Applied Physics Express

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Piezoelectric Energy Harvesting Device in a Viscous Fluid for High Amplitude Vibration Application

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Received July 24, 2008; accepted August 23, 2008; published online September 12, 2008

The fragile nature of ceramic piezoelectric cantilevers limits their ability to withstand high acceleration amplitudes in vibration energy harvesting applications. This study reports a potential solution, which is vibrating the fragile power generator in a viscous fluid. The changes in resonant frequency, Q-factor, damping ratio, and power output of a bimorph piezoelectric cantilever prototype, as well as the relationship of power output vs the vibration acceleration were investigated. The broadened application frequency spectrum and the increased maximum operational acceleration demonstrate the potential of the bimorph piezoelectric cantilever operating in a viscous fluid for high amplitude vibration applications.

DOI: 10.1143/APEX.1.098002

The concept of energy harvesting has been rapidly developing due to the increasing demands upon mobile devices, wireless sensor networks and micro-electromechanical systems (MEMS). Piezoelectric cantilevers utilized as vibration energy harvesters have been intensively studied because of their simple configuration, high conversion efficiency, and the ability for precise control of their mechanical response.[1,2] The attachment of a large proof mass to the free end tip of the cantilever has been used not only to lower the resonant frequency of a cantilever to match the exciting frequency (60–200 Hz) of many low amplitude vibration sources,[3] but also to generate a large strain or power output. However, the extremely large stresses at the clamping area that result from sudden impacts or higher vibration amplitudes such as in mobile vehicles, aircraft, and machining tools make the structures vulnerable. This is one of the reasons why few people study ceramic piezoelectric cantilevers for both low frequency and high amplitude vibration applications. A dense medium, however, can protect the fragile structure from fracture by adding damping, although the dynamic behaviors of the cantilever will be strongly influenced by the fluid physical properties.

In this study, the vibration behaviors of a piezoelectric cantilever fabricated using a bimorph bender (T215-H4-103Y, Piezo Systems, Cambridge, MA) and a tungsten proof mass in both air and Si oil environments were investigated. The schematic and notations of the device can be found in published article.[4)] Figure 1 presents the picture of the device and the schematic diagram of the device submerged in Si oil. The dimensions of the cantilever beam and the proof mass are respectively $l \times w_p \times (2t_p + t_s) = 25.2 \times 3.2 \times (2 \times 0.134 + 0.123)\,\mathrm{mm}^3$, where l and w_p are the total length and the width of the cantilever beam, t_p and t_s respectively the thicknesses of the piezoelectric material and the center shim brass, and $l_m \times w_m \times h_m = 2.73 \times 2.80 \times 2.70\,\mathrm{mm}^3$, where l_m, w_m, and h_m the length, width, and height of the proof mass.

The resonant frequency, one of the key parameters, of a bimorph cantilever generator with a nonpoint proof mass attached to the free end tip can be estimated by[4,5]

$$f_n = \frac{\nu_n^2}{2\pi}\sqrt{\frac{0.236 E_0 w_p}{(l - l_m/2)^3 (m_e + \Delta m)}}, \quad (1)$$

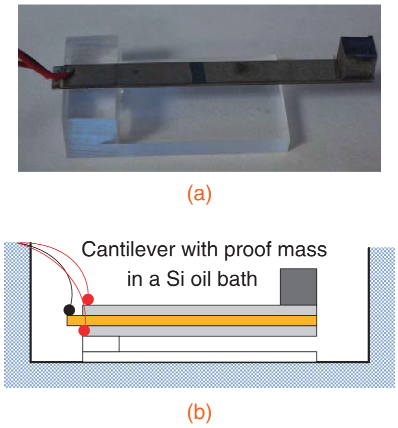

Fig. 1. (a) Picture of the device and (b) schematic diagram of the device submerged in Si oil (not to scale).

where f_n is the nth mode resonant frequency, ν_n the nth mode eigenvalue ($\nu_1 = 1.875$), E_0 a function of the Young's moduli of the two materials, E_p (PZT) and E_s (brass), and their thicknesses, $E_0 = 2E_p t_p^3/3 + E_p t_s t_p^2 + E_p t_s^2 t_p/2 + t_s^3 E_s/12$, m_e the effective mass of the cantilever beam at the middle of the proof mass, $m_e = 0.236 m w_p (l - l_m/2) + m w_p l_m/2$,[6] m the mass per unit area of the bimorph bender given by $m = 2\rho_p t_p + \rho_s t_s$, ρ_p and ρ_s respectively the densities of the piezoelectric material and the center shim brass, and Δm the mass of the proof mass.

