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Alternative Geometries for Increasing Power Density in Vibration Energy Scavenging for Wireless Sensor Networks

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Vibration energy scavenging with piezoelectric material can currently generate up to 300 microwatts per cubic centimeter, making it a viable method of powering low-power electronics. Given the growing interest in small-scale devices, particularly wireless sensor networks, concerns over how to indefinitely power them have become extremely relevant. Current limiting factors in the field of piezoelectric vibration energy scavenging include: coupling coefficients, strain distribution, and frequency matching. This paper addresses each of these three factors with a novel design and a corresponding analysis of its performance. For example, the power output of a cantilevered rectangular piezoelectric beam is limited by its uneven strain distribution under load. A prototype scavenger using a harmonically matched trapezoidal geometry solves this problem by evening the strain distribution throughout the beam, increasing by 30% the output power per unit volume. Another design is created which softens the frequency response of the generator, relaxing the constraint of frequency matching. The paper concludes that each of the three challenges to vibration energy scavenging can be met through creativity in mechanism design, making *higher power densities* possible and *broader applications* more feasible.

Nomenclature

PZT	=	lead zirconate titanate, a piezoelectric material
ε	=	strain
ω_n	=	natural frequency
k_{sp}	=	spring constant
z	=	tip deflection of cantilever beam
l,*w*,*h*	=	dimensions of cantilever beam
x,*y*	=	locators along respective dimensions *l*,*h* of cantilever beam
I	=	moment of inertia
E	=	modulus of elasticity
$d_{31,33}$	=	strain coefficient of 31,33 modes
$g_{31,33}$	=	electric field coefficient of 31,33 modes
$k_{31,33}$	=	coupling coefficient of 31,33 modes
C	=	capacitance
R	=	resistance

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P = power
'*g*' = acceleration due to gravity

I. Introduction

UBIQUITOUS networks of wireless sensors have the potential to significantly impact industry and society. Example application areas include: smart buildings, manufacturing efficiency, vehicle control, the safety of food supplies, and health care. Ultra-low-power wireless nodes are the key to realizing this potential.

While sensors and wireless electronics to support pervasive computing are becoming more prevalent, power supply to wireless sensor networks remains a challenge[1]. With the rate of improvement in batteries, it cannot be hoped that a small battery will power a wireless sensor node throughout the node's lifetime. Furthermore, if wireless sensors are to be pervasive, replacing batteries is infeasible. Energy scavenging offers an alternative to solve the energy supply problem and a number of approaches have been studied over the past years. Several researchers have pursued naturally occurring temperature variations as one potential source[2-4]. Shenck and Paradiso[5] have built shoe inserts capable of generating 8.4 mW of power under normal walking conditions. Much research has focused on solar (photovoltaic) power[6]. Although all of these power sources have an important role to play within the world of energy scavenging for wireless sensor networks, the focus of this paper is on vibrations as a power source, as this energy source is most ubiquitously available. Vibration generators based on electromagnetic[7-10], electrostatic[11-13], and piezoelectric[14-16] conversion have been suggested in the literature. After an extensive study, Roundy[17] concluded that for the size and power requirements for wireless sensor networks, piezoelectric vibration-to-electricity converters offer the most potential for meeting the demands of wireless sensor networks.

The most common type of piezoelectric generator is the piezoelectric/metal sandwich beam mounted as a cantilever, shown in Fig. 1. In this configuration, the inertial mass (M) amplifies ambient vibrations, deflecting the metal beam and straining the piezoelectric ceramic (PZT) with which it is coated. Current piezoelectric converters of this design generate about 300μW per cubic centimeter.

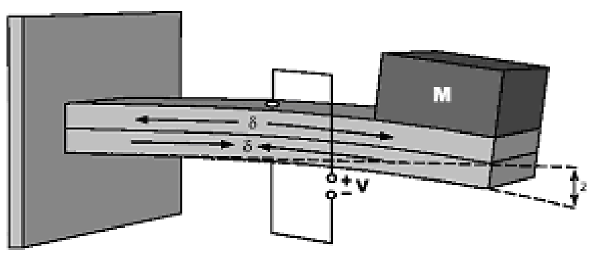

Figure 1. Schematic of traditional piezoelectric cantilever beam with inertial mass.

Prototypes have been manufactured to show that energy can indeed be scavenged from ambient vibrations, as well as to prove that this is a feasible method of powering wireless sensor networks[18]. Many scavenger devices have been demonstrated, but a number of challenges limit their application.

First, the coupling coefficient of the piezoelectric material, related to the efficiency, limits the amount of electrical energy generated per unit of mechanical energy applied. Second, the varied strain distribution in current designs leaves the piezoelectric material under-utilized, resulting in sub-optimal efficiency. Third, the constraint of matching the natural frequency of the generator to the fundamental frequency of its environment (to achieve resonance) remains a daunting challenge.

