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Using an elastic magnifier to increase power output and performance of heart-beat harvesters

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Abstract

Embedded piezoelectric energy harvesting (PEH) systems in medical pacemakers have been a growing and innovative research area. The goal of these systems, at present, is to remove the pacemaker battery, which makes up 60%–80% of the unit, and replace it with a sustainable power source. This requires that energy harvesting systems provide sufficient power, 1–3 μW , for operating a pacemaker. The goal of this work is to develop, test, and simulate cantilevered energy harvesters with a linear elastic magnifier (LEM). This research hopes to provide insight into the interaction between pacemaker energy harvesters and the heart. By introducing the elastic magnifier into linear and nonlinear systems oscillations of the tip are encouraged into high energy orbits and large tip deflections. A continuous nonlinear model is presented for the bistable piezoelectric energy harvesting (BPEH) system and a one-degree-of-freedom linear mass-spring-damper model is presented for the elastic magnifier. The elastic magnifier will not consider the damping negligible, unlike most models. A physical model was created for the bistable structure and formed to an elastic magnifier. A hydrogel was designed for the experimental model for the LEM. Experimental results show that the BPEH coupled with a LEM (BPEH + LEM) produces more power at certain input frequencies and operates a larger bandwidth than a PEH, BPEH, and a standard piezoelectric energy harvester with the elastic magnifier (PEH + LEM). Numerical simulations are consistent with these results. It was observed that the system enters high-energy and high orbit oscillations and that, ultimately, BPEH systems implemented in medical pacemakers can, if designed properly, have enhanced performance if positioned over the heart.

Keywords

piezoelectric; energy harvester; cardiac pacemaker; elastic magnifier; bistable oscillator; medical device applications

Introduction

The field of piezoelectric energy harvesting utilizes class of materials which produce current under stress or strain. The chemical composite known as PZT (lead-zirconate-titanate) is a piezoelectric class material and is most favored in industry and research. Current and previous research on piezoelectric energy harvesters was completed by Calio *et al* [1] and

Pellegrini *et al* [2]. A common piezoelectric system is the clamped-free cantilever beam. Contributions by Erturk [3] and Erturk *et al* [4] to this system built upon a distributed-parameter electromechanical model in order to accurately describe the dynamics of piezoelectric energy harvesters. They present derivations of the equations of motion as well as closed form solutions to the single-mode equations. Additionally the authors provide optimization techniques for load resistance and power generation.

Although linear energy harvesters were the first technologies to be thoroughly investigated; limitations in versatility, power density, and bandwidth prevented widespread implementation for more practical applications. A need to overcome these limitations has led to the development and investigation of other configurations [5]. Karami *et al* [6] provided a novel zigzag model in order to increase the power output of their piezoelectric energy harvester at lower excitation frequencies. Ansari *et al* [7] developed a vertically stacked fan-fold vibration based energy harvester which the authors have shown to be favorable for volumetrically limited environments.

Of more importance, however, are nonlinear systems. While these systems have a high sensitivity to initial conditions they provide a wide bandwidth and can produce more sustained. Introduction of nonlinearities can be achieved in a number of different ways. Most commonly, these configurations have taken advantage of the magnetic repulsive forces between two opposing magnets to introduce nonlinearity. The magnets force the system into monostability or bistability allowing the tip to oscillate either within the wells or between the wells. Guo *et al* [8] investigated the dynamics of a bistable piezoelectric generator. They found that the interwell limit cycle motion produced more power than interwell chaotic motion and intrawell oscillations. Zhu *et al* [9] provided the modeling and analysis of an unconventional coupled bistable structure. Stanton *et al* [10] develop a detailed derivation of the magnetic interaction of the conventional bistable piezoelectric energy harvester. The authors provide an analytic, numeric, and experimental realization of the bistable system. Karami *et al* [11] developed a unified approximation method for linear, monostable, and bistable structures which analyzed the effects of electro-mechanical coupling on each system caused by variations in damping and excitation frequency. Many other researchers have contributed to the development, understanding, and investigation of nonlinear and bistable piezoelectric energy harvesters [12–17] including work in MEMS [18] and applications for mobile devices [19].

As research and adaptation continue to give way to new methods and ideas there has been significant attention paid to systems coupled with elastic magnifiers. The elastic magnifier is intended to provide an enhancement for the bistable piezoelectric oscillator acting, ironically, like a vibration absorber. It functions by introducing high-energy orbits and amplified output performance. While there are several forms of elastic magnifiers the majority are typically modeled as a linear mass-spring-damper system placed between some substructure and the base. Vasic *et al* [20] and Zhou *et al* [21] presented novel work on multi-mode double beam harvesters. They found this magnifier to provide performance enhancement over a typical energy harvester. Tang *et al* investigated a series of multi-degree of freedom vibration based energy harvesters which resemble an elastic magnifier coupled with a cantilevered beam system [22]. What they found is that by increasing the degree of

