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Strategies for Wideband Mechanical Energy Harvester

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1. Introduction

The energy harvesting market expands day by day, this is mainly due to the number of the implemented low power sensors in different fields, such as: human body, building, car engine...etc. These sensors are in most cases powered by batteries, but the main drawback of this technique is the need of a continuous control of their state of charge, the recharge and the replacement which is in most cases expensive. Thus, in order to overcome these limitations, one of the most promising solutions is to harvest the surrounding energy beside the system to power. In our environment, we can find many types of recoverable energy, for example: mechanical energy, thermal energy and radiative energy (solar, infra-red, radio-frequency). This chapter is dedicated to mechanical energy and more particularly to mechanical vibration energy produced by cars, fridges, mechanical engines and so on. The mechanical to electrical converter can be electromagnetic, electrostatic or piezoelectric. In case of an electromagnetic conversion, the vibrations are used to create a relative movement between a coil and a permanent magnet. In case of an electrostatic transduction, the vibrations are used to create a variable capacitance. In case of a piezoelectric transduction, the vibrations are used to apply a mechanical stress on a piezoelectric material. Actually, a vibration energy harvester (VEH) features 3 main components as presented in the Figure 1.

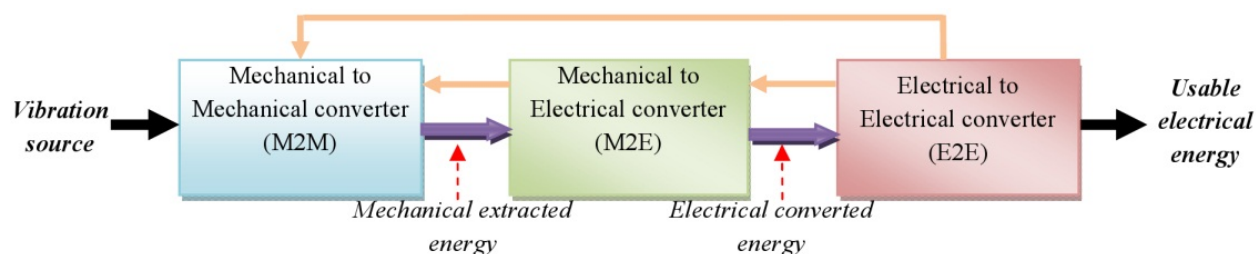


Figure 1. Vibration energy harvester conversion chain

The first stage of the conversion chain is a Mechanical to Mechanical (M2M) converter usually based on a mechanical resonator system. This converter translates the input vibration into a relative displacement between the resonator seismic mass and the vibration source. In addition, by using a resonant mechanism, the relative displacement amplitude can be larger compared to the vibration source displacement amplitude, increasing then the extracted mechanical power from the vibration source. Then, thanks to a dedicated Mechanical to Electrical (M2E) converter, which could be electromagnetic, electrostatic or piezoelectric, the amplified relative displacement is converted into electrical energy. Finally, an Electrical to Electrical (E2E) converter translates this electrical energy into a usable energy with a stable direct voltage able to supply an electrical circuit (3V for example). The efficiency of the harvester is tightly related to each stage of this chain. Moreover, as it can be noted from Figure 1, each stage has an effect on the other stages. Thus, the improvement of the VEH efficiency should take into account all the stages and also the relations between them. In what follows, more details are given for each stage.

1.1. The mechanical to mechanical converter

The first aim of this converter is to translate a vibration into a relative displacement able to actuate the mechanical input of the M2E converter. To make that, a seismic mass is required and the mechanical work that can be produced from the vibration is proportional to this mass and then to its size. This seismic mass is the main limitation in terms of power density capability for the main developed systems. In order to amplify the inertial effect of this seismic mass, it is necessary to use the resonance effect, which means using resonators. Actually, such devices are commonly modeled by a mass spring system connected to the vibration source and damped by the M2E converter and the mechanical losses. The efficiency of the M2M converter could be measured by the amplification gain of the vibration displacement amplitude (Q factor). However, the amplification gain is inversely proportional to the frequency bandwidth of the resonator making the system very sensitive to any change of the input vibration. This shift is commonly occurred especially in vibrations produced by car engine, in which case the frequency depends on the motor speed which is susceptible to change over time. In addition, the VEH resonant frequency is susceptible to change over time because of the aging of the materials. In fact, as the material of the harvester is subjected to a continuous mechanical stress, the mechanical stiffness will be altered during time and so the resonant frequency. To overcome this limitation many solutions have been proposed in literature, most of them are summarized in [1-2]. Actually, a few of these solutions allow an automatic adaptation without comprising the efficiency of the harvester neither the power balance of the VEH.

1.2. The mechanical to electrical converter

Once the vibrations are converted into an amplified relative displacement between two elements, this displacement is converted into electricity using electromagnetic, electrostatic or piezoelectric principle. The efficiency of this converter depends on its mechanical and electrical losses and also its good impedance matching with the mechanical source and electrical load. Many approaches have been developed in order to improve the efficiency of

this converter, these approaches depend mainly on the type of the converter, most of them are linked to the system size and the vibration source characteristics [3-5].

1.3. The electrical to electrical converter

The maximum of the extracted electrical power is achieved when the electrical converted power is equal to the mechanical dissipated power in the mechanical structure. However, the mechanical damping depends on the used material, while the electrical damping depends on the converted power (electromechanical coupling of the structure) and the output electrical impedance. When the vibration frequency and amplitude are known and fixed, one can design a harvester to fit the optimized conditions (in terms of resonance and damping forces). However, when the vibration magnitude changes, this equality cannot be satisfied any more since the damping forces (mechanical and electrical) have different variation profile when the vibration amplitude changes. Consequently, the VEH efficiency is decreased. At the present time, only one study has been done in this perspective, which means adapting in real time the damping forces in order to maintain an optimum point of electrical energy extraction [6].

This brief introduction highlights two improvement areas. The first one consists to ensure the tracking of the vibration frequency, while the second one consists to adjust in real time the electrical damping force with respect to the mechanical one. This chapter covers these areas of VEH efficiency improvement. At the present time, more works have been done to cover the first area of investigation than the second one.

The next part of this chapter gives an overview of the main approaches developed in the state of the art to ensure a vibration frequency tracking. These approaches are classified according to the type of vibration source: mixed frequencies vibration or vibration with a main frequency that changes over time.

2. State of the art

Many solutions have been proposed in the state of the art in order to overcome the system degradation related to the shift between the resonant frequency and the vibration one. The best way to make comparison between these solutions is to classify them according to the type of input vibration signal to which they could be subjected. Actually, two main types of vibration signals exist:

2.1. Vibration with multiple of harmonic at different frequencies

Basically, for such signals, the energy of the vibration signal is spread over a wide bandwidth. Hence, using a one degree of freedom resonator will not harvest the energy efficiently even if the resonant frequency is included in the bandwidth of the vibration signal. This kind of signal exists in staircases, buildings, train rails...etc. Solutions developed to extract the maximum of energy from such vibrations spectrum are based on systems with a wide bandwidth. Hereafter the main techniques developed in this issue:

2.1.1. High electrical damping systems

Despesse et al., [7] proposed an electrostatic converter with a high electrical coupling coefficient in order to broaden the resonance peak. The fabricated prototype is able to recover mechanical vibration below 100 Hz, with a global conversion efficiency of 60% at 50Hz. In fact, the main disadvantage of this structure is the quality factor of the converter, the resonance peak is broadened by increasing the electrical damping coefficient, the quality factor is then decreased and therefore, the quantity of the scavenged energy is relatively decreased when the input vibration frequency reaches the resonant frequency compared to a system with a high quality factor. To take advantage from this solution without decreasing permanently the quality factor and keep the same efficiency as a high quality factor system when the resonant frequency is equal to the vibration one, we should adjust in real time the electrical coupling and then the electrical damping. When the vibration input frequency is equal to the resonant frequency, the electrical coupling could be very low enabling a full resonant effect and when the vibration frequency shifts, the coupling could be increased to reach a higher output power. In fact, if the mechanical to electrical converter can reach a high electromechanical coupling, it is easy to temporarily decrease this coupling by mismatching the output electrical impedance.

2.1.2. Multi-modes systems

Shahruz et al., [8] proposed to expand the bandwidth using a multi modal structure. This structure is composed of several cantilevers. Each one has a defined resonance frequency. However, for a given vibration frequency, there is only one cantilever excited at its resonance frequency and all the others generate only a few amount of energy which limits the power density of the whole system. Another solution has been proposed by Roundy et al., [9] similar to the previous one, which consists on using a mechanical resonator composed of 3 different proof masses and four springs, they predict that the bandwidth could be multiplied by 3; nevertheless, the functionality of this system has never been experimentally verified. However, in both cases the power density of the converter is reduced since the harvested energy is proportional to the seismic mass, and in such cases only one cantilever works efficiently. Nevertheless, it can be interesting to use this technique to harvest a main vibration frequency and its harmonics that can be significantly separated in frequency but well known in advance.

2.1.3. Non-resonant system

Yang et al., [10] proposed another idea to broaden the VEH bandwidth, this idea consists to design a harvester where the effect of the air damping can be controlled. The idea is quite similar to the first one, except that for the present approach, the mechanical damping is increased instead of the electrical one, decreasing then significantly the mechanical extracted power. Actually, this solution is always less interesting than a high quality factor VEH, the bandwidth is in fact just increased because the mechanical damping limits significantly the output power when the vibration frequency fits the resonant frequency and not because it increases the output power outside the resonant frequency.

