A Device for Harvesting Energy From Rotational Vibrations

The MIT Faculty has made this article openly available. Please share how this access benefits you. Your story matters.

| Citation               | Trimble, A Zachary et al. “A Device for Harvesting Energy From Rotational Vibrations.” Journal of Mechanical Design 132, 9 (2010): 091001 © 2010 by ASME |
|------------------------|--------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------|
| As Published           | http://dx.doi.org/10.1115/1.4002240                                                                                                                                                              |
| Publisher              | ASME International                                                                                                                                                                             |
| Version                | Final published version                                                                                                                                                                          |
| Citable link           | http://hdl.handle.net/1721.1/119837                                                                                                                                                              |
| Terms of Use           | Article is made available in accordance with the publisher’s policy and may be subject to US copyright law. Please refer to the publisher’s site for terms of use.                                       |
A Device for Harvesting Energy From Rotational Vibrations

A new device designed to harvest rotational vibration energy is presented. The device is modeled as a spring-mass-damper system connected to a vibration source where a torsion rod is used as a spring element and a shearing electromagnetic induction circuit as the energy harvesting element. The device is inherently a resonant type harvester. A prototype device is tested using a purely sinusoidal vibration input and more realistic inputs consisting of wider bandwidths, multiple resonance peaks, and low amplitude noise. The performance of the prototype to realistic inputs verifies the ongoing challenge to vibration energy harvesting, namely, significant loss of performance when using broadband inputs with resonant based devices. [DOI: 10.1115/1.4002240]

1 Introduction

A significant amount of vibration in many rotating mechanical systems seems to be a viable potential source of energy for sensors and other instruments. In general, harvesting or scavenging kinetic energy associated with mechanical vibrations involves some form of electrical transducer to transform mechanical energy to electrical energy. Common forms of transduction include active materials, such as piezoelectric; and various forms of electromagnetic transduction, such as induction [1].

According to Ref. [2], regardless of the transduction method, the majority of devices described in the literature rely on an inertial proof mass connected to the vibrating environment by combinations of springs and dampers. The majority of these devices are of microscale and are designed to accept linear inputs. However, as proposed by Ref. [3], since harvesting architectures based on a rotating proof mass have less restrictive displacement constraints, rotating architectures provide potential advantages over those based on linear systems. Additionally, as shown by Ref. [4] when using electromagnetic transduction as the method for energy harvesting, a shearing magnetic circuit is more magnetically efficient than an axial or plunging circuit. A rotating proof mass naturally provides a shearing air gap taking advantage of the higher efficiency shearing magnetic circuit.

Several wrist watches have been successfully powered (or wound) by rotating proof masses [5], but in those instances, the frequency of vibration is very low (0.5–2 Hz), and the amplitude of vibration is several orders of magnitude larger than the size of the device. Thus, the majority of the work is done by the relative change in the direction of the gravitational field associated with such large amplitude changes and not the relative motion produced by inertial forces on the proof mass. To the authors’ knowledge, what is not available is a macroscale device capable of harvesting the higher frequency (5–100 Hz) but much lower amplitude vibrations of rotating machinery in varying attitudes. As explained by Ref. [6], most proposed methods of transduction have similar maximum potential power densities, and thus the choice is often application specific. Combining these results with the estimates of Ref. [4], electromagnetic induction is chosen as the best method of transduction for this device based on expected mechanical vibrations and form factor restraints. Thus, presented here are the design and performance of a rotational, macroscale, energy harvester. The harvester uses a rotating spring-mass-damper that can be directly connected to a rotational system subjected to torsional vibration and, through electromagnetic induction, harvest the potential energy of those vibrations.

