

VIBRATION ENERGY HARVESTING DEVICE BASED ON ASYMMETRIC AIR-SPACED CANTILEVERS FOR TIRE PRESSURE MONITORING SYSTEM

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Abstract: This paper reports a vibration energy harvesting device for tire pressure monitoring application based on a novel asymmetric air-spaced piezoelectric cantilever. This air-spaced cantilever structure leads to several highly desirable advantages. First, the voltage generated is increased due to the much larger distance between the piezoelectric layer and the neutral plane, and the increased voltage promises higher AC to DC conversion efficiency. Second, the overall energy conversion efficiency is further increased by enabling the majority of the mechanical energy for electricity generation. Last, the asymmetric structure makes the device more robust since the piezoelectric layer is operating in the compression mode. An analytical model was developed and the design optimization was discussed. A new prototype device was built and tested both in the lab and on the road. The experiment results show that the device has a great performance as we expected and the maximum power generated is 47 μ W with a 21.6 gram proof mass and a driving speed of \sim 50 mph.

Keywords: vibration energy harvesting, tire pressure monitoring system, air-spaced cantilever

INTRODUCTION

This work is pursued in response to a new federal rule (Federal Moving Vehicle Safety Standard 138) mandating that all vehicles sold in the U.S. after Jan. 1, 2008 will be outfitted with a tire pressure monitoring system (TPMS). Current TPMS utilizes lithium batteries as the power supply. Based on estimated usage, these batteries are expected to last as few as 6 years. The customer could lose the use of a federally mandated safety feature and would be looking at repair costs of as much as \$1000. These costs could severely impact automakers through possible warranty claims and litigation due to injury from battery failure. In addition, the environmental impact of roughly 80 million more lithium batteries being produced and disposed of each year is immense. Therefore, automotive companies are actively searching for a battery-less solution for TPMS devices. Among many battery-less solutions, piezoelectric vibration energy harvesting device is believed to be a promising technology [1].

However, for TPMS applications it is very challenging to simultaneously meet the power and reliability requirements mainly due to the large centripetal force. In this paper, we proposed an energy harvesting device based on novel piezoelectric asymmetric air-spaced cantilevers. With this structure, the amplitude of the AC voltage generated is increased, leading to a larger AC to DC energy conversion efficiency. The overall energy conversion efficiency is further increased by enabling the majority of the mechanical energy for electricity

generation. Moreover, the reliability is also improved by making the piezoelectric layer operate in compression mode.

ANALYTICAL MODEL

Fig. 1 shows the schematic of an asymmetric air-spaced cantilever with a bottom mechanical beam ($w_1 \times t_1 \times l$), a top piezoelectric beam ($w_2 \times t_2 \times l$), and a proof mass ($w_{pm} \times t_{pm} \times l_{pm}$). The distances from neutral plane to bottom and top beams are d_1 and d_2 , and the distance between top and bottom layer is D . E_1 , E_2 , I_1 , I_2 , A_1 and A_2 are Young's moduli, moments of inertia and section areas of bottom and top beams, respectively.

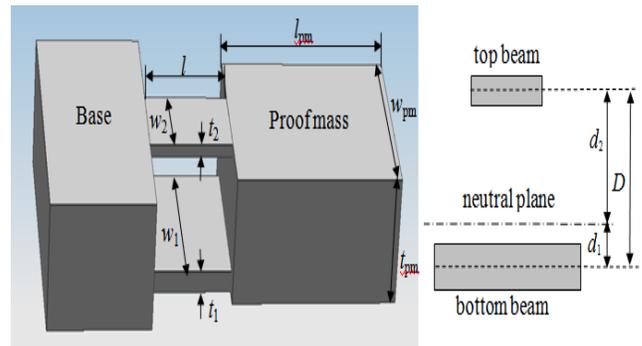


Fig. 1: Schematic structure of vibration energy harvesting device based on asymmetric air-spaced cantilever.

When acceleration a is applied in vertical direction, the proof mass will displace accordingly

and the beams will be subject to deformation. Unlike a conventional cantilever, the deformation of the air-spaced cantilever should be considered in both pure bending and S-shape modes respectively. The bending rigidities for the two modes R_p and R_s are [2]:

$$R_p = E_1(I_1 + A_1d_1^2) + E_2(I_2 + A_2d_2^2) \quad (1)$$

$$R_s = E_1I_1 + E_2I_2 \quad (2)$$

Then the spring constants of the two modes can be derived separately:

$$k_p = \frac{4R_s}{l^3} \frac{1}{\alpha^2\beta} \quad (3)$$

$$k_s = \frac{12R_s}{l^3} \quad (4)$$

where $\alpha = (l+l_{pm})/l$ and $\beta = R_s/R_p$.

