Published in final edited form as:

J Intell Mater Syst Struct. 2018 February ; 29(3): 438–445. doi:10.1177/1045389X17708344.

A sub-cc nonlinear piezoelectric energy harvester for powering leadless pacemakers

MH Ansari and M Amin Karami

Department of Mechanical and Aerospace Engineering, University at Buffalo, The State University of New York (SUNY), Buffalo, NY, USA

Abstract

A miniature nonlinear piezoelectric energy harvester is developed to power state of the art leadless cardiac pacemakers from cardiac motions. The energy harvester is integrated in the leadless pacemaker and is connected to the myocardium. The energy harvester converts myocardial motions to electricity to power leadless pacemakers. The energy is stored in a battery or supercapacitor and is used for pacing. The device is composed of a bimorph piezoelectric beam confined in a gray iron frame. The system is assembled at high temperature and operated at the body temperature. The mismatch in the coefficients of thermal expansion of the beam and the frame causes the beam to buckle in body temperature. This intentional buckling makes the beam unstable and improves the power production and robustness of the device. Having high natural frequency is a major problem in microelectromechanical systems energy harvesters. Considering the small size of the energy harvester, 0.5 cm^3 , the natural frequency is expected to be high. In our design, the natural frequency is lowered significantly using a buckled beam and a proof mass. Since the beam is buckled, the design is bistable and nonlinear, which could increase the output power. In this article, the device is analytically modeled, and the natural frequencies and mode shapes of the energy harvester are analytically derived. The terms corresponding to geometric nonlinearities are included in the electromechanical coupled governing equations. The simulations show that the device generates sufficient electricity to power leadless pacemakers.

Keywords

Sub-cc energy harvester; piezoelectricity; thermally buckled; leadless pacemaker; bimorph beam; buckling; heartbeat vibrations

Introduction

Modern cardiac pacemakers require batteries and, therefore, necessitate repeated surgical replacements over the life of the young patients who need heart rate support. Due to the size constraints, usage of batteries is more challenging in leadless pacemakers compared to

Reprints and permissions: sagepub.co.uk/journalsPermissions.nav

Corresponding author: MH Ansari, Department of Mechanical and Aerospace Engineering, University at Buffalo, The State University of New York (SUNY), Buffalo, NY 14260, USA., mansari2@buffalo.edu.

Declaration of Conflicting Interests

The author(s) declared no potential conflicts of interest with respect to the research, authorship, and/or publication of this article.

conventional pacemakers. Unlike the conventional pacemakers, the leadless pacemakers do not need any wire to connect them to the heart and they are mounted to the inner wall of the heart. The state-of-the-art leadless pacemakers are 93% smaller than the conventional pacemakers (Nippoldt and Whiting, 2014a; Williams and Jon, 2014). The miniature size of leadless pacemaker necessitates minuscule batteries. A reliable battery technology has been one of the main reasons that leadless pacemakers were not available in the market up until recently, although the idea of leadless pacemakers had been discussed more than 35 years ago (Bilitch, 1981). Our preliminary results show that heart motion energy harvesters (EHs) generate sufficient energy to power a leadless pacemaker. Also, EHs generate power continuously and never run out of power. The current leadless pacemaker batteries are depleted in less than 12 years (Nippoldt and Whiting, 2014b). This demands placement of another leadless pacemaker in the heart. This technology can, therefore, be a game changer in leadless pacemaking.

High natural frequency is a common problem in microelectromechanical systems (MEMS) EHs (Kim et al., 2012). Different methods have been used to decrease the natural frequencies of MEMS EHs in order to increase the output power of the device. Tang et al. (2013) studied different methods to increase the working bandwidth of EHs. Adding mass to the system as the tip mass or middle mass is one of the methods for decreasing the natural frequency of the EH (Kim et al., 2011; Xue et al., 2008). Another method to decrease the natural frequency of the EH is adding axial force. Shaker (1975) studies the effect of the axial force on the mode shapes and natural frequencies of beams. Bokaian (1988) investigates the effect of axial load on the natural frequencies and mode shapes of uniform beams and of a cantilevered beam with a concentrated tip mass.