2.2. Harmonic vibrations

This is the most common type of vibrations, the energy of vibration is mostly concentrated around one determined frequency. However, there are two origins to the shift between the resonant frequency and the vibration one for such type of vibration:

- The vibration frequency is susceptible to change overtime.
- The change of the VEH materials properties because of the aging.

Hence, solutions developed in this issue trend to minimize the shift between the resonant frequency and the vibration one by tuning the resonant frequency. To do so, two types of approaches have been developed:

2.2.1. *Passive tuning of the resonant frequency:*

The passive way for tuning the resonant frequency means that the adaptation system does not need to be supplied by an external source of energy. Hereafter some techniques developed in this perspective:

- *Using non linear spring*

Marzencki et al., [11] has proposed an energy harvesting device employing the mechanical non linear strain stiffness using a clamped-clamped beam. For such systems the stiffness depends on the amplitude and the frequency of the vibration source. Hence, a well design of the structure could allow an adaptation between the vibration frequency and the resonant one. In this work, they report a tuning ratio of resonance frequency of over 36% for a clamped-clamped beam device with an input acceleration of 2g. This solution allows an efficient way to make a passive dynamic adaptation of the resonant frequency. However, there are some limits of this technique: a high resonant frequency tuning ratio requires a high acceleration. In addition, the frequency response of a non linear VEH has a hysteresis aspect, which means that when the frequency exceeds a specific value, the output power drops off dramatically, and it is not possible to come back to the previous point unless the vibration frequency decreases to the start frequency of the VEH. Furthermore, the principle efficiency is very dependent on the input vibration amplitude. For low amplitudes, there is a limited non-linear effect. Inversely, for large input displacements, the non-linear effect limits the relative displacement and then the output power.

- *Manuel tuning of the resonant frequency*

Leland et al., [12] suggests another technique for tuning the resonance frequency, it consists on applying *manually* an axial preload in order to change the resonance frequency and match the frequency of the vibration source. Using this technique, the developed system is able to adjust the resonance frequency of about 24% below its unloaded resonance frequency. This solution is viable only when the vibration frequency is known a priori and is not susceptible to change over time, since it allows fitting the right resonance frequency before implementation of the VEH.

2.2.2. Active tuning of the resonant frequency:

The active tuning means that the resonant frequency is adjusted in real time using an active process. The drawback of this technology is the power required to make the dynamic tuning of the resonant frequency. Hereafter, the main active tuning techniques:

- *Application of a force*

The application of a force means that the VEH is equipped with an actuator, magnetic [13], piezoelectric [14] or electrostatic actuator [15]. The main task of this actuator is to apply an additional force on the seismic mass in the same direction as the vibration one in order to affect the mechanical stiffness of the VEH (added or substituted a force to the spring force of the resonator). This force induces a change in the effective mechanical stiffness and then the resonant frequency. Among all the works made in this perspective only a few of them report a positive power balance between the output power delivered by the VEH and the energy required to adjust in real time the resonant frequency. Eichhorn et al., [16] have presented a smart and self-sufficient frequency tunable vibration energy harvester, they report a resonant frequency tuning ratio up to 26% and a consumption of the tuning system about 10% of the VEH output power. However, the system could perform the frequency adjustment only once a 22s. Another system proposed by Lallart et al., [17] based on an original approach that permits increasing the effective bandwidth by a factor of 4 in terms of mechanical vibration with a positive power balance.

- *Electrical load adaptation*

The idea of this technique is to adjust the resonant frequency by adjusting the value of the electrical load coupled with the VEH. This technique has been used for electromagnetic harvester. Since the stiffness is related to the electrical current in the coil, by adjusting the electrical load, the current could be changed and then the resonant frequency as shown in Figure 2, [18]. However, this approach requires too much power for the implementation, about 1000 times the power generated by the VEH.

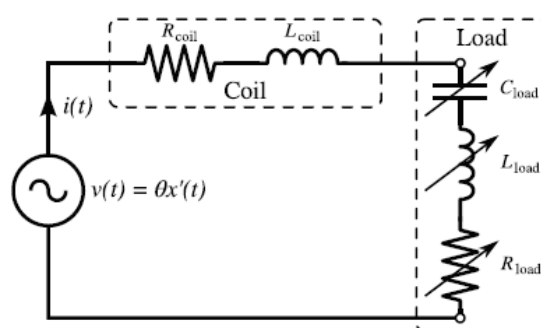


Figure 2. Method for adjusting the resonant frequency by adapting the electrical load principal scheme [18]

As it can be noted from this overview of the existed solutions, the main limit is either the lack of dynamic adaptability or the tuning ratio. To have a good dynamic of resonant frequency it is necessary to use active solutions. However, the main drawback of such

techniques is the negative power balance between the output power from the converter part and the power required to make the tuning of the resonant frequency. The works of Eichorn [16] and Lallart [17] show the possibility to develop wideband system with a positive power balance, their results are for great interest especially for autonomous system development. However, in both cases the positive balance is achieved by making a compromise with either the frequency of tuning adjustment or the tuning ratio. This brings us to conclude that it remains a large quantity of work to perform in this perspective in order to achieve a complete wideband harvester, able to be implemented in environment where the frequency vibration frequency is susceptible to change over time regardless the decrease of the output power due to the shift between the input frequency and the resonant one. Among laboratories interested by developing a viable solution using active technique there is CEA-Leti. Three different solutions have been developed in this laboratory. The next part gives details of each of these developed solutions, the first solution is based on the amplification of the generated relative displacement at off resonance, this solution is applicable for both types of vibration described before. The second and the third solutions are based on active tuning of the resonant frequency and are more applicable for the second type of vibration.

3. Solutions developed by CEA-Leti

3.1. Amplification of relative displacement at off resonance (rebound technique) [19]

In this part, a new approach for amplifying the relative movement of a cantilever system at off-resonance is presented. The aim is to broaden the resonance peak of resonators without compromising the quality factor of the system. The idea is to gather the resonance phenomenon conditions at off resonance. This could be done by adding a *rebound* mechanism to the VEH, when the speed of the vibration source reaches an extremum, the seismic mass is mechanically connected to the vibration source via a high stiffness spring, the movement of the seismic mass is then inverted and its speed is increased, which means a transfer of energy from the source to the VEH. This operation is called “*rebound*”. This approach is useful for VEH operating in environments where vibrations could be spread over a wide bandwidth or characterized by one main frequency susceptible to change over time. In the following, the principle is presented in details by giving the modeling and optimization approaches.

3.1.1. Description of the approach

The original idea of this approach is based on the principal of the elastic collision theory. When two solid bodies enter into collision there is an exchange of mechanical energy between these two bodies. After collision, the small mass goes in the opposite direction with a higher speed as shown in Figure 3. In the case of VEH, the small mass will represent the seismic mass of the harvester, while the big one will represent the mass of the vibration source.

The following calculations are intended to estimate the final energy of the seismic mass m_2 after the collision in order to compare it to its initial energy and the speed of m_1 before the collision. The objective is to deduce the energy gain of m_2 and the way to maximize it.

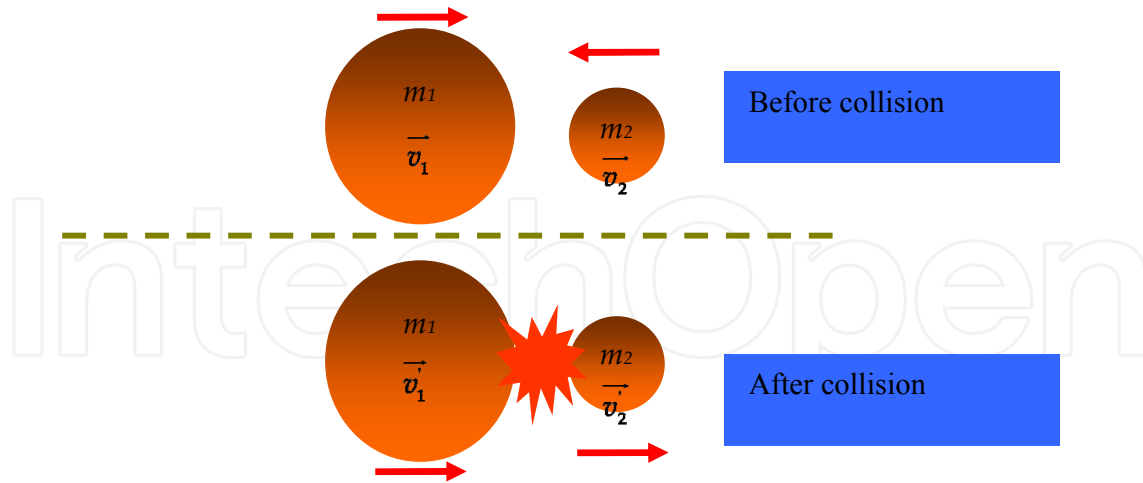


Figure 3. Illustration of the direct collision between two masses

The conservation of the kinetic energy before and after the collision leads to the following equations:

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

Where: v_1 and v_2 are the speed of the mass m_1 and m_2 , respectively, p_1 and p_2 are the quantity of movement ($m_i v_i$). This leads to the final speed v'_2 of the seismic mass:

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

Where V_I is the speed of the center of inertia of the whole moving system giving by the following equation:

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

By considering a mass m_2 much smaller than the mass m_1 the center of inertia speed becomes: $V_I = v_1$

It can be seen that the mass m_2 changes its movement direction after the collision and goes back with a higher speed (v'_2). A speed gain of twice the speed of the mass m_1 before the collision is reached. Considering the mass m_1 much greater than the mass m_2 , the achieved gain in terms of kinetic energy $E_1 - E'_1$ of the seismic mass m_2 is as follows:

Before collision:

$$E_1 = \frac{1}{2} m_2 v_2^2 \quad (4)$$

After collision

$$E_1' = \frac{1}{2} m_2 v_2'^2 = \frac{1}{2} m_2 v_2^2 + \boxed{2m_2 v_1^2 + 2m_2 v_1 v_2} \quad (5)$$

This energy gain is proportional to the square of the vibration source speed v_1 and to the product between the vibration source speed v_1 and the seismic mass speed v_2 . As soon as the seismic mass reaches a speed higher than the vibration source one, the energy gain becomes mainly related to the last term. Higher the initial speed (or initial energy) of the seismic mass is, higher the energy gain is, like a resonant mechanism on a half period.