This paper suggests three mechanical designs to meet each of these three challenges and provides analysis for fair comparison with current technology, with the end goal of making vibration energy scavenging a practical reality for powering wireless sensor networks in any environment.

II. Coupling Coefficient

One challenge to developing vibration energy scavenging has been the coupling coefficient, or the amount of electricity generated as compared with the amount of mechanical strain applied. Until now, piezoelectric vibration energy has been generated primarily utilizing the 3-1 coupling mode, as shown in Figure 2a, below. The 3-3 coupling mode (Fig. 2b) is defined by a force applied in the same direction as the piezoelectric poling, whereas the 3-1 coupling mode is utilized when a force applied is in the direction perpendicular to the piezoelectric poling. Other modes are not considered here, due to their relatively small coupling coefficients.

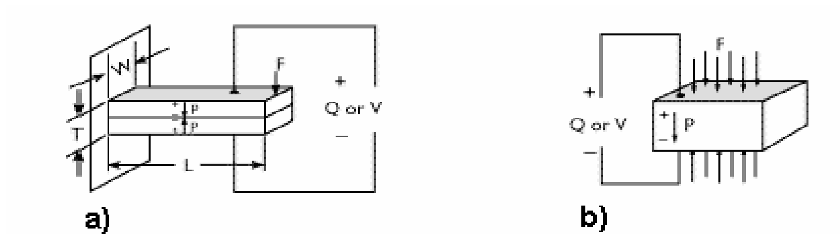

Figure 2. Schematic of a) 3-1 coupling mode, b) 3-3 coupling mode.

The coupling coefficient (*k*) is defined by Eq. (1). Table 1 lists material properties of some promising piezoelectric materials, showing that the 3-3 coupling mode has a consistently higher coupling value than does the 3-1 mode.

$$k = \sqrt{\frac{Electrical\_Energy\_Out}{Mechanical\_Energy\_In}} \qquad (1)$$

Table 1. Properties of some promising piezoelectric materials.

Property	Units	PZT	PVDF	PZN-PT
Strain coefficient (d_{31})	10^{-12} m/v	320	20	950
Strain coefficient (d_{33})	10^{-12} m/v	650	30	2000
Coupling coefficient (k_{31})	CV/Nm	0.44	0.11	0.5
Coupling coefficient (k_{33})	CV/Nm	0.75	0.16	0.91
Dielectric constant	$\varepsilon/\varepsilon_o$	3800	12	4500
Elastic modulus	10^{10} N/m^2	5.0	0.3	0.83
Tensile strength	10^7 N/m^2	2.0	5.2	8.3

Using Eq. (1), it is possible to calculate that for PZT the 3-3 coupling mode is nearly three times as efficient as the 3-1 mode. Drawn by this large efficiency difference, methods of utilizing the higher-power, 3-3 mode were investigated. A force magnification was necessary to achieve greater strain. A possible design is shown in Fig. 3.

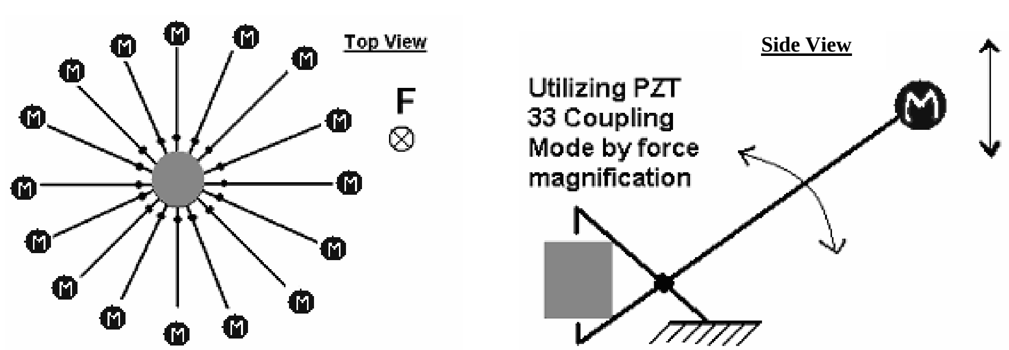

Figure 3. Schematic of piezoelectric stack design.

The slender members of this design would consist of very stiff material, with robust joints. Each member would leverage a force applied at its tip (from vibrations shaking each mass) to be much higher at the PZT stack. By adding more members in a circular fashion, the design would achieve mechanical stability and force integration.

A. Theory

In order to determine whether or not this design is an improvement in energy generation over current designs, it was compared to the traditional cantilever beam design. To ensure a fair comparison, the following terms were kept equal:

- Volume of PZT
- Force applied (inertial mass multiplied by acceleration)
- Device volume
- Maximum strain

The energy output was then calculated by multiplying the charge and the voltage produced by each design. (See Table 2.)

