

FREQUENCY TUNING OF VIBRATION ENERGY HARVESTERS USING COMPRESSIVE AND TENSILE AXIAL LOADS

Dibin Zhu*, M. J. Tudor, S. P. Beeby

School of Electronics and Computer Science, University of Southampton, Southampton, UK

*Presenting Author: dz@ecs.soton.ac.uk

Abstract: In this paper the performance in terms of resonant frequency and output power of cantilever-based vibration energy harvesters under different mechanical axial loads will be compared providing a comprehensive assessment of this method as a way of increasing the operational frequency range through tuning. Two types of axial loads were used in the test, i.e. tensile and compressive force. To eliminate the effect of any other mechanical loads on performance of the energy harvester and minimize external damping, a contactless force, i.e. magnetic force, was applied to the cantilever. It is found that although resonant frequency tuning by applying compressive force is more sensitive than tuning by tensile force, the reduction in Q-factor under compressive loads makes it less useful than tuning by applying tensile loads. Axial compressive loads can, however, be used to achieve a bi-stable structure which can also be a solution to wideband energy harvesters.

Keywords: resonant frequency tuning, vibration energy harvesting, axial loads

INTRODUCTION

With the development of low power electronics and energy harvesting technology, self-powered systems have become a research hotspot over the last decade. Some possible alternative energy sources include photonic energy, thermal energy and mechanical energy (vibration). Among these sources, photonic and thermal energy have already been widely used in power supplies. However, energy harvesting using these sources restricts the working environment of electronic systems. Vibration can be found in instances where photonic or thermal energy is not available, which makes extracting energy from vibration an attractive approach for powering electronic systems.

The most common vibration energy harvesting solution is a linear energy harvester that converts kinetic energy to electrical energy using a transduction mechanism [1]. Such harvesters produce maximum power when its resonant frequency matches the ambient vibration frequency. Furthermore, to generate maximum power, devices are typically designed to have a high Q-factor. However, in this case, output power drops dramatically if the ambient and device resonant frequencies do not match.

To solve this problem, some mechanisms have been developed to increase operational bandwidth of vibration energy harvesters [2]. One commonly used method is to tune the resonant frequency according to the vibration frequency by changing mechanical properties of the structure or adjusting electrical loads. The former method provides large tuning range but consumes much energy while the latter one has small tuning range but consumes less energy. So far, most frequency tuning methods are based on changing mechanical properties. Leland and Wright [3] successfully tuned the resonant frequency of a vibration piezoelectric energy harvester by applying axial compressive force directly on the cantilever using

a micrometer. The tuning range is from 200 to 250 Hz. Zhu *et al* [4] reported a tunable electromagnetic vibration energy harvester with a tuning range from 67.6 to 98Hz.

In this paper, the performance in terms of resonant frequency and output power of cantilever-based vibration energy harvesters under different mechanical axial loads will be compared providing a comprehensive assessment of this method as a way of increasing the operational frequency range through tuning. Two types of axial loads were used in the test, i.e. tensile and compressive force. Principles of tuning together with experiment results will be presented. Some guideline will be summarised for future design of tunable vibration energy harvesters using similar tuning methods.

PRINCIPLES

Since most vibration energy harvesters with a cantilever structure work at a resonance of mode 1, the following analysis is based on oscillation in mode 1. An approximate formula for the resonant frequency of a uniform cantilever in mode 1 with an axial load, f_r' , is given by [5]:

$$f_{r1}' = f_{r1} \cdot \sqrt{1 + \frac{F}{F_b}} \quad (1)$$

where f_{ri} is the resonant frequency in mode i without load. F is the axial load. F is positive if the load is tensile and F is negative if the load is compressive. F_b is the buckling load, i.e. the axial load that can buckle the beam.

It is found in Eq. 1 that an axial tensile load applied to a cantilever increases the resonant frequency of the cantilever while an axial compressive load applied to a cantilever decreases the resonant frequency of the cantilever. Fig. 1 shows change in

resonant frequency with different axial loads. It is found that a compressive load is more efficient in frequency tuning than a tensile load, especially when the load is close to the buckling load.

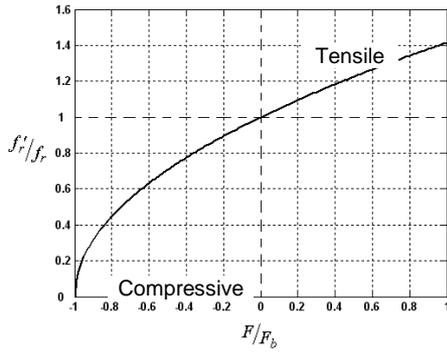


Fig. 1: Change in resonant frequency with axial loads.