Often, initial estimates of the amount of energy that can be harvested from a vibration are based on an analytic solution of the system model to a single frequency harmonic input. However, Ref. [7] has shown that for actual accelerations, small additions of bandwidth or background noise greatly reduces the actual harvestable power. Thus, because of the broad frequency spectrum and the possibly random phase of the expected vibration, our design approach is to develop an analytical model of rotational electromagnetic energy harvesters that can receive the sample inputs and can predict the energy output from the machine. The theoretical development and explanation of the analytical model and design theory are presented first. The details of a prototype rotational harvester designed based on the theory are then presented, and test details are given for each element of the analytical model tested against the prototype design for model verification. Finally, both harmonic and example signals are simulated and tested against the model for verification.
2 Modeling

Conceptual operation of the device is relatively straightforward. Figure 1 is a lumped parameter model describing the operation. The system acts like a standard spring-mass-damper with base excitation. The device is built inside a casing that is attached to a vibrating source and thus acts as the vibrating reference frame. A spring element is provided by a torsion rod, an inertial “mass” element by a rotor core, and a damping element by a combination of unavoidable losses in the support bearings and the energy removed from the system represented in the model as an electromagnetic torque. The second order governing differential equation describing the relative motion of the proof mass in terms of the system parameters and the vibration input can be written as

$$\ddot{\phi} + \frac{b}{J} \dot{\phi} + \frac{1}{J} T_e + \frac{K}{J} \phi = -\ddot{a}$$

(1)

This system is well studied in textbooks on vibration, and ideally the frequency of vibration input is matched with the resonant frequency, thus amplifying the magnitude of the relative motion and providing exponential gains in power.

Since the energy is harvested in an electromagnetic transducer, the model consists of two main components: a mechanical model and an electromagnetic model. As the magnets move relative to a coil attached to the base, they create a varying magnetic flux field relative to the base that generates an emf voltage in the coil. If a load is attached to the emf voltage potential, energy is removed from the system through the load; as a result, the current flowing through the coil to power the load generates a reaction torque that acts to dampen the vibration. The electromagnetic model calculates the varying flux based on the elements in the magnetic circuit (magnets, air gap, and back iron). Based on the flux calculation, the expected voltage in a coil attached to the base is calculated. Assuming a resistive electrical load, the reaction torque is returned to the mechanical model as an additional damping element. Key model components affecting the power output are the mechanical quality factor (Q), natural frequency ($\omega_0$), rotational inertia ($J$), and electrical damping coefficient ($b_e$).

2.1 Magnetic Model. Electromagnetically, the device is similar to a standard rotary generator (Fig. 2). An outer casing acts as focusing back iron for a surface wound coil attached to the inside of the casing (or stator). Continuing concentrically inward, an air gap separates the coil from a set of magnets attached to a rotor that acts as focusing back iron for the magnets. Rare earth permanent magnets on the rotor provide a magnetic field in the air gap between the magnets and steel casing. As the rotor moves relative to the case, the magnetic field thorough the coils changes and generates an emf voltage. According to Faraday’s law of induction, the emf voltage can be written as

$$\text{emf} = \epsilon = \frac{d\lambda}{dt} = \frac{d\lambda}{d\phi} \frac{d\phi}{dt} = \frac{d\lambda}{d\phi} \dot{\phi}$$

(2)

Under some simplifying assumptions, the quasi-static versions of Maxwell’s equations can be used to solve for $\lambda$ analytically. First, ignoring fringing fields at each end of the rotor, the 2D solutions for Maxwell’s equations can be used. Second, comparing the magnetic permeabilities and relative dimensions, the reluctance of the focusing iron is sufficiently less than the reluctance in the air gap so that the iron can be assumed to be infinitely permeable and a perfect conductor, which simplifies the boundary conditions. Under these assumptions, the solution for the magnetic flux density is shown by Ref. [8] to be

$$B_\theta = \sum_{k=1}^{\infty} \frac{2B_R}{\pi(2k-1)} r_{0\theta} \left\{ r_{u\theta} \sin[p(2k-1)\beta] \hat{e}_r \right\}$$

(3)