Table 2. Equations for comparing the piezoelectric stack to the cantilever beam.

	Cantilever Beam		Piezoelectric Stack
Volume PZT	$L_{1p}W_{1p}T_{1p}$	=	$L_{2p}W_{2p}T_{2p}$
Device Volume	$L_1W_1T_1$	=	$L_2W_2T_2$
Force applied to device	F_1	=	F_1
Force applied to PZT	F_1	Not Equal	$F_2 = \frac{R-r}{r} \times F_1$
Q (charge)	$Q = \frac{3F_1L^2d_{31}}{2T^2}$	?	$Q = F_2d_{33}$
V (voltage)	$V = \frac{3F_1Lg_{31}}{4WT}$	?	$V = \frac{F_2g_{33}T}{LW}$
Energy (Q*V)	$E = \frac{9F_1^2L^3g_{31}d_{31}}{8WT^3}$	?	$E = \frac{F_2^2g_{33}d_{33}T}{LW}$

This problem is under-constrained, and therefore many different sets of geometrical values can meet the required relations. A range of mechanically reasonable geometries were investigated.

B. Discussion

The calculations show that although the 3-3 mode has a much higher strain-to-energy conversion rate, actually inducing strain in this mode is difficult. Using the same applied force on both devices, the power output for the 3-3 piezoelectric stack is 2 orders of magnitude lower than power output in the equivalent 3-1 cantilever. However, in a high-force application, the piezoelectric stack design has two advantages over the cantilever. First, because it does not easily strain, it is much more robust. Second, it has higher efficiency.

Typical household or office vibrations have amplitudes of less than $10 m/s^2$; with such small accelerations, translating to small applied forces, the stack mechanism would be unable to generate significant energy. In environments such as heavy manufacturing facilities or in large operating machinery, a stack would stand up to a harsh environment while generating useful energy. Non-vibration applications, such as an occupant sensor in a chair leg or a wireless sensor in a mechanical stop, could also be imagined. After developing the theory for this device, it was concluded that the application space was too narrow to merit building and testing a prototype. Instead, efforts were focused on further evaluating the other two challenges to vibration energy scavenging.

III. Strain Distribution

When considering the implementation of long-term vibration energy scavengers which vibrate at resonance for maximum energy production, fatigue becomes a concern. In order to ensure that a PZT generator will maintain performance, the material strain limit of 1,000μstrain is strictly adhered to.

As shown in Eq. (1), electrical energy out is proportional to mechanical strain applied. For this reason, it is desirable to maximize the strain at each point in the beam to fully utilize the potential energy in piezoelectric material. Optimally, the strain distribution in the beam, when deflected, would be completely uniform and at the strain limit.

Unfortunately, in the traditional cantilever beam design, a simple finite element analysis (or a quick thought experiment) indicates that there will be a large stress concentration at the base. (See Fig. 4.) Whereas there exists a small area of maximum strain at the base, the tip is hardly strained at all, meaning about half of the piezoelectric material in the beam goes unutilized. By modifying the top-view footprint of the cantilever beam to allow for a more uniform strain distribution, the average strain can be raised significantly, alleviating this under-utilization problem.

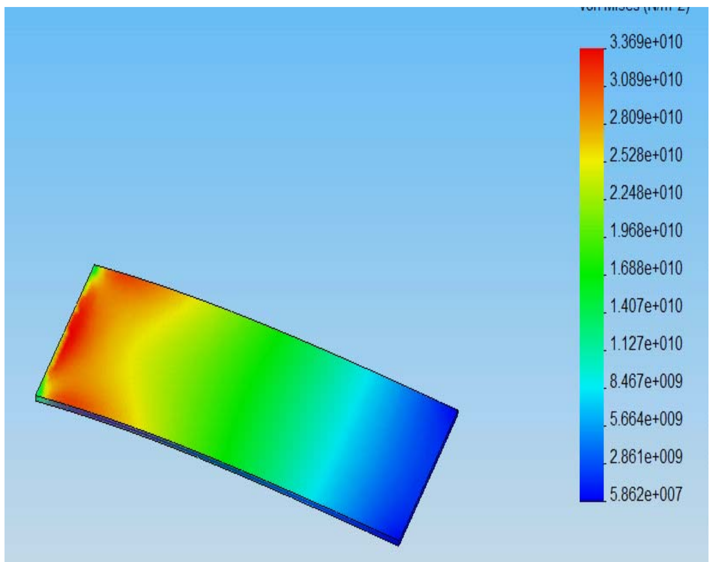

Figure 4. Finite element model of rectangular cantilever beam.