The maximum electrical power that can be extracted by a vibration energy harvester, which is given by [1]:

$$P = \frac{m \cdot a^2}{8 \cdot \omega_r} Q_{oc} \quad (2)$$

where P , m , a and ω_r are maximum power, mass, acceleration, resonant frequency and total damping, respectively. $Q = 1/2\zeta_m$ is the open-circuit Q-factor and ζ_m is the mechanical damping. Eq. 2 shows that the maximum extractable power decreases with an increase of the resonant frequency for a constant acceleration and constant damping.

EXPERIMENT AND DISCUSSION

Tuning Mechanism

Fig. 2 shows the schematic diagram of the tuning mechanism. The tuning force is provided by the repelling or attractive force between two tuning magnets with similar or opposite poles facing each other. One magnet is fixed at the free end of a cantilever while the other is placed axially in line with the cantilever and is moveable along the span of the cantilever. As the distance between the two tuning magnets is adjusted, the axial load on the cantilever changes, and hence the resonant frequency of the cantilever is tuned. The areas where the two magnets face each other are curved to maintain a constant gap between them over the amplitude range of the energy harvester.

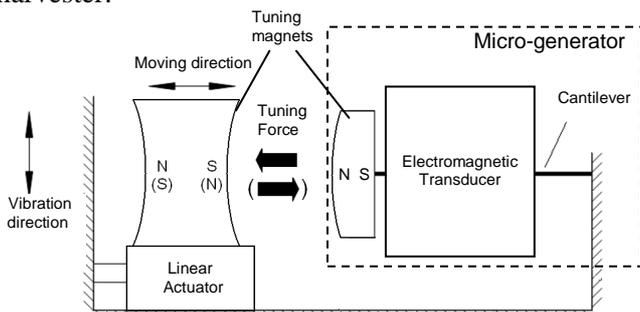


Fig. 2: Schematic diagram of the tuning mechanism.

The Tunable Electromagnetic Energy Harvester

The energy harvester used in the experiments is shown in Fig. 3. The harvester has a cantilever beam with magnets attached. The cantilever is clamped to a base where a coil is fixed. This device is similar to the one reported previously [4] although this one is slightly larger. The cantilever is made of 0.3mm-thick BeCu for its excellent fatigue characteristics and all magnets are NdFeB.

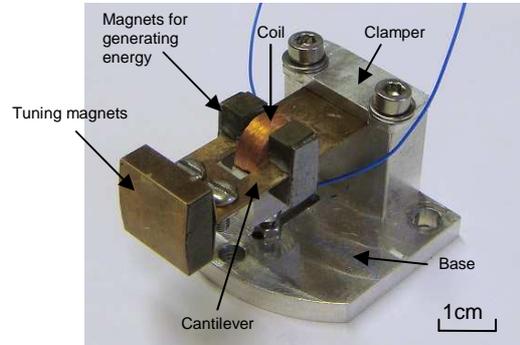


Fig. 3: Energy harvester used in the experiment.

Experimental Setup

The test platform of the system is shown in Fig. 4. Tuning magnets (5) is attached to the free end of the cantilever resonator on the energy harvester (4). The other tuning magnet (3) is fixed on a slider (2) which is able to slide along a track. The movement of the slider is controlled by the linear actuator (1). The linear actuator used here is Haydon® 21000 Series Size 8 linear actuator, E21H4(AC)-5 which has a resolution of 0.01mm/step. It is controlled by the microcontroller via two H-bridge circuits. All components in the test were powered by an external power source. In this test, the vibration acceleration was set to be $0.49\text{m}\cdot\text{s}^{-2}$, i.e. 50mg ($1g = 9.8\text{m}\cdot\text{s}^{-2}$).

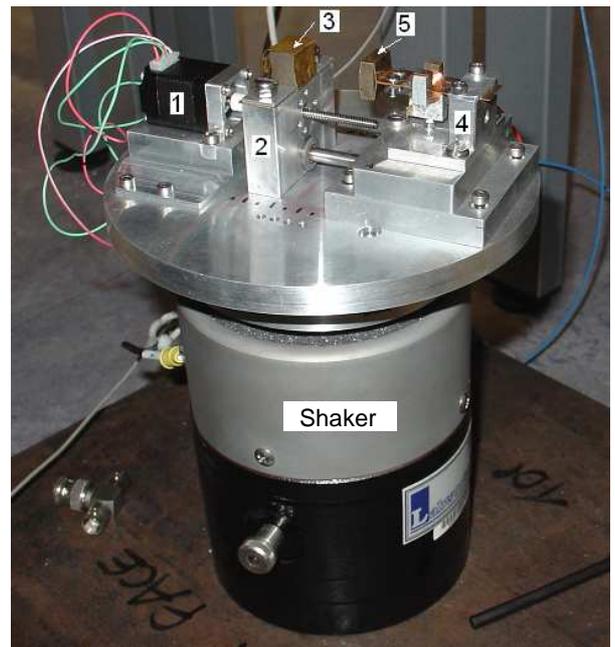


Fig. 4: Test platform in the test (1: Linear actuator; 2: Slider; 3: Tuning magnet 1; 4: Energy harvester; 5: Tuning magnet 2).

