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A bi-stable buckled energy harvesting device actuated via torque arms

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Abstract
A bi-stable switching energy harvester made from a buckled steel structure mounted with uniaxially poled piezoelectric polyvinylidene fluoride and 3D printed polylactic acid components is constructed and tested. A data collection system and frequency sweeping program is built to drive the device using a custom compression rig fitted with an accelerometer. The energy harvester is tested with the center beam compressed to different degrees of buckling, as well as in its unloaded state. Root mean square (RMS) accelerations are applied to the device in the range of 0.1–0.9 g\text{rms} by 0.2 g steps. The device is driven with a frequency between 16 and 40 Hz (by 0.5 Hz) in both forwards and backwards sweeps. Finite element modeling program ANSYS is used to model the device and determine undamped pre-stressed modal frequencies, proof mass displacements to ‘snap-through’, and static buckled profiles for the center beam. As a comparison, a doubly constrained beam (DCB) with the same width and length is constructed and tested in the same manner as the torque arm device. RMS power density for the torque actuated device compressed by 0.13 mm and frequency swept in reverse was 0.246 $\mu$W cm$^{-2}$ (3.13 $\mu$W) at 16.5 Hz and 1.5 g\text{rms} using two 0.19 g proof masses. The DCB RMS power density swept in reverse was 1.287 $\mu$W cm$^{-2}$ (6.18 $\mu$W) at 59.5 Hz and 1.5 g\text{rms} with a 1.38 g proof mass.

Keywords: bi-stable, energy harvesting, vibration, snap-through, torque arms, PVDF, VEH

1. Introduction

Vibration-based energy harvesters offer a means for converting waste mechanical energy or ambient structural oscillations into electrical power. Though the average power generated by MEMS-scale energy harvesters is not large, ~0.026 $\mu$W (0.5 g), 2.15 $\mu$W (1 g), to 60 $\mu$W (2 g) in the MEMS region [1–6] it is still sufficient enough to sample and transmit sensor data wirelessly. Macro scale devices can generate powers of 8.4 nW cm$^{-2}$ (0.12% strain per bend), 118 $\mu$W (0.2 g), and 17.3 $\mu$W (3 g) [7–9]. For hard-to-reach electrically powered devices, such as remote sensors, energy harvesters can greatly extend the time required between burdensome service visits, offering some unique benefits over other power supplies such as batteries. However, to be considered a robust, viable alternative power source, vibration-based energy scavenging devices must perform well at low frequency ranges and at driving conditions with variable spectral energy density content characteristic of real-life operating environments.

The most simplistic vibration-based energy harvesting designs utilize a mono-stable cantilever with a single minimal potential energy well, meaning the resting position has only one preferred state. These devices have narrow operating bandwidths and require proper design/tuning for their specific functional settings [10–12]. Conversely, energy harvesting systems that have nonlinear dynamic responses generally have broadband excitation characteristics, ideal for chaotic input impulse situations [13]. One method for producing nonlinear behavior is to force a system into a multi energy well arrangement. Multi energy well systems can be created through selective placement of magnets [14] or electrostatic components, mechanical orientation [15], bio inspired bi-stable structures [16], or through a mechanically bi-stable buckled structure [17–20].

Due to their interesting dynamic response, energy harvester designs incorporating buckled structures have been the focus of many recent research efforts [13, 17–22]. A more general spectrum of current bi-stable energy harvester research is covered in a recent comprehensive review by Harne et al [13]. Mono-stable energy harvesters are favorable
at low accelerations over most of the frequency range even when the frequency is swept in both directions. Intriguingly, buckled energy harvesting devices with low energy wells have larger voltage responses at super and sub-harmonic frequencies when compared to the same device in the unbuckled (low stress) state [23], if the inter-well actuation can be achieved. Another interesting behavior of buckled energy harvesters is their improved voltage output during chaotic vibrational inputs [20] and wider frequency bandwidths [19, 24]. These performance trends are enhanced during switching between stable buckled states, which involves large structure deformations during these high energy 'snap-through' events, as described by [25].

