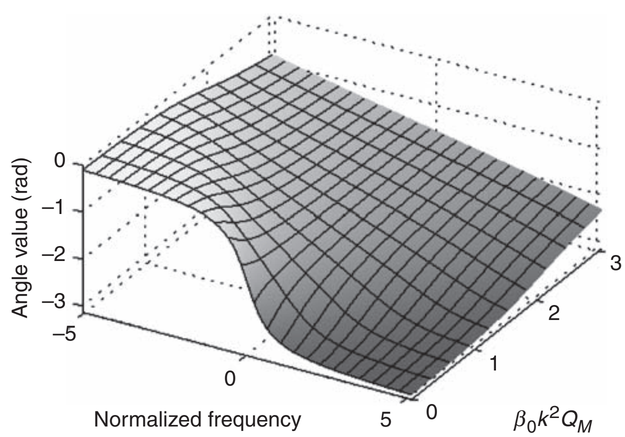

Figure 6. *Theoretical value of the angle as a function of the frequency and tuning parameters.*

Energy analysis

The purpose of this subsection is to investigate the feasibility of the proposed self-tuning method, regarding the power constraints of self-powered microgenerators. A particular emphasis will be placed on the energy required versus the energy harvested, which will be compared with the harvesting abilities of the standard approach (i.e., no tuning).

Considering both the harvesting and frequency self-tuning stages, the global electromechanical equation is given by:

$$\begin{cases} M\ddot{u} + C\dot{u} + K_E u = -\mu_1 M a - \alpha_{\text{act}} V_{\text{act}} - \alpha_{\text{harv}} V_{\text{harv}} \\ \dfrac{V_{\text{harv}}}{R_L} = \alpha_{\text{harv}} \dot{u} - (C_0)_{\text{harv}} \dot{V}_{\text{harv}} \end{cases}$$

$$\text{with } \alpha_{\text{act}} V_{\text{act}} = \begin{cases} 0 \text{ (uncontrolled system)} \\ \dfrac{{\alpha_{\text{act}}}^2}{(C_0)_{\text{act}}} \left(1 - \beta_0 \frac{2}{\pi} \frac{1+\gamma}{1-\gamma} \cos\varphi \right) \\ \text{(controlled system)} \end{cases}, \tag{16}$$

where V_{act} and V_{harv} refer to the actuator and harvester voltages, respectively, and α_{act} and α_{harv} to the actuator and harvester force factors. $(C_0)_{\text{harv}}$ and $(C_0)_{\text{act}}$ respectively denote the harvester and actuator clamped capacitance.

Without any actuation (actuator left in short circuit), the expression of the frequency domain deflection yields:

$$\frac{U(\omega)}{A(\omega)} = \frac{-\mu_1 M}{-M\omega^2 + j\omega \left(C + \dfrac{{\alpha_{\text{harv}}}^2 R_L}{1 + j(C_0)_{\text{harv}} R_L \omega} \right) + K_E} \tag{17}$$

with the power output power given as:

$$P(\omega) = \frac{1}{2} \frac{R_L \omega^2 {\alpha_{\text{harv}}}^2}{1 + \left(R_L (C_0)_{\text{harv}} \omega\right)^2} {u_M}^2 \tag{18}$$

When using the self-tuning approach, the power expression remains the same, but the global deflection is given by:

$$\begin{cases} \dfrac{U(\omega)}{A(\omega)} = \dfrac{-\mu_1 M}{-M\omega^2 + j\omega\left(C + \dfrac{{\alpha_{\text{harv}}}^2 R_L}{1 + j(C_0)_{\text{harv}} R_L \omega}\right) + K_E + \dfrac{{\alpha_{\text{act}}}^2}{(C_0)_{\text{act}}}\left(1 - \dfrac{2}{\pi}\dfrac{1+\gamma}{1-\gamma}\beta_0 \cos(\varphi)\right)} \\ \text{with } \varphi = -\arctan\left(\dfrac{\omega\left(C + \dfrac{{\alpha_{\text{harv}}}^2 R_L}{1+\left((C_0)_{\text{harv}} R_L\omega\right)^2}\right)}{-M\omega^2 + K_E + \dfrac{{\alpha_{\text{harv}}}^2 (C_0)_{\text{harv}} {R_L}^2 \omega^2}{1 + \left((C_0)_{\text{harv}} R_L \omega\right)^2} + \dfrac{{\alpha_{\text{act}}}^2}{(C_0)_{\text{act}}}\left(1 - \dfrac{2}{\pi}\dfrac{1+\gamma}{1-\gamma}\beta_0 \cos(\varphi)\right)}\right) \end{cases}, \tag{19}$$

