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## Analytical Investigation of Frequency Dependence of Average Power of a Vibration Energy Harvester

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**Abstract:** Realization of self powered wireless sensor network has become possible with the recent development in energy harvesting and micromachining techniques. The versatility of vibration energy makes it the most attractive power source for wireless sensors. In several environments, vibration energy is distributed in few harmonics. To effectively scavenge ambient energy, one has to choose the harmonic to which the harvester should be tuned. This paper analytically investigates dependence of the average power of a vibration energy harvester on the frequency of vibration. It is shown that the cubic dependence of the average power on the frequency of vibration holds only when the electromechanical coupling can be increased with the frequency. For a constant electromechanical coupling, the power increases with the fourth power of the driving frequency. When the magnitude of the base acceleration is constant, a high frequency harmonic will give more average power density than a low frequency one. *Copyright © 2012 IFSA.*

**Keywords:** Energy harvesting, Energy scavenging, Analytical analysis, Power MEMS, Wireless sensor network.

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### 1. Introduction

With the fast development in the micromachining techniques, micro-electro-mechanical (MEMS) devices are rapidly replacing conventional large-scale devices. Using the modern micro-fabrication techniques, such as CMOS-MEMS, a MEMS sensor can be monolithically integrated with signal processing circuit. Highly miniaturized devices, fabricated using these techniques, are less costly and consume very low power. Such a sensor with an incorporated wireless radio can be used for wireless

monitoring of structural health, patients status, and environmental conditions. When a wireless sensor is embedded inside the body of a structure or a patient, periodic replacement of the battery may prove dangerous for the host. In addition to that, for a large wireless sensor network, consisted of a large number of wireless sensors or spread over a large geographical area, exhaustive batteries are less suitable due to the huge cost involved in their replacement and disposal. Therefore, a large research interest in energy harvesting to power up wireless sensors has developed in recent years. Among many types of renewable energies, vibration energy harvesting has got a particular research focus due to its abundance in the target environment of the wireless sensors. For a detailed review of the techniques developed in energy harvesting, the reader is referred to the references in [1–4].

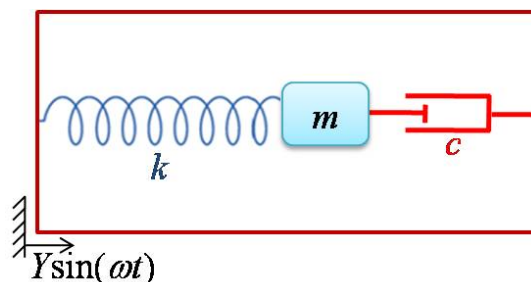
Typically, a vibration energy harvester is represented by a mass-spring-damper equivalent model whereby the linear damper represents the combined damping offered by the electrical and mechanical domains [5]. On the basis of this simple model, many researchers such as William and Yates [5], Mitcheson et al. [6], Roundy et al. [7] and N. Stephen [8] have contributed in the understanding of the behavior of a an energy harvester. Among these, the work of William and Yates, Mitcheson et al., Roundy et al. were more focused on quantification of the average power in different environments, whereas, N. Stephen investigated the role of the damping in detail and highlighted some misleading conclusions, relating to the role of the damping, drawn by some authors.

In many scenarios, energy in ambient vibrations is distributed in different harmonics. For example, the vibration in a vertical stabilizer on a PZL SW-4 helicopter occur with dominant harmonics at 30, 45 and 90 Hz with a peak acceleration value of 15.4, 8.6 and 1.5  $m/s^2$ , respectively [9]. To effectively harvest energy from such sources one need to tune the harvester with one of the dominant harmonic. Therefore, it is deemed necessary to investigate the dependence of the average of the harvester on the frequency of vibration. The aforementioned investigation has been carried out analytically in this paper.