Notable buckled energy harvesters include work by Majer et al [18] that experimented with a compressed, doubly-clamped beam with a proof mass in the middle. Their device had a wide frequency operating range, high voltage operation during bi-stable switching, and decaying output oscillation frequencies at multiples of the drive frequency right after switching. The driven electrical load had an impedance of 1 MΩ and exhibited bi-stable switching for accelerations between 4–5 g [18]. Another study by Cottone et al [21], featured an energy harvester with a proof mass positioned on a buckled beam. This design enabled a 'frequency-up' response so that the electrostatic energy harvesting core could operate at a more efficient frequency regime (162 Hz). The bi-stable oscillator device responded well to randomly generated noise in the 20–40 Hz operating frequency, with inter-well jumps (snap-through) producing high voltage responses [21]. Lui et al [17], similarly found that a bi-stable energy harvester based on a dynamic mass-spring system with flexible hinges demonstrated superior performance compared to a linear oscillator when subjected to chirp and band-limited noise accelerations [17].

The bi-stable buckled energy harvester described herein is simple, robust, and uniquely designed to operate over a range of very low driving frequencies by using torque arms (TAs) to facilitate buckled state switching. In short, an energy harvesting device was fabricated from a 100 μm thick stainless steel sheet (Precision Brand Carbon AISI/1008). Dimensions of the cut-out stainless steel section are in figure 2(a). An additional layer of polylactic acid (PLA) plastic was fabricated using a 3D printer to add stiffness to the TAs and overlapping center portion of the buckled beam, figure 2(b). Poly(lactic) acid, or PLA, is a thermoplastic polyester which is used in commercial and hobbyist 3D printing. The modulus of elasticity is around 1.7–3.6 GPa for directly extruded materials but can increase to 4.2 GPa with subsequent heat treatments [26, 27]. Information on the production and molecular structure of this polyester can be found in [28]. An industrial adhesive (JB-Weld Compound 8265-S) was used to bind the PLA layer to the bottom side of the stainless steel shim stock, figure 2(c). Simple proof masses consisting of 2–56 stainless steel hex nuts (0.19 g each) were then attached at the ends of the TAs.

Four individual strips of a commercially purchased piezoelectric polymer of polyvinylidene fluoride (PVDF) were bound to the topside of the stainless steel using the same adhesive, as shown in figures 2(c) and (d). PVDF is a ferroelectric material that, if processed correctly, can exhibit appreciable piezoelectric coefficients. While formed from a melt, PVDF usually will crystallize into the non-polar α phase which is undesirable for piezoelectric applications. The β phase of PVDF has the highest polarization potential of all the other phases and it is the most desirable in energy harvesting application. To promote the β phase from the other phases, the PVDF is drawn to about 300% of its original length while held at an elevated temperature. Gel deposition methods can also be used for MEMS scale fabrication but require polar solvents mixed in with dissolved PVDF solutions or specially catered processing parameters such as temperature, pressure, spin rates, etc [29]. Molecular structures of the α and β phases are shown in figure 1 below. More information on the molecular chains of PVDF can be found in [30–34].

The aluminum metalized piezoelectric PVDF (GoodFellow FV301960/3, \(d_{33} = 19 \text{pC N}^{-1}\), \(d_{33} = -20 \text{pC N}^{-1}\)) strips were uni-axially oriented with the beam length and had a nominal thickness of 110 μm. Two PVDF strips were attached to the front and back sections of the center beam while the other two PVDF strips were each bonded to a separate TA. Silver epoxy (Circuit Works CW2400) was used to connect enameled 30 gauge copper wire to each of the PVDF strip electrodes. The final assembled structure is shown with the electrode labeling scheme in figure 2(d).