When the cantilever is submerged in a viscous fluid, the fluid will resist its vibration due to inertial and damping forces distributed over the surface of the cantilever.[7–9] The inertial force is proportional to the acceleration of cantilever vibration with the proportionality constant, m_a, the added mass due to the fluid. The damping force is proportional to the vibration velocity. The added mass mainly serves to decrease the resonant frequency, while viscous damping contributes to the Q-factor drop.

The resonant frequency of the cantilever is therefore decreased by the added mass, and can be calculated by rewriting eq. (1) as[10]

$$f_{nf} = \frac{\nu_n^2}{2\pi}\sqrt{\frac{0.236 E_0 w}{(l - l_m/2)^3 (m_e + m_{ae} + \Delta m)}}, \quad (2)$$

where m_{ae} is the effective added mass by the viscous fluid at the middle of the proof mass, and can be calculated from

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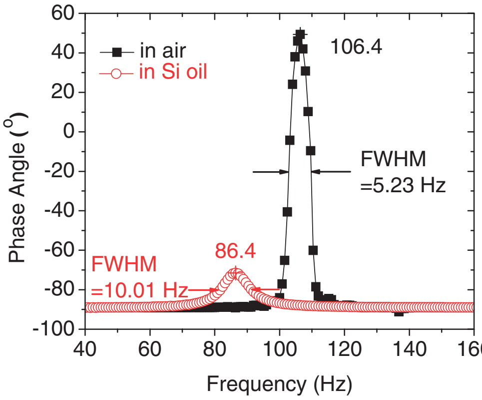

Fig. 2. Measured resonant frequency of cantilever in air and in Si oil.

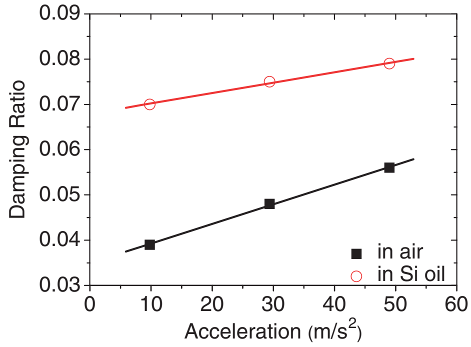

Fig. 3. Damping ratio in air and Si oil vs the exciting vibration acceleration.

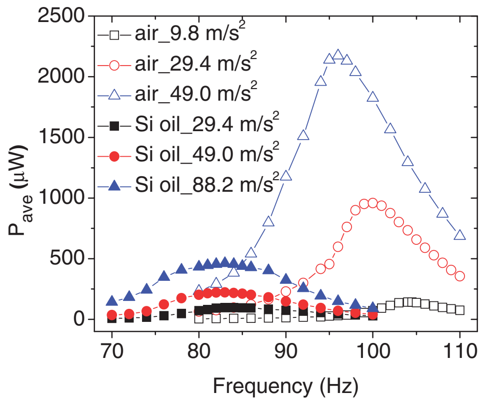

Fig. 4. Power output in air and Si oil vs the exciting vibration frequency.

the added mass, $m_{ae} = 0.236 m_a (l - l_m/2)/l + m_a l_m/(2l)$. Based on strip theory, the added mass can be roughly estimated by the density of the fluid and the geometry of the cantilever, $m_a = (\pi/4)\rho w_p^2 l$ when a slender (large aspect ratio) cantilever is submerged into a theoretically infinite fluid bath.[11]

The resonance spectra of the cantilever in air and in Si oil were measured by an impedance analyzer (Agilent 4294A), and are presented in Fig. 2. The peak value of the phase angle decreased from 106.4 Hz in air to 86.4 Hz in Si oil. In air, the theoretical value calculated from eq. (1) is 108.5 Hz resulting in a difference of 1.9%. In Si oil however, a difference of 14.8% was observed with the resonant frequency calculated from eq. (2), $f_{fluid} = 101.4$ Hz. The large discrepancy in Si oil is mainly attributed to the heavy proof mass, which appears to have weakened the effect of the inertial loading caused by the added fluid mass upon the resonant frequency of the device since estimates given by eq. (2) work well for a cantilever without a proof mass submerged in a fluid.

An alternate approach to estimating the resonant frequency in liquid is based upon[11]

$$f_{fluid} = \frac{1}{\left(1 + \dfrac{m_a}{m_b}\right)^{1/2}} f_{vacuum}, \qquad (3)$$

where m_b is the mass of a cantilever beam without a proof mass, f_{vacuum} the resonant frequency of the cantilever in vacuum, and here we substitute the resonant frequency in air, f_{air}, for f_{vacuum} because of negligible difference.