This is the basic idea of the present approach. Let us investigate in more detail how it could be possible to implement this approach with a real VEH to amplify the movement of the seismic mass on a random type vibration source.

3.1.2. Application of the rebound mechanism for VEH

The process of rebound mechanism to implement with the VEH will help to extract more energy from the environment at off resonance for harmonic signals and even for random vibration. To understand the operating principle of this technique, we will look in more detail at the mechanical behavior of the VEH at or close to the resonance. Consider the equivalent model of a converter consisting of a spring with a stiffness k , attached from one side to a seismic mass m , and from the other side to the vibration source. The mass is also attached to the vibration source via damper as shown in Figure 4. At the resonance, the speed of the source and the effort imposed by the spring on the vibration source are in phase opposition. In other words, the source provides a mechanical work to cantilever and not the reverse. Hence, maximum of mechanical energy is transferred from the source to the resonant system (resonance phenomenon). The system is then able to extract more energy from the vibration source such that the relative displacement is larger (the force exerted by the spring is important), which means that the quality factor is high. However, at off resonance, the vibration source displacement and the effort imposed by the spring on the support are not synchronized, reducing then the average power transferred to the seismic mass. Hence, only a small amount of energy is absorbed by the mass spring system from the vibration source.

By considering the previous idea, more absorption of mechanical work from the environment at off resonance can be ensured by synchronizing a rebound of the seismic mass on the vibration source when the vibration source speed is in opposite direction with the seismic mass speed, the kinetic energy gain of the seismic mass is tightly related to the vibration source speed and to the initial seismic mass speed (as given by the equation (5)), it is more convenient to synchronize the rebound to a speed extremum of the vibration source. This will amplify the relative speed of the seismic mass and then the absorbed energy from the vibration source. The energy absorbed from the vibration source increases gradually, higher the seismic mass speed becomes, higher the extracted mechanical energy is. When the speed of the seismic mass is greater, the force exerted on the source during the rebound

is higher and then mechanical work extracted from the vibration source is higher too. Obviously, there is a physical limit to this amplification mechanism, this limit is fixed by the damping coefficient of the structure; a structure with a high quality factor will allow more amplification and vice versa, like for a resonance mechanism. To validate the principle, the rebound mechanism is introduced to a mass spring harvester system as shown in Figure 4. The spring k_1 represents the system guidance that has minimal spring stiffness. The rebound is applied by connecting mechanically the vibration source to the seismic mass via a large stiffness spring k_2 . This connection is ensured by using two actuators Act_1 and Act_2 , as shown in the figure below. When these actuators are activated, the mechanical stiffness of the resonant system is modified (from k_1 to $k_1+k_2 \sim k_2$). Hence, the system will feature two natural frequencies: f_{r1} , related to the stiffness k_1 when the spring k_2 is not connected and f_{r2} , related to the stiffness k_2 when the spring k_2 is connected.

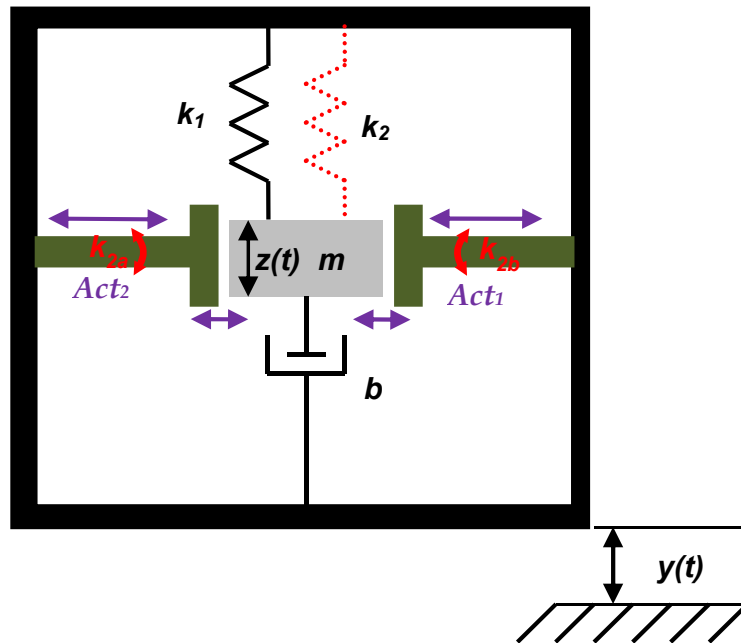


Figure 4. Equivalent VEH model with the rebound process

For a non damped collision, one can obtain a theoretical speed gain of the seismic mass twice the speed of the vibration source during the rebound, (as explained above). The next section gives an over view about the different techniques that could be used for actuating the rebound.

3.1.3. Rebound mechanism choice

Many types of mechanisms could be used for applying the rebound. This mechanism should be able to apply the rebound at any time by connecting on demand the seismic mass to the vibration source via a spring of stiffness k_2 significantly higher than k_1 . Hereafter the different types of mechanism that could be used:

- *Thermal actuation*

A current is applied in a thermal resistance when the vibration source speed reaches its maximum value; the thermal material expands and then makes a connection with the seismic mass. The main disadvantage of this technique is the reaction time of the material and its power consumption.

- *Electromagnetic actuation*

It is also possible to actuate the rebound by using an electromagnetic actuator composed of a coil and a core; such a method is used for breaking motors. When the coil is powered, the magnetic core moves and blocks the seismic mass. The main disadvantage of this approach is the power consumption which is relatively high compared to the power that can be scavenged (<1 mW for a centimeter scale device).

- *Piezoelectric actuation*

A third solution is to use a piezoelectric actuator enabling a short displacement with a high effort in a good agreement with the need to efficiently pinch the seismic mass. Furthermore the reaction time is very short compared to the blocking time (100 μ s to few ms) and its consumption is relatively low (capacitive mechanism).

The piezoelectric actuation has been adopted to implement the present idea thanks to its accuracy, time of response, low power consumption and its compatibility with the application. The piezoelectric actuators chosen are a linear actuators developed by CEDRAT Technology (APA400M) placed on each side of the seismic mass.

3.1.4. Simulations results

The time simulation diagram is presented in Figure 5. This diagram is composed of two working phases. The first one is used when the resonant frequency of the system is f_{r1} (rebound process deactivated), and the second phase is used when the resonant frequency is f_{r2} (the rebound process activated).

First, it is supposed that the system starts oscillating at a random frequency, different than the resonant one. The algorithm starts computing the speed and the displacement of the seismic mass. When a maximum speed of the vibration source is detected, the rebound is activated, the simulation phase is then changed, the simulation jumps to the second phase. This latter remains a certain period of time (the maintain of the connection of k_2 with the seismic mass). After that, the system goes back to the first phase (seismic mass connected to the spring k_1 only). This operation is repeated as often as the vibration source reaches some maximum speed, which means twice a period of vibration. The transition from one phase to another updates the initial condition of the new phase from the final position at the previous phase (initial displacement and speed, of both the seismic mass and the vibration source).

This simulation process has been used in different conditions in terms of vibration frequency, of rebound time duration and mechanical quality factor. After a deep investigation, some criteria have been established allowing an optimal amplification of the seismic mass's relative displacement:

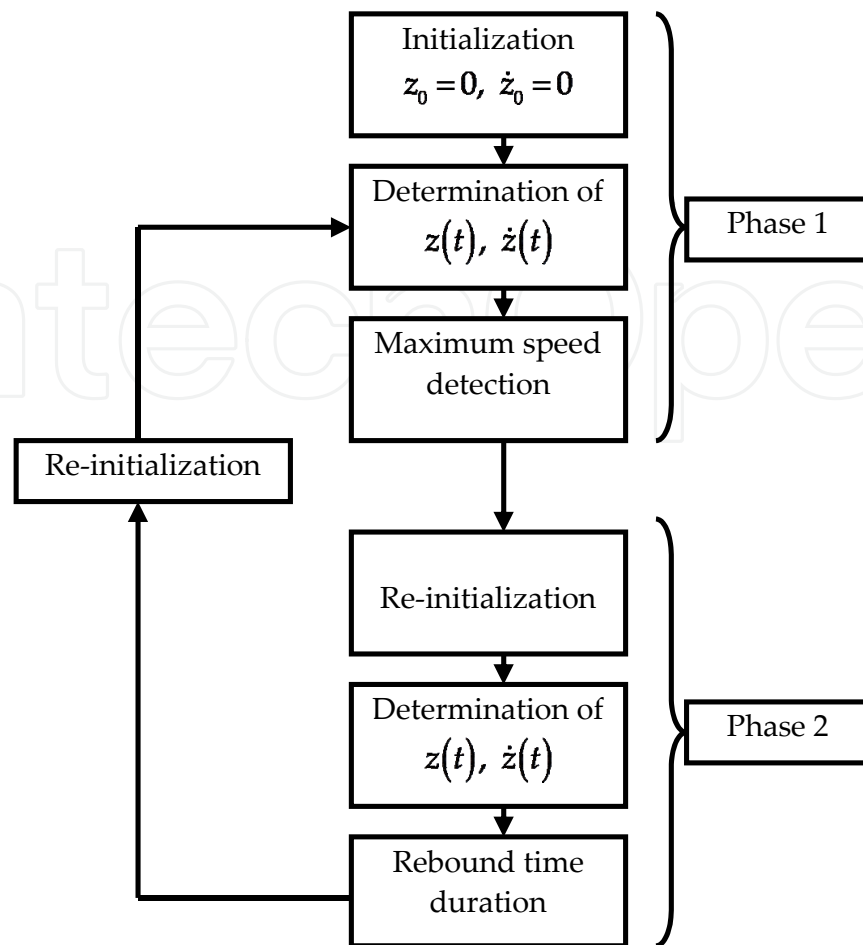


Figure 5. Rebound simulation diagram

- *The rebound time duration*

The simulation results show that the optimal rebound time duration is related to the resonant frequency f_{r2} occurring during the rebound time:

$$\Delta t = \frac{1}{2f_{r2}} \quad (6)$$

In fact, the rebound is a compression/decompression cycle of the spring k_2 , corresponding to the half of the resonant period occurring during the rebound time.

- *The second resonant frequency*

The simulation shows that the operating frequency bandwidth where there is a positive gain in terms of relative displacement is limited between f_{r1} and $f_{r2}/2$. Hence, the higher the distance between f_{r1} and f_{r2} is, the higher the bandwidth of the harvester is. To enlarge the operating frequency bandwidth, it is interesting to choose a large resonant frequency f_{r2} . However, a high resonant frequency f_{r2} implies short rebound duration, this will conduct to more difficulties to actuate the rebound. Thus, a tradeoff has been made between the bandwidth and the mechanical challenges to reduce the rebound time.

The first resonant frequency f_{r1} , while k_2 is open, has been fixed at 50Hz and the second one, during the rebound, at 200Hz.

The Figure 6 below presents the seismic mass speed amplification reached at the following conditions:

Parameter	Value
Input frequency	80Hz
Input acceleration	1g
The first natural resonant frequency (f_{r1})	50Hz
The second natural resonant frequency (f_{r2})	200Hz
Time rebound duration	2.5ms
The mechanical quality factor of the structure when the rebound process is activated	20, 30 and 40

Table 1. Simulation parameters for rebound system

The figure below shows the transient behavior of the seismic mass speed for different quality factor values. The maximal amplification is related to the mechanical quality factor. Hence, higher the quality factor is, higher the amplification gain is.

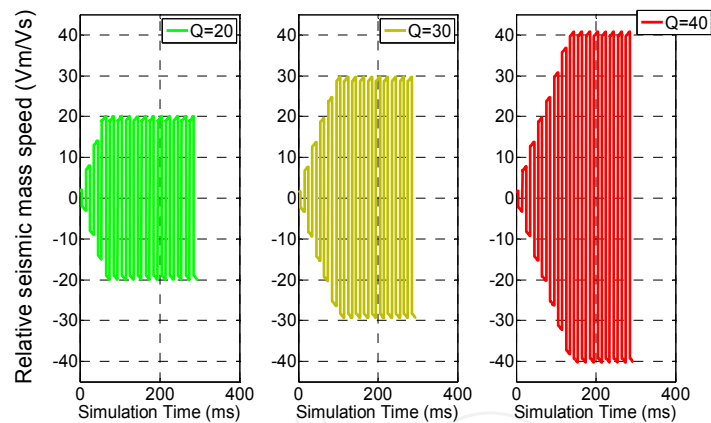


Figure 6. Seismic mass speed amplification (V_m : Seismic mass speed and V_s : vibration source speed)

The next section gives some details about the electronic attended to ensure the control of the actuators.

3.1.5. Drive electronic

The aim of this electronic is to deliver the command signals to activate and deactivate the actuators (act1 and act2). These signals are provided after measuring and processing the acceleration signal of the vibration source. Hence, the setting of electronic components is based on the following specifications:

- Output signal should be square, with an optimal duration time,
- The input vibration spectrum is comprised inside the system bandwidth (f_{r1} and $f_{r2}/2$).

The selected electronic architecture is composed of 5 stages placed in series:



Figure 7. Synoptic scheme of the selected electronic

- *Phase shift*

It is worth to remind that the rebound time duration is extremely low compared to the vibration period. The present approach needs then an extreme accuracy in terms time of activating/deactivating the actuators. However, the rebound is actuated after processing the acceleration signal. Hence, this operation will introduce a delay on the command signals. In order to overcome this limitation, a phase shift is added in order to make compensation to the delay introduced by the electronic. Hence, the information arrives to the last stage at the right moment.

- *Zero detection*

Detecting a vibration source speed extremum is equivalent to detect the zero acceleration of the input vibration. The second stage is then attended to detect zero acceleration on the vibration source. This is done by comparing the acceleration signal with zero using a comparator, the output voltage of the comparator changes from 0 to V_{cc} (for positive speed maximum) or from V_{cc} to 0 Volt (for negative speed maximum).

- *Pulse generation*

The third stage is attended to generate a pulse of an accurate duration each time the signal delivered by the previous stage change of state (rising or falling edge).

- *Power circuit*

The power circuit contains the switches to power from an external source to the actuators used for processing to the rebound.

The presented electronic was developed and tested with the mechanical system; the next part presents the experimental results

3.1.6. Experimental validation:

The experimental setup is shown in Figure 8 below. The cantilever is represented by its equivalent model, which is composed of a mass, a spring and a damper system. All these components are enclosed in the casing which is mechanically connected to the vibration source.

The device (a) is a Laser vibrometer (type: LSV250) connected to a computer in order to measure the displacement magnitude of the seismic mass. The acceleration of the vibration source is measured by an accelerometer (e), the measured signal is provided to the electronic

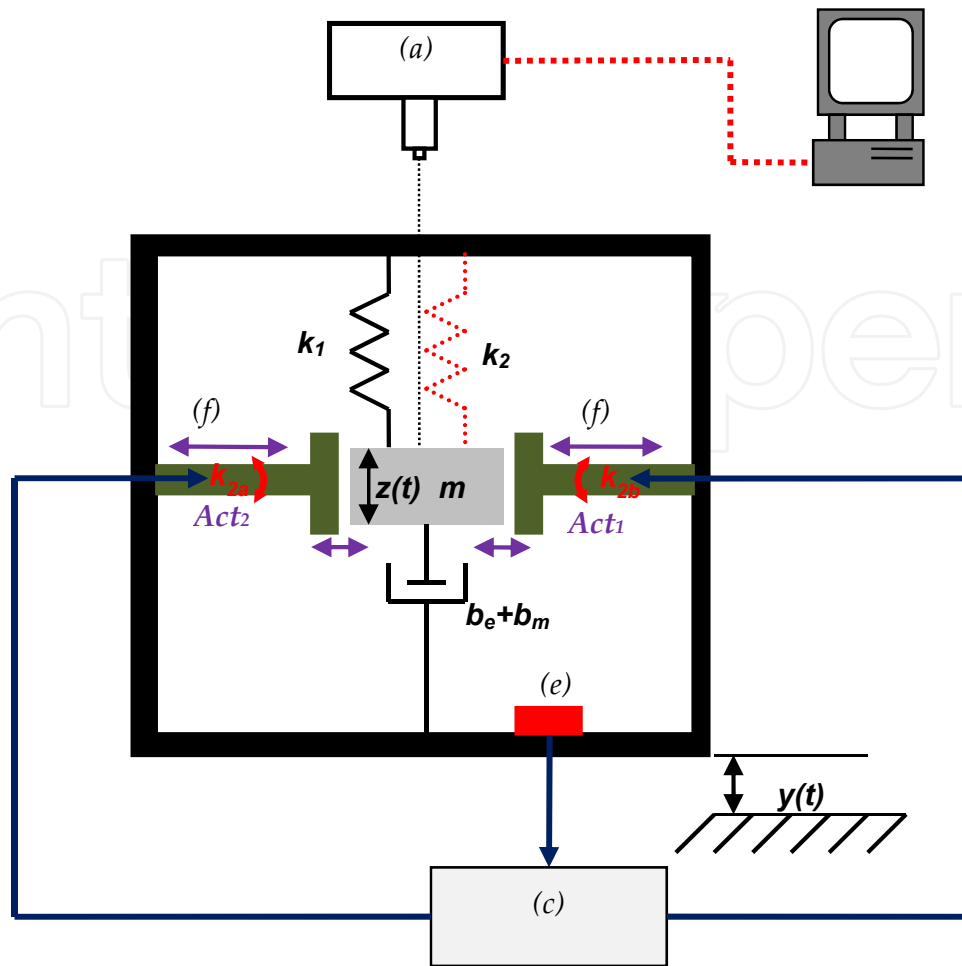


Figure 8. Experimental setup scheme for the rebound technique validation

(c) described in the previous section. This electronic processes the measured signal and generates the driving signals to the actuators (APA400M) (f) for connecting or disconnecting the spring k_2 relaying the seismic mass to the vibration source.