where $(C_0)_{\text{act}}$ refers to the actuator clamped capacitance. Hence, this expression allows assessing the power output of the microgenerator. However, in order to have a comprehensive view of the self-tuned harvester performance, the required energy for the actuation also needs to be estimated.

REQUIRED ENERGY FOR TUNING

The energy necessary to achieve the tuning can be decomposed into two parts. The first part corresponds to the power required for processing the frequency detection. This power corresponds to the requirements for computing the product of the sensors' signals. The typical consumption of such devices are within the range of a few tens to hundreds of microwatts (Vilchcs et al., 2003; Chong et al., 2005), and the power requirement can be considered as constant. Considering realistic applications (i.e., for powering up electronic devices), the harvester should be designed for supplying a few milliwatts under the considered excitation levels. Moreover, the power required for the computation strongly depends on the kind of digital circuit used. Actually, the energy required for processing would be theoretically limited (and becoming lower and lower due to the progresses in microelectronics, as demonstrated by the Phoenix processor that consumes only 30 pW in sleep mode (Seok et al., 2008), or the commercially available PIC16F688 from Microchip® featuring Nanowatt technology), as it only consists of signal processing, and therefore no energy is explicitly required (contrary to actuation where an energy conversion stage appears).

Hence, it will be considered here that the most critical part remains the power required for actuation. In order to apply the proposed non-linear stiffness tuning, it can be shown that the required power is given by (Guyomar et al., 2008):

$$P_{\text{act}} = \frac{\omega}{2\pi}\frac{1+\gamma}{1-\gamma}\frac{{\alpha_{\text{act}}}^2}{(C_0)_{\text{act}}}{\beta_0}^2 \cos(\varphi)^2 {u_M}^2. \tag{20}$$

This power is, therefore, varying with the phase between the natural (uncontrolled) frequency of the harvester and the base acceleration. Particularly, as the frequency is shifting away from the natural frequency of the harvester, the power required for tuning the device becomes more significant.

COMPARISON OF THE ENERGY HARVESTING ABILITIES

The power outputs of the unmatched and self-tuning harvesters, as well as the total energy produced by the self-tuning harvester (taking into account the actuation power requirements) are depicted in Figure 7, where the tuning coefficient β_0 is set to 1 and the inversion factor to $\gamma = 0.75$. These charts are normalized with respect to

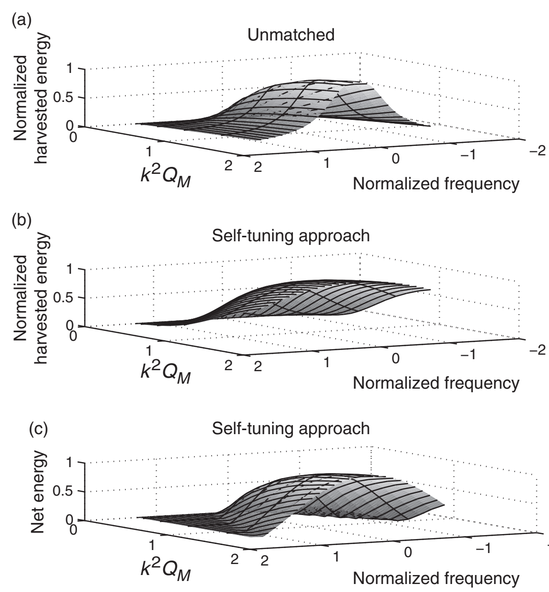

Figure 7. *Energy output and energy produced by the self-tuning harvester vs non-tuned harvester: (a) harvested energy for the uncontrolled system; (b) harvested energy for the controlled system; (c) total energy output for the controlled system.*