A number of groups studied energy harvesting of buckled piezoelectric beams. Buckling phenomenon has been used in different contexts for energy harvesting. An axial load is most often used to affect the dynamics of the piezoelectric beams and have them buckled (Cottone et al., 2012a, 2012b; Emam, 2002; Masana and Daqaq, 2011a; Porter and Berfield, 2014; Zhu and Zu, 2013). Xu and Kim (2015) utilize residual stress in micro-fabricated thin films to buckle the beam and decrease the natural frequency. Several groups have looked into thermal buckling of the beams as well (Fu et al., 2014; Li and Batra, 2007; Locke and Mei, 1990; Park et al., 2004).

Many research groups have studied energy harvesting for pacemakers. Goto et al. (1999) investigate the feasibility of using quartz watches to convert the heart motion into electricity. Karami and Inman (2012) introduce linear and nonlinear piezoelectric devices to recharge the batteries of the pacemakers continuously. The device converts the vibrations from the heartbeats to the electrical energy needed for powering the battery. Zurbuchen et al. (2013) utilize a mass imbalance to convert cardiac wall motion into electricity. Deterre (2013) presents various methods of harvesting energy to power leadless pacemakers using blood pressure changes within the heart throughout the cardiac cycle. Hwang et al. (2014) study a single crystalline piezoelectric EH to power an artificial pacemaker. Dagdeviren et al. (2014) investigate different materials and devices that convert heart and lung motion into electricity. Ansari and Karami (2016b) develop a one-cc fan-folded three-dimensional (3D) EH for

leadless pacemakers. The device consists of several piezoelectric beams stacked on top of each other.

In this article, we propose a sub-cc nonlinear piezoelectric EH for leadless pacemakers. The term sub-cc corresponds to the volume size of the EH, 0.5 cm^3 , which is less than 1 cm^3 . We use simulations to show that the proposed device generates enough energy for powering leadless pacemakers from the heartbeat vibrations.

Governing electromechanical equations

Our proposed device is a bimorph beam which is clamped at both ends. The beam is inside a gray iron frame. In order to decrease the natural frequency of the EH, the main beam is thermally buckled due to the difference in the coefficients of the thermal expansion between the beam and the rigid frame. A proof mass is added in the middle of the beam to decrease the natural frequency even more. The EH is 0.5 cm^3 which perfectly matches the miniature form factor of leadless pacemakers. Brass is chosen as the substrate due to its high density and high modulus of elasticity.

As illustrated in Figure 1, the piezoelectric bimorph beam is a clamped–clamped buckled beam. The coupled governing equation of a buckled piezoelectric beam is written as follows (Ansari and Karami, 2015)

$$\begin{aligned}
 & m \frac{\partial^2 w_{rel}}{\partial t^2} + c \frac{\partial w_{rel}}{\partial t} + EI \frac{\partial^4 w_{rel}}{\partial x^4} \\
 & + \left[P_c + K_{eq} \Delta - \frac{K_{eq}}{2} \int_0^L \left(\frac{\partial w_{rel}}{\partial x} \right)^2 dx \right] \frac{\partial^2 w_{rel}}{\partial x^2} \\
 & = -2\alpha \left[\left(\frac{d\delta(x)}{dx} - \frac{d\delta\left(x - \frac{L}{4}\right)}{dx} \right) + \left(\frac{d\delta\left(x - \frac{L}{2}\right)}{dx} - \frac{d\delta\left(x - \frac{L}{4}\right)}{dx} \right) \right] \\
 & v(t) - [m + M_{mid} \delta(x - x_{mid})] \frac{d^2 w_b}{dt^2}
 \end{aligned} \tag{1}$$

where m is the total mass per unit length of the beam, $w_{rel}(x, t)$ is the deflection along the z -axis (Figure 1), c is the damping term, EI is the equivalent bending stiffness of the beam, P_c is the first critical buckling force, Δ is the axial displacement due to the thermal compression (Figure 1), K_{eq} is the equivalent axial stiffness of the beam, α is the piezoelectric coupling coefficient, and V is the voltage across the piezoelectric elements which are connected in parallel. $\delta(x)$ is the Dirac delta function. In the above equation, M_{mid} represents the middle mass and w_b is the base displacement. Figure 2 shows the beam inside the frame for two positions. Position (1) is before the buckling and position (2) is after the beam is buckled. We assume that each end of the beam has an axial shortening of $L/2$. Using modal analysis,

