Low Frequency Vibration Energy Harvester Using Stopper-Engaged Dynamic Magnifier for Increased Power and Wide Bandwidth

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Abstract – We present a piezoelectric energy harvester with stopper-engaged dynamic magnifier which is capable of significantly increasing the operating bandwidth and the energy (power) harvested from a broad range of low frequency vibrations (<30 Hz). It uses a mass-loaded polymer beam (primary spring-mass system) that works as a dynamic magnifier for another mass-loaded piezoelectric beam (secondary spring-mass system) clamped on primary mass, constituting a two-degree-of-freedom (2-DOF) system. Use of polymer (polycarbonate) as the primary beam allows the harvester not only to respond to low frequency vibrations but also generates high impulsive force while the primary mass engages the base stopper. Upon excitation, the dynamic magnifier causes mechanical impact on the base stopper and transfers a secondary shock (in the form of impulsive force) to the energy harvesting element resulting in an increased strain in it and triggers nonlinear frequency up-conversion mechanism. Therefore, it generates almost four times larger average power and exhibits over 250% wider half-power bandwidth than those of its conventional 2-DOF counterpart (without stopper). Experimental results indicate that the proposed device is highly applicable to vibration energy harvesting in automobiles.

Keywords: 2-DOF system, Base stopper, Dynamic magnifier, Frequency up-conversion, Mechanical impact, Wideband operation

1. Introduction

During last few years, extensive efforts have been exerted to develop a wide variety of piezoelectric energy harvesters to scavenge power effectively from ambient vibration for powering low-power-consumption small wireless electronics [1-4], so that they can be operated autonomously and uninterruptedly by eliminating the need for recharging or replacing their external power sources such as batteries. Typically, a vibration energy harvester delivers maximum power to a matched load only if its resonant frequency matches ambient vibration frequency and power flow decreases as the resonant frequency decreases [5, 6]. But unfortunately, ambient vibrations are of low frequencies with eccentric nature which may drift over time [7]. A conventional piezoelectric vibration energy harvester is designed as a single-degree-of-freedom (SDOF) model in the form of a single mass-loaded cantilever beam (made of either a piezoelectric bimorph or a piezoelectric layer attached to a non-piezoelectric layer) which is efficient at its 1st resonance mode; 2nd and higher resonance modes with comparatively low response levels occur in excessively high frequencies and are generally ignored. This instinctive behavior results in a narrow operating bandwidth making itself inefficient for power

harvesting from a wide range of low frequency vibrations.

In order to overcome the narrow bandwidth problem, a number of researchers have demonstrated broadband harvester by using a multi-resonant piezoelectric beam by attaching multiple cantilever beams of natural frequencies close to the first resonance of the primary beam [8, 9]. A device with two piezoelectric beams connected with elastic springs has also been reported that improves the bandwidth by sacrificing the power [10]. A wideband piezoelectric energy harvester using a second piezoelectric beam as a stopper has also been demonstrated [11]. Beside the SDOF systems, some researchers have presented two-degree-of-freedom (2-DOF) systems using two-mass cantilever beam [12-14]. Use of an additional spring-mass system between the conventional SDOF mass-loaded piezoelectric cantilever beam and the harvester base, also known as dynamic magnifier, widens the effective bandwidth by decreasing the frequency gap between its 1st and 2nd resonance modes [15, 16]. It also increases the harvester's power output by virtue of magnifying (as the name dynamic magnifier implies) the strain experienced by the piezoelectric beam [17]. However, these harvesters cannot be considered as true wideband as the 2nd resonance peak is still considerably smaller and quite far away from the 1st resonance peak, as well as contain a deep valley between two peaks.

Besides the bandwidth issue, higher power generation under low frequency operating condition is another challenge to handle, because the output power of a vibration

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energy harvester is directly proportional to the cube of its resonant frequency [5]. Any wideband energy harvester with low operating frequency range also suffers from it. Mechanical frequency up-conversion mechanism has been proposed by a number of research groups in order to overcome this challenge. It has been done either by mechanical impact [18-20] or by non-mechanical interaction e.g., magnetic attraction/repulsion [21-23].