The manufactured structure is shown in the Figure 9 followed by a brief definition of the different components.

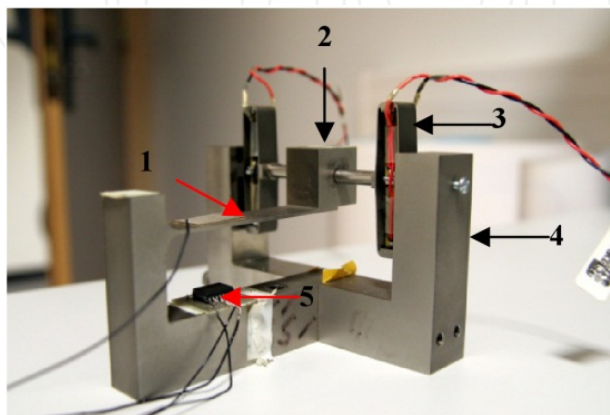


Figure 9. Picture of the fabricated structure

Components	Definition
1	Cantilever made of stainless k_1
2	Tip mass
3	Piezoelectric actuators APA 400M
4	The casing/support connected to the vibration source
5	Acceleration sensor

Table 2. Structure components definition

The Figure 10 shows the displacement gain achieved by the present approach as a function of the input vibration frequency compared to same system without the rebound mechanism. This gain is defined as the ratio between the relative displacement at off resonance obtained by using the present approach over the displacement obtained without activating the rebound when the maximum of speed is occurred. 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 because the amplitude of the relative displacement is close to the physical maximum since the vibration frequency is close the resonant one. This figure shows a difference between the theoretical expectation and the experimental results in terms of displacement gain due to the fact that in the theoretical study the damping induced by the actuators themselves was not taken into account.

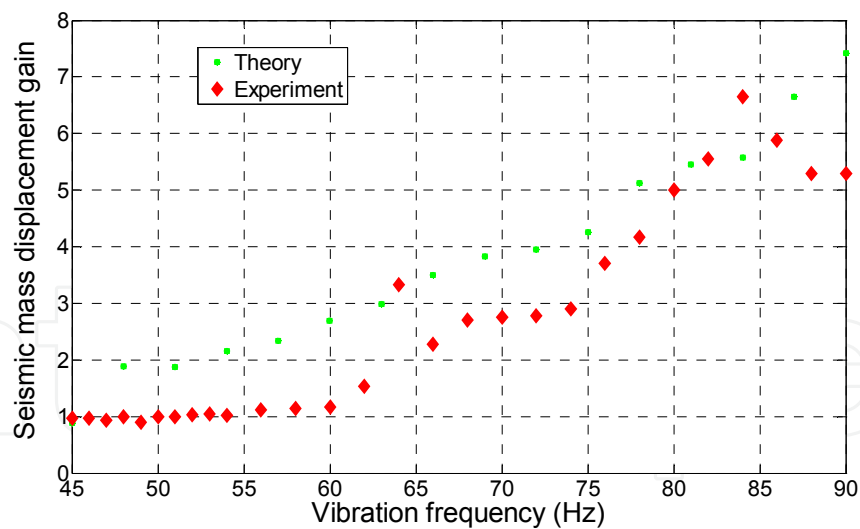


Figure 10. Experimental and theoretical results of the relative displacement gain achieved by the rebound technique

These results present a great advantage of the rebound technique for the increase of the VEH efficiency over a wide frequency band. The amplification of the seismic mass displacement allows more mechanical energy extraction when the resonant frequency is not equal to the input one. Nevertheless, the drawback that remains for the present approach is

a large power consumption to actuate the piezoelectric actuators. The energy required for each rebound is estimated at $200\mu\text{J}$. Further works are under investigation in order to reduce as low as possible this consumption.

3.1.7. Conclusions

In this part of the chapter, a new approach for amplifying the movement of the seismic mass at off resonance have been shown by theory and experiments. A relative displacement seven times higher than the one achieved with a single resonator at off resonant frequency (90Hz) was shown. This best gain occurs at twice the natural frequency of the structure. This relative displacement gain corresponds to an output electrical power gain equal to 49, which represents a good prospect in the field of energy harvesting. This technique ensures a dynamic amplification of the relative displacement in real time, with a high efficiency without the need of a control loop, a simple measure of the sign of the acceleration is sufficient to control the whole system. Nevertheless, the electrical consumption of the actuator applying the rebound is still too large to make the system completely autonomous. This first demonstrator validate the principle with an actuator over-sized, a significant reduce of its consumption promises a good perspective in this rebound technique.

If this rebound mechanism can be applied for a large number of vibration types (up to random vibrations), it is nevertheless interesting to inspect if other techniques, with narrower applications but easier to use, can be used to enlarge the frequency response. The next part presents two ways to follow a main vibration frequency that moves during the time.

3.2. Active tuning of the resonant frequency:

In the present section, two approaches for a dynamic tuning of the resonant frequency are given. These techniques could be applicable where the main vibration frequency is susceptible to change over time. It could be used for vibrations spread over a wide bandwidth as well, except that in this case the system will track only one main frequency. For both techniques given in what follows, the idea is to make a tuning of the resonant frequency by changing the stiffness of a piezoelectric material.

The most used structure shape for piezoelectric transduction in case of harvesting mechanical vibration is presented by the figure below:

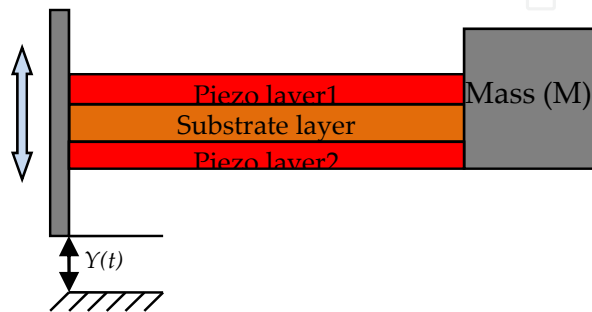


Figure 11. Piezoelectric cantilever shape

The structure is composed of three main components:

- *The substrate*: the substrate is usually added to piezoelectric harvester for two aims: enhancing the effective mechanical quality factor of the structure, removing the stress neutral line of the whole structure outside the symmetry axes of the piezoelectric part in order not to reduce the output generated power by electrical charges compensation.
- *The piezoelectric part*: stressed under mechanical stress, it converts the mechanical extracted energy into electrical one.
- *The seismic mass*: used to adjust the resonant frequency and enhance the amplitude of the relative displacement.

All these components are bounded together as shown in the previous figure.

One way to quantify the structure capability to change its resonant frequency is to estimate resonant frequency tuning ratio. For a cantilever based piezoelectric structure, the tuning ratio is given by the equation below:

$$\frac{f_{\max} - f_{\min}}{f_{\min}} = \sqrt{\frac{I_b + 2x_1x_0I_p}{I_b + 2x_0I_p} \frac{I_b \frac{L^3}{3} - 2x_0I_p \left(L^2L_p + \frac{L_p^3 - L^3}{3} - LL_p^2 \right)}{I_b \frac{L^3}{3} - 2x_1x_0I_p \left(L^2L_p + \frac{L_p^3 - L^3}{3} - LL_p^2 \right)}} - 1 \quad (7)$$

I_b and I_p represent the moment of inertia of the substrate and the piezoelectric part, respectively.

L and L_p represent the length of the beam and the piezoelectric layers, respectively.

$$x_0 = \frac{Y_{p\min}}{Y_b}, \quad x_1 = \frac{Y_{p\max}}{Y_{p\min}}$$

$Y_{p\min}$ and $Y_{p\max}$: the minimal and maximal value of the piezoelectric Young's modulus.

Y_b : the Young's modulus of the substrate

Equation (7) shows that the tuning ratio depends on the sizes of the structure, on mechanical material properties (x_0) and electromechanical properties (x_1). The choice of the piezoelectric material is based on its Young's modulus sensibility to the external conditions, a high sensibility will allow more change of the piezoelectric Young's modulus, and then a high tuning ratio.

3.2.1. Application of a DC electric field

3.2.1.1. Introduction

Mechanical and electromechanical properties of piezoelectric materials depend on external constraints. Among these properties, there is the effect of an applied electric field on the stiffness of the piezoelectric material. Thus, using a good piezoelectric material, in terms of

stiffness variation under a static electric field, a high resonant frequency tuning ratio could be achieved. Adjusting the level of the applied electric field will adjust the resonant frequency and hence a controllable resonant frequency VEH could be achieved.

3.2.1.2. Theory of the approach

The dependence that exists between the stiffness of the piezoelectric material and the strength of the applied electric field could be noted from the complete equation of piezoelectricity given by the following expression:

$$\varepsilon_{ij} = s_{ijlm}^E \sigma_{lm} + d_{ijn} E_n + \frac{1}{2} \tau_{ijlmnpq}^E \sigma_{lm} \sigma_{pq} + \frac{1}{2} a_{ijnr} E_n E_r + \kappa_{ijlmn} \sigma_{lm} E_n \quad (8)$$

This equation features two different parts, the first one relies the deflection to the stress by a constant parameter (s^E), while the second one shows a non constant coefficient of proportionality between the deflection and the mechanical stress, this non constant parameters depends on the applied electric field (E_n). This effect reflects the non linear behavior of piezoelectricity when it is subjected to a DC electric field. This effect varies from one type of piezoelectric material to another; it depends also on how the electric field is applied on the material in terms of strength and direction.