the mass normalized continuous differential equations can be discretized as follows (Ansari and Karami, 2016a)

$$\begin{cases} \ddot{T} + c\dot{T} + (K + p)T + NT^3 + \chi V(t) = \gamma \ddot{w}_b \\ C_p \dot{V} + \frac{V}{R_l} = -\chi \dot{T} \end{cases} \quad (2)$$

where T is the temporal term of the deflection, c is the damping ratio, K stands for the linear stiffness, the reduction in the stiffness coefficient due to the axial force is p , N is the nonlinear coefficient, the coupling coefficient is χ , R_l is the shunt resistor, and C_p is the internal capacitance of the piezoelectric layer. The coefficients of the governing equations are calculated as follows

$$\begin{cases} K = EI \int_0^L \phi^{(4)}(x) \phi(x) dx \\ p = (P_c + K_{eq} \Delta) \int_0^L \phi''(x) \phi(x) dx \\ N = -\frac{K_{eq}}{2} \int_0^L \phi'^2(x) dx \int_0^L \phi''(x) \phi(x) dx \\ \chi = 2\alpha \left[\left(\phi' \left(\frac{L}{4} \right) - \phi'(0) \right) + \left(\phi' \left(\frac{L}{4} \right) - \phi'(L) \right) \right] \\ \gamma = M_{mid} \phi \left(\frac{L}{2} \right) + m \int_0^l \phi dx \end{cases} \quad (3)$$

where $\phi(x)$ is the spatial term of the deflection of the beam ($w(x, t)$). The new natural frequency of the beam after being buckled is calculated as follows

$$\omega = \sqrt{(K + p) + 3NT_{st}^2} \quad (4)$$

where T_{st} is the coordinate of the temporal term for the equilibrium point (Ansari and Karami, 2015). Figure 3 shows the change in the natural frequency of a buckled bimorph beam with the change in the amount of the axial shortening of the beam (Δ). All the natural frequencies shown in Figure 3 are for the cases after the beam is buckled. The lower the delta is, the lower is the natural frequency. Theoretically, we can have zero natural frequency when the delta is zero and the beam is about to buckle ($K+p=0$) (Emam, 2002, 2013; Emam and Nayfeh, 2009; Masana and Daqaq, 2011a). Before the buckling starts, by increasing the

axial force, the natural frequency decreases to the point where the beam starts to buckle (the case with $\Delta = 0$). After that by increasing the shortening of the beam (Δ), the natural frequency rises again which is shown in Figure 3. As Masana et al. showed in their paper, we cannot reach to the point where $\Delta = 0$. They showed that before the buckling, the natural frequency is decreased to a certain point as the axial force increases. After that point, the beam buckles and the natural frequency starts to increase again. For the substrate of the beam, we use 0.01 cm thick brass sheets. Lead zirconate titanate (PZT) sheets are used as the piezoelectric element. The thickness of the piezoelectric layers (t_p) is 0.01 cm. The bimorph beam is 2-cm long and 0.25-cm wide. A 5 gr platinum mass is used in the middle of the beam as the proof mass (Figure 1). We will call this specimen *The specimen* in the article. It will be explained later why we chose these sizes for *The specimen*.

Results

The dimensions of the piezoelectric EH were optimized through numerical simulations. Different materials with different thicknesses were tested for the various amounts of the axial shortening of the beam. Table 1 shows the dimension of the optimized EH (*The specimen*). t_s is the thickness of the substrate, t_p is the thickness of the piezoelectric patches, L is the length of the beam, W is the width of the beam, and Δ is the axial shortening in the beam. L_f is the length of the frame, W_f is the width of the frame, and H_f is the height of the frame (Figure 1). The frame needs to be large enough to provide sufficient axial force for the buckling of the beam. In order to prevent charge cancellations, the piezoelectric layer is electrically divided into four segments. A middle mass is used to reduce the natural frequency and increase the generated power. Both ends of the beam are clamped inside the outer frame. For a given beam and a specific compression (Δ) in the beam, we find the first buckled natural frequency and mode shape. Then, we solve the governing equations using the heartbeat vibrations as the base acceleration of the EH and calculate the power.