In this study, we have demonstrated the details of a mechanical impact driven 2-DOF piezoelectric low frequency vibration energy harvester that has recently been proposed by our group [24]. The device consists of a stopper-engaged flexible (polymer) dynamic magnifier in order to enhance the performance, in terms of both output power and operating bandwidth. Use of polymer not only decreases the resonant frequency but also increases the displacement limit in order to enhance dynamic magnification. The dynamic magnifier engages the harvester base stopper by piecewise linear impact during its vibration, even at a wide range of low frequencies; delivers secondary shock and introduces non-linear frequency up-conversion mechanism to the power generating element which, in turn, increases the output power significantly by increasing the strain and decreasing the source resistance within it.

2. Harvester Principle and Fabrication

2.1 Device structure and its working

The schematic structure of the proposed energy harvester is shown in Fig. 1. It is consisted of two parts: the dynamic magnifier comprising a flexible primary beam with a primary mass attached to one end constituting primary spring-mass system and, a relatively smaller and

Fig. 1. Schematic of the proposed energy harvester with dynamic magnifier: (a) conventional 2-DOF structure, and (b) harvester with stopper (HSDM)

rigid secondary beam with a small secondary mass attached to one end, other end being clamped to the primary mass in a parallel manner over the primary beam, constituting secondary spring-mass system. The mass ratio is $\mu = 5.8$. The overall system is clamped to the anchor that constitutes the conventional 2-DOF system structure. A rigid stopper (also called harvester base stopper) is placed at the bottom of the overall (primary and secondary spring-mass) vibrating system at such a position that a small gap exists between the fender (lower part) of the primary mass and the base stopper that constitutes the energy harvester with stopper-engaged dynamic magnifier (HSDM). The gap is left small so that the fender can be engaged with the base stopper within its displacement over the applied frequency range.

While low frequency periodic excitation with sufficiently large acceleration is applied, the overall spring-mass system (Primary and secondary) vibrates according to the excitation frequency. The flexible dynamic magnifier allows magnifying the strain experienced by the piezoelectric bimorph beam which, in turns, increase the electrical power output of the harvester. In addition, two close resonances occur which increase the effective operating bandwidth [15]. On the other hand, the maximum displacement of the primary mass (in HSDM) is limited to the threshold value by the base stopper giving rise to non-linearity in the system. The vibration behavior deviates from its normal 2-DOF motion and results in a hindrance of the vibration motion, introducing a periodically forced piecewise linear motion (after impact on the stopper) which, in turn, broadens the operating frequency bandwidth. Detailed analysis of the dynamics of this kind of piecewise linear impact can be found in the literature [25, 26]. This impact mechanism induces a secondary shock to the secondary spring-mass system. As a result, the secondary mass vibrates at its own natural frequency (higher than that of the primary spring-mass system) with exponential decay. The average stress on the piezoelectric layer is increased due to larger displacement of the secondary mass resulting in increased output voltage or power as compared to the conventional energy harvester with dynamic magnifier (EHDM) as shown in Fig. 2.

Fig. 2. Expected output characteristics of the proposed HSDM in comparison with its conventional 2-DOF counterpart (EHDM)

Fig. 3. Equivalent mechanical model of (a) conventional energy harvester with dynamic magnifier (EHDM), and (b) energy harvester with stopper-engaged dynamic magnifier (HSDM)

2.2 Electromechanical modeling

The energy harvester with dynamic magnifier (EHDM) in Fig. 1(a) can be modeled as a conventional 2-DOF model, whereas the energy harvester with stopper-engaged dynamic magnifier (HSDM) in Fig. 1(b) can be modeled as a piecewise linear vibro-impact model for a two-mass cantilevered spring-mass system. The equivalent mechanical models for both EHDM and HSDM are shown in Fig. 3. The dynamic equations of motion of the EHDM can be expressed as [27]

$$m_p \ddot{x}_p + c_p (\dot{x}_p - \dot{x}_0) + k_p (x_p - x_0) = -m_s \ddot{x}_s \quad (1a)$$

$$m_s \ddot{x}_s + c_s (\dot{x}_s - \dot{x}_p) + k_s (x_s - x_p) = 0 \quad (1b)$$