Critical to the operation of the bi-stable energy harvester is the constraint base, which provides adjustable levels of center beam compression and clamps the side arms pinning the center beam into the ‘S’ buckled mode shape. To provide feedback for the dynamic driving routines, a 3-axis accelerometer (MPU6050) was also attached near the centroid of the constraint base, figure 3(a). By adjusting the threaded rods within the constraint base, the compression of the center buckled beam could be controlled and locked. For this work three compressive loading cases were tested; uncompressed, 0.13 mm of compression, and 0.25 mm of compression. Under the latter two levels of displacement constraint, the center buckled beam exhibited two dominant bi-stable buckled states. A visual of the device in the buckled up and down position is shown in figures 3(a) and (b), respectively. This shows the bi-stability of the proposed device. Marks and symbols on the device are for digital image correlation.

2. Materials and fabrication

The base layer of the TA actuated buckled beam energy harvesting device was fabricated from a 100 μm thick stainless steel sheet (Precision Brand Carbon AISI/1008). Dimensions of the cut-out stainless steel section are in figure 2(a). An additional layer of polylactic acid (PLA) plastic was fabricated using a 3D printer to add stiffness to the TAs and overlapping center portion of the buckled beam, figure 2(b). Poly(lactic) acid, or PLA, is a thermoplastic polyester which is used in commercial and hobbyist 3D printing. The modulus of elasticity is around 1.7–3.6 GPa for directly extruded materials but can increase to 4.2 GPa with subsequent heat treatments [26, 27]. Information on the production and molecular structure of this polyester can be found in [28]. An industrial adhesive (JB-Weld Compound 8265-S) was used to bind the PLA layer to the bottom side of the stainless steel shim stock, figure 2(c). Simple proof masses consisting of 2–56 stainless steel hex nuts (0.19 g each) were then attached at the ends of the TAs.
To provide a quantitative comparison for the TA device performance, a single doubly constrained beam (DCB) device was also constructed and tested on the same shaker table setup. This simple device, consisted of only a single beam clamped at both ends and a proof mass placed at the midpoint of the beam. The beam length and width were kept the same as that used for the torque actuated model center beam (96 mm and 5 mm, respectively). Two PVDF strips were placed on the left and right side as to be consistent with the electrode placement of the previous device. A 10–32 nut weighing 1.38 g was adhered to the middle using thin Kapton tape.
The effective stiffness of the composite beam was obtained using a specimen probe mounted on a 5 lbf load cell (Sensotec Model 11) driven by a linear actuator stage. The known stainless steel dimensions and parameters, table 1, were used to determine the material stiffness properties for the cured adhesive layer and the metalized PVDF, which were lumped to determine the material stiffness properties for the cured adhesive layer and the metalized PVDF. This information provides useful insights into electrode positioning to minimize charge neutralization.

3. Quasi-static behavior

Two experimental tests using quasi-static loading conditions were performed on the buckled energy harvester device. The first type of experiments were simple load-deflection experiments to determine the effective stiffness response of the multi-layer structure, the results of which served as material parameters for later finite element modeling. The second set of quasi-static experimental tests probed the bi-stability switching conditions for the buckled device. Both tests were performed using a specimen probe mounted on a 5 lbf load cell (Sensotec Model 11) driven by a linear actuator stage.

Prior to rigging in the constraint base, the stacked layers of metalized PVDF/adhesive/stainless steel used for the center beam were clamped as a cantilever beam. The known stainless steel dimensions and parameters, table 1, were used to determine the material stiffness properties for the cured adhesive layer and the metalized PVDF, which were lumped and treated as a single layer (210 μm in thickness) for simplicity. The effective stiffness of the composite beam was determined through load deflection data and the classic beam end deflection (1),

\[ \delta_{up} = \frac{FL^3}{3EI_{eff}}, \]

where \( F \) is the force applied at the tip, \( L \) is the length of the cantilever beam, \( E_{eff} \) is the effective Young’s Modulus of the composite beam, and \( I_{eff} \) is the effective moment of inertia for the composite beam cross-section. From the experimental data the stiffness of the composite layer was estimated, the results of which are shown in table 1. A similar experiment was then performed on a PLA cantilever beam to determine the Young’s Modulus of 3D printed PLA (JustPLA) with similar print characteristics.

