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Investigation of Mechanical Motion Amplification for Vibration Energy Harvesting

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Abstract Vibration Energy Harvesting is being investigated for autonomous sensors and actuators that mainly utilize ambient and machine induced vibrations. Recently mechanical motion amplification is incorporated for improving power to weight ratio of vibration harvesters. The present study is motivated to investigate mechanical motion amplification characteristics with different configurations. The parameters investigated are motion amplification ratio, force transmissibility characteristics, weight of the electrical generator, effective damping coefficient achieved and linear nature of damping. Numerical analysis has been performed to compare important characteristics of device operating without amplification to that of with amplification with different configuration. The study has been concluded with comments on application of suitable type of amplification mechanism depending on weight/space constraints and desired effective damping coefficient.

Keywords: motion amplification, electric power, energy harvesting, efficiency, numerical simulation.

1. Introduction

Vibration energy harvesting is being investigated as an alternate energy source to powers electronics sensors. The application includes micro-electromechanical systems as well as wearable sensors. Vibration powered devices have potential to eliminate battery as the energy source and the power cables resulting in an improves system reliability. Conventional vibration energy harvesting system has an inertial mass, spring (elastic element) and an electric generator. Generally piezoelectric or electromagnetic generators are used in vibration energy harvesting with more emphasis on electromagnetic devices for low frequency (<5Hz) and high amplitude applications. On the other hand, piezoelectric devices being preferred for high frequency applications. During the operation, the inertial mass undergoes relative motion with respect to the vibrating frame and this motion is used to drive the electric generator, which also performs the function of damper by dissipating the vibration energy into electric power.

Shahosseini and Najafi have performed simulations with velocity amplification factor of 4.0 to achieve significant improvement in the power density of the vibration harvesters [1]. Vibration energy harvesters based devices can be used for energy generation as well as for mitigating the impact factors caused by oscillations [2]. The prototype design demonstrated 15W of power with 60% harvesting efficiency. Vibration harvesters can be used to utilize human motion like foot strike and upper limb motion with piezoelectric or tribological micro generators [3]. However, there is a need for wearable device to generate watt-level of power. Low frequency resonating vibration energy harvester utilized resonance coupling to amplify the relative motion thereby improving power output [4]. Better capacitor charging rate demonstrated performance increase by 10 times than that of the conventional design. Footstep motion is a largely untapped vibration source that can be used for wearable sensors and battery charging applications. A non-linear mechanical synchronised inductor circuit amplifies the footstep motion to improve the electric power [5]. A trapezoidal slider mechanism has been used for increasing displacement in a foot vibration harvester motion amplifier [6]. Motion simulator with a gear train and electric generator has been used to demonstrate the application. Two stage compound amplification mechanism ensured higher electrical power form a piezoelectric harvester. It has been reported that the device ensured better efficiency, safety and compact size than the conventional design [7]. Mechanical motion rectifier ensured large amplitude by utilizing motion of a passing train to harvest 10-100 W of power with 74% efficiency. Simulation studies highlight the advantage of the amplifying mechanism for improving power to weight ratio [8]. A compact sized human motion harvester generated power with density of 0.62 mWcm-3. A study indicates that design parameters needs to be selected for better power output [9]. Energy extraction enhancement circuit based on piezoelectric material improves energy harvesting from human motion. Authors have reported up to 4.95 times increase in power than that of the conventional design [10]. A bistable energy harvester needs to be investigated for proper bias angle in the asymmetric potential to ensure enhanced performance [11]. A heel shaped energy harvester with two stage amplification mechanism ensured 66mW of peak power from a piezoelectric generator [12]. A simplified two stage model has been presented for human body design of 84 kg with two stage operation [13].

Attempts are being made to incorporate high energy rare earth magnets and velocity amplification to improve the electric power for ensuring better power to weight/volume ratio. Velocity amplification involves use of mechanical links, gears in various configurations. However, it also increases effective damping coefficient by the factor equal to square the velocity amplification ratio. In context to various configurations of velocity amplification mechanisms there is a need to investigate electric power, size of the electric generator and effective damping coefficient in the desired direction. the presented work focuses on numerical simulation of various configurations of amplification mechanisms with linear and rotary type of electromagnetic generators. The study presents correlation of application for the typical mechanisms to the following important performance and design characteristics of electric power harvested, effective damping coefficient achieved and size of the electric generator.

