Improving Power Output for Vibration-Based Energy Scavengers

Given appropriate power conditioning and capacitive storage, devices made from piezoelectric materials can scavenge power from low-level ambient sources to effectively support networks of ultra-low-power, peer-to-peer wireless nodes.

Shad Roundy
Australian National University

Eli S. Leland, Jessy Baker, Eric Carleton, Elizabeth Reilly, Elaine Lai, Brian Otis, Jan M. Rabaey, and Paul K. Wright
University of California, Berkeley

V. Sundararajan
University of California, Riverside

Pervasive networks of wireless sensor and communication nodes have the potential to significantly impact society and create large market opportunities. For such networks to achieve their full potential, however, we must develop practical solutions for self-powering these autonomous electronic devices.

Fixed-energy alternatives, such as batteries and fuel cells, are impractical for wireless devices with an expected lifetime of more than 10 years because the applications and environments in which these devices are deployed usually preclude changing or re-charging of batteries. There are several power-generating options for scavenging ambient environment energy, including solar energy, thermal gradients, and vibration-based devices. However, it's unlikely that any single solution will satisfy all application spaces, as each method has its own constraints: solar methods require sufficient light energy, thermal gradients need sufficient temperature variation, and vibration-based systems need sufficient vibration sources. Vibration sources are generally more ubiquitous, however, and can be readily found in inaccessible locations such as air ducts and building structures.

We've modeled, designed, and built small cantilever-based devices using piezoelectric materials that can scavenge power from low-level ambient vibration sources. Given appropriate power conditioning and capacitive storage, the resulting power source is sufficient to support networks of ultra-low-power, peer-to-peer wireless nodes. These devices have a fixed geometry and—to maximize power output—we've individually designed them to operate as close as possible to the frequency of the driving surface on which they're mounted. Here, we describe these devices and present some new designs that can be tuned to the frequency of the host surface, thereby expanding the method's flexibility. We also discuss piezoelectric designs that use new geometries, some of which are microscale (approximately hundreds of microns).

Problem overview

We first analyze the wireless sensor nodes' power requirements, and then investigate the various sources that can fill those demands.

Power demand

Assuming an average distance between wireless sensor nodes of approximately 10 meters—which means that the radio transmitter should operate at approximately 0 dBm (decibels above or below 1 milliwatt)—the radio transmitter's peak power consumption will be around 2 to 3 mW, depending on its efficiency. Using ultra-low-power techniques,[1] the receiver should consume less than 1 mW. Including the dissipation of the sensors and

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TABLE 1
Energy and power sources comparisons.

Power source	Power (μW)/cm³	Energy (Joules)/cm³	Power (μW)/cm³/yr	Secondary storage needed?	Voltage regulation?	Commercially available?
Primary battery	N/A	2,880	90	No	No	Yes
Secondary battery	N/A	1,080	34	N/A	No	Yes
Micro fuel cell	N/A	3,500	110	Maybe	Maybe	No
Ultracapacitor	N/A	50–100	1.6–3.2	No	Yes	Yes
Heat engine	1×10^6	3,346	106	Yes	Yes	No
Radioactive (^{63}Ni)	0.52	1,640	0.52	Yes	Yes	No
Solar (outside)	15,000*	N/A	N/A	Usually	Maybe	Yes
Solar (inside)	10*	N/A	N/A	Usually	Maybe	Yes
Temperature	40*†	N/A	N/A	Usually	Maybe	Soon
Human power	330	N/A	N/A	Yes	Yes	No
Air flow	380‡	N/A	N/A	Yes	Yes	No
Pressure variation	17§	N/A	N/A	Yes	Yes	No
Vibrations	375	N/A	N/A	Yes	Yes	No

* Measured in power per square centimeter, rather than power per cubic centimeter.
† Demonstrated from a 5°C temperature differential.
‡ Assumes an air velocity of 5 m/s and 5 percent conversion efficiency.
§ Based on 1 cm³ closed volume of helium undergoing a 10°C change once a day.

peripheral circuitry, a maximum peak power of 5 mW is quite reasonable.

For the radio to have a maximum data rate of 100 Kbits per second, and an average traffic load per node of 1 Kbit/sec, every node communicates for approximately 1 percent of its deployed life. During the remaining 99 percent, the only activities occurring in a node are background tasks: low-speed timers, channel monitoring, and node synchronization. If not handled appropriately, the latter is actually the node's dominant power-consuming source. Combining peak and standby power dissipation leads to an average power dissipation of approximately 100 microwatts.