The primary cause of the stress concentration at the base of the cantilever beam is the moment induced by the inertial mass. By providing a linearly increasing wider cross-section of the bender to support the increasing moment the beam feels, nearly uniform strain can be achieved. This suggests a triangular shape with a theoretical point mass, or a trapezoidal shape with a real end mass.

In order to test out the theory of the cantilever beam with a trapezoidal footprint, it was compared to a traditional rectangular cantilever. To ensure a fair comparison, the following were kept equal:

- Volume PZT
- End (inertial) mass
- Natural frequency
- Maximum strain

The traditional sandwich beam design, with a metal support surrounded by PZT layers, was used. To optimize both the rectangular and the trapezoidal beams, Eqs. (2) were used to calculate the tip displacement (z), the spring constant (k_{sp}), the natural frequency (ω_n), and the strain(ε) at each position in the beam. Results are shown in Fig. 5, demonstrating that the more triangular the shape of the beam, the more even the strain distribution, and therefore the more energy is generated per unit volume PZT.

$$\frac{d^2 z}{dx^2} = \frac{M(x)}{EI(x)}$$

$$z(x) = \frac{12F}{Eh^3}\left\{ \begin{array}{l} \frac{(a-n\ell)}{n^3}\left[-(a-nx)\log(a-nx)+(a-nx)\right] \\ +\frac{x^2}{2n} - \frac{(a-n\ell)}{n^3}\left[a - a\log(a)\right] - \frac{(a-n\ell)}{n^2}\log(a)x \end{array} \right\}$$

$$n = \frac{\Delta w}{\ell} \qquad (2)$$

$$F = k_{sp} z$$

$$\omega_n = \sqrt{k_{sp}/m}$$

$$\varepsilon(x,y) = \frac{-M(x)y}{EI(x)}$$

Where E is the modulus of elasticity, M is the moment induced by the end mass, I is moment of inertia of the beam, F is force applied to the tip (end mass multiplied by acceleration), *x* is distance along length of beam, *a* is the narrow width of the beam, *m* is the end mass, *y* is the position in the thickness of the beam, and *w* and *h* are the width and total thickness of the beam.

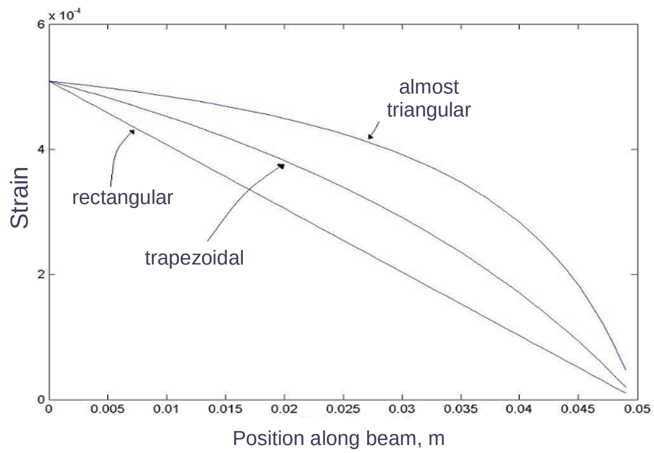

Figure 5. Theoretical results: Strain per length (at surface).

It is shown that at the realistic manufacturing limit of triangularity in the cantilever beam, the power output is 50% higher than that of a comparable rectangular cantilever.

Compressive forces generated transversely as the beam deflects are not accounted for in the model. A finite element model shows the actual strain distribution in a trapezoidal cantilever beam. (See Fig. 6.) From this figure, it is clear that although the strain distribution of the trapezoid is much improved from that of the rectangle, there is still one area of concentration. In order to have completely uniform strain, further optimization of the beam footprint would need to be performed.

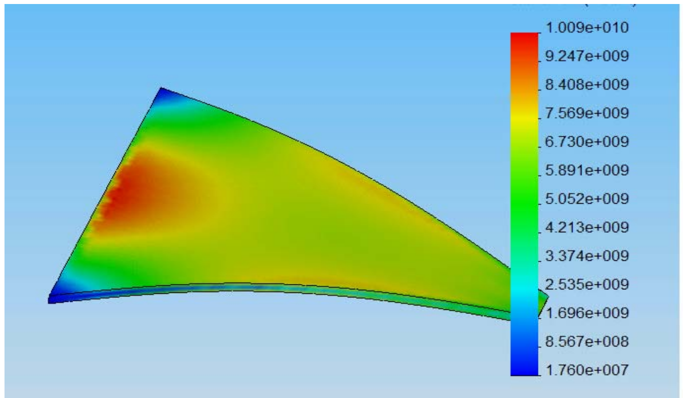

Figure 6. FEM of cantilever beam with trapezoidal footprint

However, due to its extremely brittle nature, cutting PZT is difficult, and any process more complex than a straight cut would greatly increase price, without significant improvement in power out. Therefore, the footprint has been left as a trapezoid with semi-uniform strain and good manufacturability.