Although Eq. (3) is an infinite summation as $k \to \infty$, the magnitude of the higher order harmonics tends to zero. As can be seen in Fig. 3, for the dimensions of the proposed prototype, the relative error between truncated estimates of the flux density amplitude evaluated at the casing surface quickly tends to zero. In fact, the relative error between an estimate using only the first term, $k=1$, and an estimate using the first and second terms, $k=1,2$, is less

![Fig. 1 Lumped parameter model of the harvester](image)

![Fig. 2 Cross-sectional view of the magnetic circuit (to scale)](image)

![Fig. 3 Plot of the calculated radial flux density, $B_\theta$, evaluated at $r_{oc}$. Although $B_\theta$ is an infinite summation, as can be seen, the influence of higher order terms is small and can be neglected.](image)
than 0.5%, and thus for dimensions of the proposed prototype the flux can be accurately estimated using only the first term, which greatly simplifies calculations. The magnetic flux density streamlines calculated using these simplifications are shown in Fig. 4. Assuming that all the current in the coils is concentrated on the surface of the casing, only the radial component of flux density in the air gap region evaluated at $r_{oc}$ is essential (Fig. 3). Thus, integrating the first term in the summation of the flux density at the casing surface over the area of the coils and multiplying by the number of turns in the coils results in an equation for the flux passing through the coils, $\lambda$:

$$
\lambda = \lambda_0 \sin(p(2k - 1)\phi)
$$

(4)

Substituting this result into Eq. (2) results in an equation for the emf,

$$
e = \varepsilon_0(2k - 1)\cos(p(2k - 1)\phi)
$$

(6)

The calculated emf can then be used to find an equivalent damping coefficient. Using an equivalent circuit model (Fig. 5) and assuming a resistive load, the current in the windings is $i = \text{emf}/(R_{\text{coil}} + R_{\text{load}})$. The electromagnetic torque is then given by

$$
T = i \times B
$$

(7)

Combining the previous results and integrating around the complete circle, the torque in the axial direction can be reduced to

$$
T_c = \frac{\lambda^2 \mu^2 R_{\text{load}}}{2(R_{\text{load}} + R_{\text{coil}})^2} \cos(p\phi)(\phi)
$$

(8)

from which the equivalent damping coefficient is

$$
b_e = \frac{\lambda^2 \mu^2}{R_{\text{load}} + R_{\text{coil}}} \cos^2(p\phi)
$$

(9)

Although $b_e$ is technically a function of the displacement, for the small relative displacements expected $\cos(p\phi) \approx 1$ and the damping coefficient can be approximated as

$$
b_e \approx \frac{\lambda^2 \mu^2}{R_{\text{load}} + R_{\text{coil}}}
$$

(10)

This damping coefficient can be tuned by power electronics designed to be capable of varying the load resistance. However, useful power is only harvested in the load resistance and is governed by the voltage divider equation,

$$
P_v = \left( \frac{R_{\text{load}}}{R_{\text{load}} + R_{\text{coil}}} \right) e = \frac{R_{\text{load}}^2}{(R_{\text{load}} + R_{\text{coil}})^2}
$$

(11)

The harvested power is at a maximum when $R_{\text{load}} = R_{\text{coil}}$. Thus, although the damping coefficient can be fine tuned by varying $R_{\text{load}}$, once the optimal damping coefficient is determined from the mechanics, the coils should be designed such that

$$
R_{\text{coil}} = \frac{\lambda^2 \mu^2}{2b_e}
$$

(12)

2.2 Mechanical Model. With the electromagnetic torque reduced to an equivalent damping coefficient, $b_e$, Eq. (1) describing the relative motion of the proof mass can be reduced to

$$
\frac{\dot{\phi} + b_e}{J} = \frac{b}{f} \dot{\phi} + \frac{K}{f} \phi = -\ddot{\phi}
$$

(13)