freedom and coupling they could create a larger bandwidth and, ultimately, a more practical operating region for power output. Ding *et al* also developed a theory on a beam structure with a continuously distributed spring-mass system [23]. Their goal was to parametrically model a base which supported a body of mass-spring systems such as a crowd of people on a rigid or semi-rigid structure. Aladwani *et al* presented a cantilevered energy harvester with a simple spring magnifier in order to amplify power output [24]. Using finite element theory they found that their coupled system had improved power output performance a factor of 500% higher than a system without a magnifier. Similarly, they reported a bandwidth increase of about 21%. These results, however, hinge on properly selected design parameters. Aldraihem *et al* used Lagrangian dynamics to develop a governing theory for a piezoelectric energy harvesting system coupled with a mass-spring magnifier [25]. Their results are also consistent with increased power output and broader bandwidth operation. Wang *et al* provided an accurate and efficient modelling of a piezoelectric energy harvester with a single degree of freedom elastic system [26]. The authors adjust for and include damping in the elastic magnifier which had previously been left out by others. Their theory also accounts for the strain distribution of the harvesting beam structure. Wang *et al* [27] investigated a bistable piezoelectric energy harvester with an elastic magnifier. In their work they derived equations of motions from a two degree of freedom lumped parameter model. They include damping in the model of the elastic magnifier. Perturbation methods, namely harmonic balance, are used to derive an analytical solution for the system under harmonic inputs. Damping, however, is ignored in both the analytic solution and numeric simulations. They found increased power output and increased bandwidth performance over both the bistable and linear energy harvesters under certain tuned conditions.

As it stands batteries pose an ongoing issue for any implantable device especially those surgically placed in high risk areas. Such is the case for standard and leadless pacemakers who fall under the category of ICD's. Recent studies and advances in the pacemaker field show that most cardiac pacers deplete within eight to twelve years of their installation [28–33]. Costs of replacement surgery are not always financial with risks of infection and mortality increasing with each device switchover. As the market transitions towards leadless pacemakers (a smaller version inserted directly into the heart wall) the push for alternative power sources, namely energy harvesters, now coupled with the heart is evident. Researchers continue to work towards a more thorough understanding of energy harvesters with improved power density and efficiency in designs and approaches for cardiac pacemakers [34] and a variety of implantable medical devices [35]. These include fan-folded designs [7], mass-centered clamped beams [36], and linear and nonlinear harvesters designed for heartbeat vibrations [37].

The aim of this research is to investigate the mechanics of energy harvesting systems with a focus on their implementation in lead-less pacemakers.

The elastic magnifier provides a realistic embodiment of the myocardial heart wall which is the input to the energy harvesting system as seen in figure 1. The energy harvester will be nonlinear in nature with a magnetic tip-mass. An opposing magnetic is aligned such that their poles repel each other. This effectively creating the bistable phenomenon. The nonlinear beam structure, comprised of both the substrate and PZT layers, is considered a

continuous system. The linear elastic magnifier is modeled as a coupled mass-spring-damper system. The governing equations originate from Lagrange-Hamiltonian methods and are used as a basis for the modal analysis of the system. These equations are then non-dimensionalized and used for several case studies; the first with a sinusoidal input and the second with a heart-beat input. A brief analysis of the fanfold system is also conducted. Finally, experimental results under harmonic and heartbeat signals are presented.

Modelling and governing equations

The system is shown in figure 2. This is the standard bimorph BPEH and LEM configuration. The clamped-free cantilever beam has four boundary conditions. The tip mass is a magnet whose poles are oppositely aligned with a stationary opposing magnetic. This interaction of the fields produce the magnetic repulsive forces. The two bimorph PZT strips extend the length of the substrate. The LEM is positioned under the primary beam structure. The entire system is subject to base excitations along the $\hat{z}$ -axis.

The governing equations of the beam structure are presented below. Their derivation is a simple adaptation of the methods provided in [10, 11, 14, 26] which use calculus of variations and energy methods for a continuous system.

$$m_p \frac{\partial^2 w_{\text{rel}}}{\partial t^2} + c \frac{\partial w_{\text{rel}}}{\partial t} + Y_{\text{eq}} I_{\text{eq}} \frac{\partial^2 w_{\text{rel}}}{\partial x^4} = -\alpha \nu(t) \left[\frac{\partial \delta(x)}{\partial x} - \frac{\partial \delta(x-L)}{\partial x} \right] - m_p \times \frac{\partial^2 x_e}{\partial t^2} + a_m w_{\text{rel}} \quad (1)$$

$$- b_m w_{\text{rel}}^3,$$

where m_p is the mass of the beam characterized by $(2\rho_p A_p + \rho_s A_s)$ and ρ_p , A_p , ρ_s , and A_s are the densities and area for the piezoelectric strips and substrate, respectively. c is the damping, Y_{eq} is the equivalent Young's modulus for the bimorph layer, I_{eq} is the equivalent moment of inertia for the bimorph layer, $\alpha = 2e_{13}z_m b$ is an electromechanical coupling coefficient, and ω_p is the natural frequency of the uncoupled beam. The magnetic terms, a_m and b_m , characterize the linear and nonlinear elastic forces of the opposing magnetic fields. The relative deflection, $w_{\text{rel}}(x, t)$, is along the $\hat{z}$ -axis, x_e is the displacement of the elastic system below the beam structure, and $\nu(t)$ is the voltage across the PZT patches. The layers of PZT are connected in parallel. Equation (1) is simplified by considering the separable form of the relative deflection. Note that the total deflection of the system at any position along the beam is then the summation of the relative deflection of the beam structure and the deflection of the elastic system beneath. The assumed mode shape is then the fundamental mode shape of a cantilevered beam [38]. While it is a simplified assumed mode; accuracy is recovered in corrections defined by the constants C_1 through C_4 .