A. Method

Comparable designs for cantilever beams with both rectangular and trapezoidal footprints were custom-ordered from Piezo Systems, Inc. Clamps and end masses for each beam were also designed and fabricated. Figures 7a and 7b show the two beam designs. The beams were tested using a vibrating table driven by a signal generator and an amplifier. An accelerometer was attached to the setup to ensure that both beams were driven with equal acceleration. An oscilloscope took measurements from the accelerometer and the output of each beam. To avoid parasitic capacitance, a unity gain buffer was used on the beam signal. The ideal electrical resistive load was calculated for each of the two designs, using Eq. (3). The output of the beam (through the unity gain buffer) was read across its ideal resistive electrical load, in order to ensure a perfectly fair comparison. Beams were tightly clamped and driven at their resonant frequencies. Power out was calculated using Eqs. (4).

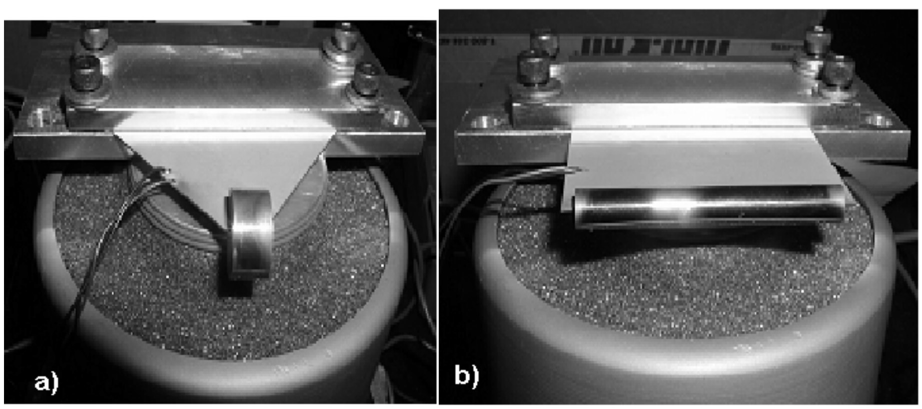

Figure 7. Cantilever beam designs with a) trapezoidal and b) rectangular footprints.

$$R_{opt} = \frac{1}{\omega_n C} \quad (3)$$

$$P_{out} = \frac{V_{rms}^2}{R_{load}}$$

$$V_{rms} = \frac{\sqrt{2} \times V_{\max}}{2} \quad (4)$$

B. Results & Discussion
Experimental results showed a *30% higher power* out in the cantilever beam with the trapezoidal footprint. This varies slightly from the theoretical 50% higher power out. Differences can be attributed to inaccuracies in the dimensions and material properties of the beams. Also contributing to discrepancies are the imperfect linear spring assumption and the neglect of the bonding layer between the metal support shim and the PZT.

The 30% gain in efficiency for the trapezoidal cantilever translates to a *smaller, cheaper generator* for the same power requirement, or an ability to use more energetically expensive sensors or wireless sensor network processes.

Ideal application spaces for the trapezoidal cantilever are the same as application spaces for the rectangular cantilever. For example, it could generate power for sensors in light manufacturing, or in HVAC air ducts. These are environments in which a high power density is required, where low vibrations exist, and no sudden impacts are likely (eliminate concern about over-strain). Additionally, it is important that the fundamental frequency of vibration of the environment exactly match the natural frequency of the cantilever beam system in order to maximize power out. Unfortunately, there exist few environments in which this last constraint can be realistically met.

IV. Frequency Matching

One difficulty with current piezoelectric generator designs is that their natural frequency must closely match the fundamental frequency of vibration of the environment in order to resonate and output maximum power. This problem is illustrated in Fig. 8. The system represented by this graph has a natural frequency of 125Hz. When the system is driven exactly at 125 Hz, it achieves resonance and outputs high power. If the system is driven at 25 or 50 Hz away from its natural frequency, its power output falls by one and two orders of magnitude, respectively.

It is difficult to closely match the fundamental and natural frequencies for several reasons. First, it requires precision manufacturing. A slight change in the material properties, or in the dimensions of the beam or inertial mass can alter the natural frequency of the generator. Second, frequencies change. A single machine may have several different fundamental frequencies corresponding to its various modes of operation. Or, as the machine wears, its fundamental frequency may drift. Third, the fundamental frequency of an environment is highly sensitive to exact location. For instance, the fundamental frequency of vibrations in the corner of an air duct is very different from the fundamental frequency measured in the center panel of the same air duct.

This need to exactly match the fundamental and natural frequencies has kept piezoelectric generators from becoming a practical reality outside of controlled environments.