Then, the total effective spring constant can be calculated by combining these two spring constants as:

$$k_E = \left(\frac{1}{k_p} + \frac{1}{k_s}\right)^{-1} = \frac{12R_s}{l^3} \frac{1}{3\alpha^2\beta + 1} \quad (5)$$

Based on Rayleigh-Ritz method, the resonant frequency is derived as:

$$f_0 = \frac{1}{2\pi} \sqrt{\frac{R_s}{ml^3} \frac{12(3\alpha^2\beta + 1)}{(3\alpha^2\beta + 1)^2 + 3\alpha^2\beta^2(\alpha - 1)^2}} \quad (6)$$

The normal strain experienced by the top piezoelectric beam is:

$$\varepsilon_2 = \frac{ma(l+l_{pm})d_2}{2R_p} \quad (7)$$

It can be observed that the normal strain of the top piezoelectric beam is proportional to d_2 which is approximately equal to the gap between the top and bottom beams for asymmetric air-spaced cantilevers. Therefore, the AC voltage generated will be significantly higher, increasing the AC to DC conversion efficiency. Also note that the average normal strain of the piezoelectric beam is only affected by the pure bending since strains due to S-shape bending are cancelled across the thickness.

Therefore, it is necessary to make pure bending mode dominant to increase energy conversion efficiency.

DESIGN OPTIMIZATION

Noting that only the energy stored in the top sensing layer by pure-stretching/compression is used to generate output voltage, here we define the mechanical energy conversion efficiency η as the ratio of the energy stored by pure-stretching/compression of the top sensing layer to the total mechanical energy, and that is [2]:

$$\eta = \frac{(1-\beta)(1-\gamma)}{1/3\alpha^2\beta + 1} \quad (8)$$

where $\gamma = d_1/D$.

It has been demonstrated that there exists an optimal γ to maximize the mechanical energy conversion efficiency when t_l and D are fixed.

$$\gamma_0 = \frac{1}{1 + \sqrt{1 + \frac{1}{C} + \frac{1+C}{C^2(3\alpha^2 + 1)}}} \quad (9)$$

where $C = t_l^2/12D^2$.

Once the optimal γ_0 has been decided, we can easily find other parameters and finalize the design. The plot of efficiency η versus γ with different C is presented in Fig. 2. As we can see in the figure, the mechanical energy conversion efficiency can reach more than 70% once optimized. It is a significant improvement compared with conventional unimorph cantilever which has less than 37.5% of mechanical energy conversion efficiency.

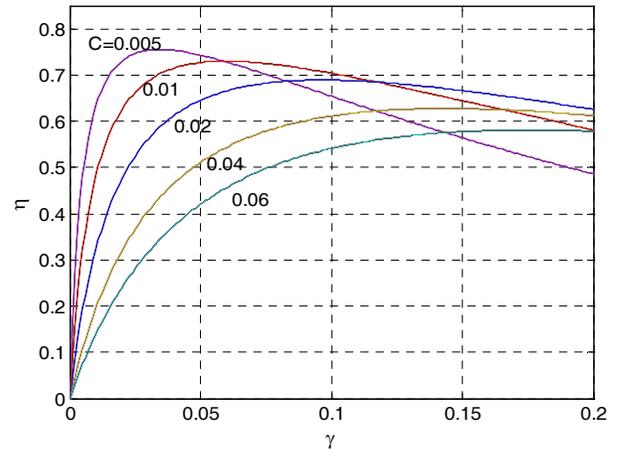


Fig. 2: Plot of efficiency η as a function of γ with different C ($\alpha=4.75$).

EXPERIMENT

A prototype vibration energy harvesting device has been developed using PZT sheet (T120-A4E-602) purchased from Piezo System, Inc., Cambridge, MA, USA. The resulting vibration energy harvesting device with bottom mechanical layer (15 mm×0.9 mm×8 mm), top PZT layer (5 mm×0.502 mm×8mm), and a proof mass (15 mm×6 mm×30mm) is shown in Fig. 3. The device was screw-mounted on a mechanical shaker, which is used to generate mechanical vibrations.

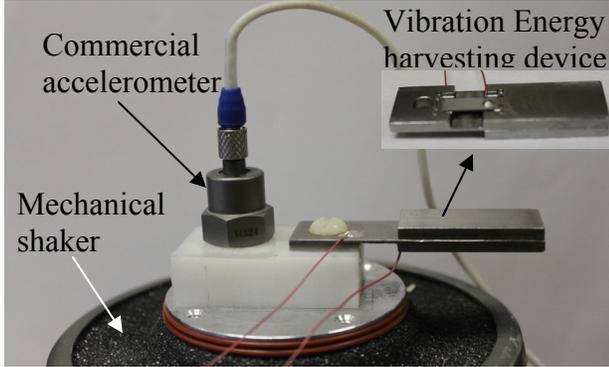


Fig. 3: Picture of a prototype vibration energy harvesting device up-side-down mounted on a mechanical shaker.

