High performances low frequency vibration energy harvester with HSLD stiffness

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Abstract. The contributions put forth by this paper first include the design and testing of a new high performance electromagnetic vibration energy harvester based on an optimized electromagnetic structure and a friction-less compliant folded-beam suspension. Then, High Static Low Dynamic (HSLD) stiffness principle is used to reduce the resonance frequency of the harvester while maintaining small static displacement.

1. Introduction
The Internet of Things (IoT) is the term used to describe a cluster of technologies enabling machine to machine (M2M) communication and machine to human interactions through the Internet [1]. Among those technologies, Wireless Sensor Networks (WSN) is a fundamental element. It is composed of wireless units able to sense a given physical phenomenon and cooperate to route the gathered data to a processing center. In some cases, these units can not be subject to maintenance on regular basis and there is a need to guarantee the energetic autonomy without battery replacement nor access to the power grid. Energy harvesting aims to benefit from the energy that can be found in the direct surrounding. Considering the various available sources (solar radiation, thermal gradient, mechanical movement, electromagnetic waves, etc.), vibration energy harvesting is a promising one [2].

Low frequency, vertical vibrations can profusely be found during road/rail transportation [3] or human motion [4]. In fact, those vibration sources’ most energetic frequencies usually lie from 1 to 5 Hz. Scavenging energy in that frequency range implies large displacements (static and dynamic) which tends to increase the difficulty to build a vibration energy harvester which is both compact and efficient. Different solutions have been proposed to overcome this issue, among them we can cite frequency-up conversion [5], parametrically excited pendulum [6] or magnetic suspension [3]. However, each of these solutions has its own drawback which makes difficult an industrial application, respectively wear, amplitude threshold and high static displacement.

In this study, we propose a new electromagnetic harvester based on High Static Low Dynamic (HSLD) stiffness solution enabling efficient, space-limited and very low frequency vibration energy harvesting. Our approach relies on two steps; the first one is the design and testing of a high performances linear vibration energy harvester and the second one is the modification of the prototype to obtain HSLD stiffness behavior.
2. Linear vibration energy harvester

2.1. Theory

Vertically oriented, the behaviour of a single degree of freedom linear electromagnetic vibration harvester loaded with a resistive impedance $R_{\text{load}}$ can be described by equations (6) and (2).

\begin{align*}
 m\ddot{x} + c_m\dot{x} + kx + \alpha \cdot i + mg &= -m\ddot{y} \\
 L\frac{di}{dt} + (R_{\text{int}} + R_{\text{load}}) \cdot i &= \alpha \dot{x}
\end{align*}

Where $m$ is the moving mass and $x$ its relative position to the base, $c_m$ the mechanical damping coefficient of the oscillator, $k$ the suspension stiffness, $g$ the gravity constant, $y$ the base excitation, $\alpha$ the electromechanical coupling term, $L$ and $R_{\text{int}}$ the electromagnetic coil inductance and impedance.

A widely accepted assumption is to neglect the coil inductance $L$. Moreover, we decide to introduce the number of turns in the coil $N$: the coil resistance $R_{\text{int}}$ is then equal to $N^2 R_0$ and the coupling coefficient $\alpha$ to $N \gamma$. $R_0$ and $\gamma$ are respectively called intrinsic resistance and coupling and only depend on the geometry and materials used in the magnetic structure (coil and magnets).

Considering that the harvester is subjected to a sinusoidal excitation $A\sin(\omega_0 t)$ tuned around its resonance frequency $\omega_0 = \sqrt{k/m}$ and that impedance matching is satisfied (equation (3)), the mean $P_{\text{mean}}$ over time of the instantaneous harvested power $P = R_{\text{load}}i^2$ is given by equation (4).

\begin{align*}
 R_{\text{load}} &= R_{\text{int}} + \frac{\alpha^2}{c_m} \\
 P_{\text{mean}} &= \frac{m^2 A^2}{8c_m} \cdot \frac{\gamma^2}{R_0 c_m} \cdot \frac{1}{1 + \gamma^2 / R_0 c_m}
\end{align*}

First, we observe that the harvested power is theoretically independent of $N$, in fact $N$ does the balance between current and tension: if $N$ is big, the harvester tends to output higher voltage and lower current.

Also, $P_{\text{mean}}$ is limited by $m^2 A^2 \cdot (8c_m)^{-1}$, to approach this limit, the harvester must be designed to maximize the term $\gamma^2 \cdot (R_0 c_m)^{-1}$.

Finally, it is worth noting that when excited, the moving mass oscillates around a non-zero equilibrium position $x_0$ such as:

\[ x_0 = -\frac{g}{\omega_0^2} \]

When harvesting high frequency vibration, this behavior can be neglected. However, dealing with very low frequency vibration, high $x_0$ values can lead to a large increase of the harvester volume and must be considered. For example, $x_0$ equals to 2.8 cm at 3 Hz.

