Design Piezoelectric Energy Harvesting Using COMSOL for Mice Telemetry Device

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Abstract: Mice are important animals used for biomedical research, and are widely used to study various disease models. When studying the behavior of mice and gathering biological data from them, miniature sensors and telemetry devices are needed. These devices must provide continuous monitoring without limiting mice mobility or behavior. One potential source of energy available in most biological systems is their natural motion. This work presents the design and simulation of a piezoelectric-based device to generate electrical power by harvesting the available energy due to the natural movement of a mouse. The mouse energy harvester design is configured as a base excitation model, which consists of a cantilever beam with a piezoelectric upper layer, and a proof mass. Maximum power is obtained when the excitation source (mouse movement) frequency corresponds to the resonant frequency of the energy harvester. A major issue is the small size of mice, and their relatively low frequency of motion. This creates challenges for the energy harvesting device design. The proposed cantilever energy harvester is designed to resonate at 11.7 Hz, which is close to the typical gait frequency while the mouse runs. The piezoelectric-based energy harvester is simulated using COMSOL that is based on the finite element method (FEM), to predict the electrical power for different mouse motion excitation frequencies with the matched load impedance. Additionally, the actual data of mouse runs that is captured by an accelerometer is used to simulate the proposed energy harvesting system.

Keywords: Energy Harvesting**,** Piezoelectric Biomedical device, Finite element analysis, COMSOL.

1. Introduction

The biomedical devices, miniature portable sensors and telemetry systems show enormous promise for monitoring animal behavior and physiological information. Often, these remote systems must operate wirelessly, since animal mobility and behavior can be adversely affected by wires and cable harnesses. A significant challenge for these devices is to provide a power source sufficient to meet the needs of the application. The process of gathering ambient energy surrounding a system, and converting it into usable electrical energy is termed as Energy Harvesting (EH) [1].

There are main ambient energy sources for EH applications such as solar, thermal, and mechanical vibration, among others [2]. Additionally, combinations of energy harvesting sources can be used to gather power from different environments. Vibration-based energy harvesting is used in numerous applications ranging from common household devices, transportation tools, industrial machinery and even human motion [3]. Significant development in EH devices using mechanical vibration has been done using various mechanisms, including electromagnetic, electrostatic, and piezoelectric [3-4]. In vibration-based EH, the optimal electrical power is achieved within a narrow frequency bandwidth, close to the resonant frequency of the device. Techniques using piezoelectric EH provide attractive features for this application, in particular: high energy density, high voltage developed, good dynamic response, and small size with the ability to scavenge energy in the range of 1 to 200 μW/ cm^3 [4-5].

The purpose of this work is to design a vibration-based piezoelectric energy harvester for small animal motion (EHAM), to generate electrical power from natural mouse movement. Figure 1 illustrates the proposed concept, where the EHAM design is mounted on a mouse head. Mice exhibit motion at low frequencies and small amplitudes, where different gaits correspond to different frequencies [6]. Of interests to this work, Figure 2 shows a spectral power density (SPD) of mouse acceleration data while it was running, in three axes of motion. The *x*-axis corresponds to front/back motion, the *y*-axis to right/left motion, and the *z*-axis to up/down motion. It was found that the predominate frequency and motion amplitude occurred along the anterior-posterior axis of the mouse, which corresponds approximately to the *x*-axis of the EHAM, as shown in Figure 1. There are a number of resonant frequencies apparent in Figure 2, corresponding to peaks of power, however, the major resonant frequency is found at 11.7 Hz.

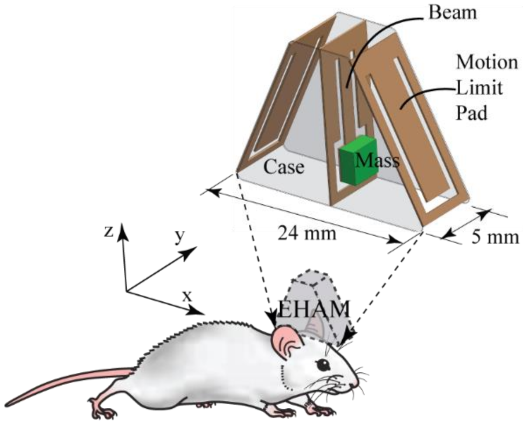

Figure 1. Illustration of energy harvester for mice motion.