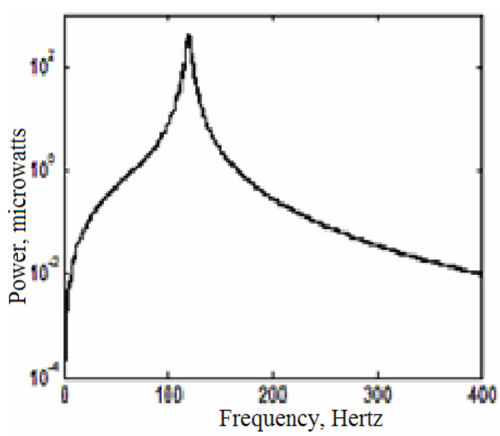

Figure 8. Frequency response of a system with ω_n=125Hz.

In response to this problem, an alternative geometry for the generator was sought which would magnify power output in the same way resonance does, without creating the tight constraint of frequency matching that resonance imposes. The chosen design puts a beam in pin-pin compression up to the critical buckling load, with a centered inertial mass, as shown in Fig. 9. It is a bi-stable device that generates power by ‘snap-through’, from one stable mode to the other. Force magnification occurs due to a constant compressive force in the structure. The ‘snap-through’ is initiated by the inertia of the mass, as the whole device vibrates with the environment. It was hypothesized that this design would relax the constraint of frequency matching.

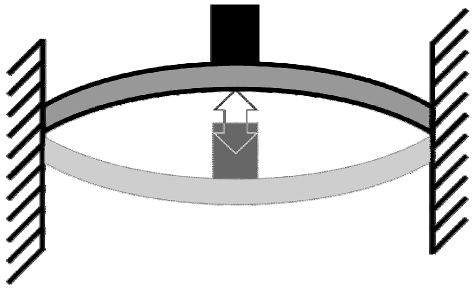

Figure 9. Schematic of bi-stable mechanism.

A. Method

In order to test the bi-stable mechanism, and to compare it to an uncompressed pin-pin-supported beam, a test-bed was designed and fabricated to support and accurately compress the beam. (See Fig. 10.) The piezoelectric beams, ordered from Piezo Systems, Inc., consist of a brass shim coated in PZT, of thickness 0.015”. A differential micrometer was used to drive the compression of the piezoelectric beam. An inertial mass of 22g was super-glued to the beam. An accelerometer was attached to the setup to ensure equal acceleration between tests. The setup was mounted to a vibrating table, which was driven by a signal generator and an amplifier. An oscilloscope read data from the accelerometer and the piezoelectric beam, which was attached to a resistive load. A force gauge read the compression force. The same beam was tested alternately in compression (as the bi-stable mechanism) and as an uncompressed beam. Data was taken at a 19 second sweep between 100Hz and 20Hz. This range was chosen because frequencies below 20Hz are rare in real-world environments; frequencies above 100Hz generated little energy for either design. This range covered the resonance frequency of each of the designs. Data was also taken in 5Hz increments over this range for direct comparison.

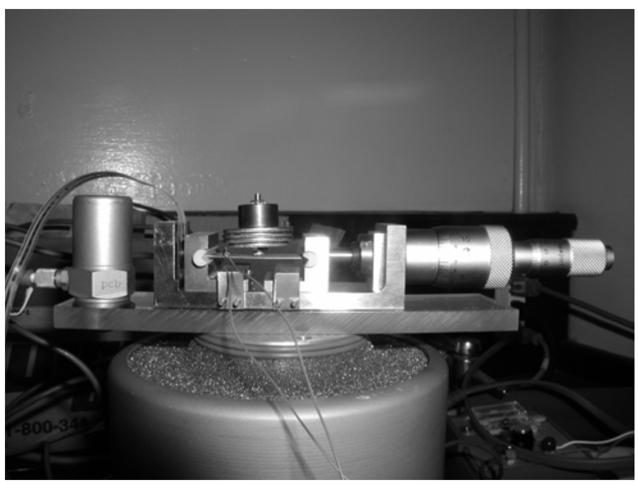

Figure 10. Setup for testing bi-stable mechanism & uncompressed beam.

B. Results

With a mass of 22g attached to the center of the beam, an acceleration of 4 ‘g’ (4 times the acceleration due to gravity) was needed to snap the beam through buckling between its two stable modes. Two representative results of the frequency sweep are shown in Fig. 11.