Initially neglecting the proof mass and only considering the loading effect of the fluid on the cantilever beam, we obtain from eq. (3) $f_{fluid} = 0.7538 \times f_{air} = 0.7538 \times 108.5 = 81.8$ Hz, and the error has been decreased to 5.6%. By considering the buoyancy effects on the whole cantilever structure, m_{ae} is reduced from 0.2559 to 0.2256 g, Δm is reduced from 0.3983 to 0.3785 g, and the f'_{air} is increased to 111.94 Hz. We, therefore, obtained $f_{fluid} = 0.7538 \times 111.94 = 84.4$ Hz. The difference from experimental value has been reduced to 2.4%.

The Q-factor calculated from Fig. 1 dropped from 20.30 to 8.65 when submerged in Si oil. The decreasing in the amplitude and the broadening in the full wave at half maximum (FWHM) are due to the added viscous damping by the fluid.[7] Broad bandwidth is favorable in energy harvesting application because a small deviation in the exciting frequency from the resonant frequency of the device will not cause such a large decrease in power output.

The damping ratio for different accelerations and different mediums were measured and calculated with the results presented in Fig. 3. Details of the measurement and calculation techniques can be found in published article.[5] The values in Si oil are much larger than those in air because of increased damping by the viscous medium. In both mediums, the damping ratio increases with acceleration due to the nonlinearity of PZT under large strain.[12] However, the slope in air is larger than that in Si oil because the displacement of the cantilever tip and the strain generated in the PZT beam are increasing faster with acceleration in air.

Figure 4 presents the optimal power output, the maximum power obtained by changing the resistive load, in air and in Si oil at different acceleration amplitudes versus vibration frequency. A summary of the comparison between a cantilever operating in air and Si oil is given in Table I. Data in air at accelerations over 68.6 m/s^2 were not available because the device experiences a dramatic power decrease due to crack formations in the ceramic PZT in the device. By utilizing Si oil as a damping medium, the applicable acceleration in Si oil becomes about 10 times higher than that in air, and the applicable frequency bandwidth in Si oil is over twice as wide as that in air when the power output levels are similar.

Table I. Comparison between different accelerations and mediums.

		Acceleration (m/s^2)					
		0	9.8	29.4	49.0	68.6	88.2
FWHM (Hz)	in air	5.23	8.83	10.62	15.27	—	—
	in Si oil	10.01	11.01	18.01	21.80	24.57	27.46
f_r (Hz)	in air	106.4	104.0	100.0	96.0	—	—
	in Si oil	86.4	85.0	83.5	83.0	83.0	83.0
P_{max} (μW)	in air	—	143	958	2174	—	—
	in Si oil	—	20	97	222	352	462

In conclusion, a piezoelectric energy harvester operating in a viscous fluid for high amplitude vibration applications was proposed. For this application condition, two advantages are presented: larger operational bandwidth and higher operational vibration acceleration limit. The resonant frequencies of a bimorph cantilever prototype with a nonpoint proof mass were modeled both in air and in fluid. Experimental results showed that the resonant frequency in air matched well with the model, but in Si oil the measured resonant frequency was lower than the predicted one based on the first model because of the underestimated liquid loading effects due to the large proof mass, however the second method based on experiential method matched the experimental results well.

Acknowledgment This work is sponsored by research grants from Industry, National Science Foundation-Division of Materials Research-0605270, and United States Department of Agriculture-Cooperative State Research, Education, and Extension Service-2006-34394-16953.

1) Z. Wang and Y. Xu: Appl. Phys. Lett. **90** (2007) 263512.
2) S. P. Beeby, M. J. Tudor, and N. M. White: Meas. Sci. Technol. **17** (2006) 175.
3) S. Roundy, P. K. Wright, and J. Rabaey: Comput. Commun. **26** (2003) 1131.
4) D. Shen, S.-Y. Choe, and D.-J. Kim: Jpn. J. Appl. Phys. **46** (2007) 6755.
5) D. Shen, J.-H. Park, J. Ajitsaria, S.-Y. Choe, H. C. Wikle, and D.-J. Kim: J. Micromech. Microeng. **18** (2008) 055017.
6) W. Y. Shih, X. Li, H. Gu, W.-H. Shih, and I. A. Aksay: J. Appl. Phys. **89** (2001) 1497.
7) C. Vancura, J. Lichtenberg, A. Hierlemann, and F. Josse: Appl. Phys. Lett. **87** (2005) 162510.
8) T. Naik, E. K. Longmire, and S. C. Mantell: Sens. Actuators A **102** (2003) 240.
9) S. Kirstein, M. Mertesdorf, and M. Schonhoff: J. Appl. Phys. **84** (1998) 1782.
10) G. A. Campbell and R. Mutharasan: Biosens. Bioelectron. **21** (2005) 462.
11) R. D. Blevins: *Formulas for Natural Frequency and Mode Shape* (Krieger, Malabar, Florida, 2001) p. 413.
12) L. Q. Yao, J. G. Zhang, L. Lu, and M. O. Lai: J. Microelectromech. Syst. **13** (2004) 645.