At an average power consumption of 100 μW (an order of magnitude smaller than any currently available node) a sensor node needs slightly more than 1 cm³ of lithium battery volume for one year of operation, assuming that 100 percent of the battery's charge is used. So, given a 1 cm³ size constraint, standard sensor node batteries must be replaced at least every nine months.[2]

Supply options

As Table 1 shows, vibration-based devices compare well to other potential energy-scavenging sources, including batteries, fuel cells, and solar, temperature, and pressure devices.[2]

Researchers have successfully built and tested vibration-based generators using three types of electromechanical transducers: electromagnetic,[3] electrostatic,[4] and piezoelectric.[5,6] The most effective transducer type depends, to a certain extent, on the specific application. Researchers often compare different methods' effectiveness on the basis of the energy storage density inherent to each transducer type. Table 2 compares three generators on the basis of two energy densities:[2]

- *Practical values* represent what is currently achievable with standard materials and processes.
- *Aggressive values* represent what is theoretically possible.

In addition to energy density, three further considerations affect transducer technology selection:

- Electrostatic transducers are more readily implemented in standard micro-machining processes.
- Electrostatic transducers require a separate voltage source (such as a battery) to begin the conversion cycle.
- Electromagnetic transducers typically output AC voltages well below 1 volt in magnitude.

Based on the data presented in Table 2, we decided to focus our work on piezoelectric generators.

TABLE 2
Energy storage density comparison.

Type	Practical maximum (millijoules/cm³)	Aggressive maximum (millijoules/cm³)
Piezoelectric	35.4	335
Electrostatic	4	44
Electromagnetic	24.8	400

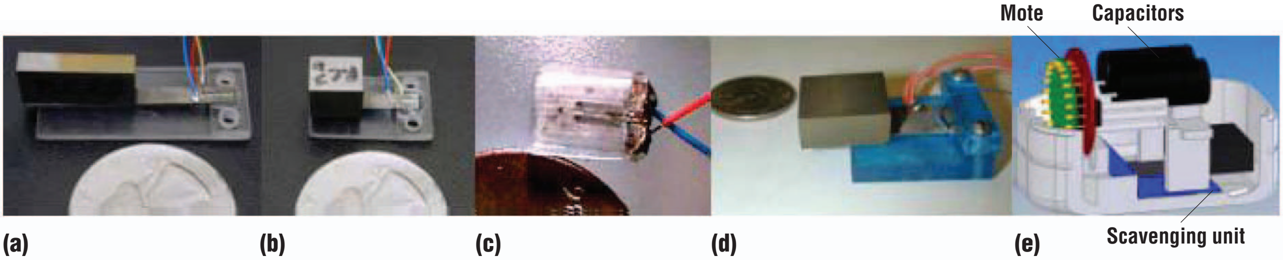

Figure 1. Meso-scale piezoelectric generators. The 1 cm^3 generator prototypes (a) and (b) powered a 1.9 GHz radio transmitter and resulted in an average power consumption of 120 μW. (c) A generator that restricts maximum deflection and stresses at the attachment point. A second test used a larger generator (d) integrated into a complete package (e) that measured and transmitted temperature readings.

Results

As the last row of Table 1 indicates, vibration sources can generate approximately 375 $\mu W/cm^3$. An initial vibration model indicates that a 1 cm^3 design can generate 375 μW from a vibration source of 2.5 m/s^2 at 120 Hz.[2] On the basis of this model, researchers fabricated some early prototypes of tiny, piezoelectric cantilevers (9 to 25 millimeters in length) with a relatively heavy mass on the free end (see Figure 1).

When affixed to a vibrating surface, such as a wooden staircase or the inside of an air-conditioning duct, these prototypes scavenged and stored enough energy on a capacitor to power sensor nodes. In recent tests, we also used the 1 cm^3 generator prototypes (Figures 1a and 1b) to power a 1.9 gigahertz radio transmitter.[1] The beacon was powered at a duty cycle of 1 percent, which resulted in an average power consumption of 120 μW. Figure 1c shows a variation of the generator that includes a casing to restrict the range of bender motion. In addition, the beam's attachment point to the support is widened to reduce stresses.

In a second test, using the larger generator integrated into a complete package (Figures 1d and 1e), a Mica2Dot "mote" (www.xbow.com) transmitted temperature readings, powered at about a 1 percent duty cycle. The "on" power of the larger node was 40 to 60 mW, and the average power was 400 to 600 μW. This TempNode unit—run entirely from scavenging—was part of a smart-building project for regulating residential temperatures.

Modeling piezoelectric energy scavengers

As Figure 2 shows, we have primarily based our piezoelectric generators on a two-layer bender (or *bimorph*) mounted as a cantilever beam. The device's top and bottom layers are composed of piezoelectric material. As the figure shows, bending the beam down produces tension in the top layer and compresses the bottom layer. A voltage develops across each of the layers, which we can condition and use to drive a load circuit. If we wire the layers in series, their individual voltages add. If we wire them in parallel, their individual currents add.