In this reduced form, the governing equation is the well studied base excitation equation. Remembering the power harvested is proportional to the relative velocity between the coils and the magnets, we see that to maximize the power harvested we must maximize the relative velocity. Using the solutions of Ref. [9], the relative velocity driven by a random input will be maximized by maximizing the product of the magnitude of transfer function and the power spectral density function of the input,

$$
E[\phi^2] = \int_{-\infty}^{\infty} |H_{\phi \phi}(\omega)|^2 S_f(\omega) d\omega
$$

(14)

For the second order model proposed and expected representative spectral densities, this maximum occurs when the resonant frequency of the system matches the frequency at the maximum amplitude of the power spectral density (PSD). Thus, the natural frequency $\sqrt{K/m}$ should coincide with the maximum amplitude of the Fourier decomposition of the input signal [10].

For a harmonic input, if $\omega_0 = \omega$, then the power becomes directly proportional to the proof mass. For a random signal, an analytical solution is not possible, but assuming superposition, this solution would still apply for the single term at the resonant frequency and would be approximately true for all terms near the resonant frequency. Thus, the power is approximately linearly proportional to the proof mass. Combining these two criteria, the proof mass should be maximized for the given volume, and then the spring constant $K$ should be chosen to provide $\omega_0 = \omega_{\phi\phi\text{max}}$.

The damping coefficient is then used to tune the match between the transfer function and the PSD. Since only the electrical damping $b_e$ is harvestable and the remaining damping is attributed to losses, the internal losses should be reduced as much as possible. The electrical damping can then be chosen to maximize the power. For some inputs, such as harmonic, the solution can be determined analytically and results in $b_e = b_h$; however, in most cases, $b_e$ should be chosen numerically by the simulation of the expected input.
3 Prototype Design and Testing

The theory was used to create a design code that enabled an experimental prototype to be designed, built, and tested. The device was required to fit into a defined cylindrical form factor of 1.25 in. (31.8 mm) diameter and 5 in. (127 mm) length. The device is comprised of a 1.25 in. (31.8 mm) diameter, 0.1 in. (2.5 mm) thick steel cylinder casing, which provides the structural ground/attachment for the device as well as acts as the stator back iron. An electrical coil wound from a 32AWG magnet wire is attached to the inner surface of the steel. A 0.9 in. (22.9 mm) outer diameter rotor assembly comprised of a steel core with neodymium-boron permanent magnets attached around the outside is located in the center of the device by ball bearings at each end. At one end of the rotor assembly, a Nitinol piano wire is rigidly affixed to the steel core and passes axially through the hollow center of the core where it is rigidly affixed to the casing and acts as the torsion spring (Fig. 6).

Although in operation the electrical and mechanical models are coupled, the prototype is designed so the electromagnetic model can be tested separately from the mechanical model. Thus, each of the model parameters can be verified independently. Starting with the electromagnetic model, the model is verified by comparing the expected emf to the measured output from the device under a constant rotational velocity input (the torsion spring is removed so the rotor can spin freely with respect to the stator). As can be seen in Fig. 7, when operating the harvester like a generator, the predicted voltage output is nearly identical to the measured voltage, suggesting that the proposed electromagnetic model is correct.

Turning now to the mechanical model, the model is verified piecewise. The moment of inertia of the rotor is determined from the solid model and is verified by comparing the predicted mass and dimensions of the solid model to the measured mass and dimensions of the device. The natural frequency of the system is verified by comparing the maximum amplitude of the open circuit voltage to a swept single frequency sinusoidal input. The mechanical Q of the system is experimentally determined by comparing the calculated open circuit voltage to the measured voltage for a single frequency sinusoidal input. The open circuit voltage is determined by solving the mechanical model (Eq. (13)) for the relative motion φ (where in open circuit b_3=0) and then using the calculated φ in the electrical model (Eq. (6)) to determine the voltage. Figure 8 is a plot of the voltage as a function of time for a single frequency harmonic input, and Fig. 9 is a plot of the voltage amplitude as a function of frequency for a single frequency harmonic input slowly swept through the frequency spectrum. The agreement of the experimental and calculated values in Figs. 8 and 9 suggests that the assumptions made to create the model are appropriate and represent the form of the equation and additionally suggests that the physical parameters have been properly calculated.

