Tunable nonlinear piezoelectric vibration harvester

S Neiss, F Goldschmidtboeing, M Kroener and P Woias
Laboratory for Design of Microsystems, IMTEK, University of Freiburg, Georges-Koehler-Allee 102, 79110 Freiburg, GERMANY
E-mail: sebastian.neiss@imtek.uni-freiburg.de

Abstract. Nonlinear piezoelectric energy harvesting generators can provide a large bandwidth combined with a good resonant power output. However, the frequency response is characterized by a strong hysteresis making a technical use difficult if the hysteresis cannot be compensated. We propose a tuning mechanism that allows both, a compensation of the hysteresis as well as maintaining the optimal work point. The compensation algorithm can reduce the hysteresis to a minimum of only 1.5 Hz and maintain a high energy oscillation in a large frequency window between 53.3 Hz and 74.5 Hz.

1. Introduction
Nonlinear mass-spring systems have found growing interest in recent research for vibrational energy harvesting [1, 2, 3, 4]. In contrast to linear vibrational generators, a nonlinear stiffness is intentionally introduced in the system. By this, such systems inherently comprise a large bandwidth combined with a good resonant power output [1, 2]. This distinguishes them from linear devices which show a contrary behavior of bandwidth and conversion efficiency [3]. As a consequence, generators with a nonlinear restoring force are better suited for energy harvesting from vibrations with a varying dominant frequency compared to a device with a linear stiffness.

2. Theory
Figure 1(a) shows an exemplary frequency response of a nonlinear piezoelectric harvester with softening characteristic. While power output can be high in a large frequency band, the response is characterized by a frequency hysteresis. If the excitation frequency is located in the hysteresis range, the oscillation will always start in the low power regime.

To overcome this limitation, a control mechanism is implemented which is schematically shown in figure 1. Starting from a low power oscillation with a phase shift between piezo voltage and excitation around 0°, the system’s stiffness is first reduced to shift the jump-up point to a lower frequency than the excitation frequency. The jump is detected by measuring the change of the phase shift which is now larger than −90°. Next, the stiffness is adjusted to have the harvester work with a higher power output at a phase shift around −135°. When the working point is reached, the control keeps the oscillation in the attractor’s high power orbit by adjusting the stiffness to maintain the phase shift between −130° and −140°. In contrast to a linear system, the working point is not set to a phase shift of −90°. In nonlinear systems, the jump-down is located at this point. Consequently, the working point is set to a point with a phase shift less than −90° to obtain a certain safety margin during operation.
The proposed mechanism uses a harvester with adjustable stiffness. Thus, we developed the harvester shown in figure 2 and 3. The transducer comprises a steel spring with a piezoceramic sheet attached to its base. In addition, an iron proof mass (2.7 g) is mounted at the tip. A magnets flux is guided through this iron mass using a magnetic circuit which generates a nonlinear softening restoring force on the latter [5]. By changing the gap size between magnetic circuit and proof mass, the strength of the magnetic spring is altered thus changing the systems linear and nonlinear spring constant [6].

3. Dynamic characterization

Figure 4 shows the influence of the excitation amplitude on the open-loop frequency and phase response of the harvester. Stronger excitation leads to an increased non-linearity of the system resulting in an increased hysteresis window. This gives further rise to the need of a control in a nonlinear harvester. On the one hand, a drop of the excitation amplitude shifts the jump-down frequency, and consequently also the optimal working point, to higher frequencies. This can lead to a drastic drop in output power if the jump-down point exceeds the excitation frequency.
Figure 4. Frequency response (left) and phase response (right) of the harvester at different excitation amplitudes (0.5 to 3 m/s²) and a gap size of 0.5 mm. Up-sweeps are indicated by dashed lines, down-sweeps by solid lines.

Figure 5. Frequency response (left) and phase response (right) of the harvester at different gap sizes (0.5 to 1.5 mm) and an excitation amplitude of 1 m/s². Up-sweeps are indicated by dashed lines, down-sweeps by solid lines.

On the other hand, an increase of the excitation frequency shifts the jump-down point to lower frequencies. Again, the work point will not be at the optimal point any longer.

In a similar sense, figure 5 shows the influence of the gap size on both, frequency and phase response. The measurement proves that the whole frequency response can be shifted along the frequency axis as needed for the implementation of the tuning mechanism by changing the gap size. The measurement also shows that small gap sizes produce a larger hysteresis window thus indicating a stronger non-linearity.

For all measurement conditions, the jump-down point is located at a phase shift of −90°. An increase of the non-linearity caused by either a change of excitation amplitude or change of gap size only alters the slope of the phase response curve close to the jump-point. Therefore, this parameter is perfectly suited to determine the work point of the system.

4. Tuning and hysteresis control

The tuning mechanism explained above is implemented by a simple algorithm. If the phase shift is larger than −130°, the frequency response has to be shifted to lower frequencies which is achieved by increasing the gap size. If the phase shift is lower than −140°, the frequency response is shifted to higher frequencies by decreasing the gap size. For the automatic adjustment of the
gap size, the harvester is mounted on a motorized linear table.