where m_p , x_p , c_p , and k_p are the mass, displacement, damping constant, and stiffness of the primary spring-mass system, respectively; m_s , x_s , c_s , and k_s are the mass, displacement, damping constant, and stiffness of the secondary spring-mass system, respectively; x_0 is the displacement of the vibrating base. On the other hand, while low frequency periodic excitation with sufficiently large acceleration a is applied to the HSDM, then in the time interval between two consecutive impacts, its dynamic equations of motion are

$$m_p \ddot{x}_p + (c_p + c_s) \dot{x}_p - c_s \dot{x}_s + (k_p + k_s) x_p - k_s x_s = m_p a \sin \omega_0 t \quad (2a)$$

$$m_s \ddot{x}_s + c_s (\dot{x}_s - \dot{x}_p) + k_s (x_s - x_p) = \delta_s(t - \tau) \quad (2b)$$

where $\delta_s(t - \tau)$ is the impulsive force generated by the secondary shock (while impact occurs) at time $(t - \tau)$, $t \gg \tau$. τ is the time between the start of the cycle (at $t = 0$) of the applied vibration and the impact (at $t = \tau$) of the primary mass on the base stopper. Then the response relative displacement of the secondary mass can be expressed by Duhamel's integral as [28]

$$x_s(t) = \frac{1}{m_s \omega_d} \int_0^t p(\tau) e^{-\zeta_s \omega_s (t-\tau)} \sin[\omega_d (t - \tau)] d\tau \quad (3)$$

where $p(\tau)$ is the arbitrarily varying excitation force at the time τ , $\zeta_s = c_s / 2\sqrt{k_s m_s}$ is the damping ratio, and $\omega_d = \omega_s \sqrt{1 - \zeta_s^2}$ is the damped natural frequency in which ω_s is the resonant frequency of the secondary beam. Then the average stress on the surface of the secondary beam (piezoelectric bimorph) can be calculated as [29]

$$\sigma_{avg} = \frac{k_s X_{max} l_s t_{eq}}{[(wt_p^3/3) + 4wt_p t_{eq}^2] + E_{sh} wt_{sh}^3/6E_p} \quad (4)$$

where X_{max} is the maximum displacement amplitude of the secondary mass, w and l_s are the width and effective length of the secondary beam, respectively; t_p and t_{sh} are the thicknesses of each piezo layer and middle shim layer of the bimorph beam, respectively while $t_{eq} = (t_p + t_{sh})/2$ is its equivalent thickness; E_p and E_{sh} are the Young's modulus of the piezo and shim materials, respectively. The output voltage generated and power delivered to a matched load resistance R_l as a function of time t can then be expressed as

$$V(t) = \frac{-d_{31} t_p \sigma_{avg}}{2\epsilon_r \epsilon_0} e^{-\xi_s \omega_s t} \sin \omega_d t \quad (5)$$

$$P(t) = \frac{1}{T} \int_0^T \frac{V(t)^2}{R_l} dt \quad (6)$$

where $-d_{31}$ is the piezoelectric strain constant, ϵ_r is the dielectric constant of the piezo material, ϵ_0 is the permittivity of free space, and T is the time period of $V(t)$.

2.3 Prototype fabrication

A macroscale prototype energy harvester was fabricated to demonstrate the proof of concept. The primary spring-mass system which acts as the dynamic magnifier, is comprised of a flexible (polycarbonate) primary beam and a primary mass (Aluminum). The secondary spring-mass system which is the main energy harvesting (power generating) element is made of a rigid secondary beam (PZT bimorph) and a small secondary mass (NdFeB cylinder), and is clamped on the primary mass. This double-beam structure is clamped to the harvester base (Aluminum) to form the EHDM (harvester without stopper). When a stopper is attached to the base (at approx. 1mm distance from the fender of the primary mass), it forms the HSDM (harvester with stopper). Fig. 4 shows the photographs of both EHDM and HSDM mounted on the vibration exciter. The material properties and the dimensions of the PZT bimorph are given in Table 1. The primary mass is made larger than the secondary mass (mass ratio $\mu = 5.8$) so that larger impulsive force could be

