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Piezomagnetic vibration energy harvester with an amplifier

João Pedro Norenberg [a], Americo Cunha Jr [b], Piotr Wolszczak [c], Grzegorz Litak [c,*]

[a] Department of Mechanical Engineering, São Paulo State University, Ilha Solteira, SP, Brazil
[b] Department of Applied Mathematics, Rio de Janeiro State University, Rio de Janeiro,RJ, Brazil
[c] Department of Automation, Lublin University of Technology, Lublin, Poland

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ABSTRACT

We study the effect of an amplification mechanism in a nonlinear vibration energy harvesting system where a ferromagnetic beam resonator is attached to the vibration source through an additional linear spring with a damper. The beam moves in the nonlinear double-well potential caused by interaction with two magnets. The piezoelectric patches with electrodes attached to the electrical circuit support mechanical energy transduction into electrical power. The results show that the additional spring can improve energy harvesting. By changing its stiffness, we observed various solutions. At the point of the optimal stiffness of the additional spring, the power output is amplified a few times depending on the excitation amplitude.

The increasing applications of electricity with the limited conditions of renewable sources have resulted in the search for more efficient uses of available ambient kinetic energy [1]. Different ways of energy transduction engage electromagnetic, piezoelectric, and magnetostrictive phenomena [2]. Account for the high energy density the piezoelectric transducers were adopted for kinetic energy harvesting. Initially, investigations were focused on linear systems, which had a good performance provided that they worked in the resonance conditions [3]. However, variability of the ambient vibration conditions, including frequency and amplitude, denies the wide applications. To overcome this deficiency, the nonlinearities were introduced to the system [4,5] to obtain a broadband frequency effect. Among the nonlinear systems bistable [4,6–8] or multistable [9] systems attracted more interest from the researchers.

In this context, multiple degrees of freedom systems were also studied [10,11]. In such a situation, frequency bandwidth is broadened by the multiple resonances appearance [12]. The other possible consequence can be the internal resonance and quasi-periodic solutions [13,14]. Based on two degrees of freedom system, the displacement amplifier to the resonator was proposed [15]. This structure can help to supply a higher amplitude of excitation dealing with bistable piezoelectric systems [4] that require a large amount of input energy to overcome the potential barrier.

In the present work, we continue this direction of study, quantifying how the amplifier stiffness affects energy harvesting. We also contribute to briefly investigating the dynamic behavior of a higher amplification factor system. Our system is composed of a piezo-magneto-elastic beam energy harvester with an additional spring and a damper (Fig. 1). Fig. 1a shows the real system, Fig. 1b illustrates the model schematics, and Fig. 1c exhibits the double-well potential of the piezo-magneto-elastic beam. The main resonator is a ferromagnetic beam with piezoelectric patches attached to the moving frame. Two magnets destabilize the vertical position of the beam by attraction to the left and right stable equilibrium positions. Two coupled masses m_1 and m_2 (Fig. 1b) are governed by the set of two differential equations (Eq. (1)). They are forced by a harmonic base excitation. Finally, the piezoelectric elements provide the electromotive force to the resistive circuit (Eq. (2)). The corresponding set of the equations of motion reads:

$$m_1\ddot{x}_1 = -c_1(\dot{x}_1-\dot{Y}) + c_2(\dot{x}_2-\dot{x}_1) - k_1(x_1-Y) + k_2(x_2-x_1) + k_{2n}(x_2-x_1)^3 + \theta v, \tag{1}$$

$$m_2\ddot{x}_2 = -c_2(\dot{x}_2-\dot{x}_1) - k_2(x_2-x_1) - k_{2n}(x_2-x_1)^3 - \theta v, \quad C_p\dot{v} + \frac{1}{R_l}v + \theta(\dot{x}_2-\dot{x}_1) = 0, \tag{2}$$

where the symbols (appearing in Eqs. (1) and (2)) represent the system parameters, and they are specified in Table 1 together with their values (see in [16,17]) used in the following simulations, while x_1, x_2 are the displacements of the first and the second masses and $Y(t)$ is the kinematic base excitation.

The original system in the set of Eq. (1) is bistable, while the additional spring and damper provide a linear coupling between the system and the excitation. The results of them are presented in the next figures. The mean power output against the additional spring stiffness value is presented in Fig. 2. We computed the system responses for three dif-

* Corresponding author.
E-mail address: g.litak@pollub.pl (G. Litak).