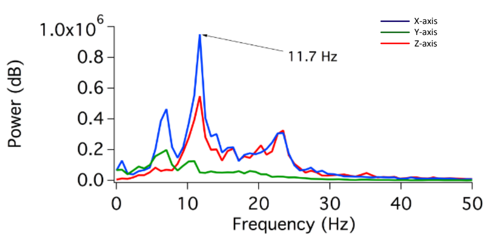

Figure 2. PSD plot of the *x*, *y* and *z* axis acceleration measurements.

Therefore, the proposed EHAM is designed to resonate at 11.7 Hz, to maximize the power harvested. Currently reported piezoelectric EHs in literature have been designed to typically operate at frequencies of 50 Hz, or higher. However, most animals exhibit typical motion at frequencies below 20 Hz, and there are no reported small-sized EH devices that operate below 36 Hz [7].

This paper is organized as follows. Section I presents the research motivation for EH devices operating a low resonant frequency for mice telemetry devices. Section II describes the theoretical operation of piezoelectric energy harvesting devices and the design of the proposed energy harvester. The simulation of the proposed energy harvester for small animal motion (EHAM) is explained in Section III. Simulation results and analysis results are presented and discussed in Section IV. Finally, concluding remarks are made in Section V.

2. Piezoelectric Energy Harvesting Design

A model of the proposed EHAM device is presented, along with the equations for the piezoelectric effect. This model is used to estimate the resonant frequency, the maximum theoretical power with the matched impedance of the EHAM. A simplified model of the EHAM, inertial-based resonant generator system is illustrated in Figure 3 [4], which can be modeled as a 1 degree-of-freedom (DOF), spring-mass-damper system. This model is configured as a base excitation, since it represents a device with a solid outer structure (black rectangle) that is shaken by an external acceleration ($\ddot{X}(t)$). According to modal analysis theory, the governing equations for such a piezoelectric EH system can be written as [8]:

$$m\,\ddot{X}_1(t) + C\,\dot{X}_1(t) + K\,X_1(t) + f_e(t) = m\,\ddot{X}(t) \tag{1}$$

where *m*, *C*, and *K* are the effective mass, mechanical damping, and stiffness, respectively; f_e is the effective force due to the piezoelectric element; $\ddot{X}(t)$ is the acceleration applied to the housing due to external ambient vibration from the environment, and $X_1(t)$ is the mechanical displacement of the proof mass. The effective transducer force f_e can be presented as [8]:

$$f_e = k_m\,V_P(t) \tag{2}$$

$$k_m\,X_1(t) - C_P\,V_P(t) = Q_p(t) \tag{3}$$

where k_m is the effective electromechanical coupling coefficient of a piezoelectric structure; $V_p(t)$ and $Q_p(t)$ are the voltage across the piezoelectric electrodes, and the charge generated on the electrodes; C_p is the capacitance of the piezoelectric material.

The main goal is to convert mouse motion energy into electrical power. Several design iterations were done whereby the design was simulated by COMSOL. The cantilever-beam structure with proof-mass was found to be the simplest oscillator design that could meet the objectives for the EHAM. A cross-sectional diagram of the proposed configuration is illustrated in Figure 4 (not to scale).

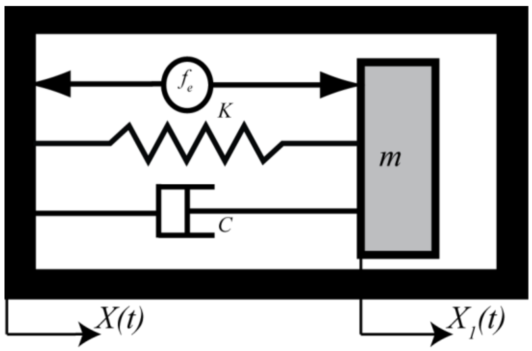

Figure 3. Models of single DOF vibration based energy harvester.

