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Buckling suspended graphene nanoribbons to harvest energy from noisy vibrations

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ABSTRACT

Most mechanical vibrations in our environment can be classified as noisy vibrations, since they have no preferred frequency and a spectrum that spreads to the low frequency range. Bistable systems have shown to be a solution to the existing frequency mismatch between the energy source and the harvester device. In this work a parametric study is carried out in order to show the dependence of these improvements with the quality factor Q of a vibrating beam and the different responses when driven by different types of model noise. Specifically, we studied Colored Gaussian Noise instead of the much more common White Gaussian Noise, considered as a reference in most studies.

1. Introduction

Mechanical vibration harvesters are mostly based on resonators capable of transducing the energy from the mechanical to the electrical domain through piezoelectric, electromagnetic or capacitive strategies [1]. The need to deal with low intensity vibrations makes the use of MEMS/NEMS suitable in order to improve the response of the device in terms of the ratio between input and output power. The reduction of the dimensions of such a resonator increases dramatically their resonant frequency [2,3]. This feature and the typical frequency selectivity of these devices make them not optimal to harvest energy from broad band noises, particularly when it extends to the low frequency range.

2. Modeling

In order to address the problems described above we propose a bistable device based on a suspended graphene nanoribbon: bistability has been demonstrated to improve the response in comparison with that of resonators when driven by noise [4]. The bistability is achieved by applying compression as it is shown in Fig. 1. Once the compression is applied the resonating behavior is broken and two new attractors appear symmetrically positioned with respect to the initial stable point ($x = 0$). The description of the graphene nanoribbon is done through the determination of the elastic potential energy through *ab initio* calculations as it is explained in [5]. Fig. 2 shows different potential energy curves that highlight the role of the compressive strain, i.e. the larger the compression, the larger the separation of the two attractors and the higher the barrier are. The inset shows the dependence of the barrier height and the minima position with the compression, which is the non-linearizing parameter. It is worth noting the strong non-harmonic shape the uncompressed case shows. Therefore, even when the system is not strained it shows a non-resonating behavior.

In this work we extend our previous report [5] by studying the response to different types of noise. Specifically, we start with a White Gaussian Noise (WGN) and then filter it to selectively eliminate the contributions of certain frequency range, as discussed below. Additionally, we focus on the dependence of the performances of the harvesting device on the quality factor Q, which is a highly variable parameter and might vary significantly from realization to realization of the device.

3. Simulation and results

To characterize the dynamics of this kind of system a Langevine differential equation of motion must be solved numerically:

$$m_{eff} \cdot x'' = -dE_p/dx - b \cdot x' + F_n(t) \quad (1)$$

where m_{eff} stands for the effective mass as it is defined in the frame of the spring-mass model [6] and x, x' and x'' are the position vector and its first and second time derivatives, respectively. The constant b accounts for the losses in the system which we assume to be dominated by friction processes, and can be expressed in terms of the quality factor as follows:

$$b = m_{eff} \cdot \omega_0/Q \quad (2)$$

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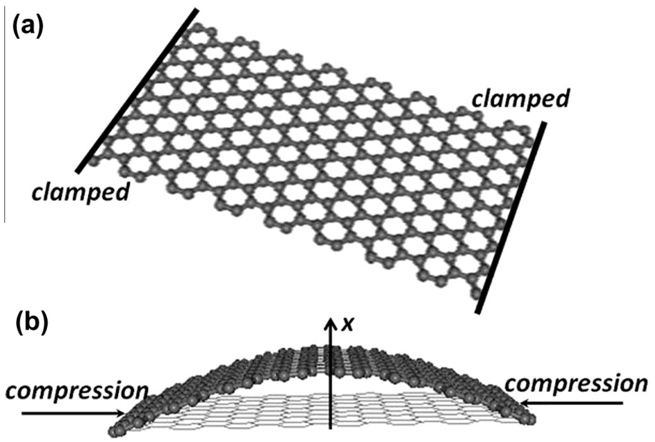

Fig. 1. (a) Scheme of the structure showing the two clamped ends and the *arm-chair* configuration along the longitudinal axis it has been considered. (b) Illustration of the compression applied in the *arm-chair* direction in order to produce the bistability. The *x*-coordinate is shown and it is considered as the only displacement direction of the mechanical system in the dynamical regime.

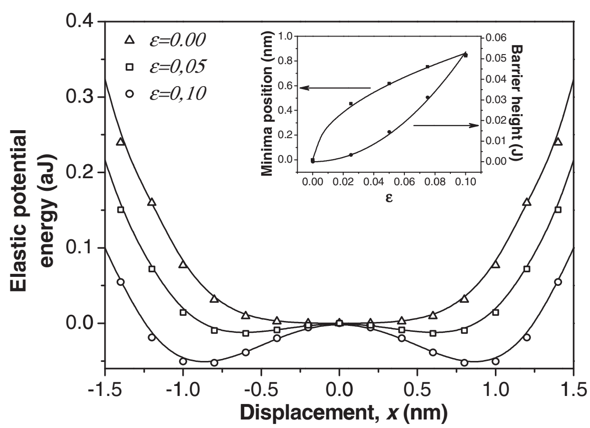

Fig. 2. Elastic potential energy for three different compressions starting from the non-stressed configuration. The inset shows the dependence of the well position and the potential barrier height between minima with the compression: $x_{\pm} = 8.64\sqrt{\varepsilon}$ Å; $\Delta V = 0.05\ \varepsilon^2 aJ$.

