Increased Bandwidth System for Mechanical Energy Harvesting

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ABSTRACT

Within this paper a new approach for amplifying relative movement of a cantilever at off-resonance frequencies is presented. The aim is to broaden the peak resonance of resonators without compromising the quality factor. We propose to do this by introducing an additional process allowing more transfer of mechanical energy from the vibration source to the converter. The amplification of the transferred energy ensured by this process is achieved by gathering the resonance phenomenon conditions at off-resonant frequencies and even for random vibrations. This approach can be used for mechanical energy harvesters and active dampers that operate in environments where the vibration frequency is not constant (such as transportation engines). In this work an amplification gain seven times greater than a simple resonator was obtained at off resonance frequencies.

Keywords: Energy harvesting, active damping, wide bandwidth, resonators, frequency tuning.

1 INTRODUCTION

There is a growing pursuit of reliable power solutions to extend the lifespan of portable and wireless electronics. Ambient mechanical source are one of the most promising sources to power such systems. Most developed mechanical harvesters are based on resonators; these devices ensure an important electromechanical coupling when their natural frequency matches the input frequency. Nevertheless, in many situations the driving frequency is not known or it might change over time. In such cases, the output power falls dramatically when the input frequency is not equal to the resonant frequency of the harvester. However, one part of efficiency of the energy harvester is the magnitude of the power harvested when the mechanical input frequency doesn't match the resonant one. By the same way, we measure the efficiency of an active damper when its resonance frequency is different from the vibration one. It's then highly desired to have an energy harvester, or active damper, able to self tune its resonant frequency to optimize its power output, or the damped displacement, for any vibration environment.

2 STAT OF THE ART

Multiple approaches have been investigated to leverage this limitation, by either tuning the resonance frequency using a system with a wide resonant peak, a passive or active adaptation:

• *Passive wider bandwidth*: the harvester system doesn't need to be assisted to match the input frequency; it recovers the maximum of energy without any exterior actuators for tuning the resonant frequency, like systems with a broad peak of resonance and multi modes systems [1-2].

• *Manuel tuning*: the manual tuning system enables to the user to adjust the system resonance frequency before each operation in order to match the input vibration frequency[3].

• *Active tuning* : the system can electrically adjust in real time its resonant frequency. The frequency adaptation system stays running continuously to match the natural frequency to the input one [4].

3 OUR APPROACH

3.1 Description

In order to understand the operation of the system that we propose, let's consider a cantilever characterized by its natural frequency f_{r1}, in addition to this resonator we add a blocking system, when the support speed reaches its maximum value we actuate a blocking between the support and the seismic mass. During the blocking time the system has another natural frequency f_{r2} higher than the first one. After a while, the mass is released and we repeat this operation at each period of operation. By doing so, the speed of the seismic mass is amplified and then more mechanical energy is stored into the resonator when the input frequency is different from the natural frequency. In the following, we will show how with a good choice of the ratio f_{r2}/f_{r1}, and the blocking time duration, one could improve the system response, we can optimize the relative displacement gain, defined as the ratio between the relative displacement amplitude of the seismic mass with the blocking system over this without blocking system, at off-resonance frequencies, over a wide bandwidth of frequency between f_{r1} and f_{r2}.

First, let's have a look on what happen when the resonance frequency is equal to the vibration one, and let's consider a resonant system composed of a seismic masse m attached to a vibration support by a spring of stiffness k and dampers b_e and b_m as shown in figure 1 bellow. At the resonance, the support movement is in quadrature phase with the movement of the seismic mass, in this case the movement imposed by the support is in phase opposition with the effort exerted by the spring on that support, and in other words it is necessary that the support brings mechanical work to the resonator system and not the opposite. Hence, more mechanical energy is transferred to the resonant system, which causes an increase of stored energy in the spring mass system, therefore, the relative displacement of the mass is increased and so the system efficiency. The effort being opposed to the actuation of support is related only to the deformation of the spring; hence the mechanical energy stored on the spring is related to the strength of the developed effort, for more stored energy, we should manage to deform the spring as much as possible. On the other hand, when the input frequency doesn't match the resonant frequency of the cantilever, there is as much work transferred towards the resonant system than this restored to the support. With the proposed system, it's possible to store more mechanical energy in the cantilever even when the input frequency is different from the resonant one.