## 2. Theory

A vibration energy harvester is typically represented by a mass-spring-damper equivalent model, as shown in Fig.1 [5]. In the mentioned figure, the dashpot  $c$  represents the damping offered by the mechanical and the electrical domains, the mass  $m$  represents the effective proof mass and the spring of constant  $k$  represents the elasticity of the beam of resonator. A linear damper representation of the electrical damping is fair for electromagnetic transduction and less so for piezoelectric transduction [8] while, electrostatic transduction is best represented by Coulomb force damping. When the 1dof mass-spring-damper system of Fig. 1 is acted upon by a sinusoidal excitation  $Y\sin(\omega t)$ , a reciprocating motion is setup in the seismic mass which is governed by the equation [8]

$$m\ddot{z} + c\dot{z} + kz = m\omega^2 Y \sin(\omega t). \quad (1)$$



**Fig. 1.** Mass spring damper equivalent of a base excited, vibration energy harvester.

The steady state solution of (1) is given as [8]

$$z = \frac{m\omega^2 Y}{\sqrt{(k-m\omega^2)^2 + (c\omega)^2}} \sin(\omega t - \varphi), \quad (2)$$

where the phase difference  $\varphi$  is given as

$$\varphi = \text{Tan}^{-1}\left(\frac{c\omega}{k-m\omega^2}\right). \quad (3)$$

Equation (2) can be expressed in terms of the damping ratio ( $\zeta = c/2\sqrt{km}$ ) and the natural frequency of the resonator ( $\omega_n = \sqrt{k/m}$ ) as

$$z = \frac{\left(\frac{\omega}{\omega_n}\right)^2 Y}{\sqrt{\left(1 - \left(\frac{\omega}{\omega_n}\right)^2\right)^2 + \left(2\zeta \frac{\omega}{\omega_n}\right)^2}} \sin(\omega t - \varphi). \quad (4)$$

The total energy absorbed by the damper in one cycle can be determined by integrating the product of the damping coefficient ( $c$ ) and the square of the velocity of the seismic mass over the time period. From the total energy absorbed by the damper, the average power flow into the device can be obtained by dividing the former with the time period, so that the latter can be expressed as [5]

$$P_{av} = \frac{\omega}{2\pi} \int_0^{2\pi/\omega} c \dot{z}^2 dt = \frac{m\zeta Y^2 \left(\frac{\omega}{\omega_n}\right)^3 \omega^3}{\left(1 - \left(\frac{\omega}{\omega_n}\right)^2\right)^2 + \left(2\zeta \frac{\omega}{\omega_n}\right)^2}. \quad (5)$$

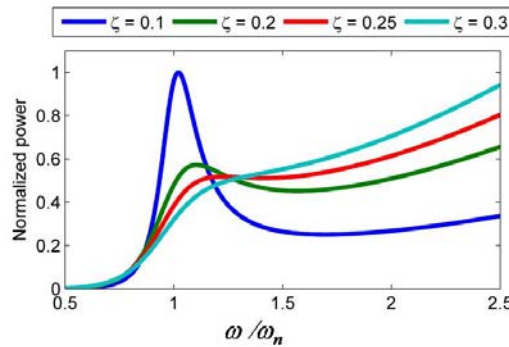
Setting the time derivative of the above equation to zero, one can show that the average power absorbed by the damper becomes maximum for a frequency ratio

$$\omega / \omega_n = \sqrt{2 - 4\zeta^2 - \sqrt{(4\zeta^2 - 2)^2 - 3}}. \quad (6)$$

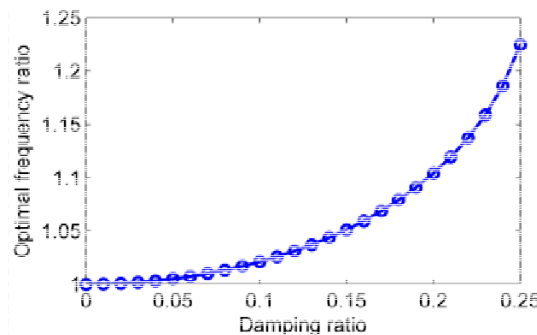
### 3. Result and Discussion

The variation of the average power with the frequency ratio ( $\omega/\omega_n$ ) at different damping levels is shown in Fig. 2. Following observations can be made. First, a small value of the damping ratio gives high average power at resonance but exhibits a very small bandwidth. Second, as the damping level is increased, the bandwidth increases however, the power at resonance drastically drops. Third, for small levels of damping the power-flow-resonance occurs at a frequency ratio close to unity. However, as the damping level is increased the power-flow-resonance frequency also increases. Forth, for off-resonance operation at a frequency higher than the natural frequency of the harvester, a highly damped system is favorable. Fifth, no resonance peak is obtained for a damping ratio greater than 0.25. This is

