MICRO BUCKLED BEAM BASED ULTRA-LOW FREQUENCY VIBRATION ENERGY HARVESTER

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ABSTRACT

We present a bi-stable buckled beam MEMS energy harvester which achieved low operating frequency, low operating amplitude and wide bandwidth, all packaged in the size of a coin. Bi-stable buckling was induced by controlling the built-in compression from the residual stress in the multi-layer beam structure, a way that is compatible to the micro-fabrication process. A fully functional piezoelectric device has been designed, monolithically fabricated, and tested to induce bi-stable buckling of $\sim 200\mu\text{m}$. The first batch device generated peak power of 85nW with 50% half-power bandwidth under 70Hz at 0.5g. Considering the figure of merit (power over device swing volume and input acceleration²), this device shows at least one order of magnitude improvement to piezoelectric MEMS harvesters ever reported.

INTRODUCTION

Vibration energy harvesting at MEMS scale will promisingly enable exciting applications such as large wireless sensor networks and Internet of Things (IoT) due to its high power-generation efficiency and small form factor. Nonetheless, the high operating frequencies of current MEMS energy harvesters have long hindered wide applications of this technology. For instance, our previous MEMS device based on nonlinear resonance has achieved high output power of 45 μW and wide bandwidth of $>20\%$, while the operation frequency and input vibration amplitude were impractically high ($>1000\text{Hz}$ at 4g) [1]. We now report that our new design addresses this major challenge to realize the MEMS-scale energy harvesting technology to a real product: energy harvesting at low frequencies ($<100\text{Hz}$) and low amplitudes ($<1\text{g}$).

Past approaches to lower the operating frequency include designing new geometries [2-7], using soft materials [7-9], and up-conversion mechanisms [10-13]. For examples, buckling has been employed as an up-conversion mechanism [10]; a rotary mass with magnetic catch-and-release has been designed [11]; and a multi-mode buckled beam has been implemented [12]. Soft materials may not be effective to absorb the input energy; most of the devices for low frequency operation are in macro scale, which inherently have low-frequency resonance; the up-conversion approaches typically need auxiliary structures such as magnets which inevitably increase the device size. The concept of bi-stable nonlinear oscillator-based energy harvesting has been previously proposed with either magneto-elastic structures [14] or axial compression-based configurations [15], which would require costly assembly and were never implemented at the MEMS-scale.

Our new design adopts the bi-stable oscillation but has been implemented at MEMS scale, demonstrating the low-frequency energy harvesting. Bi-stable oscillation, with its large-amplitude snapping motion at low frequencies was found to be suitable for low-frequency energy harvesting. To enable the bi-stable oscillation, intentional but controllable buckling was generated through mono-lithic fabrication processes. Through the extensive in-process characterization and model-based analytical composite beam design, the compression stress in the micro-beam structures could be controlled to induce the desirable buckling of the structure. The testing results validates the feasibility of this approach as described below.

DESIGN AND MODELLING

To capture the features of the newly designed energy harvester, including the low operating frequency, MEMS-fabricated multi-layer structure, and the embedded piezoelectric layer that converts the mechanical energy into electricity, we developed a theoretical framework with an electromechanically coupled lumped model, which incorporates the multi-layer structure and residual stress. The model is solved analytically with harmonic balance to obtain the frequency response of the energy harvesters, which is of our primary interest.

The energy harvester we model has a clamped-clamped beam structure of a stack of thin films including structural layer, piezoelectric layer and passivation layer (Fig.1). A heavy proof mass is concentrated at the middle of the beam to capture the external vibration and excites the whole beam to oscillate out-of-plane. Piezoelectric elements work in d_{33} mode with top interdigitated electrodes, coupling the electrical response with mechanical strain. The multi-layer beam is designed to buckle by incorporating compressive residual stress in the micro-fabricated thin films. Statically, the beam either buckles up or down (two equilibria), and the dynamics becomes complex when the system is continuously excited in post-buckling regime. To simplify the analysis of the complex problem but still capture the essence of the snapping, the beam's vibration mode is assumed, and a one degree-of-freedom model has been constructed. The non-homogeneous cross section beam structure has been taken into account by considering different thicknesses and material properties of the layers. Furthermore, residual stress of each layer is built in as part of the stiffness of the beam and induces buckling. The electrical and mechanical domains are coupled with the piezoelectric coupling, so that the generated electrical signal can be obtained.

Figure 1: Schematics of the doubly clamped beam based energy harvester. The piezoelectric element is in 33 mode with top interdigitated electrodes. The piezoelectric element and electrode span symmetrically from $-b \cdot L$ to $-a \cdot L$ and $a \cdot L$ to $b \cdot L$ on the substrate.