Figure 8. *Experimental set-up.*

the maximal output power of the untuned harvester along the z-axis, and are indexed according to the figure of merits given by the product of the mechanical quality factor by the squared coupling coefficient. The frequencies are normalized as previous. This figure demonstrates the ability of the proposed control law for magnifying the bandwidth, even when taking into account the power required for the actuation. Hence, especially for large values of k^2Q_M, the proposed technique leads to an increase of the bandwith of more than 100%, making the harvester much less sensitive to frequency drifts.

Table 1. System parameters.

Parameter	Value
Dymanic mass M	0.95 g
Structural damping coefficient C	0.025 N s/m
Short-circuit stiffness K_E	459.5 N/m
Actuator blocking capacitance $(C_0)_{act}$	120 nF
Actuator force factor α_{act}	1.1 mN/V
Harvester blocking capacitance $(C_0)_{harv}$	128 nF
Harvester force factor α_{harv}	1.9 mN/V
Correction factor μ_1	1.1
Inversion factor γ	0.95

EXPERIMENTAL VALIDATION

This section aims at experimentally validating the previously exposed approach for increasing the bandwidth of an energy harvesting system, with a particular emphasis placed on the deflection magnitude of the considered structure and harvested power.

Experimental Set-up

The experimental set-up consists of a piezoeletric bimorph, equipped with an additional piezoelectric disk for sensing the deflection of the beam, excited at $0.5\,g$ using a shaker driven by a function generator through a power amplifier, as depicted in Figure 8. One layer of the piezoelectric bimorph is used as an actuator for applying the stiffness control, while the other one is connected to a resistor of 18 kΩ,[3] which is close to the optimal load value, for energy harvesting purposes. The piezoelectric materials used to create the bimorph are $QP16n$ manufactured by Midé®, whose dimensions are $5 \times 2.5 \times 0.025$ cm^3. The supplementary piezoelectric material (made from a piezoelectric buzzer) bonded on the bimorph aims at sensing the beam deflection for deriving the value of the switching voltage V_S (together with an accelerometer giving the base acceleration). Both the displacement sensor and accelerometer are connected to a digital signal processor (DSP) (*dSpace*™ system), which also computes the value of the switching voltage V_S, and controls a digital switch for applying the non-linear stiffness control.

Preliminary measurements have been carried out in order to the identify model parameters as well, which are given in Table 1.

Results and discussion

Experimental results as well as theoretical predictions of the deflection amplitude are depicted in Figure 9. Figure 10 shows the corresponding experimental and theoretical harvested powers, whose maximal value is 0.45 mW (corresponding to a power density of 2.9 mW/cm^3/g^2 of active layers, that is, not taking into acount the beam and sensor volumes). These results, in good agreement with theoretical predictions, clearly demonstrate the ability of the proposed control scheme for efficiently increasing the -3 dB bandwidth[4] of the system by a factor up to 4 (from 4.1 to 17 Hz theoretically when $\beta_0 = 0.5$), while slightly shifting the resonance to lower frequencies (leading to a small gain in terms of

[3]The optimal load is proportional to the inverse frequency. Hence, for a frequency variation of a few percent (which is a typical and realistic value for most system), the drift in the optimal load would not lead to significant drop in the power output when using a constant load.

[4]The -3 dB bandwidth is defined as the frequency range where the deflection is greater than $1/\sqrt{2}$ of its maximal value, or equivalently the frequency range where the harvested power is greater than half its maximal value.