The power is calculated by $P_{ave} = v_{rms}^2 / R$. Where R is the load resistor parallel to the EH.

The piezoelectric patches are connected in parallel. Having them in parallel keeps the voltage constant and adds their charges. The parallel configuration gives us the maximum power in this test. The generated power can be used to power leadless pacemakers or other biomedical devices inside the body. We use the heartbeat acceleration (Figure 4) in the time domain measured by Kanai et al. (1996) to simulate the vibration to the leadless pacemaker caused by the heartbeats.

To study the motion behavior of the EH, we look at the phase portrait and the Poincaré map of two cases with different axial shortening (Δ). In one case, the axial shortening of the beam equals 0.15 μm (the same as *The specimen*), and in the second case, $\Delta = 10 \mu\text{m}$ (larger delta) which generates two orders of magnitude less energy than *The specimen* when excited with the heartbeat vibrations. For each case, the mode shapes are found analytically, and the integrals in equation (3) are calculated. The calculated coefficients are inserted in the coupled governing equations of the system (equating (2)) to find the deformation and the velocity of the beam. Figure 5(a) and (b) illustrates the phase portrait of these two cases when excited by heartbeat vibrations. The x -axis of the plot (w) shows the deformation in

the middle of the beam, and the y -axis shows the velocity of that point along the z -axis. The deformation range of *The specimen* ($\approx 0.15\mu\text{m}$) includes negative and positive positions, whereas the deformation range in the case with larger ($10\mu\text{m}$) is just around one equilibrium. In *The specimen*, the beam vibrates between the two buckled stable positions and goes from negative to positive deformation. Also, the velocity range of *The specimen* is about four times of velocity range in the second case. Figure 5(a) and (b) also shows the Poincaré map of these two cases when excited by 30 cycles of heartbeats. After each cycle, one sample is taken. Since the motion is periodic, all the 30 points are very close and all of them are seen as one point in the two plots.

Figure 6(a) and (b) shows the power output from the heartbeat vibrations versus the temperature difference and shortening of the beam. The power output is calculated for the simulated heartbeat signal (Figure 4) as the base excitation. The temperature difference is used to buckle the beam thermally. Gray cast iron is used as the material for the frame. The variation of the shortening of the beam as a function of the temperature difference is illustrated in Figure 6(c). Figure 7 shows a proposed configuration that can accurately provide a specific amount of for the beam. The active bimorph beam is clamped inside a frame. All the components of the EH including the frame and the bimorph beam will be warmed up to a higher temperature. The bimorph beam will be fixed on the structure using bolts while the components are at the high temperature. The two ends provide the clamped boundary condition on the bimorph beam. After the device cools down, there will be a mismatch between the shortening of the beam and the shortening of the frame. When the frame has a higher coefficient of thermal expansion (CTE) than the CTE of the bimorph beam, the beam will be buckled. For a given , the amount of the temperature difference is calculated as follows

$$T_{diff} = \frac{\Delta}{L(\alpha_f - \alpha_{eq})} \quad (5)$$

where L is the length of the beam, α_f is the CTE of the frame, and α_{eq} is the equivalent CTE of the bimorph beam. The frame of *The specimen* is made from gray iron with $\alpha_f = 10.8(\mu\text{m}/\text{m.K})$. The equivalent CTE of the bimorph beam is as follows

$$\alpha_{eq} = \frac{(2\alpha_p E_p A_p + \alpha_s E_s A_s)}{(2E_p A_p + E_s A_s)} \quad (6)$$

where α_p , α_s are the CTE of the piezoelectric layer and the substrate, E_p , E_s are the module of elasticity of the piezoelectric and the substrate, and A_p , A_s are the cross sections of the piezoelectric layer and the substrate. Substituting these values for the piezoelectric patches ($\alpha_p = 4(\mu\text{m}/\text{m.K})$, $E_p = 61$ GPa) and the brass substrate ($\alpha_s = 18.7(\mu\text{m}/\text{m.K})$, $E_s = 100$ GPa), the equivalent CTE of *The specimen* is $10.6(\mu\text{m}/\text{m.K})$.