2. Mathematical modelling

Three different configurations of energy harvesting damping are shown in Figures 1(a), 1(b) and 1(c). The harvesters has sprung mass (8 kg) supported to the vibrating base through a spring (with stiffness of 10497 N/m). The harvester shown in Figure 1(a) has the electromagnetic generator connected between the mass and support and operates without motion amplification. The harvester shown in Figure 1(b) is operating with motion amplification having mechanical links connected in between. Figure 1(c) shows the harvester with link based mechanism for motion amplification. The linear generator is located in horizontal direction and is driven by relative displacement between the sprung mass and vibrating base. The amplification mechanisms in Figures 1(b) and 1(c) ensure that the electric generator operates with higher relative motion than that of the sprung mass relative displacement. The factor 'K_{amp}' has been used to quantify amplification achieved with the mechanical motion amplifier ($K_{amp} = 8$) for Figure 1(b) and (c).

Referring to Figure 1 (a), electric power by the harvester (A) is given as:

$$P_{\mathrm{e}} = \left\{B^2 \frac{l_{\mathrm{coil}}^2\, d_{\mathrm{coil}}}{R_{\mathrm{coil}} + R_{\mathrm{load}}}\right\}\{\dot{z}^2\}P_e = \left\{B^2 \frac{l_{\mathrm{coil}}^2\, d_{\mathrm{coil}}}{R_{\mathrm{coil}} + R_{\mathrm{load}}}\right\}\{\dot{z}^2\}, \tag{1}$$

where B is the generator air gap flux density, l_{coil} is length of copper coils in the generator, d_{coil} is diameter of the copper coil, R_{coil} is coil resistance, R_{load} is electrical load resistance and z is the mass relative displacement (y-x).

It can be noted from Equation (1) that electrical power output can be improved by increasing first term on right side of Equation (1). However, there are limitations in improving power output by this means since there is an upper limit for the flux density during the generator design and there are limitations on increasing the conductor wire length (l_{coil}) and diameter (d_{coil}) due to size limitations. Due to the above disused limitations it is difficult to design an electrical damper (i.e. generator) to deliver the higher damping coefficient with compact construction. In fact limitations in achieving the damping coefficient with compact size and lesser weight have limited development of a commercial vibration energy harvester. Electrical power from the harvester can be improved significantly by improving relative velocity.

The electromagnetic generator can be operated with amplified relative velocity to increase power output. Electrical power with incorporation of mechanical motion amplification illustrated in Figure 1(b) is given by:

$$P_{\mathrm{e}} = \left\{B^2 \frac{l_{\mathrm{coil}}^2\, d_{\mathrm{coil}}}{2(R_{\mathrm{coil}} + R_{\mathrm{load}})}\right\}\{(K_{\mathrm{amp}}\dot{z})^2\}. \tag{2}$$

Researchers have used mechanical motion amplification for improving electrical power for vibration harvesters. The mechanical motion amplifier should be designed to provide maximum amplification and should be reliable in operation besides having better force-velocity characteristics. In case of using mechanical gears for motion amplification, presence of backlash and possibility of cracks seriously affects reliability and durability of the energy harvesters. On the other hand, link based mechanism are reliable and gives larger amplification with compact size. Therefore link based mechanism have been considered in the further analysis.