Tuning Force

Fig. 5 shows the numerical calculation of the tuning force versus distance between two tuning magnets. The compressive force and tensile force are the same in value for the same set of magnets but opposite in direction. Details of calculation can be found in [4].

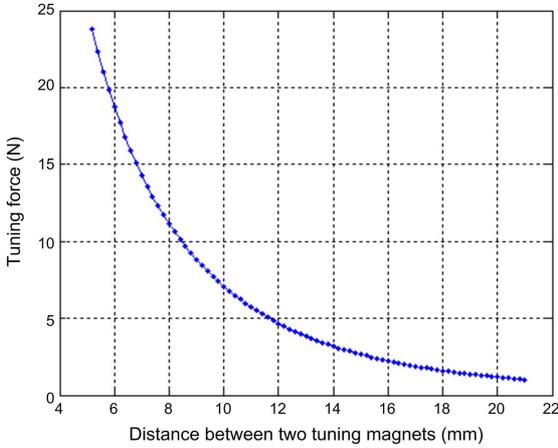


Fig. 5: Numerical calculation of tuning force vs. distance between two tuning magnets.

Resonant Frequency under Axial Loads

Fig. 6 shows the resonant frequency of the energy harvester with variation of tuning forces. Test results have been compared with the simulation results in ANSYS Workbench and found to be in reasonable agreement with the simulation. It is worth noting that only the axial load was applied in the simulation and loads on other directions were not taken into account in the simulation. The resonant frequency of the energy harvester without any tuning mechanism was measured as 27Hz.

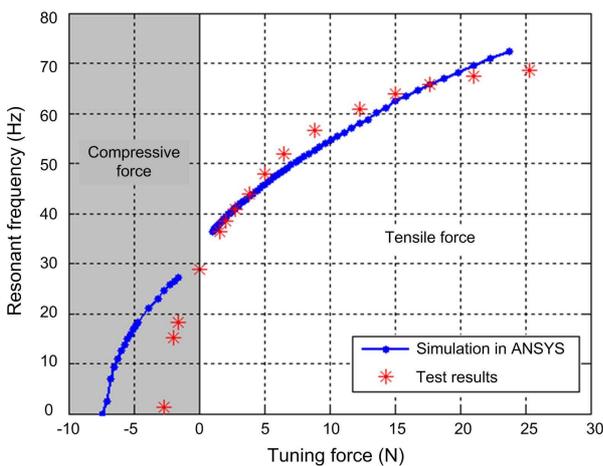


Fig. 6: Resonant frequency with variation of tensile tuning forces.

In the case of tensile force, the resonant frequency was tuned from 35Hz to 68Hz by varying axial tensile loads from 1 to 26N. When the tuning force became large, the resonant frequency was lower than expected. The reason for this is that when the tensile force much greater than the buckling force is applied to the

cantilever, the resonant frequency approaches that of a straight tensioned cable and does not increase anymore. This is because the force associated with the tension in the beam becomes much greater than the beam stiffness.

In the case of compressive force, the resonant frequency was tuned from 18Hz to 1.2Hz by varying the compressive loads from 1 to 2.8N. The test results did not match the simulation results. Experimentally, the resonant frequency of the energy harvester decreased more than expected under compressive loads. The reason for this is that in the practical situation, the cantilever was bent due to the compressive loads and the mass had an initial displacement when it was static. As the resonator started vibrating, the mass had a larger displacement in one direction than that in the other one. When the compressive load became large, the inertial displacement of the mass was so large that it hardly vibrated. However, in the simulation, only the axial load was applied while the initial displacement of the proof mass and loads on other directions were not taken into account.

Output Power under Axial Loads

Fig. 7 shows test results of the power spectra of the energy harvester with various resonant frequencies under tensile loads. It is found that the Q-factor of the energy harvester with tensile loads became higher than that of the energy harvester without any tuning mechanism. As predicted in Eq. 2, the output power reduces with the increase of the resonant frequency. Additionally, when the resonant frequency approached the upper boundary of the tuning range, i.e. when the tuning force was large, the output power of the energy harvester dropped lower than constant damping situation, which means that the total damping increased as the tuning force becomes large.