In Figure 6(a), there are some discontinuities in the power output at specific points. The difference in the power output is in the order of $0.1 \mu\text{W}$ and will not affect the final design. By plotting the deflection of the middle of the beam over the time for two of these points (one before a discontinuity ($T=25.22^\circ\text{C}$), one after that discontinuity ($T=26.63^\circ\text{C}$)), we find out that the reason is the equilibrium points. Figure 8 illustrates the deflection of these two cases in the time domain for two cycles. The first case with the lower T (the dashed line) ends up in the positive equilibrium after the first cycle, whereas the second case with the higher T (the solid line) goes to the negative equilibrium after each cycle. Thus, after beginning of a new cycle, the second case deforms more than the first case which causes the difference in the power output.

For *The specimen*, δ is assumed to be $0.15\mu\text{m}$ to maximize the power output. Using equation (4), we find the first natural frequency as 31.92 Hz . This frequency is much lower than the natural frequency of the unbuckled beam (210.68 Hz). As discussed in equation (3), the natural frequency of the beam is a function of the amount of the axial shortening in the beam. By lowering the natural frequency, we cannot conclude that the output energy increases necessarily. The maximum power happens at the point where the natural frequency of the device matches the high amplitudes frequencies of the heart vibration. As shown in Figure 4, due to the impact induced by the heart valves, the acceleration waveform of the heart does contain two significant impulses. Therefore, the heartbeat waveform consists of a very broad range of frequencies, which means that the high-amplitude frequencies might not be the same as the frequency of the heart rate (around 1 Hz for a normal heart rate). Figure 9 is the Fourier transform of heartbeat signal (Figure 4). As shown in Figure 9, most of the high-amplitude frequencies are less than 50 Hz . Thus, by lowering the natural frequency to the desired range (frequencies with high peaks in Figure 9), the amount of the average power output of the EH increases.

We consider two safety factors for our device. One is the axial safety factor ($SF_F = \sigma_{F, \max} / \sigma_y$), which corresponds to the axial stress in the beam, where $\sigma_{\max}$ is the maximum axial stress due to the axial compression force and σ_y is the yield stress. The other one is the safety factor ($SF_M = \sigma_{M, \max} / \sigma_y$) due to bending deformation of the beam. In which, $\sigma_{M, \max}$ is the maximum axial stress due to the bending. Each of these safety factors is calculated for the substrate and the piezoelectric layer to avoid any damages to the beam. In our designs, a minimum threshold of 3.0 is considered for SF_F and SF_M . Figure 10 illustrates the instantaneous output power of the EH in the time domain. The average power of the EH is calculated as $14.09 \mu\text{W}$ which is sufficient to power a leadless pacemaker. A leadless pacemaker needs about $10 \mu\text{W}$ to operate. The shunt resistor is $96 \text{ k}\Omega$ which is calculated as follows

$$R = \frac{1}{C_p \omega} \quad (7)$$

where C_p is the capacitor of the two piezoelectric layers and ω is the natural frequency of the EH. For our EH, the internal capacitance for the two piezoelectric layers is 7.83 nF .

To find the maximum power, we perform a resistor sweep test. Figure 11 illustrates the effect of the shunt resistor on the output power. The maximum power is $15.61 \mu\text{W}$, and as predicted, the maximum power occurs in the vicinity of the resistor calculated from equation (7). Figure 12 shows the effect of the heart rate on the output power. Since the device is nonlinear, it is robust to the heart rate changes. For a wide range (40–195 bpm), the generated power is sufficient for powering a pacemaker ($10 \mu\text{W}$).