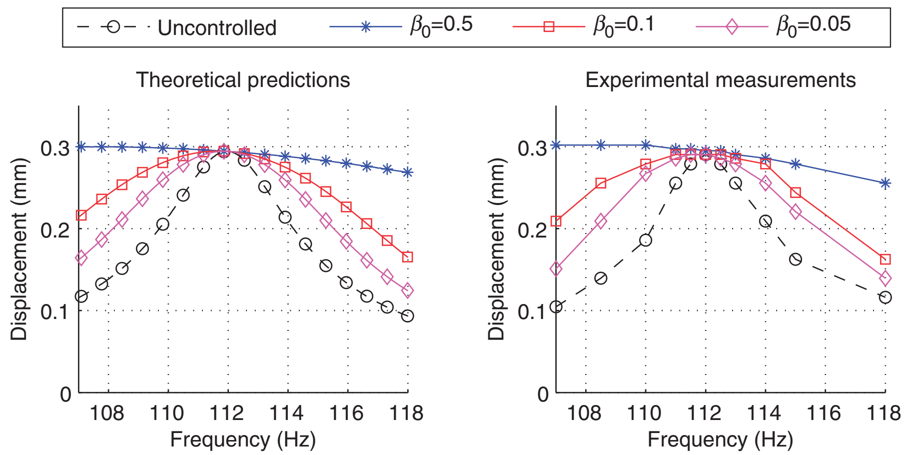

Figure 9. *Experimental measured deflection and comparison with theoretical deflection.*

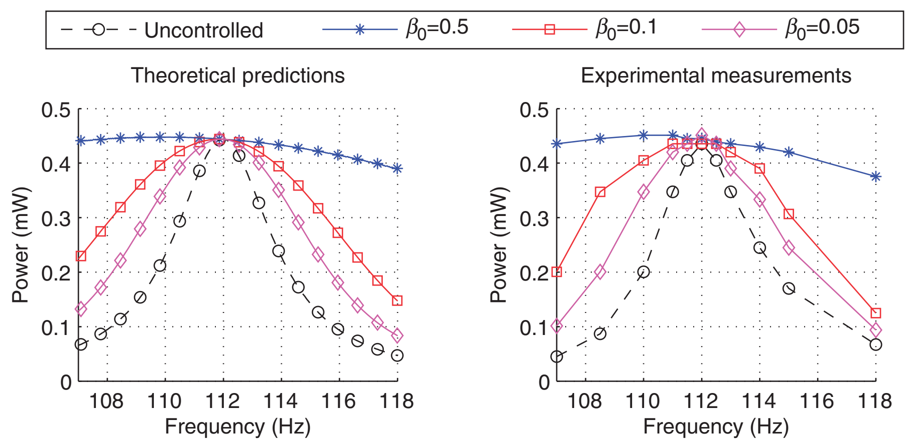

Figure 10. *Experimental maximal harvested power and comparison with theoretical maximal power as the function of the frequency.*

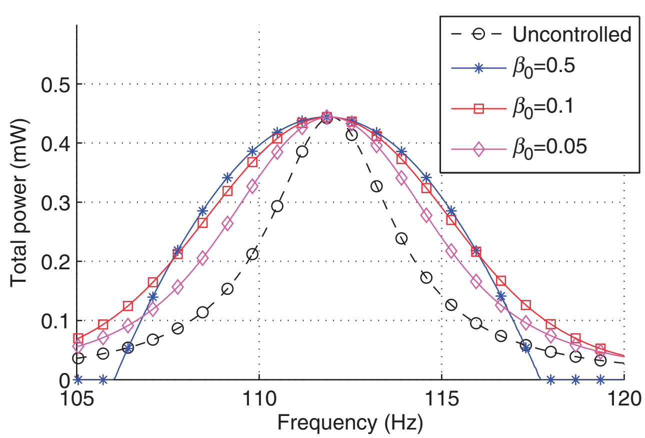

Figure 11. *Produced energy estimation from experimental parameters.*

harvested power as well), in a self-sensing fashion. Hence, the proposed approach allows a large increase in the bandwidth without compromising the deflection magnitude.