expected as (6) gives imaginary frequency ratio for damping ratio greater than 0.25. The variation of  $(\omega/\omega_n)$  with the damping ratio, as given by (6), is shown in Fig. 3. Again, for small values of the damping, the optimal damping ratio is close to unity.



**Fig. 2.** Variation of normalized average power with frequency ratio at different damping levels.



**Fig. 3.** Optimal frequency ratio Vs damping ratio [10].

In the light of the above discussion, one can substitute  $\omega/\omega_n$  with unity in (5) to get the maximum average power absorbed by a lightly damped harvester as

$$p = \frac{mY^2\omega_n^3}{4\zeta}. \tag{7}$$

The above equation apparently leads to the conclusion that the average power flow into a resonant harvester increases with the cube of the frequency, as claimed by Williams et al. [5]. However, a careful observation of the above equation shows that the cubic dependence of the average power on the frequency hold only if the mass, amplitude of base excitation, and damping ratio are kept constant. Among these parameters the damping ratio depends upon the frequency and can be expressed as

$$\zeta = \frac{c}{2m\omega_n}. \tag{8}$$

The above equation shows that, for a constant mass, damping ratio is inversely proportional to the frequency. Therefore, if the frequency changes, the only way to keep the damping ratio constant is to change the damping coefficient with the frequency in the same ratio. In the above equation, the damping coefficient  $c$  represents the total damping offered by the mechanical and electrical domains. It

means, in principle one can increase any of the mechanical or the electrical damping with the frequency to keep the damping ratio constant. However, the mechanical damping is always counterproductive [8]. Therefore, one must consider the electrical damping as the only option to be increased with the frequency. The electrical damping coefficient can be expressed as [8]

$$c_e = \frac{K^2}{R_{load} + R_{int}}, \quad (9)$$

where  $K$ ,  $R_{load}$ , and  $R_{int}$  are the electromechanical coupling coefficient, load resistance and the internal resistance of the transduction mechanism respectively. The above equation shows that the electrical damping coefficient can be increased either by increasing the electromechanical coupling or by decreasing the load resistance. Among these choices, the electromechanical coupling coefficient depends upon the size of the device, and cannot be increased from a maximum value in a given volume. For example, in an electromechanical harvester, one needs a larger coil and magnets in order to increase the electromechanical coupling coefficient ( $NBL$ ; where  $N$  is the number of turns in the coil,  $L$  is the effective length of each turn and  $B$  is the magnetic field intensity linking the coil [8]). The second option is to decrease the load resistance. But any reduction in the load resistance from its optimal value given as [8]

$$R_{load} = R_{int} + \frac{K^2}{c_m} \quad (10)$$

will reduce the average power and must be avoided. Therefore, it can be concluded that the cubic dependence of the average power on the driving frequency is valid only in the range in which the damping coefficient can be adjusted with the driving frequency. However, such an adjustment may be counterproductive.