In the next paragraph we show how we can transfer mechanical energy from one mass to another, by supposing that both masses are in movement. Then we make the link with oscillating system and we show how to use this idea for gathering the resonant conditions and transfer more mechanical energy from the vibration source to the oscillating system.

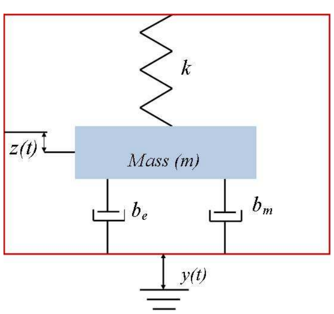

Figure 1: Equivalent electromechanical model.

With :

- z : Relative mass displacement
- y : Support displacement
- m : Effective mass of the system
- b_e : Electrical damper coefficient
- b_m : Mechanical damper coefficient
- k: Beam stiffness

3.2 Energy model

For a better understanding of the steps that led us to the present work, we propose to study what occurs when two solid bodies get in elastic collision.

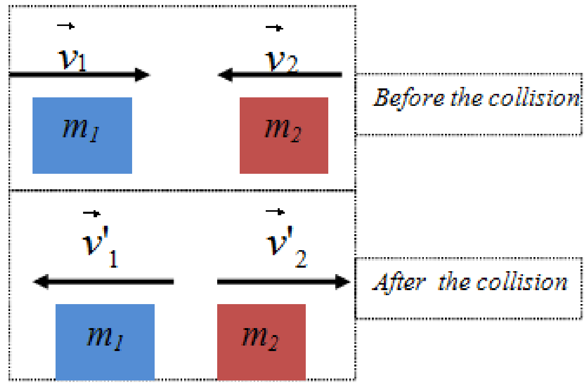

Figure 2: Illustration of the elastic collision between two masses

Consider two masses, m_1 and m_2, both masses displace before collision in two opposite direction with v_1 and v_2 their speed, respectively (Figure 2). The conservation of the momentum before and after the collision is expressed by the equations:

$$\begin{cases} \vec{p}_1 + \vec{p}_2 = \vec{p}'_1 + \vec{p}_2' \\ m_1 v_1^2 + m_2 v_2^2 = m_1 v'^2_1 + m_2 v'^2_2 \end{cases} \qquad (1)$$

With P_1 and P_2 represent the momentum of m_1 and m_2, respectively.
With v_1' and v'_2 represent the speed of m_1 and m_2 after collision, respectively.

Solving this equations system for v'_1 et v'_2 we get:

$$\vec{v}_2^{\,'} = -\vec{v}_2 + 2\vec{V}_I \qquad (2)$$

Where V_I represents the speed of the center of inertia of the whole moving system.

$$V_I = \frac{m_1 v_1 + m_2 v_2}{m_1 + m_2} \qquad (3)$$

By considering a mass m_2 much smaller than the mass m_1, the model of the center of inertia speed became:

$$V_I = v_1$$

We note from equation (2) that the mass will displace in an opposite way with a higher speed (v'_2) than this before collision. If we want to transfer more mechanical energy from the mass m_1 to the masse m_2; we should manage the collision when the speed of mass m_1 is at the maximum value.

Let's now establish the link with the mechanical resonators used in the field of mechanical vibration energy harvesting, the mass m_1 will represent the mass of the support, and m_2 will represent the seismic mass.