Conclusion

This article presents a sub-cc thermally buckled piezoelectric device to generate energy from heartbeat vibrations. The size of the device is 0.5 cm^3 . The device consists of a bimorph piezoelectric beam clamped inside a rigid frame. Due to the small size, the natural frequency of the device is high. By buckling the beam, the natural frequency is lowered to the desired range. When the device is assembled at a higher temperature and cooled down to the body temperature, the bimorph beam buckles due to the difference in the thermal coefficients of the frame and the bimorph beam. The device generates electricity when excited by the heartbeat vibrations. The generated energy can be used to power a leadless pacemaker. In this article, the device is analytically modeled, the coupled governing equations of the system are solved, and the output power is calculated for different cases. The effect of the shortening of the beam over the output power is studied. It is shown that by using the nonlinear thermally buckled piezoelectric EH, one can achieve a miniature power source pertinent to the small size of leadless pacemakers.

Acknowledgments

Funding

The author(s) received no financial support for the research, authorship, and/or publication of this article.

References

- Ansari MH, Karami MA. Energy harvesting from controlled buckling of piezoelectric beams. *Smart Materials and Structures*. 2015; 24:115005.
- Ansari, M., Karami, MA. Nonlinear thermally buckled piezoelectric energy harvester. ASME 2016 international design engineering technical conferences and computers and information in engineering conference; Charlotte, NC. 21–24 August; 2016a.
- Ansari MH, Karami MA. Modeling and experimental verification of a fan-folded vibration energy harvester for leadless pacemakers. *Journal of Applied Physics*. 2016b; 119:094506.
- Bilitch M. Leadless cardiac pacer. Google Patents. 1981
- Bokaian A. Natural frequencies of beams under compressive axial loads. *Journal of Sound and Vibration*. 1988; 126:49–65.
- Cottone, F., Basset, P., Vocca, H., et al. Electromagnetic buckled beam oscillator for enhanced vibration energy harvesting. Proceedings of the 2012 IEEE international conference on green computing and communications (Green-Com); Besancon. 20–23 November; New York: IEEE; 2012a. p. 624–627.
- Cottone F, Gammaitoni L, Vocca H, et al. Piezoelectric buckled beams for random vibration energy harvesting. *Smart Materials and Structures*. 2012b; 21:035021.
- Dagdeviren C, Yang BD, Su Y, et al. Conformal piezoelectric energy harvesting and storage from motions of the heart, lung, and diaphragm. *Proceedings of the National Academy of Sciences*. 2014; 111:1927–1932.