Combining all the components for a single frequency input of the same typical frequency and amplitude as the Fourier decomposition of an example realistic acceleration input (Fig. 10), the
model predicts that 220 mW can be harvested, and experimental
tests verify a harvested power of about 205 mW (Fig. 11). As
mentioned previously, a primary challenge to resonant based en-
ergy harvesters in application is the wide, noisy nature of realistic
vibrations. As an example, Ref. [11] presented a linear vibration
spectrum that was excited by a rotating system. The spectrum
presented shows multiple frequency peaks in close proximity,
which will cause interference with resonance. Figure 10 is a plot
of an example vibration spectrum. Given this input, the theoretical
model predicts 85 mW average power during the 5 s input, and
experimental tests verify an average harvested power of 74 mW.
Figure 12 is a plot of the voltage as a function of time with the
acceleration input from Fig. 10 and loaded by a 10 Ω resistor. As
can be seen, the model accurately predicts the voltage across the
load (and thus the power) from the wide-band signal. Notice that
although the maximum acceleration amplitude of the vibration
input is the same as the single frequency test, the harvested power
is reduced by roughly a factor of 2. This illustrates one of the
ongoing problems of resonant based harvesters. Possible solutions
to this problem in the literature include mult frequen- cro frequency inputs such as those presented in Ref. [12] or tunable harvesters such as
those presented in Refs. [13,14]; however, these approaches only
work if the input consists of highly separated distinct frequency
peaks. For noisy or closely spaced frequency spectra, a viable
solution is still needed.

4 Conclusion
As outlined, the aim of this work was twofold. First, due to the
random nature of most “real-world” vibrations, the design ap-
proach was to create and verify a detailed model that can be used
to accurately predict the power output of torsional vibration har-
esters when subjected to random inputs. For a single frequency
harmonic input, Ref. [21] shows that the maximum power that can
be harvested is

\[ P_{\text{max}} = \frac{J_{\text{tot}}^2 Q}{16 \omega_0} \]  

which for the prototype harvester with an assumed quality factor
of 60 would predict a maximum power output of 245 mW. How-
ever, when the prototype and the model are subjected to a wide-
band real-world input, the power falls off by nearly an order of
magnitude. However, the more detailed model accurately predicts
the reduced power. Based on the accuracy of the models when
compared with the prototype design, the design tools can be used
with confidence for evaluating the potential power that can be
harvested from real world signals. Additionally, with some
changes to the electromagnetic model, the basic structure can be
easily adapted to linear systems for similar evaluations.

Second, a macrosized harvester placed in the center of a rotat-
ing system designed to directly harvest torsional vibrations is a
viable method to harvest energy from rotating systems. Although
a direct comparison of a device attached to the exterior of the
rotating system and harvesting the near linear acceleration inputs
of a rotationally oscillating system at some distance from the cen-
terline with a device similar to the one presented would be ideal,
this comparison is not currently available. As a substitute, Eq. (15)
shows that normalizing the power by the inertia, acceleration am-
pitude squared, and natural frequency provides a reasonable
power density for comparison. Note that for harmonic inputs, this
is the same as normalizing by the mass, velocity, and acceleration,
which come directly from the definition of power. However, since
volume is more commonly reported and mass and volume can be
assumed to be proportional, an approximate normalization using
the device volume is used. Table 1 shows the comparison of
power densities. As can be seen, the torsional device has compar-
able power densities, suggesting that this approach is viable.