The governing equation of the mechanical domain can be obtained and written in a compact form [16],

$$m\ddot{w} + k_L w + k_N w^3 + b\dot{w} + c_N w V_N + c_L V_L = F \quad (1)$$

$$C_0(\dot{V}_L + \dot{V}_N) + \frac{V_L + V_N}{R} = I_L + I_N \quad (2)$$

where k_L , k_N , b , c_L , c_N and F are the linear stiffness and nonlinear stiffness of the beam, the mechanical damping coefficient, the linear and nonlinear electromechanical coupling, and the external excitation force respectively, C_0 and R are the internal capacitance of the piezoelectric element and the load resistance respectively. These lumped parameters are functions of the device dimensions and material properties, which are useful for device design:

$$k_L = \left[\frac{2\pi^4 W}{3l^3} \sum_{i=1}^n c_{33,i}^E (H_{U,i}^3 - H_{L,i}^3) \right] + \left[\frac{\pi^2 W}{2l} \sum_{i=1}^n T_{0,i} H_i \right] \quad (3)$$

$$k_N = \frac{\pi^4 W}{8l^3} \sum_{i=1}^n c_{33,i}^E H_i \quad (4)$$

$$c_L = \frac{\pi e_{33} W (H_{U,P}^2 - H_{L,P}^2) (\sin(2\pi b) - \sin(2\pi a))}{lg} \quad (5)$$

$$c_N = \frac{\pi^2 e_{33} W H_p (b-a)}{lg} \quad (6)$$

where W , H , l are the width, thickness and length of the beam, T_0 is the residual stress, c and e are the elastic constant and piezoelectric constant respectively, subscript i denotes the i -th layer, a and b denote the span of the electrodes on the beam (Fig.1), and g is the gap between two electrode fingers; the subscript p denotes the variable is associated to the piezoelectric layer. It should be noted that the linear stiffness has two parts: the first part is from bending of the beam and the second comes from the residual stress. More particularly, when the residual stress is negative (compressive) and large enough, the linear stiffness k_L will be negative, so that equation (1) becomes a characteristic bi-stable *Duffing* equation.

Figure 2. Simulated dynamic responses of the MEMS energy harvester with varying linear stiffness (stress) and other parameters the same. The deflection's amplitude at low frequencies increases with the linear stiffness k_L varying from positive to negative.

The nonlinear governing equations (1) and (2) are solved analytically using the harmonic balance method, so that the frequency responses are obtained. Both softening (intra-well) and stiffening (inter-well) responses are derived, but the inter-well oscillations have larger amplitude and generate more power. The enhancement of the bi-stable oscillator on the low-frequency

operation is illustrated in Fig. 2. The frequency response of the MEMS energy harvester changes with varying the linear stiffness but fixing the nonlinear stiffness, which is the case when varying only the residual stress in the beam structure. The frequency response shifts to lower frequency with higher amplitude when the linear stiffness switching from positive to negative as shown in Fig. 2. When the negative stiffness's amplitude becomes larger, the response is in ultra-low frequency range with significantly larger deflection amplitude, which characterizes the large-amplitude snap through of bi-stable nonlinear oscillators. The shift of the frequency response to desirable direction, lower frequency and larger amplitude, by tuning the stiffness provides the critical design knob for the new generation energy harvester.

To implement the residual stress induced buckling in the multi-layer beam, the linear and nonlinear stiffness are designed by matching the simulation of the model to the target operating condition. Both the residual stress and thickness as the critical parameters in the linear stiffness term (the second term in equation (3)), can then be designed to achieve the desired linear stiffness by parametric sweeping using the analytical form of the linear stiffness. However, the compressive residual stress is bi-axial and could induce the buckling in lateral direction as well. A small-width beam array (like a louver) is then devised to minimize the critical buckling load in the lateral direction to eliminate the buckling in that direction.

The newly designed bi-stable nonlinear oscillator-based MEMS energy harvester has a clamped-clamped beam structure with a stack of thin-films having 28 pairs of beams 0.4mm wide in a silicon frame of 15mm×12mm. Each beam has approximately 500 interdigitated Au fingers over 0.2 μm thick PZT. A proof mass is located at the middle, connecting the beam array to synchronize their out-of-plane motion and minimize the rotation during oscillation. The schematic of the new design is illustrated in Fig. 3.

Figure 3. (a) 3D rendition of the buckled beam based energy harvester (The buckling of the beam is exaggerated). (b) Beam composition of the MEMS energy harvester.