F_n describes the force suffered by the oscillator when it is driven by a mechanical vibration and it will have a root mean square value of 2.2 pN from now on, which corresponds to a sound pressure of approximately 200 dB.

In order to describe the dynamics of the system we considered a WGN excitation. Although real vibrations are better approximated by a colored noise, we first analyze this reference case due to its relevance to previous literature. Fig. 3 shows the dynamics in terms of trajectory and phase portrait. Three different working regimes, corresponding to different compression values, are displayed. For a non-stressed graphene system the dynamics is that of a resonator with a single attractor i.e. it oscillates around $x = 0$. For larger values of ε the trajectory shows oscillations around just one of the two wells. However, at intermediate compressions the system can suffer transitions from one attractor to the other, then increasing the root mean square of the position vector as it is shown in Fig. 4. The increase of x_{rms} is understood as an improvement with respect to the non-stressed case and it is very closely related to the capability of generating electric power when allowing the transduction [4]. In order to compute the generated power, a piezoelectric method of transduction is considered, as previously proposed in [5]. Fig. 4 shows a shift of the peak towards smaller compression values between the x_{rms} computed when no transduction is considered and when it is. It can be easily understood in terms of available energy: during transduction some of the energy is extracted, thus leaving less energy to overcome the potential barrier. Notice that there is also a difference between the optimal compression for the x_{rms} and this for the P_{rms}. Under certain conditions there can be an absolute match [7], though these cannot be applied in our particular case due to the very high time constant $\tau = RC$ of the electromechanical system [5].

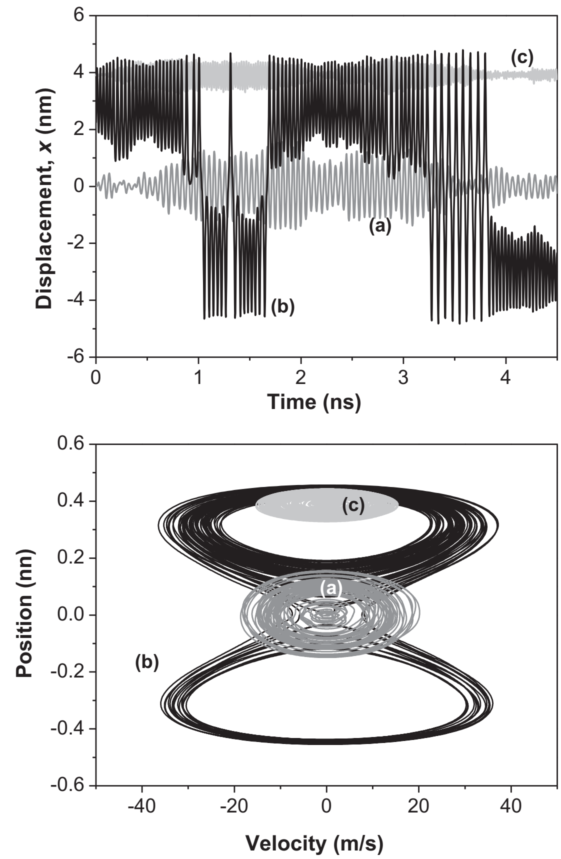

Fig. 3. Displacement vs time and phase portrait for the three different regimes: (a) non compressed case. The system oscillates around the attractor positioned at $x = 0$. (b) Medium compression applied. The excitation makes the system to cross from one well to the other. (c) Large compression. The system gets stuck in one of the two minima.

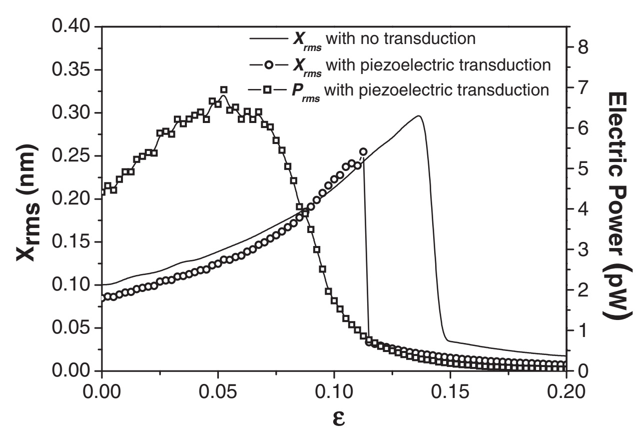

Fig. 4. Root mean square of the displacement *x* and the generated electric power as a function of the applied compression.