Table 1. The material properties of PZT and dimension of the PZT bimorph

Material properties	Young's modulus	Piezoelectric strain coefficient, $-d_{31}$	Dielectric constant	Density	Bimorph dimension	Capacitance of the bimorph
PZT	66 GPa	190 pC.N ⁻¹	1730	7800 Kg.m ⁻³	40×6×0.6 mm ³	19.7 nF

Fig. 4. Photographs of the fabricated energy harvester prototypes: EHDM (top) and HSDM (bottom)

Table 2. Dimensions of the double-beam structure used for both EHDM and HSDM

Structure	Beam dimension (mm)			Tip mass (gram)
	Length	Width	Thickness	
Dynamic magnifier	32	11	1	6.1
Harvester element	28	6	0.6	1.05

generated during impact on the base stopper, even at low acceleration. Therefore, the secondary mass displacement increases which, in turn, enhances the strain on the piezoelectric surface, resulting in an increase in the output voltage and power. The geometric parameters of the double-beam structure are shown in Table 2.

3. Experimental Results and Discussion

3.1 Experimental setup

Fig. 5 illustrates the schematic diagram of the complete experimental setup along with the photograph of the harvester prototype under test (HSDM: harvester prototype with stopper). The fabricated prototype energy harvester was mounted on an electro-dynamic shaker (TMS 2004E) which is connected to a function generator (Agilent 33120A) in conjunction with a power amplifier (TMS 2004E21) in order to provide sinusoidal excitation of various frequencies and accelerations to the harvester prototype. The amplitude of the input vibration was monitored by a measuring amplifier (Type 2525 Brüel &

Fig. 5. Schematic diagram of the experimental setup with photograph of the HSDM prototype energy harvester under vibration exciter test

Fig. 6. Peak-peak open circuit voltage responses (under 1 g peak acceleration) of the HSDM in comparison with EHDM

Kjær) connected to a reference accelerometer (Type 8305 Brüel & Kjær) that was attached to the base of the shaker. The output responses of the harvester were measured and recorded by a digital storage oscilloscope (Tektronix TDS5052B). In addition to the proposed HSDM, measurements were taken from the device without stopper that acts as a conventional EHDM in order to show the performance comparison.

3.2 Results and discussion

The frequency response of the peak-to-peak open circuit voltage of the HSDM along with the EHDM is shown in Fig. 6. It has been observed that the HSDM shows non-linear frequency response with 15 Hz $-3dB$ bandwidth while, on the other hand, the EHDM shows a 2-DOF

Fig. 7. Output voltage and power versus load resistance (at 17 Hz frequency and 1 g peak acceleration) obtained from the HSDM (top) and the EHDM (bottom)

frequency response with two consecutive resonant peaks at 17 Hz and 23 Hz. When the primary mass impacts on the stopper, the effective stiffness of the primary beam changes, allowing the primary mass to deviate from its normal motion and enables the resonance to extend over a wider range. The peak-to-peak voltage is found to be more than double than that of the EHDM at the frequency of its 1st resonance. An impulsive force during impact increases the stress on the PZT beam that allows increasing the voltage. Results show, maximum peak-peak open circuit voltages of 14.82 V and 6.94 V were generated by the HSDM and the EHDM, respectively at 17 Hz frequency and 1 g peak acceleration. The resultant maximum peak displacements of the secondary mass relative to the base (primary mass) of the secondary spring-mass system have been estimated as 590 μ m and 275 μ m, respectively.

As shown in Fig. 7, the load voltages across various load resistances have been measured to determine the maximum power delivering capability of the proposed device. It was done at the 1st resonant frequency (17 Hz) of the EHDM for convenient comparison. The generated power is experimentally equal to V_{rms}^2/R_l ; where V_{rms} is the rms voltage across the load resistance R_l . Results show that the proposed HSDM delivers maximum 21.39 μ W average power to a 30 $k\Omega$ load resistance. On the other hand, the EHDM delivers 5.48 μ W average power to its optimal load resistance, 370 $k\Omega$. Mathematically, the value of optimal R_l ($= R_s$) can be determined by [30]

$$R_l (= R_s) \propto \frac{1}{\omega_n C_p} \quad (7)$$

where R_s , ω_n , and C_p are the source resistance, natural frequency, and capacitance of the piezoelectric beam, respectively. The frequency of the secondary (piezoelectric) beam is up-converted after the impact which, in turn, reduces the source resistance. Our proposed double-beam

Fig. 8. Instantaneous voltage waveforms generated by the HSDM (top) and the EHDM (bottom) across the corresponding optimal load resistances at 17 Hz frequency and 1g peak acceleration

system is a 2-DOF system having two close resonances. Therefore, the calculated value of the source resistance R_s ($= R_l$) may be affected by some dimensionless constant related to the natural frequencies of the proposed 2-DOF system.