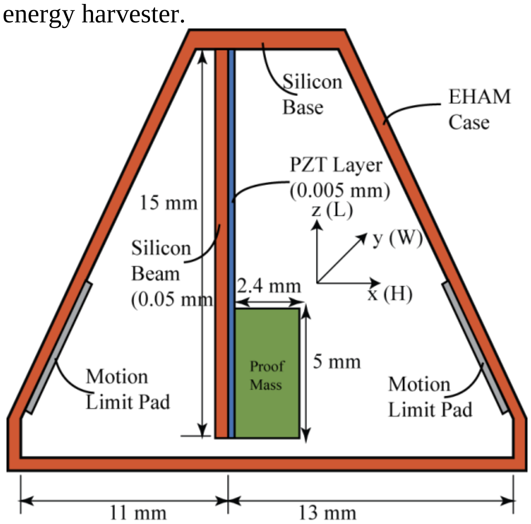

Figure 4. Illustration of dimensions for final design of EHAM cantilever (not to scale).

Given the requirement for low frequency operation at 11.7 Hz, a combination of high mass and low beam stiffness is needed. A silicon-based cantilever with a PZT layer is able to provide this 11.7 Hz resonant frequency, and is strong enough to prevent breakage of the structure. Some suggestions [8-10] were used when designing with the aim of getting a low natural frequency with maximum power generation. The cantilever is designed with the help of COMSOL, to harvest the vibration energy at 11.7 Hz. Given the ultimate strength of crystalline silicon at approximately 2.0 GPa, the maximum allowable stress at the root of the cantilever beam should be less than 600 MPa. Table 1 lists the final design parameters of the proposed EHAM.

Table 1: Design parameters for the EHAM system

Design Parameter	Description	Design Value [units]
Overall EHAM Case Size	Size of packaged EHAM $[L \times W \times H]$	$17 \times 5 \times 24$[mm]
Beam Geometry	Material: Crystalline silicon $[L \times W \times H]$	$15 \times 2 \times 0.05$ [mm]
PZT layer Geometry	Material: PZT-5 $[L \times W \times H]$	$15 \times 2 \times 0.005$ [mm]
Mass Geometry	Material: Tungsten $[L \times W \times H]$	$5 \times 4 \times 2.4$ [mm]
Operating Frequency	Natural Frequency of structure	11.7 [Hz]
Proof mass	Weight of proof tungsten mass	0.9 [g]

3. Use of COMSOL for EHAM

A computer model is developed to simulate the performance of the EHAM design as specified in Figure 4 and Table 1. The COMSOL Finite Element Analysis (FEA) package is used to estimate the EHAM performance, using four different studies. Given the symmetry of the design (constant geometry in the direction of width), the model is simulated with 2-D (two-dimensional) analysis. One challenging aspect of the FEA simulation is the considerable differences in the size between various parts, or in the parts themselves. This is a common problem is known as multi-scale FEA [11]. In this case, the piezoelectric layer is only 0.005 mm thick, yet 15 mm long, and the mass is much larger than the beam thickness. In order to guarantee mesh integrity, the aspect ratio of individual mesh elements should not be greater than three [5]. Figure 5(a) illustrates the plot of the quality mesh of the EHAM, internal COMSOL mesh quality analysis showed the mesh to be of high quality.

The base excitation is modeled using *Roller* constraint in COMSOL. Using the developed FEA model, four different studies are done to investigate the performance: (1) An *Eigenfrequency Study* to determine the first eigenfrequency (resonant), and its mode shape,

(2) A *Stationary Study* to check the total displacement and corresponding stress of the system due to gravity alone, in a static analysis, and (3) A *Time Dependent Study* (transient) to simulate the performance (displacement, stress and electrical properties) of the system, when different sinusoidal excitation frequencies are applied. (4) An additional *Time Dependent Study* to simulate the EHAM performance when mouse gait data is used as excitation for the model.

3.1 Eigenfrequency Study

The EHAM design is analyzed to determine its resonant frequencies by using FEA to find the eigenfrequencies and mode shapes of the structure. This analysis is extensively used during the geometrical design of the EHAM. The first three eigenfrequencies of the EHAM are found to be: 11.7, 133, and 683 Hz.