$$w_{\text{rel}}(x, t) = \sum_{i=1}^{\infty} \phi_i(x) \eta_i(t), \quad (2a)$$

$$\dot{w}(x, t) = \dot{w}_{\text{rel}}(x, t) + \dot{x}_e = \dot{\eta}\phi + \dot{x}_e, \quad (2b)$$

$$w(x, t) = w_{\text{rel}}(x, t) + x_e = \eta\phi + x_e, \quad (2c)$$

$$\phi_i(x) = C_1 \sin\left(\frac{\beta x}{\sqrt{C}}\right) + C_2 \cos\left(\frac{\beta x}{\sqrt{C}}\right) + C_3 \sinh\left(\frac{\beta x}{\sqrt{C}}\right) + C_4 \cosh\left(\frac{\beta x}{\sqrt{C}}\right), \quad (2d)$$

where $\phi(x)$ and $\eta(t)$ in equation (2) are the displacement and temporal coordinates for the i th mode, respectively. And where $\beta^4 = \omega^2$, $C^2 = \frac{Y_{eq} I_{eq}}{m_p}$, and C_1, C_2, C_3, C_4 are unknown coefficients associated with the beam. Using the four boundary conditions of the beam structure the four unknown coefficients can be determined numerically. β is the eigenvalue solution to the eigenvector problem outlined in [38] and a function of the natural frequency and material properties of the beam. The continuous vibrations are then simplified by considering the first mode. The governing equation changes from a PDE to an ODE and the modes are decoupled by multiplying by the j th mode and integrating equation (1) over the length of the beam, $\int_0^L \phi_j(x) dx$. Equation (3) is the simplified beam equation.

$$\tilde{m}_p \ddot{\eta}(t) = \tilde{c}_p \dot{\eta}(t) + \tilde{k}_p \eta(t) = K\nu(t) + \tilde{a}_m \eta(t) - \tilde{b}_m \eta^3(t) - \beta \ddot{x}_e. \quad (3)$$

The coupled electromechanical subsystem is:

$$c_o \frac{\partial \nu(t)}{\partial t} + \frac{\nu(t)}{R} = -\alpha \int_0^L \left(\frac{\partial^3 w_{\text{rel}}}{\partial t \partial x^2} \right) dx, \quad (4)$$

where c_o is the capacitance and R is the resistance. Again, using the separable form of the temporal coordinate equation (4) simplifies to:

$$c_o \dot{\nu}(t) + \frac{\nu(t)}{R} = -K \dot{\eta}(t). \quad (5)$$

The constants preceding the time dependent variables are determined from the mechanical and electrical properties of the materials of the system. Commonly, the magnetic coefficients $\tilde{a}_m$ and $\tilde{b}_m$ are found through experimentation, being characterized by the magnetic gap distance and properties of the magnets themselves. The gap distance, however, changes the effective natural frequency of the system and is thus an undesired technique to use in

experimentally tuning and optimizing the system. Rather than experimentally finding the linear and nonlinear magnetic coefficients a backwards solving technique is used. This method assumes the natural frequency of the bistable system and, using that, solves for the unknown coefficients. Since the system being studied is for cardiac pacing the chosen natural frequency has been optimized for a heart construct. The general form of the equation is given in equation (3).

$$\tilde{m}_p \ddot{\eta} + \dots + \tilde{k}_p \eta - \left(a_m \phi(L) \int_0^L \phi(x) dx \right) \eta = \dots \quad (6)$$

Deriving by the equivalent mass term

$$\ddot{\eta} + \dots + \left(\frac{\tilde{k}_p - a_m \phi(L) \int_0^L \phi(x) dx}{\tilde{m}_p} \right) \eta = \dots \quad (7)$$

and recognizing that the stiffness to mass ratio relates to the natural frequency of the nonlinear system:

$$\frac{K_{\text{sys}}}{M} = \frac{\tilde{k}_p - a_m \phi(L) \int_0^L \phi(x) dx}{\tilde{m}_p} = -\omega_{\text{des}}^2 \quad (8)$$

In order for the system to exhibit a nonlinear bistable behavior the linear magnetic coefficient must 'out weigh' the bending stiffness of the beam; otherwise the system will be linear or monostable. This would be reflective of gap distance that is too large. Therefore, the chosen or desired natural frequency must be negative. Solving equation (8) yields the linear elastic magnetic force.

$$a_m = \frac{\omega_{\text{des}}^2 \tilde{m}_p + \tilde{k}_p}{\phi(L) \int_0^L \phi(x) dx} \quad (9)$$

Equation (9) states that given some desired system frequency the magnetic terms and the magnetic gap distance can be determined. The magnetic gap distance, Δ , is

$$\Delta = \left(\frac{\mu_0 \mu_a \mu_b}{4\pi} \frac{12}{a_m} \right)^{\frac{1}{5}} \quad (10)$$

A more detailed derivation of the magnetic repulsive forces was done by Stanton *et al* [10]. Generally, the optimal gap distance is found through experimental validation; however, the functionality of this system encourages a design over experimentation approach. The linear and nonlinear coefficients are then:

$$a_m = \frac{\mu_0 \mu_a \mu_b}{4\pi} \frac{12}{\Delta^5}, \quad (11a)$$

$$b_m = \frac{\mu_0 \mu_a \mu_b}{4\pi} \frac{45}{\Delta^7}. \quad (11b)$$

The elastic magnifier is modeled after a coupled linear mass-spring-damper and is adapted from [27, 45].