FABRICATION

The fabrication process is based on 4" <100> silicon wafers. 300nm thermal dioxide is grown with wet oxidation, as the bottom layer of the multi-layer beam and serves as the etch stop of the final DRIE release. One LPCVD silicon nitride layer (tensile) and one PECVD silicon dioxide layer (compressive) are then deposited as the structural layer to support the proof mass. The dual-frequency plasma deposition of the ST Systems CVD enables flexible stress control in a wide range from tensile to compressive by tuning parameter is the ratio of the duration of the applied high (13.56MHz) and low (380kHz) frequency plasma. The growth rate and residual stress of the PECVD thin films have been characterized by measuring the thickness, and the wafer bow before and after the deposition. The data guides the control of the injected compression in the structure and the balance of the stress in the stack. ZrO₂, PbTi are the diffusion barrier and the seed layer which were sol-gel spin coated. The PZT solution has a composition of Pb/(Zr+Ti) of 118/100 along with a Zr/Ti ratio of 52/48, is spun on the substrate at 500 rpm for 5s and 3000 rpm for 30 s. The precursor gel film is pyrolyzed at 390 °C for 5 min on a hot plate. This deposition process cycle is repeated twice to make a PZT layer of 150nm thickness. The PZT film is then annealed at 700 °C for 1 min. PECVD silicon nitride and dioxide as the passivation layers are deposited on top to balance the stress in the stack. Reactive ion etching (RIE) is used to etch through the whole silicon dioxide and nitride stack from the top to define the beam structure and leave openings to the contact pads, and from the backside to pattern the frame and proof mass, leaving opening for the deep reactive ion etching (DRIE). Finally, etching through the whole wafer thickness from the back (DRIE) releases the device. An extra step of XeF₂ etching is used to gently remove the residual silicon. A photo of the released device is displayed in Fig. 4.

Figure 4. Photo of the released device.

EXPERIMENTAL RESULTS

The released beam show buckling that are visible to naked eyes. To accurately characterize the buckling and verify the bi-stability of the buckled structure, optical profiling of the surface was performed. Wyko NT9800 optical profilometer was used to scan the surface profile, and the stitching assembled multiple scans to cover the whole beams' surface profile. The surface profile of the whole device is shown in Fig. 5(a). The 28 beams on the same device show good consistency in buckling, and the measured surface profile matches well with the design (Figure 3). The device's surface was scanned first with top surface facing up, and then flipped so the beams buckle to the other direction and the bottom surface of the beams are scanned. Since the weight of the proof mass applies 1g loading to the beam structure, which is higher than the threshold of the snap, we could observe the bi-

stable buckling (Fig.5(b)). The buckling in both directions shows similar mid-point deflection within 5% from designed value of 200μm, which demonstrates no significant asymmetrical stress distribution that leads to only one direction buckling. The surface profile and the large buckling in longitudinal direction also proves the narrow beam width effectively eliminates the buckling in the transverse direction and preserves the buckling in the desired direction.

Figure 5. Surface profile scan: (a) Three-dimensional representation of the whole surface of the released device. (b) Surface profile of four beams showing the buckling on both sides (bi-stable) at around 200 μm.

The dynamic testing of the device further validated the design concepts that the buckled beam device could have large-amplitude oscillations with low-frequency and low-amplitude inputs (<100Hz and 0.5g). The frequency response was obtained by measuring the deformation of the beam with controlled input vibrations. The device was mounted on an electromagnetic shaker, whose vibration was monitored by an accelerometer (Analog Device ADXL335), and the input harmonic signal's frequency and amplitude are controlled by the signal generator (Prema ARB1000) and the power amplifier (Labworks PA138). A laser vibrometer (Polytec Scanning Vibrometer PSV300) was employed to measure the velocity of different spots on the device to calculate the relative movement of the beam to the frame.

The input harmonic vibrations were set at fixed frequencies below 100Hz and at different input amplitude level below 0.5g. The device was tested with an initial static state, and the input vibration's amplitude increase to reach the target value at a constant frequency. In this way, we could make the device start with lowest energy state and observe if the input energy was enough to trigger the snap. The relative displacement of the mass

to the frame was calculated by subtract the displacement amplitude of the mass by the displacement amplitude of the frame. Since the vibrometer cannot scan the two spots simultaneously but only sequentially, the phase difference cannot be obtained, especially with the abrupt bi-stable snapping that may not be perfectly the same in each cycle. The estimation is an underestimation as a result, since the phase difference between the two is not always 180° . But the trend shows large amplitude snap in a wide frequency range below 100Hz. With the proof mass of the wafer-thick central silicon proof mass ($13.9\text{mm} \times 3\text{mm} \times 525\mu\text{m}$), the frequency response is shown in Fig. 6.