When the mechanical shaker vibrates in vertical direction, the air-spaced cantilever also vibrates generating a sinusoidal voltage between top and bottom surfaces of the PZT sheet. The voltage sensitivity (V/g) at low frequency can be easily calculated since the expression of average normal strain in the top piezoelectric beam ε_2 is already known by Eq. 7:

$$V/a = \frac{\lambda d_{31}}{\varepsilon_0 \varepsilon_3} \varepsilon_2 E_2 t_2 \quad (10)$$

where d_{31} is the piezoelectric coefficient, ε_3 is the relative dielectric constant of PZT in direction 3, ε_0 is vacuum permittivity, and $\lambda=l/l_T$ (l_T is the total length of the diced PZT) is a coefficient which considers the parasitic PZT capacitances on the base and proof mass. As we can see from the bench-top characterization results shown in Fig. 4, the voltage sensitivity is 1.81 V/g at low frequency, and it is close to the theoretical one calculated by Eq. 10 which is 1.97 V/g. Also, we can find that the resonant frequency of the device is 354 Hz, and this is about 25% lower than the theoretical resonant frequency 471 Hz calculated by Eq. 6. There are several possible

reasons for this discrepancy. First, the real spring constant is smaller than the theoretical one since the proof mass is also subject to deformation in the experiment which was ignored in the analytical model. Second, the mass of the system is larger than theoretical one since we didn't consider the mass of the mechanical beam and PZT layer in the analytical model. Last, the bonding is not perfectly rigid, leading to softening effect of the boundary.

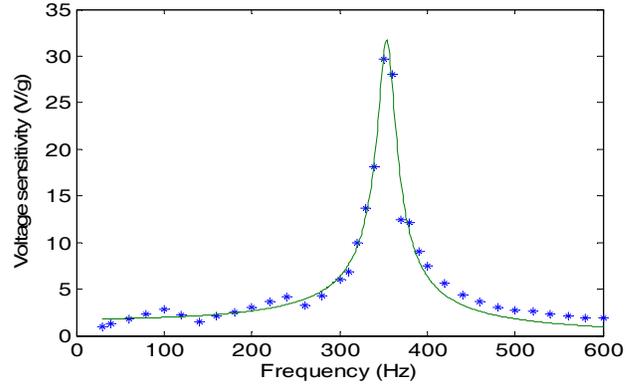


Fig. 4: Output voltage sensitivity of the vibration energy harvesting device as a function of frequency.

This prototype device was also road-tested on a vehicle (2003 Mazda Tribute). The assembly is shown in Fig. 5. The vibration energy harvesting device was mounted on the wheel up-side-down to make sure the PZT operates in compression mode. The wiring to the rotary assembly on the wheel was achieved using a slip-ring from Michigan Scientific. The two lead wires from the energy harvesting device were connected to the “rotor” of the slip-ring. There is an internal mechanism to electrically connect the terminals on the “rotor” to the “stator”. Then the signals from the “stator” were wired out through the aluminum tube and recorded by a USB-based data acquisition board (NI USB 6210) controlled by a laptop computer.

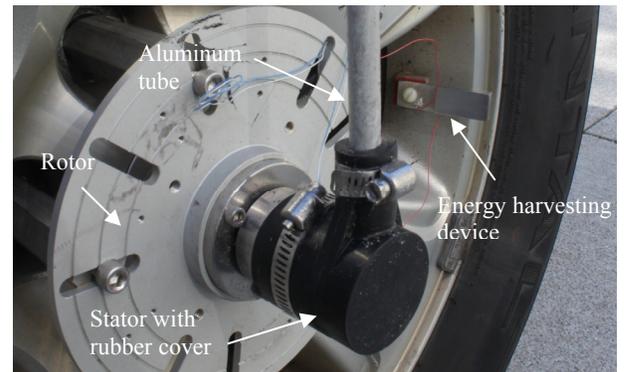


Fig. 5: The road test assembly on the wheel.

Two sets of experiments with different driving speeds and different weights of the proof mass were performed, and the power spectral density of the output voltage from the device is plotted in Fig. 6 and 7. As we can see in the figures, there are two peaks in the power spectrum plot. It is obvious that the first peak at low frequency corresponds to the rotation rate of the tire, i.e., 11 Hz in Fig. 6 is for a driving speed of 55 mph and 6 Hz in Fig. 8 is for a driving speed of 30 mph. The second peak at high frequency represents the resonant frequency of the device. Therefore, 470 Hz in Fig. 7 is for mass weight of 10.8 gram and 340 Hz in Fig. 8 is for mass weight of 21.6 gram.