The 2nd type of quasi-static experiment was performed on the entire energy harvesting device in its buckled configuration mounted within the constraint base. Tests were performed with the center beam under two different compression levels, 0.13 mm and 0.25 mm of axial displacement. At 0.13 mm of compression, the center beam just began to exhibit bi-stable buckling behavior which meant that the energy well for this configuration was the smallest that could be obtained. For 0.25 mm of compression, the center beam was also bi-stable, but required a greater applied force to switch between stable buckled states, figure 4. The force and displacement required to induce switching between buckling stability states in each direction was determined by either pushing or pulling on the two proof masses simultaneously using a custom fixture connected to the same 5 lbf load cell and linear actuator stage as the previous experiments. The results of the quasi-static bi-stability switching tests are shown in table 2 below.

A finite element model of the buckled energy harvesting system was meshed using SHELL281 and MASS21 elements. Using the ANSYS simulation software, the buckled conditions were applied to discern the resting displacement profile, quasi-static stability switching displacements, and the pre-stressed undamped modal frequency of the structure. As shown in figure 5(a), the post-buckling displacement profile of the structure was determined for both stable states using an analysis which included the effects of gravitational forces. The displacement of the proof masses needed to induce switching between stable buckled states, or ‘snap-through’, was found from this FEA model. A plot of the typical load-displacement behavior to the snap-through load is shown in figure 5(b).

In addition to the buckled switching behavior, the static post-buckled profile of the beam found via the FEA model was compared with a 3D experimental scan of the structure. The experimental buckled profile was measured by taking approximately 40 high resolution images of the structure from various angles and building a reconstruction of the specimen surface using software called 123D Catch. A comparison of the maximum and minimum out-of-plane displacements found for the buckled devices in their different configurations via both ANSYS and the experimental scan are included in table 3. A typical profile of the center buckled beam derived from ANSYS is shown in figure 5(c).

The ANSYS model results could also be used to determine the static axial strain state spatially throughout the center buckled beam. The results for the 0.25 mm compressed case, figure 5(d), show the location of strain sign change for the top (PVDF) surface. This information provides useful insights into electrode positioning to minimize charge neutralization.

4. Dynamic behavior

The energy harvesting potential of the device was tested using a custom designed shaker system capable of both variable acceleration and frequency control. The shaker system consisted of an anchor platform mounted on a 42 W speaker driven using an audio amplifier (Kinter MA-150) capable of delivering 30 W of peak power. An Arduino Nano (V3.0) was used to read the accelerometer and send its value to a LabVIEW program. The LabVIEW program compared root mean square (RMS) acceleration values to a setpoint value and then adjusted input parameters for the proportional-integral-derivative (PID) controller so that the sine wave audio signal minimized the RMS acceleration input error. To perform constant acceleration sweeps, the LabVIEW program would adjust the frequency of the speaker while tuning the RMS value of the acceleration until an acceptable tolerance
threshold was met (<2.5%). When this tolerance was met for a certain amount of time (\(\sim 5\) s), data was logged at that frequency and acceleration. Then the frequency was increased to the next increment and the stabilization process was repeated. During testing, the entire constraint base containing the buckled energy harvester device was mounted on the anchor platform and driven at various accelerations and frequencies.

All grounds (bottom portion) of the PVDF strips were connected together while the top portions of the four individual strips went to independent circuit inputs for logging. All four channels of the devices PVDF strips where fed into a unity gain amplifier (LM224) while driving a load of 3.3 \(\Omega\). The output of this amplifier was then put into another op-amp (LM224) which converted a \(\pm 10\) V signal to a 0–5 V which is utilized by the Arduino’s 10 bit analog inputs. Calibration of the conversion circuit was done with an oscilloscope to verify that \(-10, 0,\) and \(10\) V input produced 0, 2.5, and 5 V output, respectively. The LabVIEW program along with the Arduino recorded the acceleration in the vertical direction and all four voltage channels at a frequency of 2000 Hz.