3.2 Stationary Study

A static FEA analysis is done to determine the displacement and stress of the EHAM, for the nominal case where only gravity acts on the mass. Figure 5(b) shows the tip displacement result, which is about 0.34 mm. This would correspond to the case where the mouse is at rest, with gravity acting in the -L direction. In this position, the proof mass would be located mid-way between the upper and lower motion limit pads. At this position the maximum Von Mises stress is found to be 10 MPa near the root of the beam, as shown in Figure 5(c).

3.3 Transient Study (Sinusoidal Excitation)

In this section, the EHAM is being subjected to sinusoidal excitation at different frequencies using the matched impedance of the EHAM. In order to model this situation, a transient (time domain) FEA study is done, where the study is computed for a four second duration with a time step of 0.002 seconds. Figure 6 shows the maximum displacement result during the sinusoidal excitation at the resonant frequency of 11.7 Hz, which is about 10 mm.

The nature (magnitude and frequency) of the excitation force applied to the EHAM is important, since it determines the maximum displacement of the EHAM proof mass and the corresponding power generation. The EHAM would be mounted on the head of the mouse, as illustrated in Figure 1, and is excited by the displacement ($X(t)$) of the moving head. This is a case of base excitation, whereby the acceleration ($\ddot{X}(t)$) acts on the housing structure of the EHAM. The excitation displacement, $X(t)$, is applied in the *x*-axis, the EHAM is excited by 1.25 mm amplitude displacement.

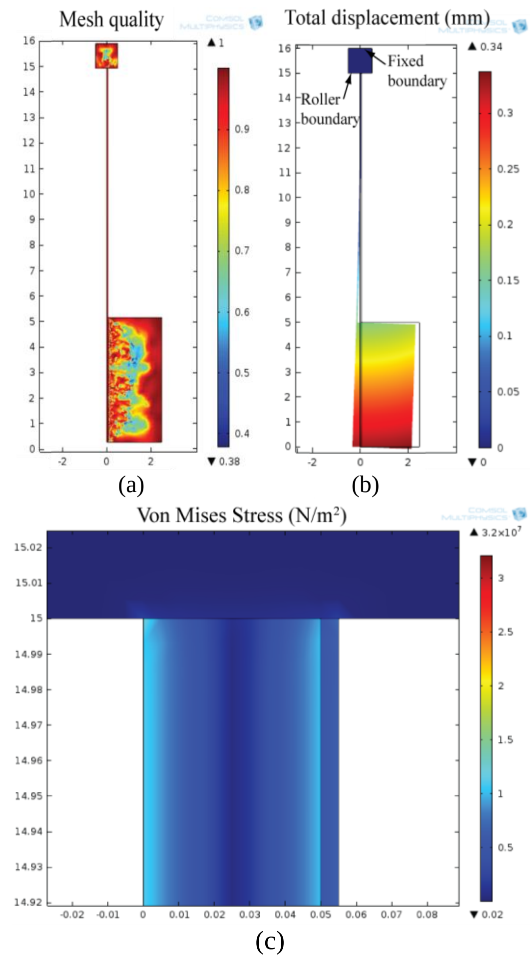

Figure 5. Simulation results of Stationary study; (a) mesh quality, (b) total displacement, (c) Stress at the root.

3.4 Transient Study (Arbitrary waveform Excitation)

In this section, the actual acceleration data from a mouse during running was imported in COMSOL to excite the EHAM. However, the simulation results (displacement, voltage) of the EHAM had accumulation error and did not follow the pattern of the excitation actual acceleration data, as the tip displacement of the cantilever was increasing. Since the basic partial differential equations of the FEM is based on the force and the displacement [12], so the EHAM should be excited by the displacement. Therefore, the actual acceleration data was integrated twice, to get the displacement of the mouse head while the mouse running. We removed the random errors and the noise from the acceleration data during the integration using "moving average curve" technique [5, 13-14].

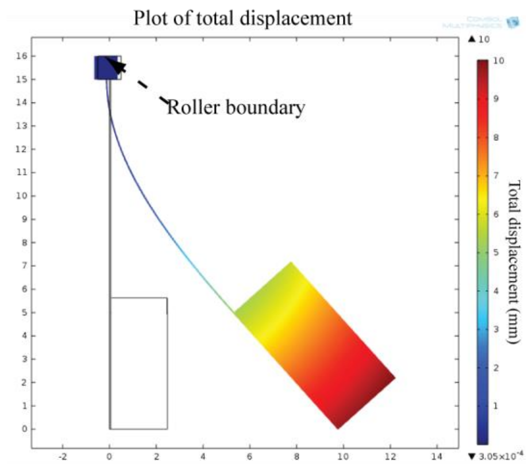

Figure 6. Total displacement in the sinusoidal excitation.