All the generators in Figure 1 use this basic type of construction and operation. Also, although Figure 2 doesn't show a neutral central layer (typically made of metal), most of our prototypes contain such a layer. This central elastic layer adds robustness, as the ceramic is very brittle; if we carefully choose the relative layer thickness, the central layer can also improve overall electromechanical coupling.

Power output

When vibrations drive the device, the generator provides an AC voltage. When we connect a resistor across the piezoelectric electrodes, a simple resistance-capacitance (RC) circuit results. By combining the standard beam equations with

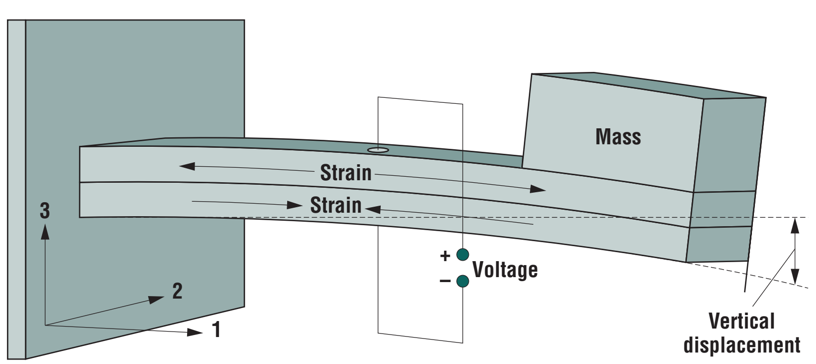

Figure 2. A two-layer bimorph mounted as a cantilever. The top and bottom layers are piezoelectric; bending the beam creates tension in the top layer and compresses the bottom layer.

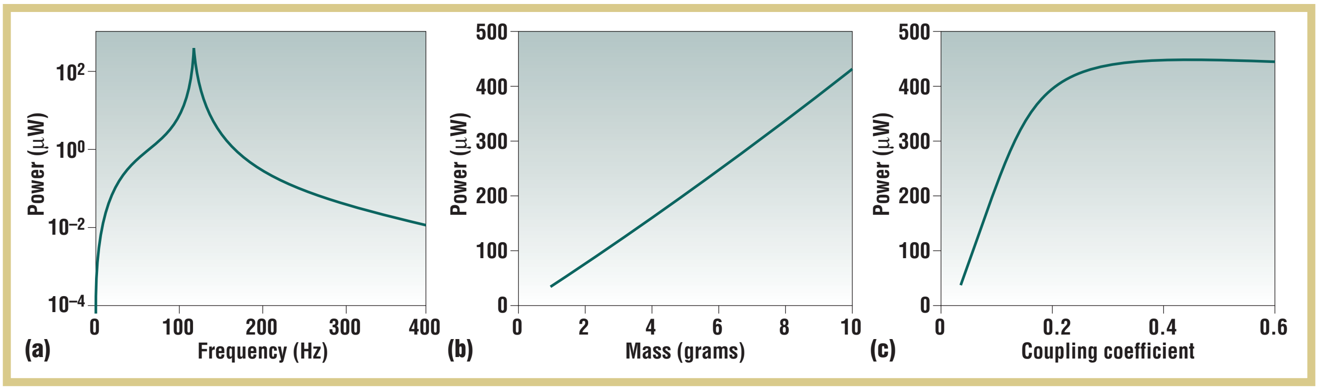

Figure 3. Simulations for a 1 cm^3 piezoelectric scavenger driven by vibrations of 2.5 m/s^2. (a) Power output versus driving frequency. The design's resonance frequency is 120 Hz, and its proof mass is 9 grams. (b) Power output versus proof mass. All parameters are constant except piezoelectric-beam thickness; maximum deflection is approximately 90 μm. (c) Power output versus coupling coefficient. The proof mass is 9 grams, and the maximum deflection is approximately 90 μm.

the constitutive piezoelectric equations and circuit equations, we can derive a relationship for power output as a function of input vibration amplitude and frequency.[2] Equation 1 shows this relationship:

$$P=\frac{V^2}{2R}=\frac{1}{2R}\times\frac{\left(\frac{2k_{31}t_c}{k_2}\right)^2\frac{c_p}{\varepsilon}A_{in}^2}{\left[\frac{\omega_n^2}{\omega RC_b}-\omega\left(\frac{1}{RC_b}+2\zeta\omega_n\right)\right]^2+\left[\omega_n^2\left(1+k_{31}^2\right)+\frac{2\zeta\omega_n}{RC_b}-\omega^2\right]^2} \qquad (1)$$

where

- ω is the frequency of driving vibrations.
- ω_n is the resonance frequency of the generator.
- c_p is the elastic constant of the piezoelectric ceramic.
- k_{31} is the piezoelectric coupling coefficient.
- t_c is the thickness of one layer of the piezoelectric ceramic.
- k_2 is a geometric constant that relates average piezoelectric material strain to the tip deflection.
- ε is the dielectric constant of piezoelectric material.
- R is the load resistance.
- V is the voltage across the load resistance.
- C_b is the capacitance of the piezoelectric bimorph.
- ζ is the dimensionless damping ratio, which represents the viscous loss from the system.