Frequency sweeps at constant acceleration were performed on the uncompressed, 0.13 mm compressed, and 0.25 mm compressed beam configurations over the range of 16–40 Hz using 0.5 Hz increments, figures 6 and 7. From the experimental frequency sweeps, the natural frequency of the device was found and compared with the undamped natural frequencies found from the ANSYS models, table 4. Acceleration was controlled via a PID tuning method that modified the amplitude of the output audio signal. The sweeps were

<table>
<thead>
<tr>
<th>Compression (mm)</th>
<th>FEA displacement (mm)</th>
<th>Experimental displacement (mm)</th>
<th>% Difference</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.13, push down</td>
<td>5.36</td>
<td>6.35</td>
<td>-18.50</td>
</tr>
<tr>
<td>0.25, push down</td>
<td>7.83</td>
<td>9.33</td>
<td>-19.12</td>
</tr>
<tr>
<td>0.13, push up</td>
<td>6.35</td>
<td>5.70</td>
<td>10.19</td>
</tr>
<tr>
<td>0.25, push up</td>
<td>8.71</td>
<td>10.28</td>
<td>-17.99</td>
</tr>
</tbody>
</table>
performed at constant acceleration levels from 0.1 to 0.9 g$_{\text{rms}}$, increasing by 0.2 g increments. Though testing up to 1.5 g$_{\text{rms}}$ were performed on the 0.25 mm compression case to evaluate the acceleration needed for bi-stable switching. The tolerance to determine acceptable acceleration to initiate data logging was 2.5% of the set point acceleration. Logged data included the acceleration of the rigid base structure with respect to time, the RMS value for the acceleration for $\sim1$ s, the

![Figure 6. Peak-peak voltages for ‘forward’ frequency sweeps of the ‘front’ and ‘back’ center beam PVDF sections under different constant RMS acceleration levels.](image)

![Figure 7. Peak-peak voltages for ‘backward’ frequency sweeps of the ‘front’ and ‘back’ center beam PVDF sections at different constant RMS acceleration levels.](image)

<table>
<thead>
<tr>
<th>Compression (mm)</th>
<th>ANSYS simulation results</th>
<th>Experimental scan measurements</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Back channel (mm)</td>
<td>Front channel (mm)</td>
</tr>
<tr>
<td>0</td>
<td>0.04</td>
<td>−0.10</td>
</tr>
<tr>
<td>0.13, buckled up</td>
<td>−1.13</td>
<td>1.09</td>
</tr>
<tr>
<td>0.25, buckled up</td>
<td>−1.56</td>
<td>1.53</td>
</tr>
<tr>
<td>0.13, buckled down</td>
<td>1.09</td>
<td>−1.12</td>
</tr>
<tr>
<td>0.25, buckled down</td>
<td>1.53</td>
<td>−1.55</td>
</tr>
</tbody>
</table>

Table 3. Maximum and minimum displacement points for FEA and experimental results.
peak-peak value for the acceleration, an averaged peak-peak value for acceleration (2048 data points split into 10 sub-arrays), and similar data for each voltage channel of the PVDF strips. After a forward (increasing) frequency run, a similar sweep of the setup was run in reverse (frequency descending from 40 to 16 Hz). All sweeps were run with the TAs starting in the buckled down position, figures 4(b) and 3(c). When run in a compressed state, the device was actuated manually to ensure that it remained bi-stable both before and after each frequency sweep.

\( V_{\text{rms}} \), for all channels during 0.5\( g_{\text{rms}} \) and 0.9\( g_{\text{rms}} \) (before and after inter-well actuation threshold at 0.13 mm compression level) are shown in figure 8 below. Snap-through was observed at about 0.7\( g_{\text{rms}} \) for the 0.13 mm compressed center beam state and about 1.3\( g_{\text{rms}} \) for the 0.25 mm case. An observable benefit from the device at higher compression levels is the obvious reduction in operating frequency, which can be seen from figure 8. Once actuation between the two energy wells is reached, the operating frequency band is widened and the device can maintain high voltage operation as long as there is enough energy in the system to continue snap-through actuation as the frequency is swept. The higher compression value causes the device to kick out of inter-well actuation because the stored energy during actuation is not enough to cause snap-through to perpetuate as the frequency meanders out of the resonance range. To widen the frequency band at this compression level, higher accelerations would have to be used. For all loading cases, the output from the TA PVDF strips (right and left channels) was comparatively low due to the higher structure stiffness provided by the thick PLA brace.