- Deterre, M. PhD Thesis. University of Paris-Sud; Orsay: 2013. Toward an energy harvester for leadless pacemakers.
- Emam, SA. PhD Dissertation. Virginia Polytechnic Institute and State University; Blacksburg, VA: 2002. A theoretical and experimental study of nonlinear dynamics of buckled beams.
- Emam SA. Approximate analytical solutions for the nonlinear free vibrations of composite beams in buckling. *Composite Structures*. 2013; 100:186–194.
- Emam SA, Nayfeh AH. Postbuckling and free vibrations of composite beams. *Composite Structures*. 2009; 88:636–642.
- Fu Y, Wang J, Hu S. Analytical solutions of thermal buckling and postbuckling of symmetric laminated composite beams with various boundary conditions. *Acta Mechanica*. 2014; 225:13–29.
- Goto H, Sugiura T, Harada Y, et al. Feasibility of using the automatic generating system for quartz watches as a leadless pacemaker power source. *Medical & Biological Engineering & Computing*. 1999; 37:377–380. [PubMed: 10505390]
- Hwang GT, Park H, Lee JH, et al. Self-powered cardiac pacemaker enabled by flexible single crystalline PMN-PT piezoelectric energy harvester. *Advanced Materials*. 2014; 26:4880–4887. [PubMed: 24740465]
- Kanai H, Sato M, Koiwa Y, et al. Transcutaneous measurement and spectrum analysis of heart wall vibrations. *IEEE Transactions on Ultrasonics, Ferroelectrics and Frequency Control*. 1996; 43:791–810.
- Karami MA, Inman DJ. Powering pacemakers from heartbeat vibrations using linear and nonlinear energy harvesters. *Applied Physics Letters*. 2012; 100:042901.
- Kim I-H, Jung H-J, Lee BM, et al. Broadband energy harvesting using a two degree-of-freedom vibrating body. *Applied Physics Letters*. 2011; 98:214102.
- Kim S-G, Priya S, Kanno I. Piezoelectric MEMS for energy harvesting. *MRS Bulletin*. 2012; 37:1039–1050.
- Li S, Batra R. Thermal buckling and postbuckling of Euler-Bernoulli beams supported on nonlinear elastic foundations. *AIAA Journal*. 2007; 45:712–720.
- Locke J, Mei C. Finite element, large-deflection random response of thermally buckled beams. *AIAA Journal*. 1990; 28:2125–2131.
- Masana R, Daqaq MF. Electromechanical modeling and nonlinear analysis of axially loaded energy harvesters. *Journal of Vibration and Acoustics*. 2011a; 133:011007.
- Nippoldt D, Whiting J. Micra transcatheter pacing system: device volume characterization comparison (Medtronic data on file). 2014a
- Nippoldt D, Whiting J. Micra transcatheter pacing system: FIH announcement (Medtronic data on file). 2014b
- Park J-S, Kim J-H, Moon S-H. Vibration of thermally post-buckled composite plates embedded with shape memory alloy fibers. *Composite Structures*. 2004; 63:179–188.
- Porter DA, Berfield TA. A bi-stable buckled energy harvesting device actuated via torque arms. *Smart Materials and Structures*. 2014; 23:075003.
- Shaker, FJ. Technical report TN-8109. Cleveland, OH: NASA Lewis Research Center; 1975. Effect of axial load on mode shapes and frequencies of beams.
- Tang, L., Yang, Y., Soh, CK. Broadband vibration energy harvesting techniques. In: Elvin, N., Erturk, A., editors. *Advances in Energy Harvesting Methods*. New York: Springer; 2013. p. 17–61.
- Williams E, Jon W. Micra transcatheter pacing system: size comparison (Medtronic data on file). 2014
- Xu R, Kim S. Low-frequency, low-G MEMS piezoelectric energy harvester. *Journal of Physics: Conference Series*. 2015; 660:012013.
- Xue H, Hu Y, Wang Q-M. Broadband piezoelectric energy harvesting devices using multiple bimorphs with different operating frequencies. *IEEE Transactions on Ultrasonics, Ferroelectrics and Frequency Control*. 2008; 55:2104–2108.
- Zhu Y, Zu JW. Enhanced buckled-beam piezoelectric energy harvesting using midpoint magnetic force. *Applied Physics Letters*. 2013; 103:041905.

Zurbuchen A, Pfenniger A, Stahel A, et al. Energy harvesting from the beating heart by a mass imbalance oscillation generator. *Annals of Biomedical Engineering*. 2013; 41:131–141. [PubMed: 22805983]

Author Manuscript

Author Manuscript

Author Manuscript

Author Manuscript

Figure 1.
Schematic representation of the thermally buckled beam clamped at both ends in a rigid frame.

Figure 2. Axial shortening of the beam (). Position (1) shows the un buckled beam and position (2) shows the buckled beam.

Figure 3.
The effect of shortening of the beam on its natural frequency.

Figure 4.
The heartbeat waveform in the time domain.

Figure 5. Phase portrait and the Poincaré map for 30 cycles of excitation when (a) $\Delta = 0.15 \mu\text{m}$ and (b) $\Delta = 10 \mu\text{m}$.

Figure 6.

(a) The effect of the temperature difference on the power output from the heartbeat vibrations. (b) The effect of the shortening of the beam on the power output from the heartbeat vibrations. (c) The variation of the shortening of the beam (Δ) with respect to the temperature difference.

Figure 7.
Proposed design for the EH device.

Figure 8. Midpoint deflection of the active beam in the time domain for two different cases. Solid blue line for $T=26.63^{\circ}\text{C}$ and dashed red line for $T=25.22^{\circ}\text{C}$.

Figure 9.
Fourier transform of the heartbeat.

Figure 10.
Instantaneous power of *The specimen* from heartbeat vibrations.

Figure 11.
Effect of the shunt resistor on the output power of *The specimen*.

Figure 12.
The output power of *The specimen* for different heart rates.

Table 1

The optimized dimensions of *The specimen*.

H_f (mm)	W_f (mm)	L_f (mm)	t_f (mm)	t_f^s (mm)	t_f^y (mm)
5	5	25	0.15	2.5	20
0.1					0.1