Equation (8) is the general piezoelectric equation, but considering our cantilever design, some assumptions can be taken into account: the mechanical behavior of the cantilever is elastic, only one component of the electric field vector of the stress and strain tensors are taken into account. Hence, this leads to the following simplified equation relating the stress to the strain and the applied electric field:

$$\varepsilon_x = s_{11} \sigma_x + d_{31} E_3 + \frac{1}{2} a_{113} E_3^2 + \kappa_{113} \sigma_x E_3 \quad (9)$$

This leads to the following expression of the mechanical stress:

$$\sigma_x = \frac{1}{s_{11} + \kappa_{113} E_3} \varepsilon_x - \frac{d_{31} E_3 + \frac{1}{2} a_{113} E_3^2}{s_{11} + \kappa_{113} E_3} \quad (10)$$

The expression relating the stress to the strain shows clearly the dependence that exists between the applied electric field and the stiffness of the piezoelectric material. However, it is difficult to determine the parameters appearing in this model, in most cases they are not provided by suppliers. Another simple equivalent model has been proposed by Thornburgh et al [20], reflecting the same effect by using a simple relation. This model is based on the linear constitutive equations of the piezoelectric material, we keep only the first and the second terms of the equation (9). Except that the effect of the DC electric field is reflected on the piezoelectric coefficient (d_{31}). The value of this one depends on the level of the applied electric field. The equation showing this dependence is given by (11), where the d_{31} is the piezoelectric strain coefficient at $0kV/cm$ and q_{31} is called piezoelectricity coefficient:

$$d_{31}^* = d_{31} + q_{31}\varepsilon_x \quad (11)$$

Using this relation, the stress function of the strain can be expressed as:

$$\sigma_x = Y_p' \varepsilon_x - \beta \quad (12)$$

Y_p' : is the expression of the piezoelectric effective Young's modulus as a function of the applied electric field and is expressed as follows:

$$\text{With } Y_p' = \frac{1 - q_{31}E_3}{s_{11}} \text{ and } \beta = d_{13}E_3$$

For a bimorph cantilever shape, the resonant frequency is then expressed as follows:

$$f_r \frac{1}{2\pi} = \sqrt{\frac{w}{4(M + 0.24M_b)L^3} (Y_p(1 - q_{31}E_3)(6t_s^2t_p + 12t_s t_p^2 + 8t_p^3) + Y_s t_s^3)} \quad (13)$$

The optimization of the tuning ratio involves in first step the choice of a piezoelectric material. This material should allow a high resonant frequency shift under a low applied DC electric field without compromising the efficiency of the mechanical-to-electrical conversion. Finding a piezoelectric material having a high electric coupling can induce a material with a low quality factor reducing the harvesting power, it is then important to find a material with a good compromise, we introduce then a new figure of merit for choosing the piezoelectric material taking into account the following parameters:

- *The electromechanical coupling coefficient k_{31} :*

It is necessary to choose a material with a high electromechanical coupling coefficient, this will improve the efficiency of the electromechanical conversion.

- *The maximum electrical field supported by the piezoelectric material ($E_{\max}$ and $E_{\min}$):*

As it can be seen from the expression of the resonant frequency (13), the highest the supported electric field is, the highest and the tuning ratio is. It is then important to take into account the limits of the applied electric field imposed by the piezoelectric material.

- *The coefficient of piezo-elasticity:*

The effect of the applied electric field on the stiffness of the piezoelectric material is described by the coefficient of the piezo-elasticity q_{31} as shown by equation (11). This means that a material with a high coefficient of piezoelasticity will provide a high resonant frequency tuning ratio.

- *Dielectric losses coefficient:*

The limiting factor of the present approach is the dielectric losses of the material. A material with a high dielectric losses coefficient presents more leakage current. This will induce higher power consumption for the management electronic of the system. It is then important to choose a material with low losses.

The figure of merit taking into account the different constraints above, can be expressed as:

$$\lambda_{p-1} = \frac{k_{31}^2 (E_{\max} - E_{\min})}{\tan(\delta)} q_{31} \quad (14)$$

After an overview of the most used piezoelectric material, it was found that the best material for this application is the PZN-PT, allowing the best compromise between the resonant frequency tuning and the electromechanical conversion. Despite the best performance of this material, it remains actually quite expensive compared to others.

3.2.1.3. The experimental validation of the approach:

- *The manufactured device*

The Figure 12 is a picture of the fabricated structure, it consists on a bimorph cantilever shape. The structure sizes and the main electromechanical properties are presented in the table below:

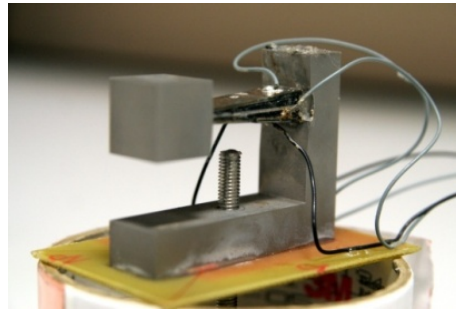


Figure 12. Picture of the fabricated structure

Parameter	Value
<i>Substrate material:</i>	
Y_b (GPa)	210
Density (kg.m ⁻³)	7500
Length x width x thickness (mm)	30x10x0.8
<i>Piezoelectric material (PZN-PT):</i>	
$Y_{p-\max}$ (GPa)	100
Density (kg.m ⁻³)	8800
k_{31}	0.9
relative dielectric constant (ϵ)	7500
Length x width x thickness (mm)	30x10x1

Table 3. Characteristics of the fabricated structure

- *The experimental results:*

The Figure 13 presents the relative displacement amplitude as a function of the input frequency for different applied DC voltage on the piezoelectric layers. It presents also the structure resonant frequency as a function of the applied DC voltage. A tuning ratio of the

resonant frequency up to 20% has been obtained by applying an electric field from -1 to +6 kV/cm. Theory shows that a tuning ratio of 26% should be reached in these conditions, this discrepancy is due to the errors introduced during the fabrication process. The bounding of the different layers has been done using an adhesive. The given theoretical model does not take into account the parameters of this adhesive, which explain for the main part the difference between the experimental and theoretical results.

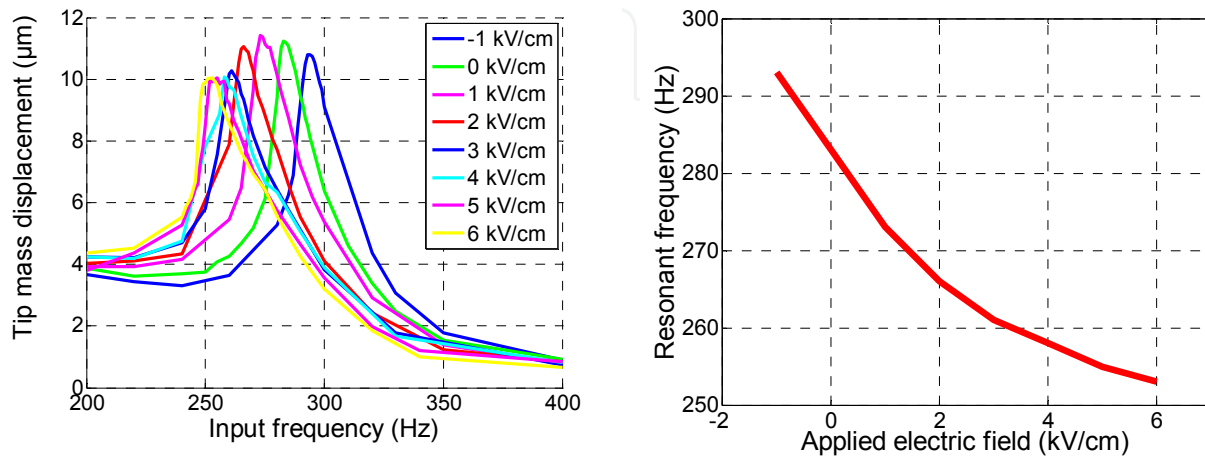


Figure 13. The effect of the applied DC electric field on the resonant frequency

The most important challenge while designing a VEH able to adjust its resonance frequency automatically, is the power balance between the converter output power and the power required to drive the frequency tuning. As explained before, most of developed wideband VEH have a negative power balance. In the next section, a new low power consumption electronic is proposed, this electronic is under development within the *CEA-Leti*, it allows a dynamic tuning of the resonant frequency by tracking the maximum output power point.

3.2.1.4. The resonant frequency tuning electrical circuit

The drive electrical circuit is composed of two principal parts, the power circuit and the control circuit. The first one allows the flow of the power between an electrical energy storage element and the piezoelectric material in order to apply a DC electric field, while the second one controls the level of the applied electric field according to the shift that exists between the vibration frequency and the resonant one.