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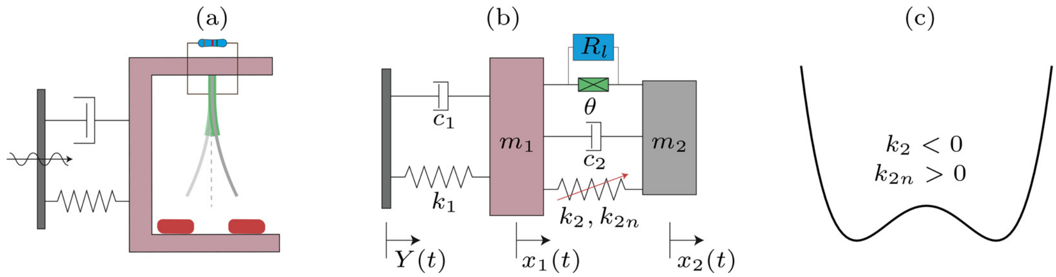

Fig. 1. Piezo-magneto-elastic beam energy harvester with displacement amplifier mechanism: (a) real system schematic, (b) electromechanical model, and (c) general shape of double-well potential. Symbols and coordinates are defined in Eqs. (1) and (2) and Table 1.

Table 1
System parameters.

Mass of the amplifier, m_1	66 g
Mass of the beam, m_2	22 g
Damping coef. of the amplifier, c_1	0.1 N·s/m
Damping coef. of the beam, c_1	0.125 N·s/m
Stiffness of the amplifier, k_1	0.1-10^6 N/m
Linear stiffness of the beam, k_2	-63.451 N/m
Nonlinear stiffness of the beam, k_{2n}	634 509 N/m^3
Effective piezoelectric coupling, θ	-4.57 m·N/V
Effective piezoelectric capacitance, C_p	43 nF
Electrical loading, R_l	5 kΩ

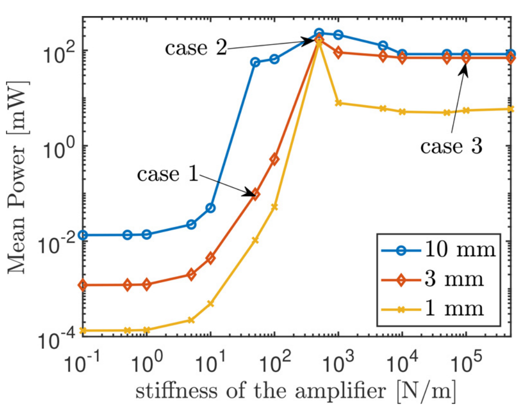

Fig. 2. Mean power recovered against the stiffness values of the amplifier mechanism under different excitation displacements when the excitation frequency is 10 Hz. The results are obtained for a single set of initial conditions of minimum energy. The three cases highlighted will be discussed later.

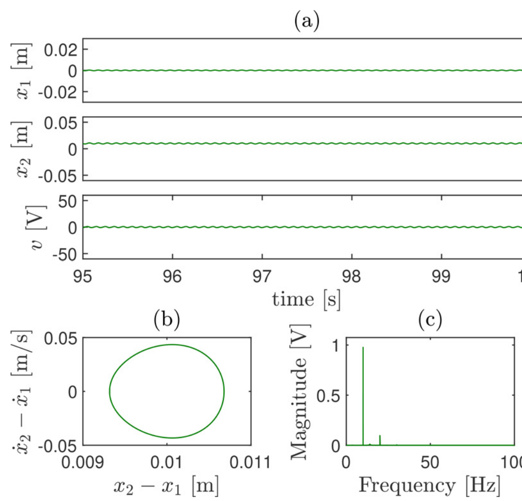

Fig. 3. Dynamic behavior of case 1 that the stiffness is 50 N/m: (a) serial time of the amplifier displacement, beam displacement, and the recovered voltage, (b) phase portrait of the beam, and (c) Fourier transform of the voltage.

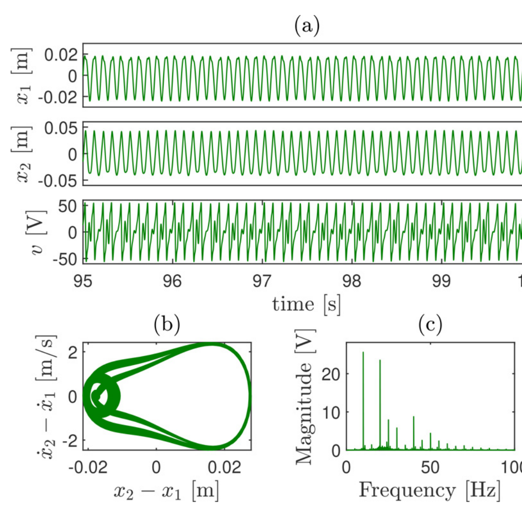

Fig. 4. Dynamic behavior of case 2 that the stiffness is 500 N/m: (a) serial time of the amplifier displacement, beam displacement, and the recovered voltage, (b) phase portrait of the beam, and (c) Fourier transform of the voltage.

ferent excitation amplitudes (1, 3, 10 mm) for an excitation frequency of 10 Hz. The power outputs are represented by non-monotonic curves with a maximum value of stiffness of 500 N/m. Note that in the lower stiffness limit, the system is very flexible, suppressing the energy transfer, while in the higher stiffness limit, the additional spring is very rigid, leading to the original system [4]. The horizontal flat line represents this limit. The logarithmic scale is used to show the amplification more clearly. In this limit power output of the energy harvester is broadband in terms of frequency as was studied in the previous papers [4,7,8]. Still, the amplitude amplification mechanism does not work.