The complicated second term represents the square of voltage across the load resistor as a function of input vibrations and frequency. While the RC circuit is oversimplified, it does give a reasonably good approximation of the amount of power generated.

If we assume that the device is operating at resonance (that is, the driving frequency, ω, matches the natural frequency ω_n), then we could rewrite Equation 1 as

$$P=\frac{m^2}{2k}\times\frac{RC_b^2\left(\frac{2k_{31}t_c}{k_2}\right)^2\frac{c_p}{\varepsilon}A_{in}^2}{(4\zeta^2+k_{31}^4)(RC_b)^2k+4\zeta k_{31}^2RC_b\sqrt{km}+4\zeta^2m}, \qquad (2)$$

where $\omega_n^2 = km^{-1}$, substituting k, the equivalent spring stiffness. This equation is similar to the one developed by Shad Roundy, Paul Wright, and Jan Kalaley,[2] except that we've used mass and stiffness in place of natural frequency, and the coupling coefficient k_{31} in place of the strain coefficient d_{31}.

Implications

It directly follows from Equation 1 that power output is maximized when the natural frequency ω_n is equal to the driving frequency ω. In fact, power output drops off dramatically as ω_n deviates from ω, as Figure 3a shows. Equation 2 indicates that power is dependent on the proof mass m. As Figure 3b shows, this relationship is close to linear in the target region. Because natural frequency depends on the stiffness and mass ($\omega_n = (k/m)^{1/2}$), an increase in mass will necessitate an increase in stiffness to maintain the natural frequency. We typically accomplish this increase in stiffness by making the piezoelectric beam thicker or wider, thus increasing the amount of piezoelectric material. So, a mass increase is typically accompanied by an increase in the volume of piezoelectric material.

Figure 3c graphically indicates the relationship between the power and coupling coefficient. As the figure shows, with increasing coupling, power increases quickly up to a point, beyond which the increase is quite modest. Nevertheless, because system-level coupling is usually below the "knee" in Figure 3c (especially for microfabricated devices), improving the material coupling coefficient is an important research area. All of the parameter values in Figures 3 are typical of devices that we've built and tested. Furthermore, vibrations of 2.5 m/s^2 at 120 Hz are typical of many low-level vibrations we've measured.[2]

Table 3 summarizes the design rela-

TABLE 3
Design relationships and potential improvements for piezoelectric vibration-based scavengers.

Design relationship from Equations 1 and 2	Current designs	Design strategy/improvement
Power vs. resonance frequency	Operation limited to a narrow frequency band	Design for adaptive self-tuning of the resonance frequency
Power vs. mass	Power limited by proof mass at the end of the cantilever	Explore designs that improve the strain from a given mass
Power vs. piezoelectric coupling coefficient	System coupling coefficient is below "knee" in power vs. coupling curve	Improve designs for better system coupling, and improve thin-film piezoelectric-material properties
System integration (constraints implied by equations)	Limited by hand assembly	Possibly use MEMS (microelectromechanical systems) designs to integrate with sensors and a CMOS (complementary metal-oxide semiconductor)

tionships that our model highlights. The data in the table also serves as the basis for our design improvements and research directions for piezoelectric energy scavengers. The design of optimal power electronics to efficiently convert power from a high-impedance AC source (the piezoelectric element) to a low-impedance DC supply is also an active research area[7] but is outside the scope of this article.

Improving energy harvesting

We can increase the energy that can be scavenged from vibration sources by either passively or actively altering the bender configuration, or by modifying the bender geometry.

Several possible designs could obtain more energy from a given source.

Self- and adaptive-tuning energy sources

Vibration-based energy-scavenging devices generate the maximum power when their resonance frequency matches the driving frequency. Under many circumstances, we know the driving frequency before we design and fabricate the device and can thus build in the appropriate resonance characteristics. In other situations, however, we either don't know this frequency a priori or it might change over time. Given this, it would clearly be advantageous to have a single design that operates effectively over a range of vibration frequencies. Two possibilities exist here: we can develop actuators to alter the scavenger's resonance frequency or develop designs for scavengers with wider bandwidths.