The block diagram of the control circuit is given by Figure 14 below. The aim of this electronic is to determine how much is the resonant frequency lower (or higher) than the vibration one. First of all, this shift is determined by measuring the phase difference between the acceleration of the vibration source and the piezoelectric voltage. At resonance the phase shift between these two signals is equal to a quarter of a period, when the resonant frequency is higher than the vibration one, this phase shift is higher than this quarter of a period and inversely when the resonant frequency is lower than the vibration one. Thus, after measuring this phase shift, two cases may occur: (i) the phase shift is lower than a quarter of a period, this means that the voltage already applied on the piezoelectric material is higher than the

voltage that should be applied across the material ($f_r < f_{vib}$). (ii) the phase shift is higher than a quarter of a period, this means that the voltage already applied on the piezoelectric material is lower than the voltage which should be applied across the material ($f_r > f_{vib}$). In the first case, energy is transferred from the electrical energy storage element into the piezoelectric material, the switch k_1 is closed first, the close time of both switches should correspond exactly to the energy expected to be injected into the piezoelectric capacitance to reach the right DC voltage across the piezoelectric material. After switching on k_1 during the right time, the expected energy is stored into the magnetic core, it is then switched off and k_2 is switched on, the stored energy in the magnetic core is then injected in the piezoelectric capacitance and the voltage applied across the piezoelectric material attends its intended value. The process for the second case is the same as the first one, except in this case the energy transfer is made in the other direction in order to decrease the voltage across the piezoelectric material by closing first k_2 and then k_1 . The energy is then restored to the electrical energy storage. The power used to maintain the right voltage across the piezoelectric material is just the losses that occur during the power transfer in the fly-back converter.

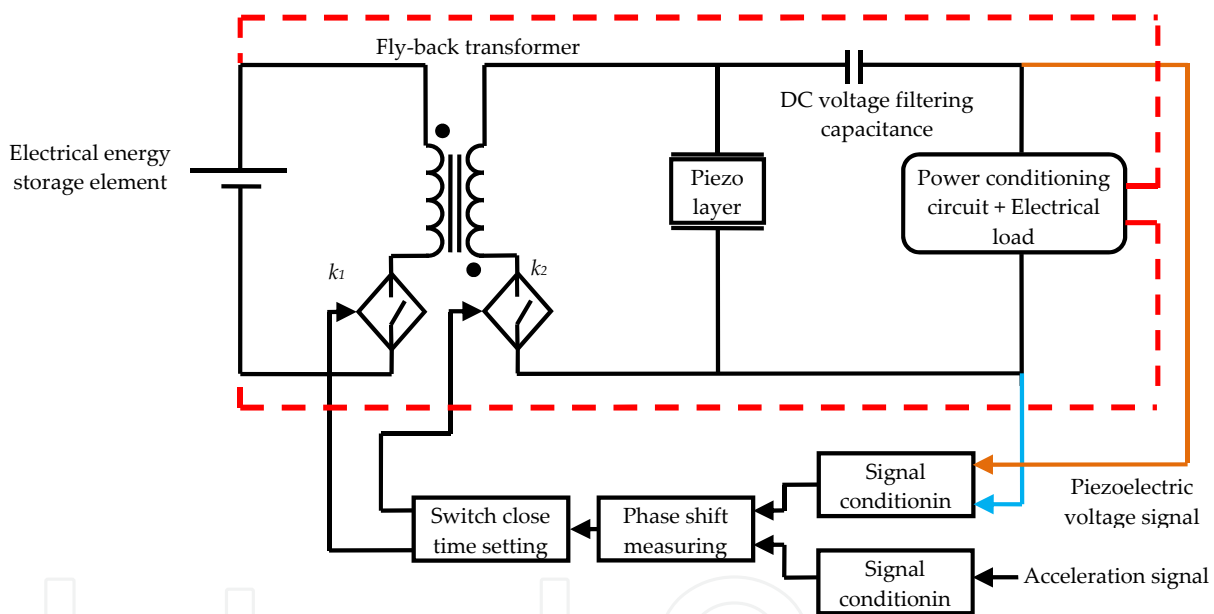


Figure 14. Resonant frequency adaptation circuit

3.2.1.5. The power balance

The Figure 15 below shows the power balance of the whole VEH including the resonant frequency tuning system. It is assumed that the resonant frequency is adjusted by step of 0.1Hz and it can be changed up to 10 times a second. The accelerometer consumption is not taken into account.

The mechanical, dielectric and electrical losses are investable, these losses are common for all VEH systems. The most important point shown in this power balance is the positive net output power. The whole system is self sufficient including the power required by the electronic for racking the vibration frequency and also the power needed to apply the right DC voltage across the piezoelectric material.

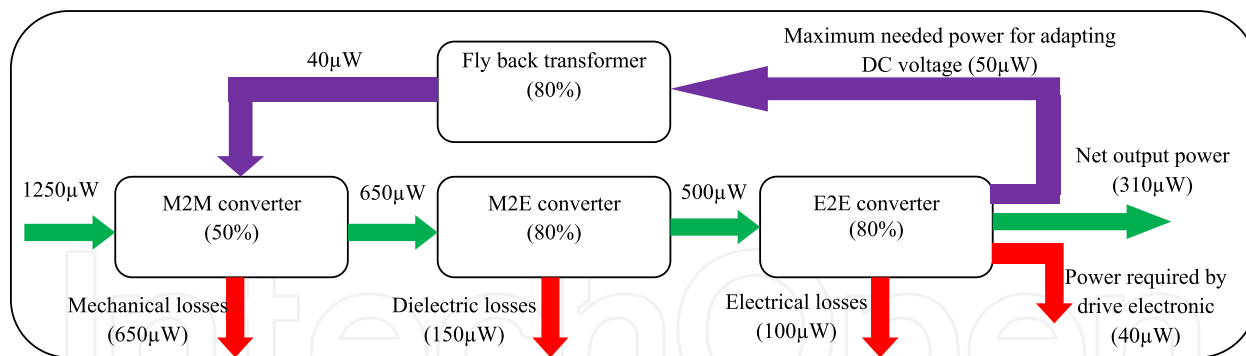


Figure 15. Complete power balance of the whole system

This result is very promising for the next generation of piezoelectric VEH, it shows that the real time resonant frequency tuning can be energetically positive. Nevertheless, the actual piezoelectric material which is in good agreement with this technique is still expensive (few 100€ per cm³) and their quality factor is quite limited (< 100).

In the next part of this section, another solution for a dynamic resonant frequency adjustment is given. This solution is quite similar to the previous one, it consists to adjust the electrical load connected to the piezoelectric material in order to adjust the electrical stiffness of the piezoelectric material.

3.2.2. Adaptation of the electrical load [21]

This subpart presents a declination of the previous solution for dynamic resonant frequency tuning. This one is based on the dependence that exists between the mechanical stiffness of a piezoelectric material and the electrical conditions to which it is subjected. In fact, the deflection of a piezoelectric bulk material under an applied mechanical force is higher when it is placed in short circuit (zero electric field) than in open circuit (no charge displacement), this means that the stiffness is lower in case of short circuit conditions than in open circuit conditions. The relation between the stiffness at short and open circuit is given by (14).

$$s^D = s^E - d^2 / \epsilon^T \Leftrightarrow s^E = \frac{s^D}{1 - k^2} \quad (15)$$

with

- s^D : Mechanical compliance of piezoelectric at open circuit conditions
- s^E : Mechanical compliance of piezoelectric at short circuit conditions
- d : Piezoelectric coefficient
- ϵ : Dielectric permittivity
- k : Electromechanical coupling coefficient.

One way to obtain a variable stiffness between these two limits (s^D and s^E) is to connect the piezoelectric material with an adjustable non-dissipative electrical load. However, this electrical load should not affect the quality factor of the structure. Thus, connecting a variable capacitor seems to be a good compromise, able to change the resonant frequency

with less power losses. The value of the capacitance set the effective dielectric permittivity of the piezoelectric material and then the piezoelectric material stiffness as it is shown by the equation (16), where Y_p : is the effective Young's modulus for the piezoelectric, C_p is the piezoelectric capacitance, C_{sh} the capacitance connected with the piezoelectric material, and A is the effective section of the piezoelectric material. Hence, by adjusting the value of the connected capacitance (C_{sh}), it is possible to adjust the value of the piezoelectric stiffness and then the resonant frequency.

$$Y_p = \left(s^E - \frac{d_{33}^2 A}{t_p (C_p + C_{sh})} \right)^{-1} \quad (16)$$

In case of a cantilever shape like in the Figure 11, the resonant frequency of the harvester becomes:

$$f = \frac{1}{2\pi} \sqrt{\frac{3 \left(2 \left(s^E - \frac{d_{31}^2 A}{t_p (C_p + C_s)} \right)^{-1} I_p + Y_b I_b \right)}{L^3 M}} \quad (17)$$

With M : the effective mass, t_p : the thickness of the piezoelectric layers and L : represents the beam length.

3.2.2.1. Choice of the piezoelectric material

As for the previous technique, to choose the suitable material we define a new figure of merit. This one is based on a compromise between the following parameters:

- *High electromechanical coupling coefficient k_{ij} :*

For the present approach, the electromechanical coupling has a significant effect on the resonant frequency tuning ratio as shown by the equation (14). It is better to choose a material with a high electromechanical coupling.

- *Low dielectric losses:*

In order to reduce as much as possible the power losses, it is necessary to choose a material with a low dielectric losses. But, unlike the previous approach, the coefficient of dielectric losses will have no effect on the power consumption of the power management electronic.

- *The effective permittivity:*

For the present approach, it is better to have a material with a high electrical capacitance, which means high dielectric permittivity because it minimizes the effect of the parasitic capacitances on the adjustment of the resonant frequency. If the parasitic capacitance is at the same order as the piezoelectric one, the variation of the shunt capacitance will have a minor effect on the tuning ratio of resonant frequency.