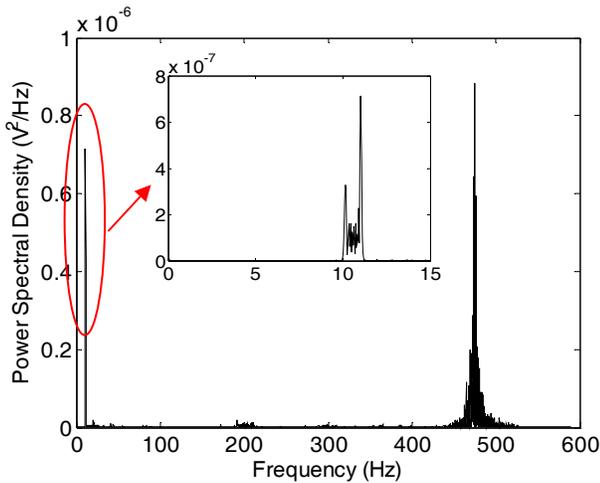


Fig. 6: Power spectral density of the output voltage in road test when driving speed is 55 mph and proof mass is 10.8 gram. Two peaks: 11 Hz (inset) and 475 Hz.

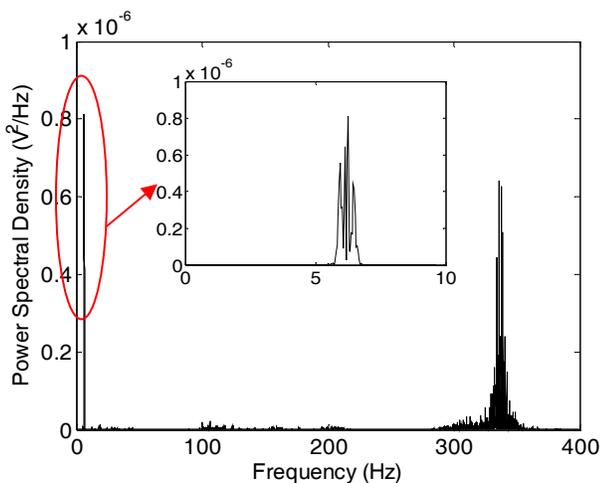


Fig. 7: Power spectral density of the output voltage in road test when driving speed is 30 mph and the proof mass is 21.6 gram. Two peaks: 6 Hz (inset) and 336 Hz.

The voltage generated by the vibration energy harvesting device was used to charge a 32 μF capacitor through a bridge rectifier. The result with ~ 50 mph driving speed and 21.6 gram of proof mass is shown in Fig. 8. As we can see in the figure, it takes about 35 seconds to charge the capacitor to around 8 V, and the maximum power is calculated to be 47 μW .

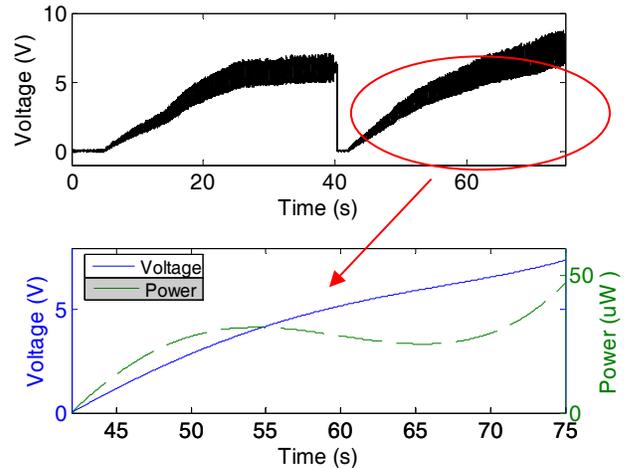


Fig. 8: Top: the voltage of a 32 μF capacitor charged by the energy harvesting device. Bottom: voltage of one charging cycle and corresponding power. Driving speed is ~ 50 mph, and the mass of the proof mass is 21.6 gram.

CONCLUSION

Vibration energy harvesting device based on asymmetric air-spaced cantilevers for tire pressure monitoring system has been demonstrated. Analytical model with the method of decomposing the deformation of the cantilever into pure bending and S-shape bending modes has been developed. Design was optimized to achieve more than 70% of mechanical conversion efficiency. A prototype device was then developed and tested both in the lab and on the road. The output voltage sensitivity and resonant frequency show good agreement with theoretical values calculated using our analytical model. The road test preliminarily demonstrated the feasibility of vibration energy harvesting device based on asymmetric air-spaced cantilevers for powering the tire pressure monitoring system.

REFERENCES

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