Resonance-tuning actuators. We classify potential resonance-tuning actuators into two categories:

- *Active tuning actuators* run continuously to match the generator's resonant frequency to the source's driving frequency. Electronic springs are a good example here.
- *Passive tuning actuators* tune the generator then turn off while maintaining the new resonance frequency. An example here is a moveable clamp that changes the length (and thus the stiffness) of a flexure mechanism.

An active tuning actuator provides a force that is proportional to displacement (alters the effective stiffness), proportional to acceleration (alters the effective mass), or proportional to velocity (alters the effective damping). Roundy has mathematically shown that, as long as a second-order mechanical system models the scavenger system reasonably well, an active actuator will never improve power output.[8] In other words, the actuator will always require more power than its operation ultimately provides.

A passive tuning actuator must be able to alter the resonance frequency and then cut power to the actuator while maintaining the new resonance frequency. One possible method to alter the stiffness of the beam structure is to apply destabilizing axial loads.[9] Figure 4a illustrates this idea. We modeled the beam as a simply supported beam with a proof mass in the middle (in reality, there would be another proof mass on the top of the beam, but we removed it here for the sake of simplicity). The beam's apparent stiffness is a function of the axial compressive preload; it theoretically reduces to zero as the axial preload approaches the critical buckling load. We could easily apply the preload by set screws or other devices that push on the clamps at either end of the beam. While this would require a significant amount of tuning-phase power, once the proper preload is applied, we can turn off all power to the tuning operation.

Figure 4b shows how the resonance frequency of Figure 4a's beam decreases as a function of preload. We can reduce the resonance frequency by approximately 40 percent by using a preload equal to half of the critical buckling load. Furthermore, the actuator's response is fairly linear in the region up to half of the critical buckling load.

Wide bandwidth designs. A second approach is to design a structure with a wider inherent bandwidth. Again, using the model in Figure 3, we can prove the bandwidth to be approximately $2\zeta\omega_n$. If the bimorph resonates at 120 Hz and the damping ratio is 0.025 (which corresponds with our measurements), the bandwidth is 6 Hz, or +/– 3 Hz. This implies that to get a high coupling

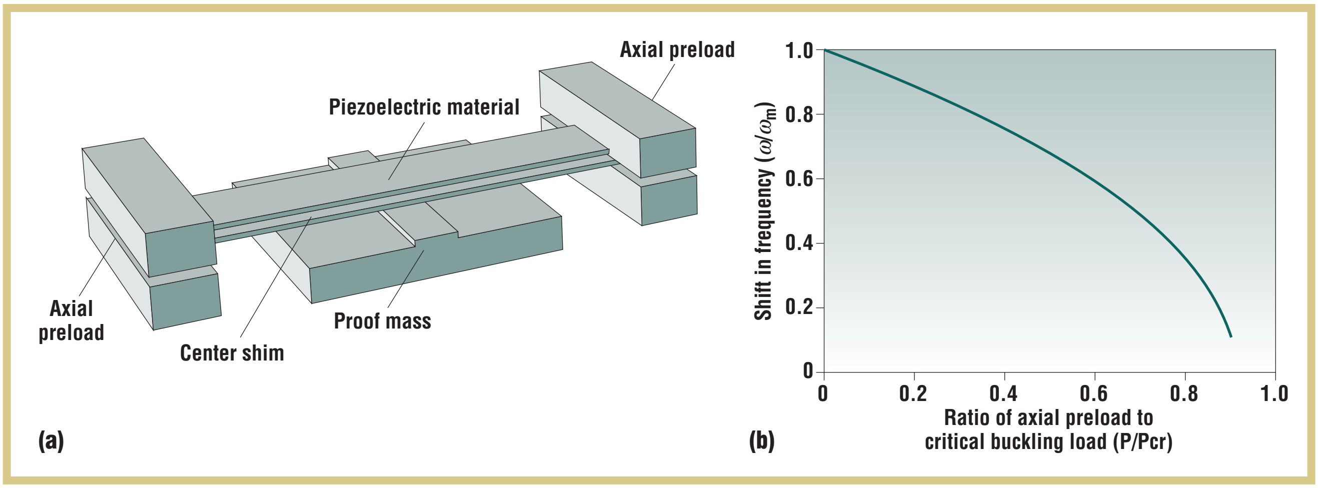

Figure 4. Applying axial loads to alter beam stiffness. (a) Schematic of a simply supported piezoelectric beam scavenger with an axial preload (with the top half of proof mass removed for simplicity's sake). (b) Ratio of tuned resonance frequency to original resonance frequency as a function of the axial preload.

between the source and the piezoelectric transducer, we must tune the scavenger to within +/– 3Hz. Practically, this can be quite difficult because the scavenger geometry and material properties show enough variation to push the device off resonance. One way to design scavengers with a wider bandwidth is to connect N spring-mass-damper systems (Figure 5a shows a system with three masses and four springs). This locates the $i + 1$th system's resonance frequency one bandwidth away from that of the ith system. The various spring-mass subsystems' resonant frequencies thus almost overlap such that at least one element is in resonance over the desired frequency range.