Figure 11 depicts an estimation of the total harvested power (i.e., taking into account actuation cost) from the identified model parameters, showing the effectiveness of the proposed technique for increasing the power bandwidth by a factor of 2 (from 4.1 to 8.1 Hz), without compromising the maximal power that can be harvested from the microgenerator (i.e., no damping effect is induced by the control). It can be noted that, for large values of β_0, the net power becomes negative for frequencies far away from the resonance frequency (cropped to 0 in Figure 11, as this means that the microgenerator cannot work in a self-powered fashion), the actuation cost being higher than the harvested power in this case.

CONCLUSIONS

This article proposed a complete, fully working self-sensing method for ensuring the resonance excitation of vibration-based seismic energy harvesters, as well as a comprehensive analysis of the sensing and actuation mechanisms. The total energy that can be harvested, taking into account the actuation cost, has also been estimated, exhibiting a significant increase of the harvester robustness facing a frequency drift, without compromising the deflection magnitude and voltage levels. Based on low-power frequency sensing relying on the phase between the base acceleration and piezoelement deflection, as well as a cost-effective stiffness tuning using switched piezomaterial, it has been demonstrated that the exposed technique allows a fine tuning of the resonance frequency on a wide range, whatever the excitation frequency is (i.e., higher or lower than the uncontrolled system), while ensuring a net positive energy output that features a wider bandwidth than the uncontrolled system.

ACKNOWLEDGMENTS

The authors would gratefully acknowledge of the support of the U.S. Department of Commerce, National Institute of Standards and Technology, Technology Innovation Program, Cooperative Agreement Number 70NANB9H9007.

REFERENCES

Anton, S.R. and Sodano, H.A. 2007. "A Review of Power Harvesting using Piezoelectric Materials (2003–2006)," *Smart Mater. Struct.*, 16:R1–R21.

Badel, A., Guyomar, D., Lefeuvre, E. and Richard, C. 2005. "Efficiency Enhancement of a Piezoelectric Energy Harvesting Device in Pulsed Operation by Synchronous Charge Inversion," *J. Intell. Mater. Syst. Struct.*, 16:889–901.

Badel, A., Lagache, M., Guyomar, D., Lefeuvre, E. and Richard, C. 2007. "Finite Element and Simple Lumped Modeling for Flexural Nonlinear Semi-passive Damping," *J. Intell. Mater. Syst. Struct.*, 18:727–742.

Challa, V.R., Prasad, M.G., Shi, Y. and Fisher, F.T. 2008. "A Vibration Energy Harvesting Device with Bidirectional Resonance Frequency Tenability," *Smart Mater. Struct.*, 17:015035.

Chong, K.-S., Gwee, B.-H. and Chang, J.S. 2005. "A Micropower Low-voltage Multiplier with Reduced Spurious Switching," *IEEE Trans. Very Large Scale Integr. (VLSI) Syst.*, 13: 255–265.

Erturk, A. and Inman, D.J. 2008. "Issues in Mathematical Modeling of Piezoelectric Energy Harvesters," *Smart Mater. Struct.*, 17:065016.

Glynne-Jones, P., Beeby, S.P. and White, N.M. 2001. "Towards a Piezoelectric Vibration-powered Microgenerator," *Sci. Meas. Technol., IEE Proc.*, 148:68–72.

Guyomar, D., Badel, A., Lefeuvre, E. and Richard, C. 2005. "Towards Energy Harvesting using Active Materials and Conversion Improvement by Nonlinear Processing," *IEEE Trans. Ultrason., Ferroelect., Freq. Contr.*, 52:584–595.

Guyomar, D., Lallart, M. and Monnier, T. 2008. "Stiffness Tuning Using a Low-cost Semi-active Nonlinear Technique," *IEEE/ASME Trans. Mech.*, 13:604–607.

Guyomar, D., Sebald, G., Pruvost, S., Lallart, M., Khodayari, A. and Richard, C. 2009. "Energy Harvesting from Ambient Vibrations and Heat," *J. Intell. Mater. Syst. Struct.*, 20:609–624.