The harvester with an added tungsten mass was also connected to a load resistor ($1\text{M}\Omega$) to measure the power consumed as an indication of the generated power. The voltage across the resistor is measured at a sample rate of 5.2 kHz. The power consumed by the resistor is calculated as $P = V^2/R$. During a period of 3.2s, the peak voltage is identified for each fixed frequency and amplitude and is used to calculate the peak power. At 0.5g, the peak power spectrum shows 50% of the jump-down frequency half-power bandwidth from 30Hz to 70Hz (Fig. 11).

Figure 6. (a) Dynamic testing: displacement of the mass relative to the frame at ultra-low frequencies (silicone mass). (b) The peak power on a $1\text{M}\Omega$ load resistor (with added 0.22g tungsten mass).

CONCLUSION

Bi-stable nonlinear oscillator-based energy harvester is realized and demonstrated at MEMS scale for the first time. The MEMS buckled beam oscillator utilizes the large-displacement snapping motion in a wide bandwidth at low frequencies to generate power. Residual stress is intentionally introduced and controlled along the MEMS monolithic fabrication processes of 10 thin film layers. Compression induced buckling of the release device was about $200\mu\text{m}$, which is within 5% of the designed amount. Static testing demonstrated the bi-stability of the buckled beam. Dynamic testing of the first-batch fully functional energy harvester with PZT thin film demonstrates the state-of-the-art operating conditions of MEMS energy harvesters of 50% bandwidth below 70Hz at 0.5g input with a peak power of 85nW.

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REFERENCES

- [1] A. Hajati and S.-G. Kim, "Ultra-wide bandwidth piezoelectric energy harvesting," *Appl. Phys. Lett.*, vol. 99, no. 8, p. 83105, 2011.
- [2] Kim, In-Ho, et al. "A performance-enhanced energy harvester for low frequency vibration utilizing a corrugated cantilevered beam." *Smart Materials and Structures* 23.3 (2014): 037002.
- [3] Xu, Chundong, et al. "Cantilever driving low frequency piezoelectric energy harvester using single crystal material 0.71 Pb (Mg1/3Nb2/3) O3-0.29 PbTiO3." *Applied Physics Letters* 101.3 (2012): 033502.
- [4] Massaro, Alessandro, et al. "Freestanding piezoelectric rings for high efficiency energy harvesting at low frequency." *Applied physics letters* 98.5 (2011): 053502.
- [5] Deng, Jiadong, et al. "Topology optimization and fabrication of low frequency vibration energy harvesting microdevices." *Smart Materials and Structures* 24.2 (2014): 025005.
- [6] Liu, Huicong, et al. "A new S-shaped MEMS PZT cantilever for energy harvesting from low frequency vibrations below 30 Hz." *Microsystem technologies* 18.4 (2012): 497-506.
- [7] Liu, Wen, et al. "Low frequency wide bandwidth MEMS energy harvester based on spiral-shaped PVDF cantilever." *Science China Technological Sciences* 57.6 (2014): 1068-1072.
- [8] Edamoto, M., et al. "Low-resonant-frequency micro electret generator for energy harvesting application." *Micro Electro Mechanical Systems, 2009. MEMS 2009. IEEE 22nd International Conference on. IEEE, 2009.*
- [9] Xie, Mengying, Kean C. Aw, and Wei Gao. "A multi-layered polydimethylsiloxane structure for application in low-excitation, broadband and low frequency energy harvesting." *Sensors and Actuators A: Physical* 222 (2015): 140-148.
- [10] Jung, Seok-Min, and Kwang-Seok Yun. "Energy-harvesting device with mechanical frequency-up conversion mechanism for increased power efficiency and wideband operation." *Applied Physics Letters* 96.11 (2010): 111906.
- [11] Pillatsch, Pit, Eric M. Yeatman, and Andrew S. Holmes. "A piezoelectric frequency up-converting energy harvester with rotating proof mass for human body applications." *Sensors and Actuators A: Physical* 206 (2014): 178-185.
- [12] Burgueno R, Lajnef N 2013 *United States patent application* US 13/960,356
- [13] Halim, Miah A., and Jae Y. Park. "Theoretical modeling and analysis of mechanical impact driven and frequency up-converted piezoelectric energy harvester for low-frequency and wide-bandwidth operation." *Sensors and Actuators A: Physical* 208 (2014): 56-65.
- [14] A. Erturk, J. Ho mann, and D. J. Inman, "A Piezomagnetoelastic Structure for Broadband Vibration Energy Harvesting," *Applied Physics Letters*, 94(25):254102, 2009.
- [15] F. Cottone, et al., "Piezoelectric Buckled Beams for Random Vibration Energy Harvesting," *Smart Materials and Structures*, 21(3):035021, 2012.
- [16] R. Xu, and S.-G. Kim, "Modeling and Experimental Validation of Bi-Stable Beam Based Piezoelectric Energy Harvester." *Energy Harvesting and Systems* 3.4 (2016): 313-321.

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