Figure 5b shows the frequency response for Figure 5a's mass position amplitudes, where k_1 = 568 N/m, k_3 and k_2 = 776 N/m, and k_4 = 488 N/m. The masses are m_1 = 2.8125 grams and m_3 = 2.268 grams, while m_2 = 30 grams. We added a damping constant of 0.025. As the figure shows, the overall bandwidth is 17.5 Hz, with two of the four springs contributing to the output at any given time.

Of the three possible approaches to increasing the generator's operation frequency, the most feasible seem to be multimass, multimode resonators, which can

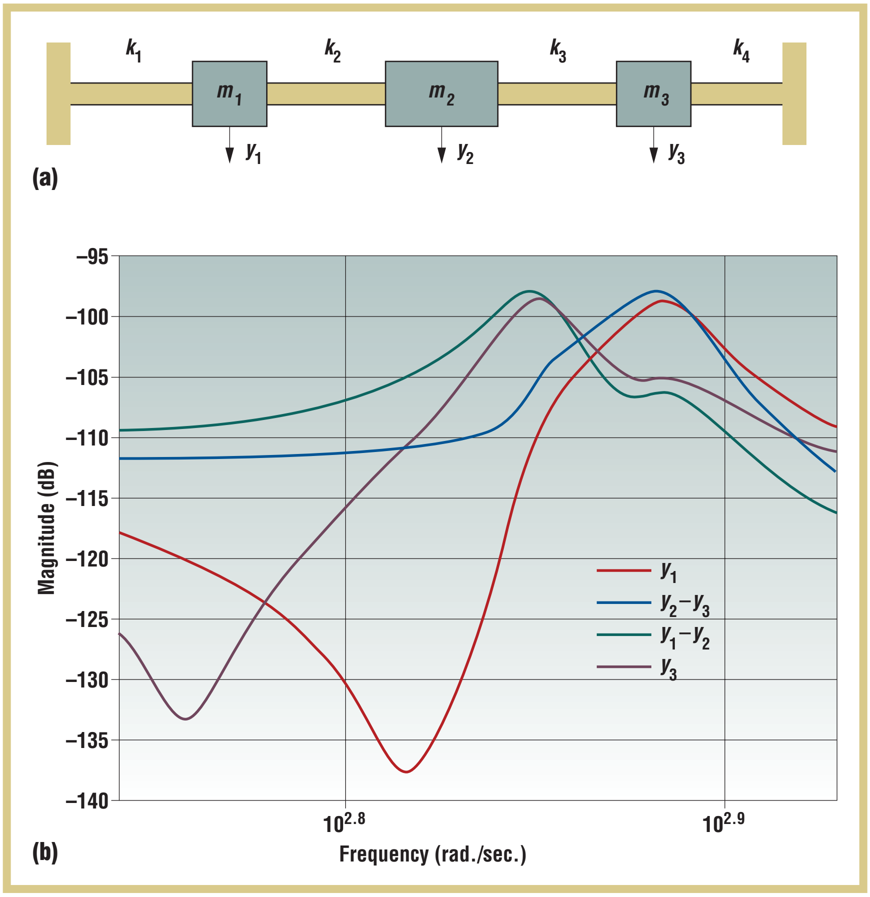

Figure 5. Multi-degree-of-freedom bimorph designed for higher bandwidth. (a) A system comprising a piezoelectric beam with three masses (m_1 ... m_3) and four springs (k_1 ... k_4); (y_1 ... y_3) represents the downward deflection of masses. (b) The frequency response for the position amplitudes of the masses.

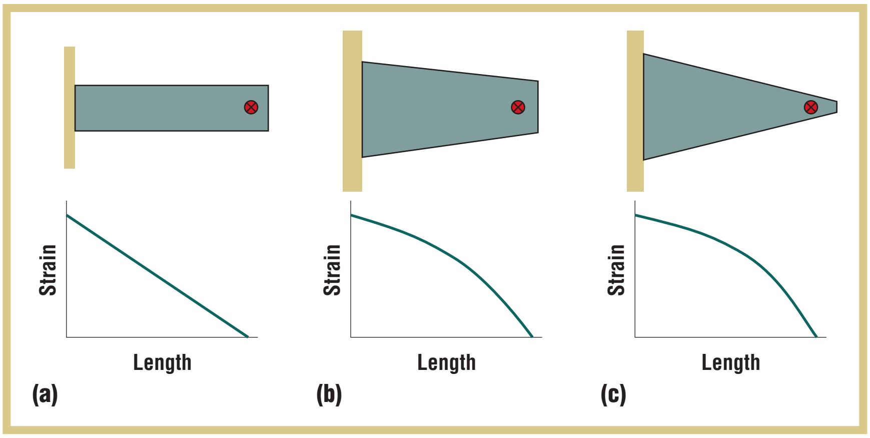

Figure 6. Relative bending energies and strain profiles for alternative beam geometries. (a) $U = 1$; (b) $U = 1.54$; and (c) $U = 2.17$. The red circles indicate the point at which load is applied.

easily triple the generator's bandwidth, and passive tuning actuators, which can reduce frequency by 50 percent. Also, because a passive tuning actuator requires no power once the frequency is tuned, over time the total energy output will increase.