Time variant graphs of the acceleration and voltage for all four PVDF channels are shown in figure 9 below. The frequencies for each sub-graph are picked to correspond to the highest RMS voltage in the back channel during a backwards sweep and a set point acceleration of 0.5 and 0.9\( g_{\text{rms}} \). From figure 9 it is seen that the front channel voltage (center beam part opposite the TA placement) is mostly 180° out of phase with the other channels. Given a little bit of compression, figures 9(c) and (d), the device exhibits snap-through if the base acceleration is high enough. This causes higher frequency voltage content to appear in all of the channels post-interwell actuation. It also causes very high \( V_{p-p} \) as shown in figures 9(c) and (d). The accelerometer mounted in constraint base detects reaction forces due to the bi-stability switching, which can also be seen in figures 9(c) and (d).

It was also observed in preliminary experiments involving a single impulse acceleration applied over a baseline low-level acceleration that snap-through actuation could be induced and maintained, even when the baseline acceleration level was below the typical threshold to induce bi-stable switching. This is a similar effect to that seen in [14] and [35] where a perturbation or impulse could force the system into high energy orbits and allow it to output more power given the same vibrational power input. This behavior is desirable for chaotic vibration loading characteristics found for many real-life energy harvesting operating conditions.

The doubly constrained device, shown in figure 10(a), was driven from 20 to 80 Hz by 0.5 Hz increments using accelerations similar as those used for the previous tests. Afterward testing the uncompressed state, the device was buckled until it was just bi-stable at 0.73 mm of axial compression. Again the device was driven with a sinusoidal signal in the range of 20–80 Hz forwards and then backwards in frequency. Total RMS power results from the compressed forward and reverse sweeps shown in figures 10(b) and (c). The setup for the DCB was heavily biased in the buckled down position due to the weight of the proof mass. In the acceleration ranges tested, bi-stable snap-through did not occur continuously.

A comparison of the power output results from the dynamic experiments is shown in table 5. The full-width half-max (FWHM) measurements are used to quantify the range of usability in the frequency spectrum. Peak power is calculated by taking the absolute maximum voltage generated during a frequency sweep at a given RMS acceleration and converting it to power using the 3.3 MΩ load. Area used to construct the device was the factor governing power density. For the DCB, it was simply the length multiplied by the width which came out to be 4.8 cm². The TA device required more space, 12.75 cm², for the elaborate geometry needed to transfer torque and create a pseudo pin. If the entire rectangular area was used to calculate power density, then a value of 58.56 cm² should be used in the calculation of power density. FWHM measurements for the buckled TA device could not be fully measured in some cases due to the operation of the device falling below the 16 Hz limit of the shaker system. Below 16 Hz the sinusoidal wave from the accelerometer begins to indicate distortions from the speaker. Thus the

Table 4. Undamped FEA and experimental natural frequencies.

<table>
<thead>
<tr>
<th>Compression (mm)</th>
<th>Mode 1, ANSYS, no damping (Hz)</th>
<th>Experimental natural frequency (Hz)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>0.1 g</td>
<td>0.3 g</td>
</tr>
<tr>
<td>0</td>
<td>25.79</td>
<td>22.50</td>
</tr>
<tr>
<td>0.13, buckled up</td>
<td>24.48</td>
<td>19.50</td>
</tr>
<tr>
<td>0.25, buckled up</td>
<td>27.74</td>
<td>25.49</td>
</tr>
<tr>
<td>0.25, buckled down</td>
<td>27.89</td>
<td>18.50</td>
</tr>
</tbody>
</table>

* Bi-stable actuation, value reported is average of broadband frequency.
FWHM of these devices are almost certainly greater than indicated in table 5.