- *The coupling mode:*

As the vibrations are supposed to be straight and unidirectional, two modes could be used for ensuring an efficient electromechanical coupling, the longitudinal mode (polarization and mechanical stress axes are collinear), or the transverse mode where the polarization and mechanical stress axes are perpendicular. As the electromechanical coupling is higher in the longitudinal mode, this one enables a better tuning ratio.

Finally, by taking into account all these parameters and their effect on resonant frequency tuning, the figure of merit is:

$$FOM = \frac{k_{3j}^2 \cdot \epsilon_{33}^T}{\tan(\delta)}, j = 1, 3 \quad (18)$$

F.O.M(x1e3)	PZT	PMN-PT	PZN-PT
Transversal mode (31)	39.85	104.54	204.12
Longitudinal Mode (33)	136.96	447.7	618.52

Table 4. Comparison between the different piezoelectric materials

The table above shows the *FOM* of three different piezoelectric materials in two different modes:

It can be noted from this table that the PZN-PT in longitudinal coupling mode presents the best figure of merit and seems to be the suitable material for this method of resonant frequency tuning.

The next part presents the experimental results obtained with this technique on a structure prototype.

3.2.2.2. *The experimental validation:*

- *The manufactured structure*

The piezoelectric prototype developed by CEA-Leti is presented Figure 16. It consists on a bimorph piezoelectric cantilever, each piezoelectric layer is composed of a number of subparts mechanically bounded together in series. The polarization axis of each subpart is oriented on the direction of the resulted mechanical stress in order to work in the longitudinal mode. The electrodes of the subparts are connected in parallel, in order to obtain the highest equivalent capacitance.

This structure has been mounted on a shaker, the piezoelectric part has been coupled with a variable capacitance, and the obtained experimental results are presented in the next section.

- *The experimental results*

The first measurements show that the resonant frequency is equal to 208 Hz at short circuit condition and 294 Hz at open circuit condition, which represents **41% of tuning ratio**. The

Parameter	Value
<i>Substrate material:</i>	
Y_b (GPa)	210
Density (kg.m^{-3})	7500
Length x width x thickness (mm)	30x10x1
<i>Piezoelectric material (PZN-PT):</i>	
$Y_{p-\text{max}}$ (GPa)	100
Density (kg.m^{-3})	8800
k_{33}	0.94
relative dielectric constant (ϵ)	7500
Length x width x thickness (mm)	30x10x1

Table 5. Characteristics of the fabricated structure

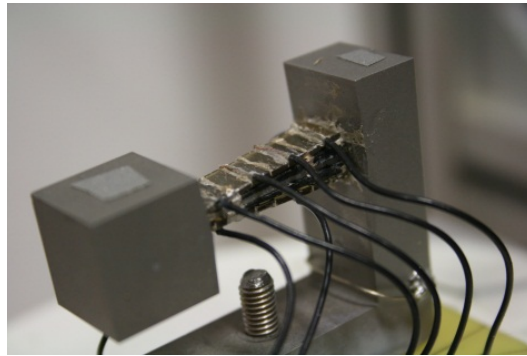


Figure 16. Picture of the fabricated structure

Figure 17 below presents the resonant frequency as a function of the shunt capacitance. The capacitance value is normalized to the blocked capacitance of the piezoelectric material (C_s/C_p). The blocked piezoelectric capacitance C_p is equal to 1nF.

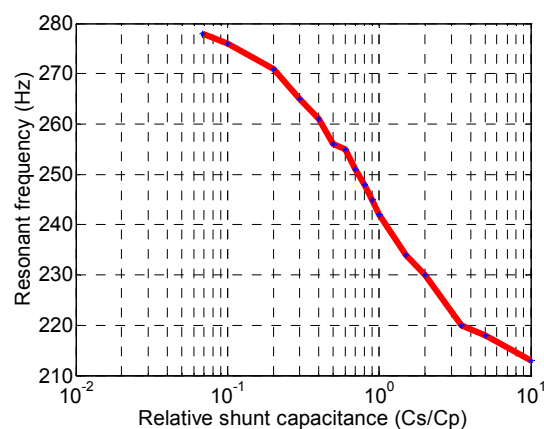


Figure 17. Resonant frequency as a function of the shunt capacitance

A tuning ratio up to 31% is noted when the shunt capacitance varies between $0.07C_p$ and $10C_p$. The power extracted using an optimal load is equal to $320\mu\text{W}$ for $0.1\text{g}@250\text{Hz}$.

3.2.2.3. Resonant frequency tuning drive electronic

As mentioned before, the ultimate goal of the work is to develop an automatic system able to adjust in real time the harvester resonant frequency to the main frequency of the vibration source. The idea here is to couple the VEH with an adjustable capacitive load. The block diagram of the drive electronic is given by the following figure. The VEH is first connected to the adjustable capacitive load composed of two capacitances, C_1 and C_2 . It is supposed that the power conditioning circuit requires a very low voltage ($\approx 3V$) compared to the piezoelectric output voltage ($>10V$). Hence, the equivalent shunt capacitance that affects the piezoelectric stiffness is the sum of C_1 with C_2 . Nevertheless, the capacitance C_2 enables a power transfer from the piezoelectric material to the power conditioning circuit and has more effect on the extracted electrical energy and then it enables an adjustment of the electrical damping.

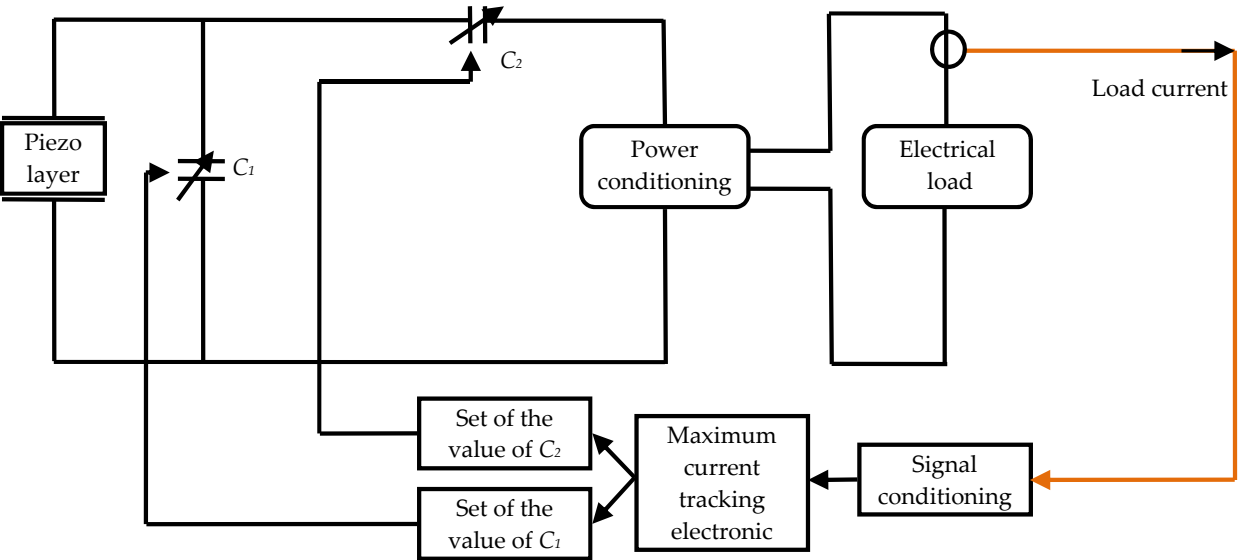


Figure 18. Block diagram circuit for the resonant frequency tuning by adapting the electrical load

The objective of the drive electronic is to track the maximum output power flowing through the electrical load by adjusting the electrical damping (adjusting C_2) and the resonant frequency (adjusting C_1 and C_2). Hence, the reaction time of a change of C_2 is shorter than a change of C_1 , since the first has an effect of the transfer of energy while the second one on the resonant frequency. This electronic is composed of two loops, a slower one for adjusting the capacitance C_1 and a faster one to adjust the capacitance C_2 .

The first measurements show that the whole electronic consumes about $30\mu W$ which represents only 10% of the $300\mu W$ generated power.

4. Conclusions

Through this chapter, it has been demonstrated that there is a real need to ensure a real time tracking of the vibration frequency. The issue of the adaptation between the input vibration characteristic and the mechanical characteristics of the VEH is capital for the development of

robust and efficient VEH. Within this chapter, a special focus has been made on the solutions developed in *CEA-Leti* laboratory. Three solutions have been presented, the first one expects to amplify the relative displacement of any vibration type (random) occurring on a wide width frequency bandwidth, the second one consists to apply a DC electric field on the piezoelectric layer in order to adjust its stiffness and then the resonant frequency of the structure. The third solution consists to adjust the electrical load coupled with the piezoelectric material in order to adjust its stiffness and then the resonant frequency of the structure. Each solution has been validated experimentally, the first one enables an operation over one octave of frequency (50Hz to 100Hz), the second one a frequency tuning from (250Hz to 300Hz) and the last one from (210Hz to 280Hz). This chapter introduces also the drive electronic for each strategy, the drive electronic part is the most critical point for active techniques since it requires in most cases a huge amount of energy. First results obtained in CEA-Leti laboratory show that the resonant frequency tuning can significantly increase the net output power without consuming too much power to be managed (about 10% of the converter output power). Anyway, the successful of VEH systems is tightly related to the frequency bandwidth. So, the commercialization of such systems at large scale could not be done until overcoming efficiently the limits imposed by the bandwidth issue. Active techniques allowing positive power balance let us hope a large flexibility in the near future of VEH.

Author details

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