Alternative mechanical structures

Most researchers have focused on traditional or slightly varied cantilever beams. A cantilever beam has several advantages: it produces relatively low resonance frequencies and relatively high average strain for a given force input. It is also easily realizable in a microfabrication process. However, a broader consideration of potential design geometries can increase scavenger performance.

A design geometry should try to address at least one (if not all) of the following goals:

- Maximize the piezoelectric response for a given input. This can be accomplished either by maximizing the material's average strain for a given input or by changing the design to use direct coupling rather than the transverse coupling that piezoelectric bimorphs use.
- Improve scavenger robustness by reducing stress concentration.
- Minimize the losses (damping) associated with the mechanical structure.
- Improve the scavenger's manufacturability.

There are a few potential designs that address one or more of these goals. Improving the geometry of the scavenger's piezoelectric bimorph is itself a major improvement area. Bulk material properties impose a strain limit of 500 microstrain on the bimorph to avoid brittle fracture. To prevent overstrain and maximize energy, we must design geometries that produce uniform and large strain below the 500 microstrain limit. With the same volume of lead zirconium titanate (PZT, the most common type of piezoelectric material) and an increasingly triangular trapezoidal profile (rather than a rectangular cantilever), we can distribute the strain more evenly, such that maximum strain is reached at every point in the bimorph. A trapezoidal geometry can supply more than twice the energy (per unit volume PZT) than the rectangular geometry, reducing both the bimorph's size and cost. Figure 6 shows the bending energy value relative to a cantilever of uniform width for three alternative beam geometries.

In addition to modifying the cantilever beam's profile, we are exploring

- *A simply supported beam*. This design exploits the trapezoidal shape, while letting us apply an axial load at the clamped ends (which provides for frequency tuning).
- *A curved compressor*. In this design, we utilize curvature in the generator to ensure that the acting force induces compressive rather than tensile stress in the piezoelectric material. This extends fatigue life—the number of bending cycles the structure can withstand before failing—while maintaining the same power output level as an equivalent tensile stress. Again, we can apply a lateral force to tune the natural frequency of the generator to its environment.
- *A compliant mechanism and stack*. This design translates relatively low-force, high-displacement proof mass oscillations into smaller-displacement, higher-force motion around a PZT stack. This capitalizes on the direct-coupling mode's higher efficiency without the usual high actuation force requirement.

Developing alternate structures can significantly increase a device's longevity. This, in turn, lets the device operate at larger vibration amplitudes and thus generate more energy from the environment.

Vibration-based energy scavenging is a viable means of obtaining the small quantities of energy necessary to power wireless sensor nodes. Current technology, however, focuses mostly on rectangular cantilever bulk PZT structures. Such structures have low bandwidth, inefficiently utilize PZT material, and can't be integrated with standard fabrication methods. To overcome some of these problems, we're pursuing three approaches to improving vibration-based energy scavengers:

- *Frequency tuning*. Current designs must resonate at the driving frequency to generate significant power. We are pursuing designs incorporating tuning actuators that will tune the generator's natural frequency to match the driving

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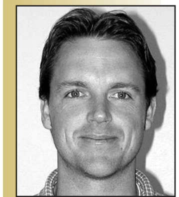
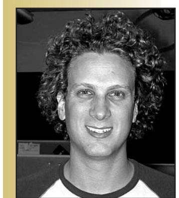


Shad Roundy is chief architect for energy scavenging at LV Sensors. His research interests include energy scavenging, MEMS, and smart material actuators. He received his PhD in mechanical engineering from UC Berkeley, where he was the recipient of the Intel Noyce Fellowship and Department of Energy Integrated Manufacturing Fellowship. Also, MIT's *Technology Review* recently named him one of the world's 100 top young innovators. Contact him at LV Sensors, 1480 6th St., Suite 175, Emeryville, CA 94608; sroundy@lvsensors.com.



Eli S. Leland is a PhD student in mechanical engineering at UC Berkeley. His research interests include vibration energy scavenging and energy policy. He received a BS in mechanical engineering from Princeton University. He is a National Defense Science and Engineering Graduate fellow and a member of Sigma Xi, the Scientific Research Society. Contact him at 2111 Etcheverry Hall, UC Berkeley, Berkeley, CA 94720; eli@kingkong.me.berkeley.edu.