$$m_e \ddot{x}_e + c_e(\dot{x}_e - \dot{x}_b) + c_p(\dot{x}_e - w(x, t)) + k_e \times (x_e - x_b) + k_p(x_e - w(x, t)) + \alpha v \quad (12) \\ \times \left[\frac{\partial \delta(x)}{\partial x} - \frac{\partial \delta(x - L)}{\partial x} \right] = 0,$$

where m_e is the modal mass of the elastic system plus the mass of the beam system, and c_e and k_e are the damper and spring coefficients of the elastic system, respectively. Consider the relative form of the temporal coordinate of the elastic magnifier and the relative and separable form of the beam temporal coordinate. Again, decoupling the modes by multiplying by the j th mode and integrating over the length of the beam equation (12) can be reduced.

$$x_e = x_{e_{\text{rel}}} + x_b, \quad (13a)$$

$$\tilde{m}_e \ddot{x}_{e_{\text{rel}}} + \tilde{c}_e \dot{x}_{e_{\text{rel}}} + \tilde{k}_e x_{e_{\text{rel}}} - \tilde{c}_p \dot{\eta} - \tilde{k}_p \eta + K\nu = -\tilde{m}_e \ddot{x}_b. \quad (13b)$$

Equations (3), (5), and (13b) comprise the equations of motion for the system. We nondimensionalize the system by the nonlinear magnetic coefficient, $\tilde{b}_m$, since it is a large order of magnitude term. Redefined terms can be found in the appendix.

$$\ddot{X}_p(\tau) + \mu \dot{X}_p(\tau) + (1 - \bar{a}_m) X_p(\tau) + X_p^3(\tau) = V(\tau) - \tilde{\beta} \ddot{X}_e, \quad (14a)$$

$$\dot{V}(\tau) = -\theta V(\tau) - \tilde{K}^2 \dot{X}_p(\tau), \quad (14b)$$

$$\ddot{X}_e(\tau) + \frac{r_c}{r_m} \mu \dot{X}_e(\tau) + \frac{r_k}{r_m} X_e(\tau) - \frac{1}{r_m} \mu \dot{X}_p(\tau) - \frac{1}{r_m} X_p(\tau) = -\frac{1}{r_m} V(\tau) - \dot{X}_b(\tau). \quad (14c)$$

Equations (14a)–(14c) are the nondimensionalized equations of motion for the beam, electromechanical system, and linear elastic magnifier, respectively. In order for the system to exhibit nonlinear bistability the linear magnetic coefficient, μ , in equation (14a), must be greater in magnitude than one. In equation (14c) the terms r_m , r_c , and r_k are the mass, stiffness, and damping ratios of the elastic magnifier to beam systems, respectively.

The coupled system can be considered at open circuit in order to analyze the effective potential. The system can be put into state-space. The state-space form can be found in the appendix. Accounting for only the displacement terms the effective potential is given as:

$$U = \frac{1}{2} X_p^2 - \frac{1}{2} \bar{a} X_p^2 + \frac{1}{2} \frac{\tilde{\beta}}{r_m} X_p^2 + \frac{1}{4} X_p^2 + \frac{1}{2} \frac{r_k}{r_m} \tilde{\beta} X_e^2. \quad (15)$$

If we assume that $r_m \gg r_k \gg 1$ we recover the bistable potential:

$$U \approx \frac{1}{2} (1 - \bar{a}) X_p^2 + \frac{1}{4} X_p^2. \quad (16)$$

The equilibrium points are found by taking the derivative of the potential equal to zero and solving for the roots. One root is found at $X_p = 0$ and the other two are found at $X_p = \pm \sqrt{\bar{a} - 1}$. These three roots characterize bistability. The height of the potential well is simply the difference between the potential function at the upper equilibrium point and one of the lower equilibrium points:

$$\Delta U = U(0) - U(\pm \sqrt{\bar{a} - 1}) = \frac{1}{4} (\bar{a} - 1)^2. \quad (17)$$

As seen in figure 3 the system exhibits a supercritical pitchfork bifurcation at the bifurcation point.

Nondimensionalized numerical simulation case study

The nature of the elastic magnifier is that it can either produce high orbit oscillations with improved power performance or it can suppress system performance. In order to explore the interaction between the BPEH and the LEM a case study of different ratio parameters was

examined. Similar case studies were conducted by Galbier *et al* [40] and Wang *et al* [27]. Since the purpose of the LEM is to place the system in high energy orbits we will focus on the optimal $r_m = r_k$ case determined from the case studies provided in [27, 40]. In order to justify the performance enhancement of the BPEH + LEM we compare this configuration alongside a PEH + LEM system, a BPEH system, and a PEH system under varying base excitation amplitudes. Unlike Wang *et al* [27] this case study includes the damping of both the elastic magnifier and the beam composites. Similarly, the PEH + LEM system is included as well as the power-time plots and average powers of each system. As a way to highlight the differences between each system and effectively characterize their respective behaviors we use parameters directly from [27] where F_0 is the amplitude of the base excitation. The parameters used, while not explicitly reflective of measured quantities, are simply a tool to exploit the fundamental nature of each system. Designing a comparison case study to capture the performance of scale heart-beat harvesters would require characterizations of the forcing behavior of the heart which exhibits different quasi-periodic motion over variable cardiac cycles. A single forcing magnitude generalizing the heart-beat would be inadequate to effectively draw any conclusions about the differences in the sinusoidal input case study and the heart input case study. In order to more accurately capture system behavior under that condition, however, we will consider each system subject to a realistic heart-beat signal in the next section.