Nomenclature

- $\mathbf{B}$ = magnetic flux density
- $\mathbf{B}_a$ = magnetic flux density in air gap region
- $B_R$ = remanence of permanent magnets
- $|H_{a \rightarrow \mathbf{0}}|$ = magnitude of the complex frequency response
- $J$ = mass moment of inertia of proof mass
- $K$ = torsion spring constant
- $N$ = number of turns in the coil
- $Q$ = mechanical quality factor
- $R_{\text{coil}}$ = electrical resistance in coil
- $R_{\text{load}}$ = electrical resistance of load
- $S_a$ = power spectral density of the input acceleration
- $T$ = torque
- $T_e$ = electrical transducer torque
- $b_i$ = internal damping coefficient
- $h$ = axial dimension of the coils and magnets

Figure 10 Example real-world signal the device is expected to
produce energy from. Shown are the time and frequency de-
compositions of the desired signal and the actual signal. The
input vibration was provided to the device without feedback
control; however, as can be seen, the actual input signal is still
representative of a wide-band vibration as commonly
encountered.

Figure 11 Plot of the power versus frequency for a single
frequency input at approximately the same amplitude as the ex-
ample realistic signal shown in Fig. 10.

Figure 12 Plot of the voltage across a load resistor as a function
of time for the real-world input shown in Fig. 10.
Table 1 Comparison of prototype device to other devices in the literature (note: the presented device is to the authors’ knowledge the only rotational harvester and is thus compared against a selection of linear harvesters)

| Device                   | Volume (cm³) | Input acceleration amplitude (g) | Input frequency (Hz) | Power (mW) | Power density (mW Hz/cm³/(rad/s²)) |
|--------------------------|--------------|---------------------------------|----------------------|------------|----------------------------------|
| Presented prototype      | 80           | 150 (rad/s²)                    | 16                   | 205        | 9 (mW Hz/cm³/(rad/s²))           |
| Perpetuum [15]           | 130          | 0.14                            | 100                  | 3.5        | 1.37                             |
| Perpetuum [15]           | 130          | 1.4                             | 100                  | 40         | 0.15                             |
| Ferro solutions [16]     | 75           | 1.0                             | 21                   | 9.3        | 2.60                             |
| MIT [17]                 | 23.5         | 0.3                             | 2                    | 0.4        | 0.0037                           |
| Southampton [18]         | 0.24         | 10                              | 322                  | 0.83       | 0.11                             |
| Southampton [19]         | 0.84         | 5.4                             | 322                  | 0.037      | 0.0048                           |
| Volture [20]             | 40           | 0.7                             | 50                   | 12         | 0.30                             |

\[ \ddot{\mathbf{i}} = \text{coil current} \]
\[ p = \text{number of magnetic pole pairs} \]
\[ r_{a_0} = \text{nondimensional magnetic scaling factor (common)} \]
\[ r_{a_1} = \text{nondimensional magnetic scaling factor (radial direction)} \]
\[ r_{a_2} = \text{nondimensional magnetic scaling factor (theta direction)} \]
\[ r_{c_i} = \text{radial dimension to the inner surface of magnets} \]
\[ r_{c_o} = \text{radial dimension to the outer surface of magnets} \]
\[ \bar{\alpha} = \text{reference frame acceleration} \]
\[ \phi_{0_i} = \text{amplitude of the acceleration input if } \bar{\alpha} \text{ is harmonic} \]
\[ e = \text{emf voltage} \]
\[ \lambda = \text{magnetic flux} \]
\[ \lambda_0 = \text{amplitude of magnetic flux} \]
\[ \phi = \text{relative rotation} \]
\[ \dot{\phi} = \text{relative velocity} \]
\[ \ddot{\phi} = \text{relative acceleration} \]