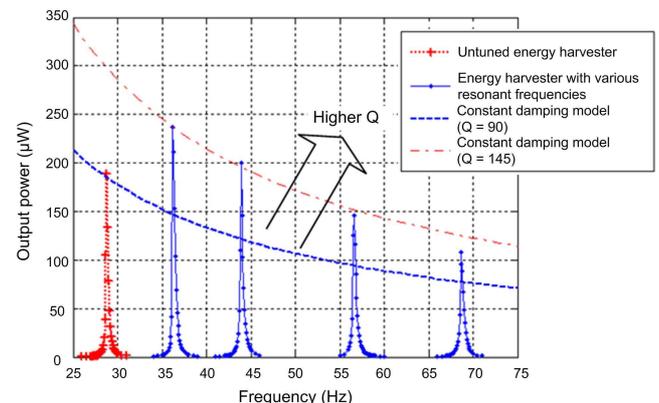


Fig. 7: Power spectra of the energy harvesters with various resonant frequencies under tensile loads.

Fig. 8 shows test results of the power spectra of the energy harvester with various resonant frequencies under compressive loads. It is found that the Q-factor of the energy harvester decreased significantly when compressive loads were applied. As mentioned earlier,

the energy harvester must have a high Q-factor to produce more output power. Although tuning by applying compressive force is more sensitive than tuning by tensile loads, this characteristic makes frequency tuning by applying compressive loads less useful than tuning by applying tensile loads.

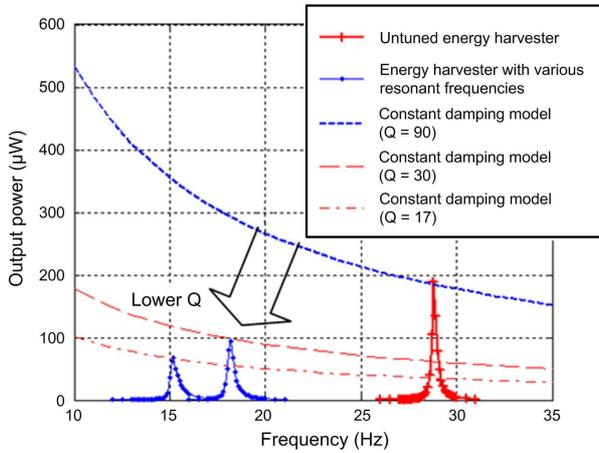


Fig. 8: Power spectra of the energy harvesters with various resonant frequencies under compressive loads.

CONCLUSION

This paper compared the performance in terms of resonant frequency and output power of cantilever-based vibration energy harvesters under different axial loads. The axial loads were provided by the interacting force between two tuning magnets. An axial tensile load can increase the resonant frequency while an axial compressive load can decrease the resonant frequency. The theoretical analysis showed that the resonant frequency of a cantilever structure is more sensitive to compressive load than tensile load, i.e. resonant frequency changes more by applying the compressive load than by applying the same amount of tensile load.

A prototype of an energy harvester has been tested. The energy harvester consisted of a cantilever, an electromagnetic transducer and a tuning magnet attached to the free end of the cantilever. The untuned resonant frequency is 27Hz. The other tuning magnet is fixed on a slider which can move axially along the cantilever span. The distance between two magnets can be manually adjusted and thus the tuning force.

Experimentally, it was proved that the resonant frequency of the energy harvester could be tuned by applying axial load. Under tensile loads, the resonant frequency has been tuned from 35Hz to 68Hz by axial tensile loads from 1 to 26N. It is found that, when the tuning force became large, the resonant frequency was lower than expected because, when a tensile load much greater than the buckling force is applied to a beam, the resonant frequency approaches that of a straight tensioned cable and does not increase anymore because the force associated with the tension in the beam becomes much greater than the beam stiffness. Furthermore, the Q-factor of the energy harvester with tensile loads became higher than that of the energy

harvester without any tuning and the output power reduces with the increase of the resonant frequency as predicted. As long as the tensile load is applied, the Q-factor stayed constant for most of the tuning range. However, the Q-factor decreased when the tensile load became large.

Under compressive loads, the resonant frequency has been tuned from 18Hz to 1.2Hz by varying compressive loads from 1 to 2.8N. The resonant frequency decreased significantly for a small tuning force. It showed a larger decrease than the simulation results suggested. Furthermore, the Q-factor of the energy harvester dropped a lot as the resonant frequency decreased, i.e. the compressive load increased, which is a fatal drawback for this method as a useful tuning mechanism.

In conclusion, although resonant frequency tuning by applying compressive force is more sensitive than tuning by tensile loads, the reduction in Q-factor which occurs under compressive loads makes it less useful than tuning by applying tensile loads. Conversely, an axial tensile load increases the Q-factor of the energy harvester. Therefore, applying axial tensile loads to a cantilever is the preferred method to tune the resonant frequency of a vibration energy harvester. When an axial compressive load exceeds the buckling force of the cantilever, a bi-stable structure [6] can be achieved, which can also be a potential solution to wideband energy harvesters.

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