Using the same method outlined in the previous section; equations of motion are derived for the three systems mentioned above. In comparison to the BPEH + LEM system the PEH + LEM is simply the removal of the linear and nonlinear magnetic terms:

$$\ddot{X}_p(\tau) + \mu \ddot{X}_p(\tau) + X_p(\tau) = V(\tau) - \tilde{\beta} \ddot{X}_e, \quad (18a)$$

$$\dot{V}(\tau) = -\theta V(\tau) - \tilde{K}^2 \dot{X}_p(\tau), \quad (18b)$$

$$\ddot{X}_e(\tau) + \frac{r_c}{r_m} \mu \dot{X}_e(\tau) + \frac{r_k}{r_m} X_e(\tau) - \frac{1}{r_m} \mu \dot{X}_p(\tau) - \frac{1}{r_m} X_p(\tau) = -\frac{1}{r_m} V(\tau) - \ddot{X}_b(\tau). \quad (18c)$$

The BPEH is simply the removal of all elastic terms:

$$\ddot{X}_p(\tau) + \mu \dot{X}_p(\tau) + (1 - \bar{a}_m) X_p(\tau) + X_p^3(\tau) = V(\tau) - \tilde{\beta} \ddot{X}_b, \quad (19a)$$

$$\dot{V}(\tau) = -\theta V(\tau) - \tilde{K}^2 \dot{X}_p(\tau) \quad (19b)$$

and the PEH is the removal of all elastic terms and magnetic terms:

$$\ddot{X}_p(\tau) + \mu \ddot{X}_p(\tau) + X_p(\tau) = V(\tau) - \tilde{\beta} \ddot{X}_b, \quad (20a)$$

$$\dot{V}(\tau) = -\theta V(\tau) - \tilde{K}^2 \dot{X}_p(\tau). \quad (20b)$$

Figures 4(a)–7(a) show a comparison of phase-portraits for each of the three systems. For each case, as the base amplitude increases, it is evident that the BPEH + LEM oscillates in a higher energy orbit. The system, in general, experiences larger tip-deflections and is encouraged into limit cycle behavior. Depending on the input amplitude the BPEH either exhibits single-well low energy oscillations; figure 4(a), chaotic large amplitude oscillations; figure 5(a), or high energy-oscillations; figures 6(a)–7(a). The PEH oscillates, with little variation, in low-energy limit-cycle periodic orbits. The PEH + LEM oscillates about the zero-position in a periodic orbit similar to the PEH. The elastic magnifier, however, allows the system to maintain a higher energy limit cycle. The voltage history; figures 4(b)–7(b), and power signal; figures 4(c)–7(c), confirm the performance enhancement of the BPEH + LEM. As a contrast to the results in [27] it is observed that the inclusion of the damping brings the BPEH + LEM closer to the BPEH. The damping is less effective, however, as the force increases and the BPEH + LEM maintains a higher energy orbit than the BPEH. The inclusion of the damping is, nonetheless, important for accurately understanding the nature of each system.

Numerical simulation with heart beat

While harmonic inputs provide an understanding of the response of a system our application is being positioned over the heart. The heart provides constant quasi-periodic pulses to the energy harvesting system. A numeric heart beat signal was constructed from data acquired from a humans beating heart and adapted to the case study used in the previous section. The parameters are consistent with case study used above provided by [40]. We consider the heartbeat vibrations as the base acceleration of the system. Leadless pacemakers are typically implanted on the interventricular septum. The motion of the myocardial heart wall at that location was measured by Kanai *et al* [39]. Figures 8(a) and (b) show the signal in the time and frequency domain, respectively. Most of the high energy beating and, therefore, the optimal harvesting region is within the 50 Hz or less range.

At the bifurcation point $\tilde{a}$ will be greater than or equal to ω_n^2 . This indicates the bistable system is indeed exhibiting bistability. Figure 9 shows the power signal of each system where the input is given as the heart acceleration and where the magnetic gap distance is $\tilde{a} \geq \omega_n^2$. The average power is calculated for each system and summarized in table 1. The BPEH + LEM harvests more than one order of magnitude greater than each system. As the magnetic gap distance is increased the effects of the nonlinearities and bistable phenomena effectively disappear. Figure 10 shows this case. The power signals for each system become

almost indistinguishable with the BPEH + LEM approaching the PEH + LEM and the BPEH approaching the PEH. The average power is calculated for each system for the case of the large magnetic gap distance and summarized in table 2.

Fan-fold

Although the focus of this work is primarily on the analysis and implementation of cantilevered energy harvesters it is important to comment on the fan-fold configuration. The fan-fold or zig-zag structure consists of several linear cantilevered beams connected vertically or horizontally. The vertically stacked fanfold configuration is of particular interest and was thoroughly analyzed by Ansari *et al* [7]. Because of the coupling between each segment this configuration can maintain relatively high harvesting potential with the ability to lower the natural frequency and create a broader bandwidth. The reason its inclusion is important is primarily because leadless pacemakers, inherently, are forcing designs to become increasingly volume sensitive. The volumetric restrictions of these devices are, thus, requiring unconventional designs with high power density and broad bandwidth. A tip-mass fan-fold design is entirely magnet free, thus MRI compatible, and can be tuned to exhibit either linear or nonlinear properties by adjusting the ratios between link masses. This makes it extremely versatile; reaping the benefits of nonlinearities without the drawbacks of magnets. The fan-fold configuration has showed high promise as a practical candidate for the implementation of energy harvesters in leadless pacemakers and is a natural extension of cantilevered beam structures. A brief analysis of the frequency response of the system with and without the elastic magnifier will provide insight into the reasonable functionality of the fan-fold heartbeat harvester.