Hamakawa, Y. 2003. "30 Years Trajectory of a Solar Photovoltaic Research," In: *3rd World Conference on Photovoltaic Energy Conversion*, Osaka, Japan.

Keawboonchuay, C. and Engel, T.G. 2003. "Electrical Power Generation Characteristics of Piezoelectric Generator Under Quasi-static and Dynamic Stress Conditions," *IEEE Trans. Ultrason., Ferroelect., Freq. Contr.*, 50:1377–1382.

Krikke, J. 2005. "Sunrise for Energy Harvesting Products," *IEEE Pervasive Comput.*, 4:4–35.

Lallart, M., Garbuio, L., Petit, L., Richard, C. and Guyomar, D. 2008b. "Double Synchronized Switch Harvesting (DSSH): A New Energy Harvesting Scheme for Efficient Energy Extraction," *IEEE Trans. Ultrason., Ferroelect., Freq. Contr.*, 55:2119–2131.

Lallart, M. and Guyomar, D. 2008. "An Optimized Self-powered Switching Circuit for Non-linear Energy Harvesting with Low Voltage Output," *Smart Mater. Struct.*, 17:035030.

Lallart, M., Guyomar, D., Jayet, Y., Petit, L., Lefeuvre, E., Monnier, T., Guy, P. and Richard, C. 2008a. "Synchronized Switch Harvesting Applied to Selfpowered Smart Systems : Piezoactive Microgenerators for Autonomous Wireless Receiver," *Sens. Actuators A*, 147:263–272.

Lefeuvre, E., Badel, A., Richard, C., Petit, L. and Guyomar, D. 2006. "A Comparison Between Several Vibration-powered Piezoelectric Generators for Standalone Systems," *Sens. Actuators A*, 126:405–416.

Leland, E.S. and Wright, P.K. 2006. "Resonance Tuning of Piezoelectric Vibration Energy Scavenging Generators Using Compressive Axial Preload," *Smart Mater. Struct.*, 15:1413–1420.

Peters, C., Maurath, D., Schock, W., Mezger, F. and Manoli, Y. 2009. "Closed Loop Wide Range Tunable Mechanical Resonator for Energy Harvesting Systems," *J. Micromech. Microeng.*, 19:094004.

Priya, S. 2005. "Modeling of Electric Energy Harvesting Using Piezoelectric Windmill," *Appl. Phys. Lett.*, 87:184–101.

Roundy, S., Wright, P.K. and Rabaey, J. 2003. "A Study of Low Level Vibrations as a Power Source for Wireless Sensor Nodes," *Comput. Commun.*, 26:1131–1144.

Roundy, S. and Zhang, Y. 2005. "Toward Self-tuning Adaptive Vibration-based Microgenerators," In*: Proceedings of SPIE Smart Structures, Devices, and Systems II*, Vol. 5649, pp. 373–384.

Seok, M., Hanson, S., Lin, Y.-S., Foo, Z., Kim, D., Lee, Y., Liu, N., Sylvester, D. and Blaauw, D. 2008. "The Phoenix Processor: A 30 pW Platform for Sensor Applications," In: *2008 IEEE Symposium on VLSI Circuits*, Honolulu, HI, pp. 188–189.

Shearwood, C. and Yates, R.B. 1997. "Development of an Electromagnetic Microgenerator," *Electronics Lett.*, 33:1883–1884.

Sodano, H.A., Inman, D.J. and Park, G. 2004. "A Review of Power Harvesting from Vibration using Piezoelectric Materials," *Shock and Vib. Digest*, 36:197–205.

Vilchcs, A., Fobelets, K., Michelakis, K., Despotopoulos, S., Papavassiliou, C., Hackbanh, T. and Konig, U. 2003. "Monolithic Micropower Amplifier Using SiGe n-MODFET Device," *Electronics Lett.*, 39:884–886.

Zhu, D., Roberts, S., Tudor, J. and Beeby, S. 2008. "Closed Loop Frequency Tuning of a Vibration-based Micro Generator," *Proceedings of PowerMEMS 2008*, Sendai, Japan, pp. 229–232.