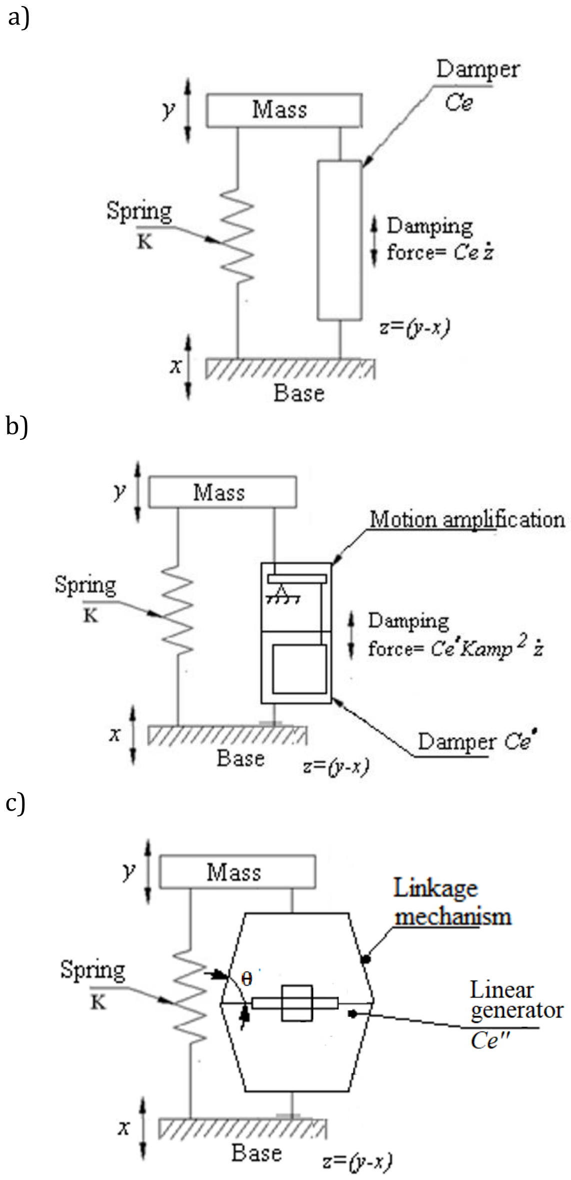


Figure 1. a): Harvester without motion amplification (A), b) Harvester with type 1 motion amplification (B), c) Harvester with type 2 motion amplification (C).

A case study has been demonstrated to evaluate application of motion amplification in a vibration energy harvester, with three different configurations illustrated in Figures 1 (a), (b) and (c). Governing equations for the configurations A, B and C are written as:

$$m\ddot{z} + kz + C_{\mathrm{e}}\dot{z} = mX_0(2\pi f)^2\sin(2\pi ft), \tag{3}$$

$$m\ddot{z} + kz + K_{\mathrm{amp}}^2 C_{\mathrm{e}}'\dot{z} = mX_0(2\pi f)^2 \sin(2\pi ft), \tag{4}$$

$$m\ddot{z} + kz + K_{\mathrm{amp}} C_{\mathrm{e}}''\sin(\theta_{\mathrm{t}})\cos(\theta_{\mathrm{t}})\dot{z} = mX_0(2\pi f)^2\sin(2\pi ft), \tag{5}$$

where z is the relative displacement of the mass (y-x), y is the displacement of the mass, x is support excitation, f is excitation frequency with amplitude of X_0, k is spring stiffness, C_{e} is the electrical damping coefficient of the linear electrc generator used in Type A, C_{e}' is electrical damping coefficient for Type B and

C_e'' is electrical damping coefficient for Type C. θ is the angle made by links as illustrated in Figure 1(c). It can be noted that Type (A), (B) and (C) energy harvesters will need different values of optimum damping coefficient because of different arrangements used for motionamplification.

K_{amp} is amplification factor achieved with the mechanical motion amplifier (K_{amp} = 8). It is the ratio of electrical generator to the inertial mass relative motion. Numerical simulation is performed with equations (3-5) to determine optimum value of the electrical damping coefficient to derive maximum electric power from the harvester, which comes at C_e=93 N-s/m, C_e'=1.45 N-s/m and C_e''=46 N-s/m. Design parameters of the electric generator have been calculated from Equation 4 and reported in Table 1.