By tuning the length of each beam and the link mass between each segment, shown in figure 11, the natural frequency of the system can be lowered. The interaction of the beams also create a frequency-robust system allowing harvesting to take place under varying input excitations. This is particularly relevant for the case of a cardiac device which would experience a range of uncontrolled inputs as the heart rate variability and heart beat amplitude change with activity and cardiac cycles.

The governing equations of the fan-fold piezoelectric energy harvester configuration are virtually the same in appearance to a PEH. As a result the fan-fold design coupled with a linear elastic magnifier (FF + LEM) will have the same governing equations and analytic solution as the PEH + LEM. The differentiation between the fan-fold and linear harvester comes from the boundary conditions which, in turn, create a unique set of coefficients for the general solutions of the mode shapes. The result is a distinct and fundamental difference in behavior and response from the PEH. In order to validate the FF + LEM; initial conditions and parameters are pulled from [7]. The first four modes are considered. The response is then the summation of these modes over a 300 Hz range, shown in figure 12.

The system response of the fan-fold structure with the elastic magnifier is similar in nature to the fan-fold structure without the magnifier. The first resonant peak around 30 Hz is the same for each system. Beyond the first resonance the elastic magnifier encourages the fan-fold system into higher amplitude oscillations. What results is a widened bandwidth and

higher voltage values at each subsequent resonance. While the uncoupled fan-fold configuration drops off considerably after the second mode the elastic magnifier permits the coupled system to have higher mode harvesting potential.

Experimentation and application

The experimental setup is shown in figure 13. A hydrogel was manufactured for the LEM. Cross-linker ratios and other chemical properties were designed so that the hydrogel maintained mechanical properties similar to the myocardial heart wall [41]. The proper ratio of chemicals were selected to reflect the size, density, and elasticity of the desired 0.1524 m (6 inch) diameter and 0.0127 m (0.5 inch) thickness hydrogel. The chemicals comprising the hydrogel included urea (29.029 g), buffer (57.916 ml), TEMED (0.116 ml), ammonium persulfate (0.232 g), acrylamide (38.358 ml), bis-acrylamide (7.672 ml), and deionized water (107.401 ml). Once the ratios and quantities were confirmed and measured the chemicals were combined incrementally, allowed to set for two hours under a fume hood, and then placed in a 4 °C refrigeration chamber until testing. The hydrogel is assumed to be a linear damper [42]. A harmonic sinusoidal sweep signal provided by the shaker software is used as the base excitation of the system. The base and BPEH configuration in figure 13(a) rests freely on the hydrogel, figure 13(b), which sits atop the shaker. The whole system is subject to vertical excitations while motion is tracked by an optical laser, base acceleration monitored by an accelerometer, and voltage monitored via the shaker controller software. A 20 k Ω resistor was found to be the optimal load resistance and was used between the PZT leads and the software acquisition.

Table 3 shows the properties of the system used in experimentation. Before experiments of coupled structures were completed a brief analysis of the hydrogel was conducted. In order to verify and understand its mechanical properties compression tests were conducted on sections of the hydrogel. This is primarily to validate the gels elastic properties and to ensure the hydrogel mimics the heart.

Figure 14 is a snap-shop of a video capturing the compression test. A video analysis of the compression test using numerical methods in MATLAB allowed the strain and stress to be determined. In figure 14, the red component was subtracted from the grayscale image in order to extract the red components in the primary image. A median filter was used to filter out noise and the resulting grayscale image was converted into a binary image. All of the connected components of the image were labeled together and an image blob analysis was conducted giving a set of properties for each labeled region. The strain was then found by looking at variations in the distance of the centroid to the red reference point throughout the duration of the test. The strain relationships from the compression test are shown in figure 15. An elastic modulus on the order of 10×10^3 (Pa) was measured from the compression test. This result is within accepted values for the elasticity of the myocardial heart wall [43, 44].

Experiments were conducted for the BPEH + LEM, PEH + LEM, BPEH, and PEH cases. The frequency response of each system is given over a range of 1–100 Hz.

The green line in figure 16, indicating the BPEH + LEM, has a much broader bandwidth than any of the other systems and, similarly, gives the largest voltage amplitude. This is a result of the nonlinearities created by the magnets and the high orbit oscillations from the elastic magnifier. While the PEH + LEM has a large voltage output it has a small bandwidth, in comparison, due to the lack of the nonlinear magnetic restoring forces of the magnets. The asymmetric behavior observed with the PEH + LEM has been reported among several publications featuring elastic magnifiers [22, 24–26]. The reason for the asymmetry is because the elastic magnifier behaves akin to a vibration absorber. In the operational range of the vibration absorber the absorber mass has relatively large motion and thus, within these parameters, acts to enhance the broadband and magnitude of the harvester. In general though, rather than having two resonant peaks which stretch the effective bandwidth region the linear systems only have a single resonant peak. Based on these results it appears as though the BPEH is exhibiting monostable behavior. This is likely resulting from a combination of the magnetic gap distance, strength of the magnets, and insufficient base force to allow the strength of the magnets to overcome the buckling forces of the beam. It is seen, however, that under the same conditions the BPEH + LEM is, in fact, in the bistable regime.

The same configurations were tested again with the heartbeat acceleration replacing the harmonic sweep, shown in figure 17.