References

[1] Arnold, D., 2007, “Review of Microscale Magnetic Power Generation,” IEEE Trans. Magn., 43(11), pp. 3940–3951.
[2] Mitcheson, P., Green, T., Yeatman, E., and Holmes, A., 2004, “Architectures for Vibration-Driven Micropower Generators,” J. Microelectromech. Syst., 13(3), pp. 429–440.
[3] Yeatman, E., 2008, “Energy Harvesting From Motion Using Rotating and Gyroscopic Proof Masses,” Proc. Inst. Mech. Eng., Part C: J. Mech. Eng. Sci., 222(1), pp. 27–36.
[4] Jonnalagadda, A., 2007, “Magnetic Induction Systems to Harvest Energy From Mechanical Vibrations,” MS thesis, Massachusetts Institute of Technology, Cambridge, MA.
[5] Chapuis, A., and Jaquet, E., 1956, The History of the Self-Winding Watch, Roto-Sadag SA, Geneva.
[6] Roundy, S., 2005, “On the Effectiveness of Vibration-Based Energy Harvesting,” J. Intell. Mater. Syst. Struct., 16(10), pp. 809–823.
[7] Halvorsen, E., 2008, “Energy Harvesters Driven by Broadband Random Vibrations,” J. Microelectromech. Syst., 17(5), pp. 1061–1071.
[8] Haas, H., and Melcher, J., 1989, Electromagnetic Fields and Energy, Prentice-Hall, Englewood Cliffs, NJ.
[9] Innman, D. J., 2001, Engineering Vibrations, 2nd ed., Prentice-Hall, Englewood Cliffs, NJ.
[10] Newland, D., 1984, An Introduction to Random Vibrations and Spectral Analysis, Longman, New York.
[11] Johnson, T., Charnegie, D., Clark, W., Buric, M., and Kusic, G., 2006, “Energy Harvesting From Mechanical Vibrations Using Piezoelectric Cantilever Beams,” Proc. SPIE, 6169, p. 61690D.
[12] Yang, B., Lee, C., Xiang, W., Xie, J., He, J., Kotlanka, R., Low, S., and Feng, H., 2009, “Electromagnetic Energy Harvesting From Vibrations of Multiple Frequencies,” J. Micromech. Microeng., 19, p. 035001.
[13] Challa, V. R., Prasad, M. G., and Fisher, F. T., 2009, “Towards a Self-Tunable, Wide Frequency Bandwidth Vibration Energy Harvesting Device,” ASME 2009 International Mechanical Engineering Congress and Exposition, pp. 57–65.
[14] Challa, V. R., Prasad, M. G., Shi, Y., and Fisher, F., 2007, “A Wide Frequency Range Tunable Vibration Energy Harvesting Device Using Magnetically Induced Stiffness,” ASME 2007 International Mechanical Engineering Congress and Exposition, pp. 387–395.
[15] Perpetuum, energy harvesting microgenerator product data sheet.
[16] Solutions, F., 2009, http://www.ferrosi.com/energy-harvesters.html
[17] Amirtharajah, R., and Chandrakasan, A. 1998, “Self-Powered Signal Processing Using Vibration-Based Power Generation,” IEEE J. Solid-State Circuits, 33(5), pp. 687–695.
[18] El-hamri, M., Glynn-Jones, P., White, N., Hill, M., Beeby, S., James, E., Brown, A., and Ross, J. N., 2001, “Design and Fabrication of a New Vibration-Based Electromechanical Power Generator,” Sens. Actuators, A, 92, pp. 335–342.
[19] Glynn-Jones, P., Tudor, M. J., Beeby, S. P., and White, N. M., 2004, “An Electromagnetic, Vibration-Powered Generator for Intelligent Sensor Systems,” Sens. Actuators, A, 110(1–3), pp. 344–349.
[20] Volture Technologies, 2010, http://www.mide.com/pdfs/voltage_specs_piezo_properties.pdf
[21] Trimble, A. Z., 2007, “Downhole Vibration Sensing by Vibration Energy Harvesting,” MS thesis, Massachusetts Institute of Technology, Cambridge, MA.