A sample of *x*-axis acceleration data is used to obtain the actual displacement. The accelerometer hardware sampling time for the actual acceleration data is 0.005 seconds. For the simulation, we interpolated that data 5× to a time step of 0.001 seconds, as shown in Figure 7(a). It is found that the larger sample size of the moving average curve technique, the less error is obtained in the integration [14]. We used 500 and 1000 sample sizes to obtain the velocity and the displacement, as shown in Figure 7(b&c). The actual displacement that is shown in Figure 7(c) was imported in COMSOL to excite the EHAM. Figure 9(a) shows the *x*-component displacement result, when exciting the EHAM with actual displacement data. It is noticed that the maximum displacement is 10 mm, so the proposed cantilever will not hit the motion limit pads of the EHAM, as shown in Figure 4.

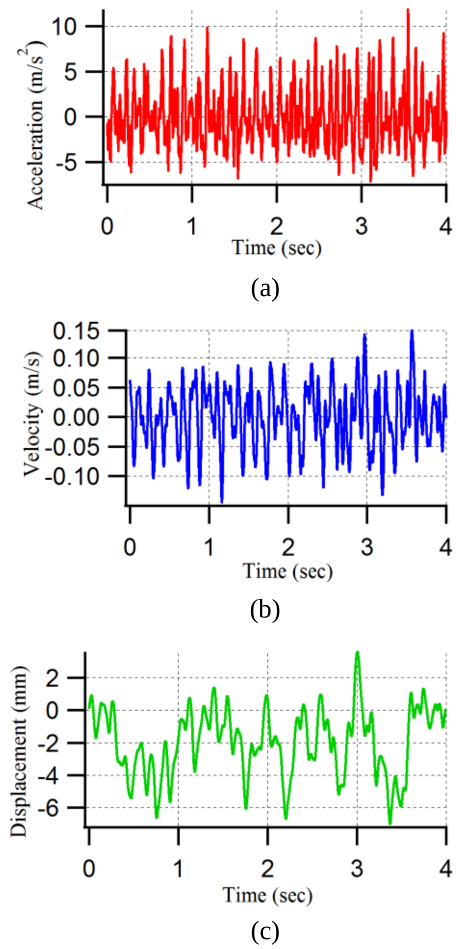

Figure 7. Actual data of mouse head movement, (a) acceleration, (b) velocity, (c) Displacement.

4. Results and Discussion

The simulation results of the proposed EHAM design are analyzed and discussed. The response of the EHAM to various excitation frequencies is done with an extensive simulation, which consisted of performing a transient study for each excitation frequency between 9 Hz to 13 Hz, in 0.2 Hz increments. This is accomplished using the *Parametric Sweep* function within COMSOL. The electrical power generated by the piezoelectric layer of the EHAM is analyzed for the same range of applied excitation frequencies. For the piezoelectric harvester devices, there is an ideal circuit load. The ideal circuit load must be matched to the impedance of the proposed EHAM to maximize the output power. The matched impedance is computed as [5]:

$$R_{matched} = \frac{1}{2\pi f_{res} C_p} \quad (4)$$

where C_P is the capacitance of the piezoelectric layers, and f_{res} is the resonant frequency of the EHAM. The capacitance of the piezoelectric material depends on the layer dimensions, including in-plane area and thickness, and is given by:

$$C_p = \frac{\varepsilon_r \varepsilon_o W L}{H} \quad (5)$$

where W is the width (2 mm), L is the length (15 mm), H is the thickness (0.005 mm), ε_o = 8.854 × 10^{-12} F/m is the electrical permittivity of free space, and ε_r relative dielectric constant of the piezoelectric layers = 1730 F/m. Therefore, the ideal matched load, $R_{matched}$, for the proposed EHAM is 222 kΩ.