2.2. Design

To assess the design rule extracted from theory (mainly maximizing $\gamma^2 \cdot (R_0 c_m)^{-1}$), we decide to design a linear vibration energy harvester not accounting for the constraint of harvesting the very low frequencies.

The optimized design process is based on two steps, the first one is to choose and size the suspension and guiding of proof mass. This part not only has an influence on the mechanical damping $c_m$ but also on the intrinsic coupling $\gamma$: it will be high if the airgap between magnet and coil is small and, this airgap can be minimized if the guiding has sufficient lateral stiffness to allow rectilinear motion. Among the numerous configurations available, we chose to use a
folded beam suspension which allowing highly rectilinear motion together with linear stiffness; furthermore, to allow high displacement and low damping, it has been machined out of a single piece of Beryllium Copper.

The second test is to design the electromagnetic coupling structure. We chose to base our optimization on the work done by Spreeman and Manoli [7] and their “magnet across coil with back iron” structure shown on Figure 1. Within a given volume of $35 \text{ mm} \times 40 \text{ mm} \times 50 \text{ mm}$, the objective is to maximize $\gamma^2 \cdot (R_0)^{-1}$ on a sufficient stroke of the moving mass while avoiding magnetic flux leakage. The main challenge being to find the right division between back iron, coil and magnet. Using the finite element software called FEMM, the final structure is made out of soft iron, Neodymium Iron Bore magnets and a copper coil counting 1000 turns (to obtain sufficient voltage). The equivalent scheme of the final prototype is given on Figure 2.

2.3. Experimental testing

The linear prototype has been tested on a high force shaker and several parameters have been identified. It has a resonance frequency of 15.3 Hz, the friction-less suspension gives a mechanical damping as low as 0.06 %. The coupling coefficient $\alpha$ is equal to 11.15 N.A$^{-1}$ and the internal resistance $R_{int}$ to 28 Ω. These characteristics give great energy harvesting performances, mostly when the excitation amplitude is low: for an excitation of 5 mg$^{\text{RMS}}$, the output power is 2.7 mW which correspond to 169 mW.g$^{-2}$cm$^{-3}$. Comparing it to some harvesters of the literature [8] (Figure 3), the great performances of our prototype can be assessed.

3. HSLD stiffness vibration energy harvester

Our linear prototype shows good performances but does not comply to our objective consisting in harvesting the very low frequency. Lowering the resonant frequency (e.g. by decreasing the stiffness) would lead to a high static displacement not withstanded by the folded beam suspension. The solution that we introduce here is to use a HSLD stiffness.

3.1. Theory

Oscillators with HSLD stiffness is a technique used by vibration isolator designers [9]; it consists in adding a negative stiffness component to the linear oscillator around its equilibrium position to lower the dynamic stiffness while maintaining the static stiffness. Equation (6) becomes:

$$m \ddot{x} + c \dot{x} + kx - k_1(x - x_0) + k_3(x - x_0)^3 + \alpha \cdot i + mg = -m \ddot{y}$$

(6)
Hence, the harvester keeps its static stiffness $k$ and its equilibrium position $x_0$. The dynamic stiffness is given by $k - k_1$ and the resonant frequency is $\sqrt{(k - k_1)/m}$. By correctly sizing the negative stiffness, the resonant frequency can therefore be tuned to the desired frequency.

3.2. Design
To add the negative stiffness to our linear prototype, we chose to use two parallel bistable buckled beams. The buckled beams machined out of Beryllium Copper sheets using EDM technology, they have been sized using a model by Cazottes et al. [10] based on the minimization of the internal energy of the beams. The equivalent scheme of the obtained prototype is given on Figure 4.

![Figure 4. Vibration energy harvester with HSLD stiffness.](image)

3.3. Experimental testing
Like the linear harvester, the HSLD harvester has been tested; the resonant frequency has been successfully decreased down to 5.2 Hz while maintaining the equilibrium position of the linear harvester. However, the mechanical damping is increased to 1.15 % (multiplied by 19). As could be expected, for the same excitation amplitude 5 mg$_{RMS}$, the harvested power falls to 252 μW (15.6 mW.g$^{-2}$.cm$^{-3}$). Refering on Figure 3, the HSLD prototype still gives good performances but in the same range as litterature.

4. Conclusion
While the designed linear harvester prototype shows high performances above its counterparts from the literature, it could still be improved by decreasing its volume. Moreover, tests using electronic circuitry to store the scavenged energy may be done. About the HSLD harvester, even if it shows good performances at low frequency and the concept has been proven, work is still to be done to further decrease the resonant frequency while limiting the impact on mechanical damping.

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