Form (4), when driven at the natural frequency, the amplitude of vibration of the seismic mass with respect to the base of the harvester can be expressed as

$$z_o = \frac{Y}{2\zeta}. \quad (11)$$

Using the above equation (7) can be rearranged as

$$p = m\zeta z_o^2 \omega_n^3. \quad (12)$$

The above equation still has  $\zeta$ , which can be eliminated using (8) to express the average power as

$$P = \frac{c z_o^2 \omega_n^2}{2}. \quad (13)$$

From the above equation, before concluding that the power increases with the square of the frequency, one must consider that for a constant damping coefficient, the amplitude is not independent of the frequency. Rearranging (8) and (11), the relation between the amplitude and the frequency can be shown to be

$$z_o = \frac{m\omega_n Y}{c}. \quad (14)$$

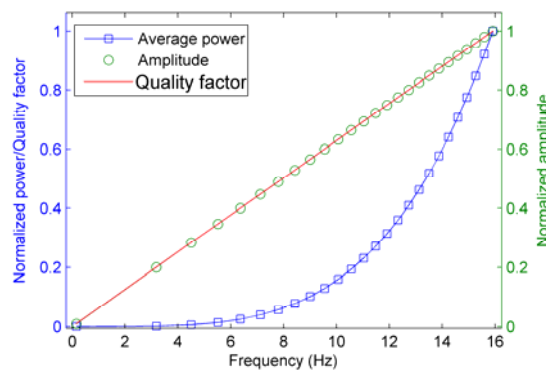
The above equation shows that, for a constant damping coefficient, a high frequency scavenger will have higher amplitude and vice versa. Therefore, it can be concluded that, when excitation level is considered constant, the average power does not increase with the square of the frequency rather the former will increase more rapidly with the latter. However, if the excitation level is not considered to be constant and the amplitude is kept constant with the increase in the frequency by continuously reducing the amplitude of the base excitation then the power increases with the square of the frequency.

One may consider that the maximum amplitude is limited by the device geometry and the elastic limits of the spring and replace the amplitude in (13) with the maximum displacement limit. In that case, the power increases with the square of the frequency. However, a high frequency harvester will give the same power with smaller amplitude of the base vibration as a low frequency oscillator will give with larger amplitude of the base vibration. Therefore, the volume figure of merit [1] will increase with the frequency.

Rearranging (7) and (8) the average power can also be expressed as

$$p = \frac{m^2 Y^2 \omega_n^4}{c} \tag{15}$$

The above equation shows that the power increases with the fourth power of the frequency. However, it may be noted that when the mass, the amplitude of the base excitation, and the damping coefficient will be kept constant, from (14), the amplitude will increase linearly with the frequency. Variation of the average power, quality factor, and amplitude with the frequency under the aforementioned conditions is shown in Fig. 4. It may be noted that both the amplitude and the average power increase with the frequency. As the plot is obtained keeping the damping coefficient and the mass unchanged so an increase in the frequency causes the quality factor to increase and hence results in an increased amplitude, this increased amplitude in association with the increased frequency results in an increased velocity so that the average power absorbed by the damper increases.



**Fig. 4.** Variation of normalized power, normalized quality factor, and normalized amplitude with frequency at resonance when the mass of the resonator is kept constant.

Equation (15) can be rearranged using  $k = m\omega_n^2$  to express the power in the form

$$p = \frac{(kY)^2}{2c} \tag{16}$$

The above equation shows that if the damping coefficient is kept constant, the power is primarily a function of the spring constant and the amplitude of the base excitation. As the frequency of a resonator can be changed by varying the mass, while keeping the spring constant unchanged, one obvious conclusion that can be made from the above equation is that the average power is independent of the frequency. However, it may be noted that the low frequency oscillator will have a larger mass, lower damping ratio and larger amplitude. Therefore, the total volume will increase for a low frequency harvester. To look into the issue more closely, consider two scavengers  $H_h$  and  $H_l$  such that  $H_h$  is working in a high frequency environment while  $H_l$  is working in a low frequency environment. If the spring constants of the two scavengers are considered as the same, scavenger  $H_h$  will have a small seismic mass while  $H_l$  will have a large seismic mass. The ratio of these masses can be written as

$$\frac{m_h}{m_l} = \left( \frac{\omega_l}{\omega_h} \right)^2. \quad (17)$$

Similarly, with the same damping coefficient the relation between the damping ratios can be expressed as

$$\frac{\zeta_h}{\zeta_l} = \frac{\omega_h}{\omega_l}. \quad (18)$$

Using  $m = k / \omega_n^2$ , the ratio of the amplitudes of the seismic masses with respect to the base can be written as