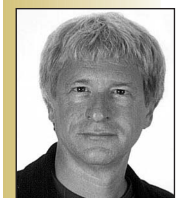
Jessy Baker is a second-year graduate student and National Science Foundation fellow at UC Berkeley. Her research interests are in alternative energy and sustainable technologies. She received her BS in mechanical engineering from MIT. Contact her at 2111 Etcheverry Hall, UC Berkeley, Berkeley, CA 94720; jbaker@berkeley.edu.

Eric Carleton is a PhD candidate in materials science at UC Berkeley. His research interests include ferroelectric materials, thin-film science, MEMS, and energy scavenging. He received his MS in ceramic engineering from the University of Missouri-Rolla. Contact him at 210 HMMB #1760, UC Berkeley, Berkeley, CA 94720-1760; carleton@berkeley.edu.

Elizabeth Reilly is a PhD student in mechanical engineering at UC Berkeley, where her research focuses on piezoelectric thin-film vibrational energy scavenging. She received her MS in mechanical engineering from UC Berkeley. Contact her at 2111 Etcheverry Hall, UC Berkeley, Berkeley, CA 94720; beth@kingkong.me.berkeley.edu.

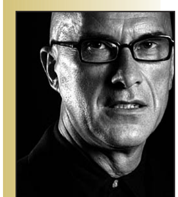
Elaine Lai is a graduate student in the Berkeley Manufacturing Institute at UC Berkeley. Her research interests include vibration-based energy scavenging and MEMS structures. She received her BS in mechanical engineering from MIT. Contact her at 1930 Hearst Ave., Apt. E, Berkeley, CA 94709; emlai@me.berkeley.edu.

Brian Otis is a graduate student in the Department of Electrical Engineering and Computer Science at UC Berkeley. His research interests include analog IC design, bioelectronics, RF circuit design, and IC-MEMS interfaces. He received his MS in electrical engineering and computer sciences at UC Berkeley. Contact him at 511 Cory Hall, UC Berkeley, Berkeley, CA 94720; botis@eecs.berkeley.edu.



Jan M. Rabaey is the Donald O. Pederson Distinguished Professor in the Department of Electrical Engineering and Computer Sciences at UC Berkeley. He's also the scientific codirector of the Berkeley Wireless Research Center and the director of the Gigascale Systems Research Center. His research interests include the conception and implementation of next-generation integrated wireless systems, including the analysis and optimization of communication algorithms and networking protocols, the study of low-energy implementation architectures and circuits, and the supporting design automation environments. He received his EE and PhD degrees in applied sciences from Katholieke Universiteit Leuven. Contact him at 511 Cory Hall, UC Berkeley, Berkeley, CA 94720; jan@eecs.berkeley.edu.

V. Sundararajan is a faculty member in the Department of Mechanical Engineering at the University of California, Riverside. His research interests include sensor networks, environmental monitoring, energy harvesting, computational geometry, collaborative processing, and manufacturing systems. He received his MS and PhD in mechanical engineering from UC Berkeley. Contact him at A317 Bourns Hall, Univ. of California, Riverside, Riverside, CA 92521; vsundar@engr.ucr.edu.



Paul K. Wright is associate dean of the College of Engineering, cochair of the Management of Technology Program, and codirector of the Berkeley Manufacturing Institute at UC Berkeley. He is also working with various UC Berkeley colleagues to design and prototype "batteryless" wireless systems for demand-response power management throughout California. He received his PhD in industrial metallurgy from the University of Birmingham, England. Contact him at 2117 Etcheverry Hall, UC Berkeley, Berkeley, CA 94720; pwright@kingkong.me.berkeley.edu.

frequency. Additionally, designs incorporating multiple proof masses can moderately increase scavenger bandwidth from around 6 Hz to perhaps 24 Hz.

- *Alternative design geometries*. We've considered several potential alternative design geometries. These designs focus on improving design robustness or a given input's power output.
- *Developing processes to allow MEMS (microelectromechanical systems) implementation and integration with sensors and electronics*. We have recently created prototypes of thin-film PZT structures. Initial calculations show that such structures can generate an areal power density of 5 $\mu W/cm^2$ and a volume power density of 80 $\mu W/cm^3$. Recently, we patterned and dry-etched beams using standard microfabrication processes. We are currently attempting selective wet-etching processes to release the cantilever beams from the Si wafer.

In addition, at least two other outstanding research areas exist for improving vibration-based energy scavenging: optimizing the power circuitry and improving microscale generators' PZT microfabrication and integration capabilities. ■

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