5. Discussion

The difference in mass displacement required to switch between bi-stable states, shown in table 2, can be attributed to imperfections in the constructed device. The device fabrication process involved cutting the steel using shearing forces, which could potentially leave residual stresses near these edges. The FEA models do not account for these effects, though the overall influence should be minimized due to the constraint on the thin steel layer provided by the PLA brace and the compression rig.

Another insight provided by the FEA model was the strain profile for the topside PVDF layers. Ideally, to minimize potential charge neutralization, the PVDF strips should be placed further in on the center beam starting at about 1/6 of the center beams constrained length and terminating at the center point. Our setup had the strips epoxied the full half length of the constrained beam; thus there will be charge

Figure 8. \( V_{rms} \) device output values for backward and forward sweeps of all channels at driving accelerations of 0.5 and 0.9\( g_{rms} \).
neutralization due to opposing strain. Future iterations will have four channels for the center beam and two for the outer TAs. From figure 5(d), it is evident some charge neutralization will be occurring given our current electrode placement. Correction of this issue should only improve the device power output.

It should also be noted that the device in its unbuckled state generated large voltages at higher frequencies when compared to the buckled cases. Even uncompressed the device shows 'hard' nonlinear behavior and, eventually, 'soft' nonlinear behavior after the beam is compressed [36]. During the onset of a bi-stable compression (~0.13 mm) the device is actuated between states at low g levels and exhibits a wide frequency response range compared to the other configurations at the same g level. Lower compression levels allow the device to traverse between energy wells quite easily. Only a relatively small amount of energy is needed to 'push' the device over the energy well even when the operation frequency is meandering past the natural frequency. At the 0.25 mm compressed state, the g level needed to onset a bi-stable actuation is much higher and the broadening effect is realized at higher g ranges such as 1.3–1.5g_{rms} (not shown). This higher compression level gives impressive V_{p-p} but near the voltage limits of the constructed DAQ. Even more interesting is amount of energy needed to continue bi-stable oscillation in the 0.25 mm compressed state. If the device is perturbed in such a way during bi-stable actuation as to diminish energy in the system (post-resonance), then the system will kick out of bi-stable oscillation and enter the mono-stable oscillation state. Conversely, if the system is near resonance in the mono-stable state and is perturbed (say, by a static spike applied in the audio line), the device will

Figure 9. Time variant results at frequencies corresponding to the maximum back channel RMS voltage during backwards sweeps on the uncompressed beam at (a) 0.5g_{rms} and (b) 0.9g_{rms}, the 0.13 mm compressed beam at (c) 0.5g_{rms} and (d) 0.9g_{rms}, and the 0.25 mm compressed beam at (e) 0.5g_{rms} and (f) 0.9g_{rms}. 

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enter the bi-stable mode and often remains that way until the system goes out of the resonance region or is perturbed negatively as to diminish the systems overall energy. The higher compression level, 0.25 mm, allows the observer to see this behavior much more easily than the 0.13 mm state during large device oscillations.

Figure 10. Doubly constrained beam in the unbuckled state (a), and RMS power results from (b) a forward sweep, and (c) a backwards sweep.

Table 5. Power comparison of the TA device and DBC.

<table>
<thead>
<tr>
<th></th>
<th>RMS acceleration (g)</th>
<th>Resonance frequency RMS (Hz)</th>
<th>Peak RMS power (μW)</th>
<th>FWHM RMS power (Hz)</th>
<th>Resonance frequency V_{p-p} (Hz)</th>
<th>Peak power (μW)</th>
<th>FWHM peak power (Hz)</th>
<th>RMS power density, DevArea (μW cm^2)</th>
</tr>
</thead>
<tbody>
<tr>
<td>DCB, 0 mm Comp, Fw</td>
<td>0.7</td>
<td>61.5</td>
<td>0.45</td>
<td>4.76</td>
<td>61.5</td>
<td>1.29</td>
<td>4.16</td>
<td>0.094</td>
</tr>
<tr>
<td></td>
<td>1.1</td>
<td>63.5</td>
<td>0.63</td>
<td>6.62</td>
<td>63.5</td>
<td>1.94</td>
<td>5.55</td>
<td>0.132</td>
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<tr>
<td></td>
<td>1.5</td>
<td>68.0</td>
<td>1.71</td>
<td>5.02</td>
<td>68.0</td>
<td>6.42</td>
<td>4.35</td>
<td>0.356</td>
</tr>
<tr>
<td>DCB, 0.73 mm Comp, Fw</td>
<td>0.7</td>
<td>64.0</td>
<td>2.75</td>
<td>5.44</td>
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^a indicates that actual value may be greater due to limited frequency range tested.
Post-interwell actuation generates higher frequency content in the voltage signal as shown in figures 9(c) and (d). Utilization of sub-harmonics has been seen in [22] to harvest energy off of lower frequencies for a bi-stable buckled harvester. The super harmonics seen in the present device post-actuation could lead to better performance given the right conditions like ambient vibrations that match these sub/super harmonics to assist maintaining high energy orbits.