For a non damped collision, which endures during an infinitesimal time, one can obtain a theoretical speed gain of the seismic mass twice the speed of the mechanical support, (as explained in the previous mathematical model). In our case, the system hasn't the same nature of movement as the system shown in the described model, it's an oscillating system, therefore, by report to the model of the collision occurred by two masses that have a free movement, there are two extra important parameters that should be taken into account and added to the previous models. The first one is the resonant frequencies with and without blocking, the second one is the time of blocking, the dynamic study will take into account the relationship between these resonant frequencies, as well as the blocking time duration. These parameters are optimized in order to achieve a good efficiency and then a good transfer of the mechanical input energy towards the converter. It is more advantageous if the rebound takes place when the support speed reaches the maximum value, in order to give maximum transfer of energy to the cantilever (maximum gain of mass speed), but an intermediate speed can be satisfactory. In all cases it is necessary that the rebound takes place when the support moves in direction opposed to that of the moving part. Thus it is necessary to detect the direction of the displacement of the support; and the maximum support speed. Many approaches could be proposed for blocking the seismic mass, either by electrostatic mean, electromagnetic or piezoelectric, we propose to actuate the rebound by using piezoelectric actuators thanks to their accuracy, time of reaction, the low power consumption and the compatibility with our system.

3.3 Dynamic model

The aim of this part is to establish the temporal equation of mass displacement and speed in both situations blocked on and blocked off conditions. These equations will be used for modeling the rebound and then evaluated the displacement and speed gain of the seismic mass.
The most common way for studying such systems is to establish the equivalent electromechanical model based on a seismic mass attached to the vibration source by a spring and two dampers as shown in figure 1.
By applying Newton's law, one can develop the following differential equation that governs the present system:

$$m\ddot{z} + (b_e + b_m)\dot{z} + kz = -m\ddot{y} \tag{4}$$

Let's consider the following function of the mechanical support movement:

$$y(t) = Y.\sin(\omega t) \tag{5}$$

Hence the induced displacement is as follows:

$$z(t) = e^{-(\xi\omega_n t)}\left[A\cos\left(\omega_n\sqrt{1-\xi^2}.t\right) + B\sin\left(\omega_n\sqrt{1-\xi^2}t\right)\right] - \frac{Y_0\cos(\omega t+\varphi)}{\sqrt{(\omega_n-\omega)^2+(2\xi\omega_n\omega)^2}}$$

(6)

With: $\xi = \xi_e + \xi_m = \dfrac{(b_e + b_m)}{2m\omega_n}$ $\quad \omega_n = \sqrt{\dfrac{k}{m}}$

Where ***A*** and ***B*** are the integration constants, their values depend on the initial conditions, the initial conditions are computed whenever the stiffness get changed, and so the integration constants.

Theoric study has shown that the maximum gain is achieved when the blocking duration is equal to half the period of oscillation at the high resonant frequency (*1/2.*f_{r2}). By keeping the seismic mass blocked during the half period we allow maximum transfer of energy from the vibration source towards the resonant system. Two cases could be occurred. If the time blocked duration is lower than this optimal time, we transfer less energy from the source towards the resonant system, if it's higher; we restore to the source a part of the stored energy in the resonant system.

From these equations we can also deduce the frequency bandwidth covered by our system. The amplification gain falls when the input frequency is higher than half the resonant frequency at the blocking conditions, so we limit the badwdth from f_{r1} to $f_{r2}/2$.

In the next section, we present the implemented system used for the first validation of the present approach.

4 EXPERIMENTAL STUDY

4.1 Experimetal setup

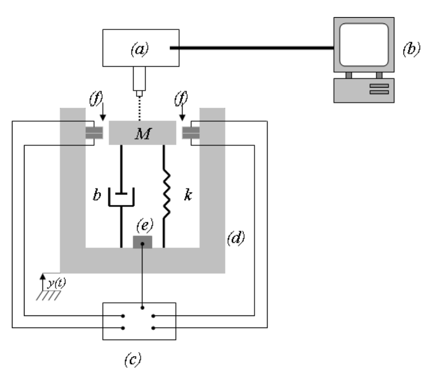

Figure 3: Experimental setup.