Investigations are presented with use of motion amplification in an energy harvester using linear electric generator as the damping element. Simulations have been performed with linear generator having 450 numbers of 0.4 mm diameter copper wire turns and 20 Ω electrical load resistance. Simulation results for the necessary average coil diameter, linear damping coefficient of the linear generator and peak electric power delivered by the three types of harvester is reported in Table 1. It can be noted that implementation of mechanical motion amplification in case of Type B and Type C harvesters results in significant increase in electric power or reduction in the electrical generator size leading to compact and lightweight construction of the energy harvester. Comparison of Type A and Type B arrangements reveals that the motion amplification will results in significant reduction in the electric generator armature diameter while maintaining equal electric power to that of the Type A arrangement without motion amplification. Larger size of the armature coils with 380 mm average diameter, in case of the arrangement without amplification will result in heavy and bulky design of the generator which will make the arrangement impractical for a commercial implementation.

In case of Type C harvester, K_{amp} is equal to 8 and the angle θ is varying between 80°-90°, which will result in the generator size for maximum power to remain identical to that of the arrangement without motion amplification. However, there will be 370% increase in the peak electric power than that of the Type A and B harvesters.

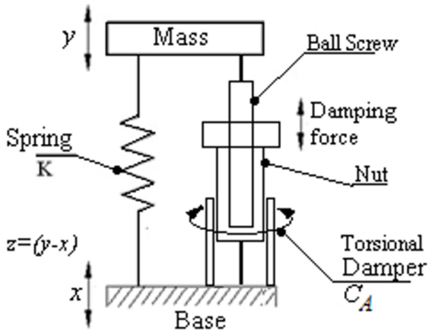
It can be concluded from the above analysis that the arrangement shown for Type B will ensure compact and lightweight construction of the vibration energy harvester whereas Type C harvester will ensure increase in the electric power output. Choice of the appropriate type of velocity amplification will be governed by size and weight requirements of the energy harvester.

Table 1. Design parameters of the electrical generator coil.

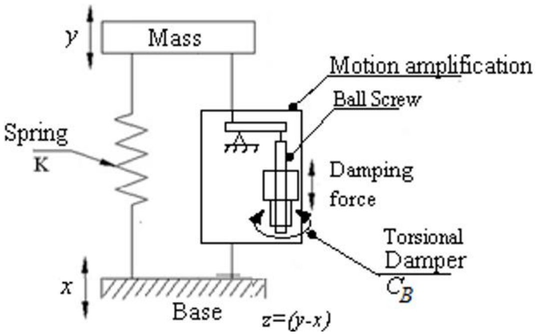
	Armature coil average diameter (mm)	Damping coefficient of the linear generator (N·s/m)	Peak electric power (W)
Type A (design without motion amplification)	380	93	$93\dot{z}^2$
Type B (design with motion amplification)	67	1.45	$93\dot{z}^2$
Type C (design with motion amplification)	380	93	$344\dot{z}^2$

In case of low frequency energy harvesting applications (frequency < 3Hz), ball screw-nut based devices use rotary electric generators as the damping and energy generating elements. Investigation is presented with different arrangements for implementation of mechanical motion amplification in the ball screw based energy harvesters, as shown in Figures 2(a), 2(b) and 2(c). Figure 2(a) shows the energy harvester where the ball screw arrangement is used without motion amplification and the linear relative motion of the vibrating and made to rotate as it moves along the screw axes. Further, the rotary motion of the nut is used to drive a rotary electric generator having damping coefficient of 'C_A'. Type-B harvester shown in Figure 2(b) is operated with amplification mechanism such that the screw moves with amplified displacement and velocity with respect to the nut. K_{amp} is the ratio of the inertial mass relative motion (i.e. input to the amplification mechanism) to output motion at the amplification mechanism which drives the ball screw-nut arrangement. Rotating motion of the nut is used to drive an electric generator having damping coefficient of 'C_B'.

a)



b)



c)

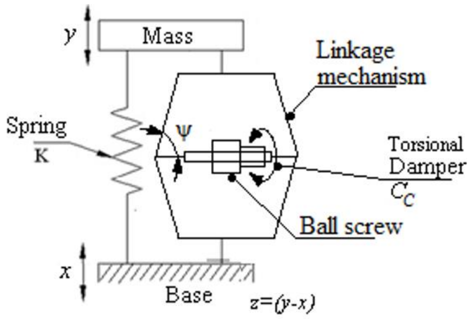


Figure 2. a): Ball screw based harvester without amplification, b): Ball screw based harvester with amplification, c): Ball screw based harvester with linkage mechanism for amplification.