Fig. 8 shows the instantaneously generated voltage waveform across the corresponding optimal load resistances of the HSDM and EHDM at 17 Hz frequency. The peak-to-peak voltage of the HSDM right after an impact is quite high, as compared to the EHDM, but decays exponentially with time due to damping and becomes almost zero before the next impact occurs. As a result, the value of rms voltage (0.8 V) is very low than its peak-to-peak value (7.35 V). The up-converted frequency of the decaying waveform is 375 Hz, obtained by Fast Fourier Transform (FFT). This fast exponential decay can be reduced by changing the characteristic (e.g. stiffness, damping coefficient) of the power generating element. It is to be noted that, part of the impulsive force generated by the impact between the primary mass and the harvester base is transferred to the shaker that results in fickle (shown inset) in the acceleration waveform. As seen, both the waveforms contain harmonics, resulting in the waving of the corresponding outputs. It happens because the flexible dynamic magnifier i.e., the primary spring-mass system, vibrates at low frequency with large displacement. As a result, vibration of the primary spring-mass system (that works as the base for the secondary spring-mass system) superimposes on the vibration of the secondary spring-mass system. Therefore, waving of the generated output voltage waveforms is a superposition of the mutual vibration of the two systems. The harmonic of EHDM introduces a slight change in the frequencies between its generated voltage waveform and the applied acceleration waveform. Also reduces the rms value of the generated

Fig. 9. Measured average powers delivered to respective optimum load resistances of the HSDM and the EHDM as a function of frequency under 1g peak acceleration

Fig. 10. Output voltage and power delivered to the corresponding optimum load resistances of the HSDM (top) and the EHDM (bottom) at 17Hz under various base excitations

voltage than that obtained from a proper sinusoid.

Fig. 9 shows the average power delivered to the respective optimum load resistances as a function of the measuring frequency for both the devices. The EHDM generated maximum 5.48 μW average power at its 1st resonance (17 Hz) and 1.32 μW average power at its 2nd resonance (23 Hz). According to literature, its effective bandwidth is 6 Hz (from 17 Hz to 23 Hz) [15]. On the other hand, the HSDM generated maximum 21.39 μW average power with a half-power bandwidth of 15 Hz (from 9 Hz to 24 Hz) which is 250% larger than that of its conventional counterpart, EHDM. Moreover, the average value of the average power from 11 Hz to 19 Hz was 18.4 μW that is almost four times higher than that of the EHDM at its 1st resonance. The changes in output voltage and power delivered to the corresponding load resistances of the HSDM and the EHDM at 17Hz excitation frequency (first resonance of the EHDM) with the change in base acceleration is shown in Fig. 10. Results show that both voltage and power increase linearly as the acceleration amplitude increases.

The experimentally obtained results show that the HSDM is a highly effective way of increasing both the operating bandwidth and the output power in low frequency vibration energy harvesting, specifically, in harvesting energy from automobile vibrations. Running engine in a stationary vehicle and a moving vehicle on the road produce low frequency vibrations. Depending on engine's rpm, vehicle speed, suspension system of the vehicle, road roughness etc., characteristics (frequency and amplitude) of generated vibrations are different. Moreover, occasional braking and speed-breakers on the road allow the vehicle to experience shock. The HSDM is applicable to both general vibration and shock vibration generated by vehicle. Use of flexible (polymer) beam material and mechanical stopper offers its reliable operation even when subjected to shock vibration. Further analysis and optimized design e.g., mass ratio, stopper distance, generating elements characteristics etc. and fabrication through MEMS process will improve its performance which are of our next interest.