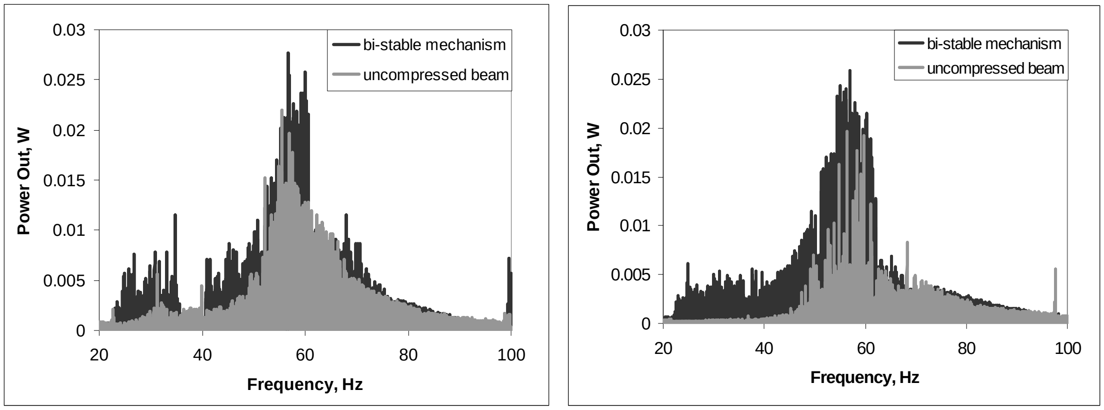

Figure 11. Experimental results: Power out vs. frequency (2 tests).

Readings were taken for distinct frequencies for each beam, all at equal acceleration. The ratio of power output from the bi-stable mechanism to the uncompressed beam were calculated. Two representative results are provided in Fig. 12.

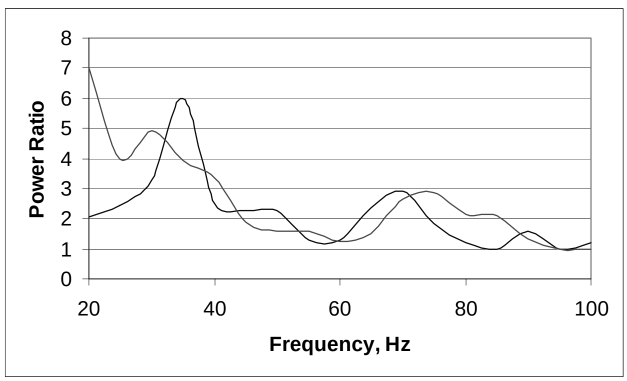

Figure 12. Plot of power ratio vs. frequency (2 tests).

C. Discussion

Figure 11 demonstrates the difference in frequency response between the bi-stable mechanism and the uncompressed beam, showing that the bi-stable mechanism has a comparatively wider and gentler peak power. By defining the available power as the integral of the power out over the relevant frequency sweep of 20 to 100 Hz, the universality of the designs can be compared. The bi-stable mechanism has a wide range in performance, but consistently has more available power – 30% to 100% more - than does the uncompressed device.

Figure 12 further demonstrates the higher available power of the bi-stable mechanism, with the ratio of power out almost always favoring the bi-stable mechanism. At resonance, the two designs have similar power outputs. However, if the fundamental frequency falls more than 5 Hz away from the beam's natural frequency in either direction, the bi-stable mechanism's power output is much higher. At low frequencies, the bi-stable mechanism generates 4 to 7 times more power than the uncompressed beam. The results warn that the power output, despite being cumulatively more across a broad sweep of frequencies, is still low for the bi-stable mechanism when off-resonance. Still, for applications in which fundamental frequency cannot easily or economically be matched, this design is a simple solution for at least doubling the power output over current designs.

1. An integrated design

Rectangular piezoelectric beams were used for these experiments because they are easy to acquire. Combining the lessons of the trapezoidal cantilever and the bi-stable mechanism, it is clear that the footprint of the beam could be optimized to increase the power out by another 30%. (See Fig. 13.)

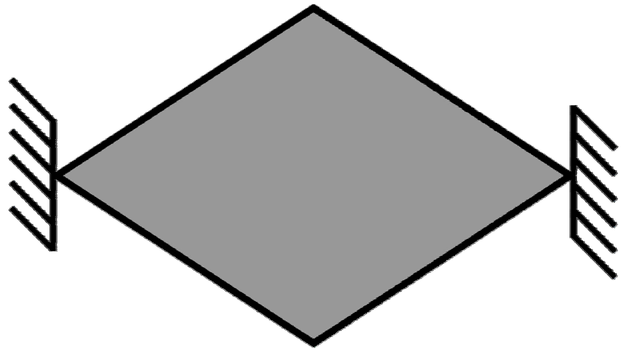

Figure 13. Schematic of improved footprint of bi-stable mechanism.

It should be noted that the required driving acceleration of 4 ‘g’ is a very large acceleration, found in select but important environments such as industrial, aerospace and automotive applications. The conclusion is drawn that the bi-stable mechanism is a *superior design in applications of uncertain or changing fundamental frequency*, the most common real-world scenario.

V. Conclusion

Many of the current challenges in vibration energy scavenging with piezoelectric material can be solved by utilizing new mechanical designs, and considering real-world, rather than ideal, environments.