The BPEH + LEM system has larger spikes in power throughout the time interval but struggles to maintain high energy orbits as a result of the nonlinearities and input. It still outperforms each subsequent system, though, by a smaller margin. The BPEH + LEM harvested an average power of $49 \mu\text{W}$ while the PEH + LEM, BPEH, and PEH harvested average powers of $41 \mu\text{W}$, $23 \mu\text{W}$, and $19 \mu\text{W}$, respectively. These experimental results confirm the enhanced power output performance and versatility found in the numeric analysis.

In order to justify and further validate the results the frequency response of each system is plotted with its numeric and analytical solutions. Numerical solutions were obtained using an ordinary differential equation solver to solve state space equations of each system. In MATLAB the function handle is ODE45 which utilizes a four point Runge–Kutta method with a variable time step. Comparisons between numeric and analytical are generally identical if the analytical solution is accurate and valid. Differences arise with the experimental solution, however, due to the inability to exactly capture all of the physical phenomena occurring and could be a consequence of inaccuracies in measurements, noise, magnetic coupling, and fabrication errors of the beam structure and hydrogel.

In general, the frequency response of the PEH is closely captured throughout the 100 Hz range with resonance occurring around 48 Hz for each simulation shown in figure 18. In figure 19, the numerical simulation of the BPEH seems to be exhibiting its nonlinearity and bistability around the 43 Hz range. As a result of the physical model, weak coupling of the magnets, and insufficient forcing; the experimental simulation seemed to closely resemble the monostable case. This indicates that the buckling forces of the beam overcame the magnetic repulsive forces of the opposing magnets. The magnetic interaction does, however,

take effect in lowering the natural frequency of the system and slightly widening the bandwidth. In the PEH + LEM case, figure 20, the numeric and analytical solutions match closely with the exception of the systems response below 10 Hz. The peak voltage at resonance is consistent between all three simulations. The small voltage phenomena observed from 50 to 75 Hz was not captured by the numeric and analytical solutions. This is likely a result of an over simplified mathematical model of the hydrogel. Similarly, for the case of the BPEH + LEM, figure 21, the effects of the nonlinearities are observed as the bandwidth is widened to the right. Any errors observed in the PEH + LEM system would be present more-so in the nonlinear system. Since the system is highly dependent on initial conditions its time evolution and frequency response would hinge highly on a deeper understanding of the mechanical properties and behavior of the hydrogel-heart subsystem. Nonlinear matching is difficult numerically as noise and marginal errors are exploited as the system runs. The coupling of the elastic system further accentuates this.

Conclusion

An analysis of several energy harvesting systems coupled with an elastic magnifier were compared to cases without the elastic magnifier. The goal was to model pacemaker energy harvesters over the heart by developing numerical, experimental, and analytical simulations of systems with an elastic magnifier mimicking the hearts mechanical properties. An analysis of the resulting data and figures showed the BPEH + LEM to exhibit high-energy orbits and broader bandwidth. This configuration was shown out to outperform the PEH + LEM, BPEH, and PEH systems under identical conditions. Comparisons of the phase-portraits, power-time, and voltage-time histories of these systems in several numeric case studies confirmed these results. Experimental results also proved to validate the large amplitude and high energy performance. Improvements to power and performance were observed with both a harmonic input and a heart-beat input. A similar analysis and result was observed for a fan-fold configuration with and without the magnifier. Analysis, case studies, and simulations have created a reasonable foundation into understanding the interaction of these harvesters and the heart. In general, the elastic magnifier provides a platform in which to enhance the power and performance of oscillators and encourages high energy orbits and increased bandwidth of nonlinear bistable harvesters.