4. Conclusion

In summary, a piezoelectric vibration energy harvester with stopper-engaged dynamic magnifier (HSDM) exhibits significantly improved performance in terms of both output power and wideband operation in low frequency energy harvesting, compared to a conventional energy harvester with dynamic magnifier (EHDM). Engagement of the base stopper with the mass of the flexible dynamic magnifier transforms an impulsive force generated by impact to the secondary power generating element, introducing a frequency up-conversion mechanism and non-linear motion of the magnifier's mass as well. Non-linearity results in wideband operation and frequency up-conversion allow the power to increase by increasing stress and decreasing source resistance in the generating element. As compared to its conventional counterpart, the device with stopper (HSDM) offers 250% increased bandwidth (15 Hz) and almost four times increased average power (21.39 μW). Optimization through further analysis and fabrication through MEMS process can further improve its performance.

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References

- [1] Z. Xiao, T. Q. Yang, Y. Dong, and X. C. Wang, Energy harvester array using piezoelectric circular diaphragm for broadband vibration, *Appl. Phys. Lett.*, Vol. 104, No. 22, pp. 223904-1–223904-4, Jun. 2014.
- [2] E. K. Reilly, F. Burghardt, R. Fain, and P. Wright, Powering a wireless sensor node with a vibration-driven piezoelectric energy harvester, *Smart Mater. Struct.*, Vol. 20, No. 12, pp. 125006, Nov. 2011.
- [3] A. Mathers, K. S. Moon, and J. Yi, A vibration-based PMN-PT energy harvester, *IEEE Sens. J.*, Vol. 9, No. 7, pp. 731-739, Jul. 2009.
- [4] S. R. Anton and H. A. Sodano, A review of power harvesting using piezoelectric materials (2003-2006), *Smart Mater. Struct.*, Vol. 16, No. 3, pp. R1-R21, May 2007.
- [5] C. B. Williams and R. B. Yates, Analysis of a micro-electric generator for micro systems, *Sens. Actuators A*, Vol. 52, No. 1-3, pp. 8-11, Mar.-Apr. 1996.
- [6] H. Kulah and K. Najafi, Energy scavenging from low-frequency vibrations by using frequency up-conversion for wireless sensor applications, *IEEE Sens. J.*, Vol. 8, No. 3 pp. 261-268, Mar. 2008.
- [7] A. F. Arrieta, P. Hagedorn, A. Erturk, and D. J. Inman, A piezoelectric bistable plate for nonlinear broadband energy harvesting, *Appl. Phys. Lett.*, Vol. 97, No. 10, pp. 104102-1–104102-3, Sep. 2010.
- [8] S. Qi, R. Shuttleworth, S. O. Oyadiji, and J. Wright, Design of a multi-resonant beam for broadband piezoelectric energy harvesting, *Smart Mater. Struct.*, Vol. 19, No. 9, pp. 094009, Aug. 2010.
- [9] H. Xue, Y. Hu, and Q. -M. Wang, Broadband piezoelectric energy harvesting devices using multiple bimorphs with different operating frequencies, *IEEE Trans. Ultrason. Ferroelectr. Freq. Control*, Vol. 55, No. 9, pp. 2104-2108, Sep. 2008.
- [10] Z. Yang and J. Yang, Connected vibrating piezoelectric bimorph beams as a wide-band piezoelectric power harvester, *J. Intell. Mater. Syst. Struct.*, Vol. 20, No. 5, pp. 569-574, Jan. 2009.
- [11] H. Liu, C. Lee, T. Kobayashi, C. J. Tay, and C. Quan, Investigation of a MEMS piezoelectric energy harvester system with a frequency-widened-bandwidth mechanism introduced by mechanical stoppers, *Smart Mater. Struct.*, Vol. 21, No. 3, pp. 035005, Feb. 2012.
- [12] X. Tang and L. Zuo, Enhanced vibration energy harvesting using dual-mass systems, *J. Sound Vib.*, Vol. 330, No. 21, pp. 5199-5209, Jun. 2011.