- Although the piezoelectric stack design is more efficient and more robust than the traditional cantilever beam, its application space is extremely narrow, requiring very high forces for power generation.

- Changing the top view footprint of a piezoelectric beam can even the strain distribution, leading to a 30% increase in power output per unit volume PZT. This is applicable to any current cantilever scavenger.

- Using a bi-stable mechanism can relax the tight constraint of frequency matching, and create a more universal solution to micro-power generation. By optimizing the footprint of the bi-stable mechanism, the power output and universality of the design can be further augmented. *This makes powering wireless sensor networks with vibration energy scavenging a practical reality* outside the lab.

Acknowledgments

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References

[1]B. Otis and J. Rabaey, “A 300µW 1.9GHz oscillator utilizing micro-machined resonators,” *IEEE Proceedings of the 28th ESSCIRC*, vol. 28, Sept. 2002.

[2]Strasser, M., et al. (2002*) Sens. Act. A*, Vol. 97-98C, 528-535.

[3]Pescovitz (2002) “The Power of Small Tech.” Smalltimes, 2(1).

[4]Stordeur, M., Stark, I., “Low Power Thermoelectric Generator – self-sufficient energy supply for micro systems,” *16th Int. Conf. on Therm*, 1997, pp. 575-7.

[5]Shenck, N. S., Paradiso, J. A., 2001. “Energy Scavenging with Shoe-Mounted Piezoelectrics,” *IEEE Micro*, 21 (2001) 30-41.

[6]Randall, J.F. *On ambient energy sources for powering indoor electronic devices* Ph.D. thesis Ecole Polytechnique Federale de Lausanne, Switzerland May 2003

[7]Shearwood, C., Yates, R.B., 1997, Development of an electromagnetic micro-generator, *Electronics Letters*, vol.33, (no.22), IEEE, 23 Oct. 1997. p.1883-1884.

[8]Amirtharajah, R., Chandrakasan, A.P., "Self-Powered Signal Processing Using Vibration-Based Power Generation," *IEEE JSSC*, Vol. 33, No. 5, p. 687-695.

[9]El-hami, M., Glynne-Jones, P., White, N.M., Hill, M., Beeby, S., James, E., Brown, A.D., and Ross, J.N., 2001. "Design and fabrication of a new vibration-based electromechanical power generator." *Sensors and Actuators A (physical)* 92 (2001) 335-342.

[10]Ching N.N.H., Wong H.Y., Li W.J., Leong P.H.W., Wen Z., 2002. "A laser-micromachined multi-modal resonating power transducer for wireless sensing systems." *Sensors and Actuators A (physical)* 97-98 (2002) 685-690.

[11]Meninger, S., Mur-Miranda, J.O., Amirtharajah, R., Chandrakasan, A.P., and Lang, J.H., 2001. "Vibration-to-Electric Energy Conversion." *IEEE Trans. VLSI Syst*., 9 (2001) 64-76.

[12] Miyazaki M., Tanaka H., Ono G., Nagano T., Ohkubo N., Kawahara T., and Yano K., 2003. "Electric-Energy Generation Using Variable-Capacitive Resonator for Power-Free LSI: Efficiency Analysis and Fundamental Experiment." ISLPED 2003, August 25-27, 2003, Seoul Korea.

[13]Mitcheson, P.D., Green, T.C., Yeatman, E.M., and Holmes, A.S., "Architectures for Vibration-Driven Micropower Generators", *Journal of Microelectromechanical Systems*, Vol. 13, No. 3, (2004) 1-12.

[14]Glynne-Jones P., Beeby S.P., Jame E.P., and White N.M., 2001. "The Modelling of a Piezoelectric Vibration Powered Generator for Microsystems." *Transducers '01 / Eurosensors XV*, Munich, Germany, June 10-14, 2001.

[15]Ottman G. K., Hofmann H. F., Lesieutre G. A., Optimized piezoelectric energy harvesting circuit using step-down converter in discontinuous conduction mode. *IEEE Trans on Power Elect*, vol.18, no.2, 2003, pp.696-703.

[16]Roundy, S., Wright, P .K., and J. Rabaey, <u>Energy Scavenging for Wireless Sensor Networks with Special Focus on Vibrations</u>, Kluwer Academic Press, ISBN Number 1- 4020-7663-0.

[17]Roundy, S. J., 2003. Energy Scavenging for Wireless Sensor Nodes with a Focus on Vibration to Electricity Conversion, Ph.D Thesis, University of California at Berkeley, Berkeley CA, May 2003.

[18]Eli S. Leland, Elaine M. Lai, Paul K. Wright. "A Self-Powered Wireless Sensor for Indoor Environmental Monitoring." *WNCG 2004 – Wireless Networking Symposium.* Oct 20-22, 2004, Austin, TX.