The figure 3 shows the experimental setup, the device *(a)* is a Laser vibrometer (type: LSV250) connected to a computer *(b)* in order to measure the displacement magnitude of the seismic mass. We measure also the acceleration of the support by an accelerometer, the acceleration signal is provided to the drive electronic *(c)*. This electronic deliver the driving signals for the actuators (APA400M) *(f)* for blocking on or blocking off the seismic mass *(M)*.

The fabricated structure is shown in the figure 4 followed by a brief definition of the different components.

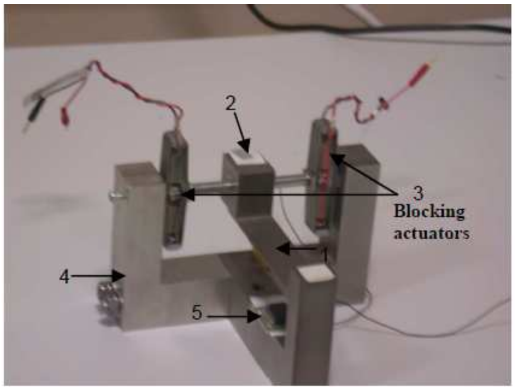

Figure 4: Designed structure.

Components	Definition
1	Cantilever made of stainless
2	Tip mass
3	Piezoelectric actuators APA 400M
4	Support fixed to the vibration source
5	Acceleration sensor

Table 1: Definition of the components.

In the table bellow, we show the main parameters of the structure, the theoritical and experimental ones. We note a slight divergence of the resonant frequency in blocking on conditions, this could be due to the effect of the actuators which add a not insignificant stifness to the system.

	Blocking on state		**Blocking off state**
Theory	200 Hz		45 Hz
Experiment	**Sweep up**	**Sweep down**	45 Hz
	190 Hz	175 Hz	

Table 2 : Main structure modes.

The figure bellow (Figure 5) shows the gain achieved by the present approach compare to results obtained when the cantiliver oscillates withoutblocking system. As expected by theory, the gain depends effectively on the input frequency, the gain is more important for frequencies much higher than the first resonant frequency. We note also that there is a difference between the theoretical expectation and the experimental results in terms of displacement gain; this could be due to the fact that in the theoretical study doesn't take into account the damping introduced by the actuators.

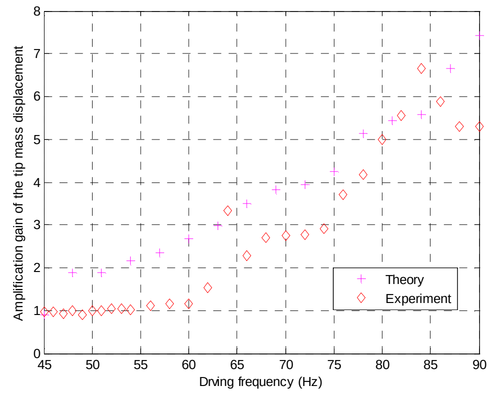

Figure 5: Displacement gain.

5 CONCLUSION

Within this paper, we've shown by theory and verified experimentally an original approach for amplifying the efficiency of selective electromechanical resonators without compromising their quality factor. We first gave an analytical model that describes and show the operation of the approach. A gain seven times greater than simple resonator was reached over more than one octave. Theory shows that output power is proportionel to square displacement of seismic mass, which means that with present approach we could achieve a power gain up to 49 at 90 Hz, which is a good result in the field of energy harvesting and active damping operating over a wide band of frequency. Our approach ensures a dynamic amplification of the relative displacement, and then the output power, without introducing a closed loop, but just a simple measure of the support acceleration that could be done by piezoelectric part of the structure. The next work will be alow power electronic integration for control in order to design a complete autonomous mechanical energy harvester with an increased bandwidth.

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NSTI-Nanotech 2011, www.nsti.org, ISBN 978-1-4398-7142-3 Vol. 1, 2011