Figure 2(c) shows Type-C harvester, which has link based mechanism for motion amplification. Mechanical links have been connected between the inertial mass and vibrating support and the ball screw-nut arrangement is mounted in horizontal direction. Vertical vibrations are transmitted in horizontal direction and the nut is rotating as it moves linearly along the screw axes, where the nut is driving an electrical generator having damping coefficient of 'C_C'.

Governing equations for the vibration harvesters shown in Figures 2 (a), (b) and (c) are given as:

$$m\ddot{z} + kz + [\frac{2\pi C_A}{{l_\mathrm{b}}^2}]\dot{z} = mX_0(2\pi f)^2 \sin(2\pi f t), \tag{6}$$

$$m\ddot{z} + kz + (\frac{2\pi K_{\mathrm{amp}}^2}{{l_\mathrm{b}}^2})C_B\dot{z} = mX_0(2\pi f)^2 \sin(2\pi f t), \tag{7}$$

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$$m\ddot{z} + kz + (\frac{2\pi sin2\psi K_{\mathrm{amp}}}{{l_{\mathrm{b}}}^2})C_{\mathrm{c}}\dot{z} = mX_0(2\pi f)^2\sin(2\pi ft). \tag{8}$$

Numerical simulations are performed with Equations 6-8 in order to compare values of optimum torsional damping coefficient required for the electrical generator and peak electrical power delivered for the above mentioned three arrangements. The simulation results are reported in Table 2 which indicates that there will be significant difference in the necessary values of the damping coefficient and peak power delivered by the harvesters. Comparison for Type-A and Type-B harvester reveals that Type-B will require lower value of the damping coefficient and peak power delivered by both the types is identical. Lower electrical damping will make it possible to have compact and low weight of the electrical generator components in Type-B harvester. On the other hand, electric damping coefficient of Type-C harvester will be close to that of Type-A (since $K_{\mathrm{amp}}\sin\psi \approx 1$) and it will deliver higher electric power output than Type-A and Type-B harvesters (since $K_{\mathrm{amp}} / \sin\psi \approx 4$).

Table 2. Numerical results.

	Rotary electrical damping coefficient (N-s-m/rad)	Peak electrical power (W)
Type-A	$\left[\frac{{l_{\mathrm{b}}}^2}{2\pi}\right]C_{\mathrm{eq}}$	$\frac{C_{\mathrm{eq}}\dot{Z}^2}{2\pi}$
Type-B	$\left[\frac{{l_{\mathrm{b}}}^2}{2\pi}\right]\frac{1}{K_{\mathrm{amp}}^2}C_{\mathrm{eq}}$	$\frac{C_{\mathrm{eq}}\dot{Z}^2}{2\pi}$
Type-C	$\left[\frac{{l_{\mathrm{b}}}^2}{2\pi}\right] sin2\psi K_{\mathrm{amp}}C_{\mathrm{eq}}$	$\frac{C_{\mathrm{eq}}\dot{Z}^2}{2\pi}\frac{K_{\mathrm{amp}}}{sin2\psi}$

3. Conclusions

In the presented work investigation of mechanical motion amplification with use of linear and rotary electric generator having different configurations for velocity amplification is discussed. It has been demonstrated that the mechanical motion amplification reduces size of the electric generator or significantly increases power output of the vibration energy harvester. Numerical simulations are performed for the different arrangements with linear and rotary electric generator to derive the damping coefficient for better power output and peak power harvested. It is observed that the optimum value of the damping coefficient of the electric generator depends on configuration of the amplification mechanism. Presented conclusions for the different configurations of vibration energy harvesters can be useful for selection of suitable type of amplification mechanism depending on weight/space constraints and desired effective damping coefficient.

Additional information

The author(s) declare: no competing financial interests and that all material taken from other sources (including their own published works) is clearly cited and that appropriate permits are obtained.

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