In the next three figures (Figs. 3-5), we study with more detail the cases marked in Fig. 2 by arrows and numbered 1–3, respectively. Fig. 3 shows the low stiffness case. The power and voltage outputs are small. Furthermore, the solution of the resonating beam corresponds to intra-well oscillation. The relative displacement $x_2 - x_1$ is periodic, with a leading frequency coinciding with the excitation frequency and a small super-harmonic 2 component. In contrast to these results, Figs. 4 and 5 show large orbit solutions with inter-well oscillations of the resonator. The highest power output is visible in case 2 for the additional spring stiffness of 500 N/m (see Fig. 2) and studied with details in Fig. 4. This is the optimum stiffness case. The thick line indicates the presence of some fluctuation in the main path on the phase portrait. This effect could appear because of the vicinity of the bifurcation point concerning the stiffness value. This solution is rich in frequencies and possibly possesses small irrational components leading to quasi-periodicity or chaotic behavior. In this case, we observe two leading peaks for the excitation and double-excitation frequencies. There are also many medium and smaller features in the frequency spectrum.

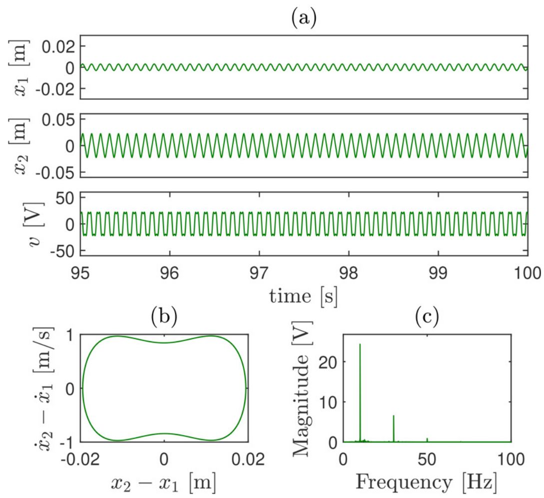

Fig. 5. Dynamic behavior of case 3 that the stiffness is 10^5 N/m: (a) serial time of the amplifier displacement, beam displacement, and the recovered voltage, (b) phase portrait of the beam, and (c) Fourier transform of the voltage.

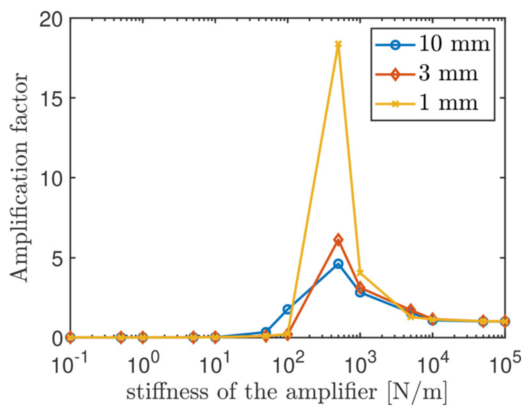

Fig. 6. Amplification factor against the additional stiffness values under different excitation displacement when the excitation frequency is 10 Hz.

At last, Fig. 5 shows the original solution of the reduced system [4]. There is also a high value of power output and a larger orbit with passing through the potential barrier. Here the leading frequency in the frequency spectrum coincides with the excitation. However, the superharmonic 3 component is also significant. This frequency modifies the shape of the resonator phase portrait. To show the system responses in the best condition, we estimate the amplification factor for the three cases of amplitude excitation in Fig. 6. The amplifier demonstrated more efficiency in low excitation amplitude conditions. This result is essential because it can improve the power recovered by the bistable oscillator in its critical situation.

Finally, we want to draw attention to the advantages of the proposed system. Our results show that the power output can be amplified significantly. Furthermore, the optimum condition is robust against the excitation amplitude variation. From the preliminary calculations, the optimum condition would be shifted accordingly by changing the excitation frequency. The bifurcations point were found and will be discussed in the next works with more detail. In the limit of high stiffness, the additional degree of freedom is unimportant, and our system reproduces the original system without the amplifier [4]. The quasi-periodic solution presented in Fig. 4 deserves more explanation. In that system, various solutions could appear. Their properties can be investigated further using nonlinear time series methods as the Lyapunov exponent [18] and multi-scaled entropy [19]. It should also be noted that the optimal amplifier stiffness should be determined after more systematic studies with various initial conditions and excitation frequency sweeps.

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Data Availability Statement

The data supporting this study's findings is openly available in STONEHENGE GitHub repository [20].

Declaration of Competing Interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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