$$\frac{z_h}{z_l} = \frac{\omega_l}{\omega_h}. \quad (19)$$

As the two scavengers are considered to have springs of equal stiffness and damping coefficients therefore, from (16) they will absorb the same power. However, the scavenger  $H_h$  will have certain advantages on the scavenger  $H_l$ . From (17), the harvester  $H_h$  will have a very small mass as compared to  $H_l$ , from (18), the damping ratio will be higher for the former, from (19) the high frequency harvester will have smaller amplitude, and hence will give the same power in a smaller volume as compared to the latter.

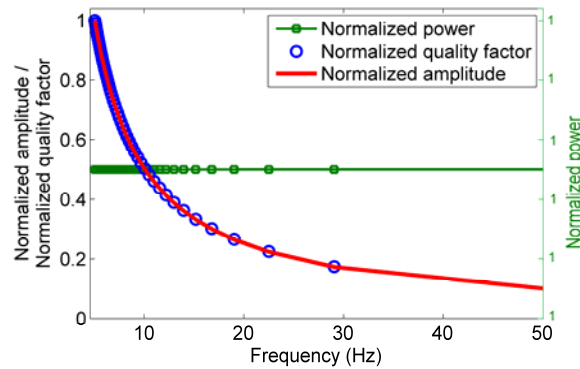
Variation of the average power, quality factor, and amplitude with frequency whereby variation in the frequency is achieved by varying the seismic mass is shown in Fig. 5. While the amplitude reduces with the frequency, the power does not change. This is expected because, with a decrease in the mass, the quality factor reduces, which tends to decrease the velocity, but the increase in the frequency counterbalance the effect and hence the velocity and the power remains unaffected.

Conventionally, vibration at a certain frequency is measured in terms of the acceleration instead of the amplitude. The peak value of base acceleration can be related to the amplitude of the base vibration as

$$Y = \frac{a_o}{\omega^2}. \quad (20)$$

Therefore, from (7), the average power at resonance can be written as

$$p = \frac{ma_o^2}{4\zeta\omega_n}. \quad (21)$$



**Fig. 5.** Variation of normalized average power and amplitude with frequency when stiffness of the spring is kept constant.

The above equation apparently leads to the conclusion that if the base acceleration is constant, the average power flow into a resonant harvester decreases with the frequency, as claimed by Roundy et. al in [7]. However, it may be noted that the mentioned inference holds only when the mass is kept constant, and from (8), when the mass is kept constant, the damping ratio is inversely related to the natural frequency. Therefore, the inverse relation between the power and the frequency holds only when the damping coefficient is increased with the frequency in the same ratio. As mentioned earlier, under the explanation of (7), this strategy will have practical limitations and may prove counterproductive.

Using (20), (14) can be rearranged as

$$z_o = \frac{ma_o}{c\omega_n}. \quad (22)$$

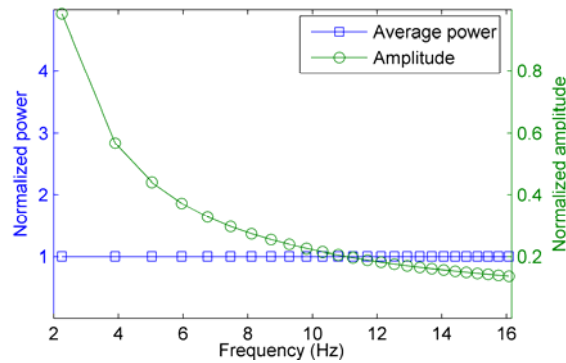
Equation (21) can be rearranged using (11) and (20) as

$$p = \frac{mz_o a_o \omega_n}{2}. \quad (23)$$

The above equation shows that when the amplitude of the seismic mass oscillation is limited by the device geometry, the power increases linearly with the frequency. In addition, it may also be noted that, from (22), one needs a larger damping at low frequencies to confine the seismic mass oscillation. Therefore, contrary to what has been claimed when the acceleration is constant, a harvester should be tuned with a higher frequency harmonic, and when the acceleration is decreasing with the frequency the harmonic having highest value for the product of the acceleration and the frequency should be selected. Variation of the power and the amplitude with the frequency, at constant damping ( $c$ ) and mass, as predicted by (23) and (22), respectively, is shown in Fig. 6.