The performance of the EHAM is simulated with the matched resistor of 222 kΩ, and the applied sinusoidal displacement. Figure 8 illustrates a plot of the electrical power produced, over a range of excitation frequencies from 9 Hz to 13 Hz, in 0.2 Hz increments. It is noteworthy that the maximum power occurs at 11.7 Hz, which is 68 µW. The maximum stress at the root of the beam is determined to be 0.53 GPa. The EHAM can harvest 37.5 µW, when the mouse is running (importing the actual displacement). Figure 9(b) shows the stress at the root of the beam, when the EHAM is excited by the actual data. The maximum stress is 0.5 GPa. Since the ultimate stress (failure stress) of the silicon is approximately 2 GPa and higher, this indicates the cantilever is in a safe range of operation.

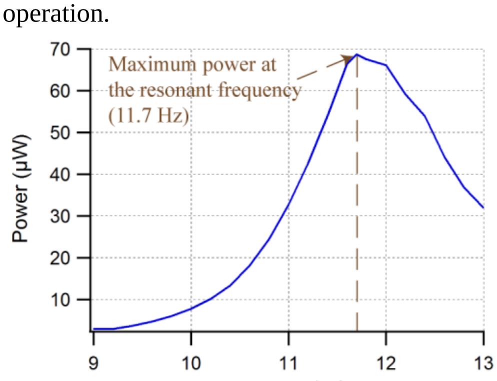

Figure 8. Plot of electrical power of sinusoidal excitation versus excitation frequency.

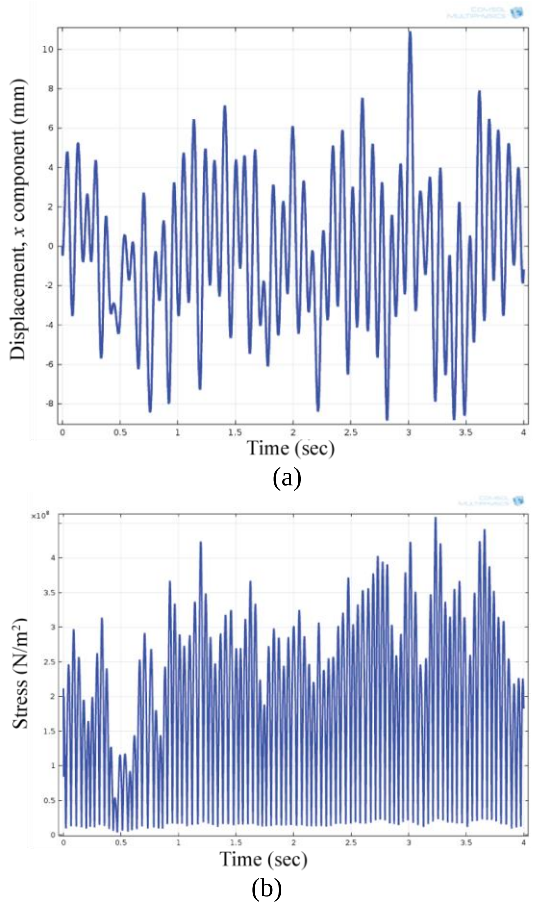

Figure 9. Simulation results of exciting the EHAM by the actual displacement data, (a) *x*-component displacement of EHAM, (b) Stress at the root of the EHAM.

5. Conclusions

The proposed EHAM model is proposed for harvesting motion energy from mouse gait. The EHAM model is configured as a base excitation model. It has been designed with the final design parameters of Table 1, which provides for realistic dimensions that can be tolerated by a mouse, along with a simple and manufacturable configuration. The simulation results confirm that the maximum power with the matched load is 68 µW, when the EHAM is subjected to sinusoidal excitation at the resonant frequency 11.7 Hz. The actual acceleration data was used to excite the proposed EHAM. In order to eliminate the simulation error from exciting the EHAM by the acceleration, the actual acceleration data was integrated twice to obtain the actual displacement of mouse gain. The EHAM harvested 37 µW, when the actual displacement excited the EHAM. The harvested power inherently provides low power, but may be of use in a low-power analog application circuit. Due to the intermittent motion of mice, an energy cell is proposed as an energy buffer to provide a realistic power supply for an application circuit.

9. References

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