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Appendix

$$\tilde{m}_p = m_p \int_0^L \phi^2(x) dx, \quad (A1)$$

$$\tilde{c}_p = c_p \int_0^L \phi^2(x) dx = 2\zeta\omega_p, \quad (A2)$$

$$\tilde{k}_p = Y_{eq} I_{eq} \int_0^L \phi^{(4)}(x) \phi(x) dx, \quad (A3)$$

$$\beta = m_p \int_0^L \phi(x) dx, \quad (\text{A4})$$

$$K = \alpha[\phi^{(1)}(L) - \phi^{(1)}(0)], \quad (\text{A5})$$

$$\tilde{a}_m = a_m \phi(L) \int_0^L \phi(x) dx, \quad (\text{A6})$$

$$\tilde{b}_m = b_m \phi^3(L) \int_0^L \phi(x) dx, \quad (\text{A7})$$

$$\tilde{m}_e = m_e \int_0^L \phi(x) dx, \quad (\text{A8})$$

$$\tilde{c}_e = c_e \int_0^L \phi(x) dx, \quad (\text{A9})$$

$$\tilde{k}_e = k_e \int_0^L \phi(x) dx, \quad (\text{A10})$$

$$\tau = \omega_p t, \quad (\text{A11})$$

$$\omega_p = \sqrt{\frac{\tilde{k}_p}{\tilde{m}_p}}, \quad (\text{A12})$$

$$\bar{b}_m = \sqrt{\frac{\tilde{k}_p}{\tilde{b}_m}}, \quad (\text{A13})$$

$$\bar{a}_m = \frac{\tilde{a}_m}{\tilde{k}_p}, \quad (\text{A14})$$

$$\tilde{K}^2 = \frac{K^2}{c_o \tilde{k}_p}, \quad (\text{A15})$$

$$\theta = \frac{1}{c_o R \omega_p}, \quad (\text{A16})$$

$$\tilde{\beta} = \frac{\beta}{\tilde{m}_p}, \quad (\text{A17})$$

$$\mu = \frac{\tilde{c}_p}{\tilde{m}_p \omega_p}, \quad (\text{A18})$$

$$r_m = \frac{\tilde{m}_e}{\tilde{m}_p}, \quad (\text{A19})$$

$$r_c = \frac{\tilde{c}_e}{\tilde{c}_p}, \quad (\text{A20})$$

$$r_k = \frac{\tilde{k}_e}{\tilde{k}_p}, \quad (\text{A21})$$

$$\nu(t) = \frac{\tilde{k}_p \bar{b}_m}{K} V(\tau), \quad (\text{A22})$$

$$\eta(t) = \bar{b}_m X_p(\tau), \quad (\text{A23})$$

$$x_e(t) = \bar{b}_m X_e(\tau), \quad (\text{A24})$$

$$x_b(t) = \bar{b}_m X_b(\tau), \quad (\text{A25})$$

$$\begin{aligned}
 & \begin{bmatrix} \dot{x}_1(t) \\ \dot{x}_2(t) \\ \dot{x}_3(t) \\ \dot{x}_4(t) \\ \dot{x}_5(t) \end{bmatrix} = \\
 & \begin{bmatrix} x_2 \\ \frac{1}{m_e}(-\tilde{k}_e x_1 - \tilde{c}_e x_2 + \tilde{k}_p x_3 + \tilde{c}_p x_4 - K x_5 - \tilde{m}_e x_b) \\ x_4 \\ -\left(\sqrt{\frac{\tilde{k}_p}{\tilde{m}_p}} + \frac{\tilde{k}_p \beta}{\tilde{m}_e} - \tilde{a}_m\right)x_3 - \tilde{b}_m x_3^3 - \left(1 + \frac{\beta}{\tilde{m}_e}\right)\tilde{c}_p x_4 + \left(1 + \frac{\beta}{\tilde{m}_e}\right)K x_5 - \frac{\beta}{\tilde{m}_e}(\tilde{k}_e x_1 + \tilde{c}_e x_2 + \tilde{m}_e \ddot{x}_b) \\ -\frac{K}{c_o}x_4 - \frac{1}{Rc_o}x_5 \end{bmatrix} .
 \end{aligned}
 \tag{A26}$$

Figure 1.
Application of heart beat harvesters.

Figure 2.
BPEH + LEM system configuration.

Figure 3.

Effective potential of the bistable system at open circuit when $r_m \gg r_k \gg 1$.

Figure 4.
(a) Phase portrait (b) voltage-time history and (c) power signal for $F_0 = 0.01$.

Figure 5.

(a) Phase portrait (b) voltage-time history and (c) power signal for $F_0 = 0.04$.

Figure 6. (a) Phase portrait (b) voltage-time history and (c) power signal for $F_0 = 0.08$.

Figure 7.

(a) Phase portrait (b) voltage-time history and (c) power signal for $F_0 = 0.12$.

Figure 8. Heart beat acceleration in the (a) time domain and (b) frequency domain.

Figure 9.

(a) Power signal at bifurcation point with heartbeat input, (b) its log, and (c) a snap-shop of the log figure from 0 to 1.8 s.

Figure 10.

(a) Power signal at large gap distance with heartbeat input, (b) its log, and (c) a snap-shot of the log figure from 0 to 1.8 s.

Figure 11.
Vertically stacked bimorph fan-fold energy harvester [7].

Figure 12.
Frequency response comparison of the fan-fold design and FF + LEM.

(a)

(b)

(c)

Figure 13. Experimental setup of (a) (left) the BPEH (b) (right) the hydrogel (c) (bottom) analysis and data acquisition.

Figure 14.
Compression test of hydrogel.

Figure 15.
Stress–strain results of compression test.

Figure 16.
Frequency response of various systems from 1 to 100 Hz.

Figure 17.

(a) Experimental power signal with heartbeat input, (b) its log, and (c) a snap-shot of the log figure from 0 to 1.8 s.

Figure 18.

Numeric, experimental, and analytical frequency response of the PEH.

Figure 19.
Numeric and experimental frequency response of the BPEH.

Figure 20.
Numeric, experimental, and analytical frequency response of the PEH + LEM.

Figure 21.
Numeric and experimental frequency response of the BPEH + LEM.

Table 1

Average power with heartbeat input at bifurcation point.

System	Average power (μW)
BPEH + LEM	0.663 31
PEH + LEM	0.032 151
BPEH	0.037 315
PEH	0.027 357

Table 2

Average power with heartbeat input at bifurcation point.

System	Average power (μW)
BPEH + LEM	0.032 294
PEH + LEM	0.032 151
BPEH	0.027 206
PEH	0.027 357

Table 3

Physical properties of the experimental setup.

Description	Value
Length of substrate and PZT, (L)	22.99 (mm)
Width of the substrate and PZT (b)	27.06 (mm)
Thickness of substrate (t_s)	101.63×310^{-3} (mm)
Thickness of PZT (t_p)	1273×310^{-3} (mm)
Young's modulus for substrate (Y_s)	1103×310^9 (Pa)
Young's modulus for PZT (Y_p)	623×310^9 (Pa)
Young's modulus for LEM (Y_o)	103×310^3 (Pa)
Density of substrate (ρ_s)	2329 (kg m ⁻³)
Density of PZT (ρ_p)	7700 (kg m ⁻³)
Piezoelectric strain coefficient (d_{31})	-1903×310^{-12} (V m ⁻¹)
Capacitance (c_0)	933×310^{-9} (F)
Tip mass (m_t)	8.4 (g)