The algorithm in LabVIEW to test the usability of this approach. The algorithm itself is known in a similar implementation from linear devices [7]. However, the phase response shows a varying slope around $-135^\circ$ depending on the excitation frequency and gap size. As a consequence, safety margin between operating point and jump-down frequency becomes small if either the gap or the excitation amplitude are small. This might be a problem since the oscillation becomes unstable the closer the frequency is located at the jump-down frequency. This gives rise to basically two questions which have to be answered to prove the usability of this method for nonlinear devices as well. First, a sudden change of the spring constant could destabilize the oscillation and lead to a sudden jump-down as soon as the motor is active. In a similar sense, the vibration that is induced by the motor itself could lead to a destabilization as well. This problem can be compensated if the safety margin is increased and the work point is set to a larger phase shift. However, this leads to an unwanted drop in power output.

Figure 6 shows the frequency response of the harvester with activated control at an excitation amplitude of $1 \text{ m/s}^2$. The hysteresis is reduced to a minimum of $1.5 \text{ Hz}$ and the high energy oscillation is maintained between $53.3 \text{ Hz}$ and $74.5 \text{ Hz}$. The noise in the frequency response is caused by the control itself. If the gap size is changed, the piezo voltage shows a step caused by the change of the working point. However, it is not possible to reduce the safety margin between operating point and jump-down frequency any further. During tests with a work point located at $-125^\circ \pm 5$ degree, the activation of the motor occasionally lead to a destabilization of the oscillation. The measurement also shows another advantage of using a nonlinear device if a tuning mechanism is included. The configurations with strong non-linearity need less tuning activity compared to systems with weaker non-linearity.

In the single measurements without control presented above, the power dissipated in a resistive load with 300kΩ ranged from $\approx 130 \mu\text{W}$ at a gap of 0.5 mm to $\approx 1020 \mu\text{W}$ at a gap of 1.5 mm. With activated control, the power drops to $\approx 90 \mu\text{W}$ at a gap of 0.5 mm to $\approx 880 \mu\text{W}$ at a gap of 1.5 mm. This corresponds to a drop of power of about $15 - 35 \%$. It is obvious that the drop is larger for systems with a smaller gap. Theses systems show a stronger non-linearity and consequently, the safety margin between operating point and jump-down point is larger resulting in an emphasized drop of the voltage amplitude.

Whenever a tuning mechanism is included in an energy harvesting device, the tuning energy consumption is a major figure that has to be evaluated and compared to the power delivered by the harvester itself. However, the presented system was built to test the usability of this approach and uses a LabVIEW based control in combination with a linear motor. Therefore, we limit the comparison at this point to the evaluation of the energy that is needed to move the harvester against the pulling force of the magnetic spring. Figure 7 shows the measured
Figure 7. Force acting on the mass in beam direction, i.e. the tuning direction. The energy needed for one tuning cycle from 0.5 mm to 1.5 mm is given by the filled area.

force acting on the mass in beam direction, i.e. the force against which the harvester has to be moved. From this measurement, the energy that is needed to move the harvester from 0.5 mm to 1.5 mm is calculated to 780 µJ. However, the energy that is needed in a real system will be higher due to the limited efficiency of the actuation mechanism. With a reasonable efficiency of around 20%, the energy would be in the range of a few Millijoule. Compared to the output power of the system, the energy for a full tuning cycle would be harvested in 40 s at maximum. We emphasize at this point that this is a theoretical estimation. Nevertheless, it proves the suitability of the magnetic springs in combination with this tuning mechanism to build a tunable nonlinear energy harvester.

5. Conclusion
The results of the measurements prove the proposed tuning and compensation mechanism to be working. The frequency hysteresis is nearly completely compensated and the harvester works always at an optimal working point. In addition the chosen implementation achieves both, a compensation of the hysteresis as well as maintaining the work point at a phase shift of around −135°. In addition, the tuning energy is in the same order as the power delivered by the harvester. This allows for an implementation of an autonomous control if a suitable actuation mechanism replaces the current linear motor.

Acknowledgments
We gratefully acknowledge the financial support from the German Research Association (DFG) within the Research Training Group (GRK) 1322 ”Micro Energy Harvesting”.

References
[1] Ramlan R 2009 Potential benefits of a non-linear stiffness in an energy harvesting device (University of Southampton: PhD thesis).
[2] Mann B, Barton D and Owens B 2012 Uncertainty in performance for linear and nonlinear energy harvesting strategies J. Intell. Mater. Syst. Struct. 23 pp 1451-60.
[3] Zhu D, Tudor M and Beeby S 2010 Strategies for increasing the operating frequency range of vibration energy harvesters: a review Meas. Sci. Technol. 21 pp 022001.
[4] Ferrari M, Ferrari V, Guizzetti M, And B, Baglio S and Trigona C 2010 Improved energy harvesting from wideband vibrations by nonlinear piezoelectric converters Sensor. Actuat. A Phys. 162 pp 422-31.
[5] Furlani E P 2001 Permanent Magnet and Electromechanical Devices (London: Academic Press).
[6] Neiss S, Kleber J, Woias P and Kroener M 2012 Reluctance Springs for Nonlinear Energy Harvesting Generators Proc. PowerMEMS (Atlanta, GA) Dec 02-05 2012 pp 153-156.
[7] Eichhorn C, Tchagsim R, Wilhelm N and Woias P 2011 A smart and self-sufficient frequency tunable vibration energy harvester J. Micro mech. Microeng. 21 pp 104003.