- [13] Q. Ou, X. Chen, S. Gustschmidt, A. Wood, N. Leigh, and A. F. Arrieta, An experimentally validated double-mass piezoelectric cantilever model for broadband vibration-based energy harvesting, *J. Intell. Mater. Syst. Struct.*, Vol. 23, No. 2, pp. 117-126, Jan. 2012.
- [14] H. Wu, L. Tang, Y. Yang, and C. K. Soh, A novel two-degrees-of-freedom piezoelectric energy harvester, *J. Intell. Mater. Syst. Struct.*, Vol. 24, No. 3, pp. 357–368, Jan. 2013.
- [15] O. Aldraihem and A. Baz, Energy harvester with a dynamic magnifier, *J. Intell. Mater. Syst. Struct.*, Vol. 22, No. 6, pp. 521-530, May. 2011.
- [16] W. Zhou, G. R. Penamalli, and L. Zou, An efficient vibration energy harvester with a multi-mode dynamic magnifier, *Smart Mater. Struct.*, Vol. 21, No. 1, pp. 015014, Jan. 2012.
- [17] A. Aladwani, M. Arafat, O. Aldraihem, and A. Baz, Cantilevered piezoelectric energy harvester with a dynamic magnifier, *J. Vib. Acoust.*, Vol. 134, No. 3, pp. 031004, Apr. 2012.
- [18] O. Zorlu, E. T. Topal, H. and Kulah, A vibration-based electromagnetic energy harvester using mechanical frequency up-conversion method, *IEEE Sens. J.*, Vol. 11, No. 2, pp. 481-488, Feb. 2011.
- [19] H. Liu, C. Lee, T. Kobayashi, C. J. Tay, and C. Quan, Piezoelectric MEMS-based wideband energy harvesting systems using a frequency-up-conversion cantilever stopper, *Sens. Actuators A*, Vol. 186, pp. 242-248, Feb. 2012.
- [20] M. A. Halim and J. Y. Park, Theoretical modeling and analysis of mechanical impact driven and frequency up-converted piezoelectric energy harvester for low-frequency and wide-bandwidth operation, *Sens. Actuators A*, Vol. 208, pp. 56-65, Jan. 2014.
- [21] I. Sari, T. Balkan, and H. Kulah, An electromagnetic micro power generator for low-frequency environmental vibrations based on the frequency up conversion technique, *J. Microelectromech. Syst.*, Vol. 19, No. 1, pp. 14-27, Feb. 2010.
- [22] Q. C. Tang, Y. L. Yang, and X. Li, Bi-stable frequency up-conversion piezoelectric energy harvester driven by non-contact magnetic repulsion, *Smart Mater. Struct.*, Vol. 20, No. 12, pp. 125011, Nov. 2011.
- [23] P. Pillatsch, E. M. Yeatman, and A. S. Holmes, A piezoelectric frequency up-converting energy harvester with rotating proof mass for human body applications, *Sens. Actuators A*, Vol. 206, No. 1, pp. 178-185, Feb. 2014.
- [24] M. A. Halim and J. Y. Park, Performance enhancement of a low frequency vibration driven 2-DOF piezoelectric energy harvester by mechanical impact, in Proc. PowerMEMS, 2014.
- [25] S. W. Shaw and P. J. Holmes, A periodically forced piecewise linear oscillator, *J. Sound Vib.*, Vol. 90, No. 1, pp. 129-155, Sep. 1983.
- [26] S. W. Shaw, Forced vibrations of a beam with one-sided amplitude constraint: Theory and experiment, *J. Sound Vib.*, Vol. 99, No. 2, pp. 199-212, Mar. 1985.
- [27] L. Tang and Y. Yang, A multiple-degree-of-freedom piezoelectric energy harvesting model, *J. Intell. Mater. Syst. Struct.*, Vol. 23, No. 4, pp. 1631-1647, Jun. 2012.

- [28] C. Lalanne, Mechanical Vibration and Shock Analysis-volume 2, *John Wiley & Sons*, pp. 51-94, Jan. 2010.
- [29] L. Gu, Low frequency piezoelectric energy harvesting prototype suitable for the MEMS implementation, *Microelectr. J.*, Vol. 42, No. 2, pp. 277-282, Feb. 2011.
- [30] S. Adhikari, M. I. Friswell, and D. J. Inman, Piezoelectric energy harvesting from broadband random vibrations, *Smart Mater. Struct.*, Vol. 18, No. 11, pp. 115005, Sep. 2011.

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