Comparison of the TA device to the DCB is difficult because the actuation mechanisms are very different. Ultimately it was decided to keep the beam length and width the same for the DCB but using the same mass value pushed the device into a higher frequency range (>100 Hz). The larger 1.38 g mass was selected to help push the frequency down to the ~60 Hz region where it would be more comparable to the TA device. The DCB did provide more power than the TA device, but the resonance frequency was almost three times higher. When the DCB was compressed until just bi-stable the RMS power output, peak power, increased but the operating frequency decreased. Compression of the TA device tended to decrease the RMS power output and operating frequency of the entire device but increased the peak power generated. Both the TA and DCB seemed to favor reverse sweeps in generating the most power.

Two favorable outcomes from these experiments are low operating frequency of the TA device compared to the DCB and the increased peak power generation. Also, if high energy orbitals can be maintained then a frequency broadening effect for the TA device can be realized in the $V_{p-p}$ data as shown in figures 6 and 7. It is important to note that the load impedance used in these experiments are not optimal. In future iterations, an optimal load circuit using a digital pot will be created so that fast resistance and short band frequency sweeps can be used to target the driving load impedance.

Indeed, buckling tended to broaden the operating bandwidth of the devices. Even though the confirmation of frequency broadening on the TA device could not be realized at higher accelerations, it is evident from the voltage graphs that the peaks become broader as acceleration is increased. A solution to determine this for the TA device would be to remove the masses and drive it at much higher accelerations using a more powerful shaker table. Other improvements include replacing the PVDF strips with a more efficient piezoelectric material such as PZT or aluminum nitride.

6. Conclusions

A bi-stable energy harvester utilizing PVDF and buckled stability state switching has been fabricated and demonstrated in a low frequency, low acceleration testing range. Larger $V_{p-p}$ were obtained during a backwards sweep for the back and front PVDF channels with a 0.13 mm compression. A larger peak–peak voltage was obtained for the front voltage channel only with a 0.13 mm compression during a forward sweep and the back PVDF channel was comparable to the uncompressed case. Larger applied compression levels (0.25 mm), well beyond the minimum required to induce the bi-stable buckled profile, were found to be detrimental to the device performance at the reported accelerations. Bi-stable switching broadened the frequency response of the device and was seen at around 0.7 g for the 0.13 mm compression case and 1.3 g for the 0.25 compression case. A slight compression of 0.13 mm caused the center beam to have a broader operating frequency. The average natural frequency response for the device was found to be 22.2 Hz, 19.0 Hz, and 18.4 Hz for the uncompressed, 0.13 mm, and 0.25 mm compression cases, respectively. FEA predicted snap through distance for proof masses within 20% of experimental results for all cases. FEA maximum and minimum displacement of the center beam profile was within 35% of experimental results for all cases. Buckling of the DCB and TA until just bi-stable generally decreased the resonance frequency by an average of 3.17 and 27% respectively. The effect of buckling on the TA device lowered the RMS power and resonance frequency of the device. The unbuckled TA device generated the most RMS power at 1.5$g_{rms}$, 3.3 MΩ, and 24.5 Hz.

Acknowledgements

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