In the light of discussion made above, a possible design procedure can be adopted as below:

- The resonator should be designed to have the lowest possible mechanical damping ratio.
- The mass and the transduction mechanism should have the highest value, possible in the finite geometry.
- With the achievable electromechanical coupling coefficient and the mechanical damping coefficient, the load resistance should be decided which will in-turn decide the optimal electrical damping.



**Fig. 6.** Variation of normalized average power and amplitude with frequency when the base acceleration is kept constant.

- The spring constant should be increased to increase the natural frequency of the resonator to match with the frequency of vibration. As the frequency is being increased at constant damping coefficient, the power will increase with the fourth power of the frequency. However, the amplitude of the seismic mass with respect to the base will also increase.
- If the combination of the optimal electrical damping and parasitic damping is not sufficient to confine the seismic mass displacement within allowed geometry, any further increase in the frequency must be carried out keeping the damping ratio constant. As the mechanical damping is always counterproductive the electrical damping should be increased with the spring constant in the same ratio so that the damping ratio and hence the amplitude remains unaffected. Such an increase in the frequency results in an increase of the average power with the cube of the frequency. This is the case considered by C. B. Williams et al. in [5].
- If the device geometry does not allow an increase in the electromechanical coupling coefficient the last option is to reduce the seismic mass, which will result in an increased damping ratio and hence decreased amplitude and broaden frequency response but the average power will not be affected.
- When the magnitude of the base acceleration is constant, a harvester should be tuned with a higher frequency harmonic, and when the acceleration is decreasing with the frequency the harmonic having highest value for the product of the acceleration and the frequency should be selected.

## 4. Conclusion

In conclusion, we have shown that cubic dependence of the average power on the frequency of vibration, as claimed in literature, holds only when the electromechanical coupling coefficient can be increased with the frequency. For a constant electromechanical coupling coefficient, the power increases with the fourth power of the driving frequency. When the magnitude of the base acceleration is constant, contrary to what has been claimed, a high frequency harmonic will give more average power density than a low frequency one.

## Acknowledgements

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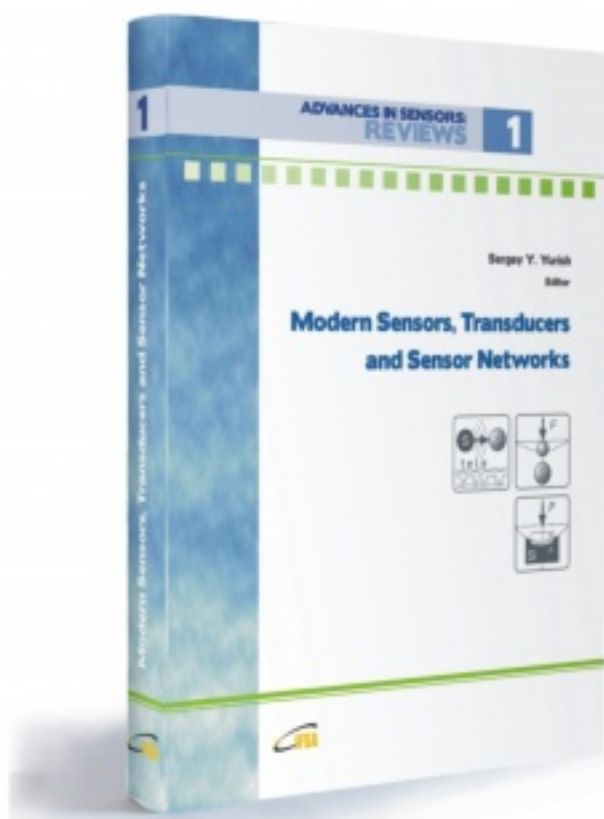
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