

Article

Experimental Comparison of Frequency Tuning Strategies for Piezoelectric Cantilever Beam: Implications for Bridge Vibration Harvesting

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Abstract

Piezoelectric cantilever beam harvesters are widely considered for self-powered bridge monitoring, yet their performance is often constrained by resonance detuning under low-frequency ambient vibrations. This issue is particularly pronounced in bridge environments, where the dominant vibration frequencies are typically low and narrowly distributed. While several frequency tuning strategies have been proposed, their relative effectiveness under bridge-relevant conditions has not been systematically evaluated within a unified experimental framework. This study experimentally evaluated four tuning strategies for cantilever piezoelectric energy harvesters, i.e., spring tuning, magnetic tuning, tip mass adjustment, and beam length modification, to identify effective methods for matching the dominant frequency of bridge deck vibrations. A unified test platform using a common harvester configuration was established, and performance was quantified by resonant frequency alignment, maximum output voltage, and -3 dB bandwidth. Among the four methods, root-based spring tuning showed the best overall performance, achieving frequency matching while retaining strong electrical output, with a maximum voltage of 9.01 V and a bandwidth of approximately 1.5 Hz. Magnetic tuning also provided accurate frequency control, but reduced voltage by 15–25%. By contrast, tip mass and beam length tuning produced larger resonance shifts but caused voltage reductions of up to approximately 50%.

Keywords: piezoelectric cantilever beam; frequency tuning strategies; bridge vibration harvesting



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1. Introduction

In recent decades, the progressive aging of large-scale bridge infrastructure, coupled with increasing traffic demands and cumulative environmental effects, has led to growing concerns regarding structural safety and serviceability. To address these challenges, real-time and long-term structural health monitoring (SHM) has become an indispensable tool in modern bridge management, enabling continuous condition assessment, early damage detection, and timely maintenance interventions. However, the widespread deployment of sensor networks on bridges is fundamentally constrained by power supply limitations. Conventional power solutions based on wired connections or replaceable batteries are often impractical for large or long-span bridges due to installation complexity, vulnerability to harsh environmental conditions, limited operational lifespan, and the substantial main-

tenance burden associated with periodic battery replacement, particularly in remote or hard-to-access locations.

On this account, ambient energy harvesting has emerged as a promising and increasingly necessary solution for powering bridge SHM systems. By converting ambient energy sources, most notably mechanical vibrations generated by traffic flow, wind excitation [1,2], and other operational or environmental loads, into electrical energy, energy harvesting technologies offer a sustainable alternative to traditional power supplies. Note that bridge structures inherently experience persistent low-frequency vibrations throughout their service life, representing a largely untapped and reliable energy resource. Harnessing such vibration energy enables sensor nodes to operate autonomously, thereby reducing reliance on external power infrastructure and significantly lowering long-term maintenance costs. Moreover, the adoption of ambient energy harvesting aligns with the broader objectives of sustainable and resilient infrastructure development. Self-powered sensing systems can support continuous, distributed monitoring without intrusive installation procedures or frequent human intervention, enhancing both the durability and scalability of SHM deployments.

Among the available mechanisms for converting ambient mechanical energy into electrical energy, piezoelectric energy harvesting has received considerable attention due to its high-power density, compact configuration, and strong suitability for miniaturized and integrated sensing systems. This technology exploits the direct piezoelectric effect, whereby mechanical strain generated within a piezoelectric material under external excitation induces an electrical charge. When subjected to vibrational loading, this strain-driven electromechanical coupling enables efficient energy conversion. Within this framework, the cantilever beam configuration remains the most fundamental and widely adopted structural form in piezoelectric vibration energy harvesting. Its simple geometry, well-established analytical modeling, and ability to concentrate bending strain near the fixed end allow for effective electromechanical coupling and efficient energy extraction [3,4].

Despite these advantages, most existing cantilever piezoelectric vibration energy harvesters (PE-VEHs) typically exhibit a narrow effective operating bandwidth, given that these PE-VEHs generally behave as lightly damped linear resonators with a high-quality factor (Q) [5]. Under excitation near resonance, the cantilever undergoes large bending deformation, producing high curvature and strain in the piezoelectric layer and thereby enabling efficient energy conversion. However, the frequency response peak of a high- Q system is inherently sharp, such that the resonance amplification is confined to a limited frequency interval. Once the excitation frequency deviates from the natural frequency, the vibration amplitude and bending strain decrease rapidly, leading to a pronounced drop in electromechanical conversion efficiency and electrical output [6,7].

It is noteworthy that bridge vibrations relevant to energy harvesting are predominantly low-frequency in nature. Existing studies reported that the fundamental global frequencies of bridge vibration typically range between 1 and 50 Hz [8,9] with lower values for longer and more flexible systems. On the other hand, cantilever PE-VEHs operate most effectively near their fundamental bending resonance, and their working frequency is therefore determined primarily by the stiffness–mass characteristics of the cantilever system. As has been well-documented in the literature, these devices can operate in the tens to hundreds of hertz range. This creates a persistent mismatch with bridge applications, i.e., bridge vibrations are concentrated in a much lower frequency regime than that of many conventional piezoelectric cantilever devices. Under such circumstance, frequency tuning becomes an essential design strategy for bridge-oriented piezoelectric harvesting, as it enables the harvester resonance to be shifted toward the low-frequency bridge–vibration

regime and thereby improves the stability and effectiveness of electromechanical energy conversion [10–16].

Existing frequency tuning strategies for PE-VEHs are primarily aimed at modifying the system's effective stiffness or inertia in order to shift its natural frequency toward the dominant excitation frequency. The approaches reported in the literature can be broadly classified into stiffness-based, magnetic force-based, mass-based, and geometry-based tuning methods. Stiffness-based tuning methods typically rely on auxiliary mechanical components to regulate the equivalent stiffness of the cantilever system. Rajarathinam and Ali et al. [17], for example, introduced additional springs attached to the cantilever beam, enabling frequency adjustment through controlled variation in spring stiffness. Such approaches offer predictable and continuous tuning capability. However, they often increase structural complexity and may introduce additional damping or mechanical losses that reduce overall energy conversion efficiency. Magnetic force-based tuning has received extensive attention due to its non-contact nature and capability for both linear and nonlinear stiffness modulation. It is well-documented that magnetic tuning introduces non-linearity into piezoelectric cantilever harvesters through the strongly position-dependent nature of magnetic interaction. Unlike conventional linear stiffness elements, magnetic forces vary nonlinearly with the separation distance and relative arrangement between magnets, such that the restoring force acting on the harvester changes continuously with structural displacement. As a result, the equivalent stiffness of the system becomes displacement dependent, which can alter the resonance condition, distort the frequency response curve, and in some cases broaden the effective operating range. This nonlinear behavior is further influenced by the dynamic process of magnet engagement. In particular, when the oscillating beam moves into and out of the region of strong magnetic interaction, the response depends not only on the instantaneous relative position of the magnets but also on the relative velocity of the moving component during this interaction [18]. Li et al. [19] and Jung et al. [20] proposed a rotary self-tuning harvester, in which the resonance frequency is adjusted by varying the relative position between a tip-mounted magnet and an external magnet, thereby modifying the magnetic interaction force. Challa et al. [21] further exploited attractive and repulsive magnetic forces by altering magnet polarity and spacing, generating tunable nonlinear stiffness effects that enable wider frequency adjustment. Bouhedma et al. [22] extended this concept by adopting an axial magnetic configuration, in which the magnetic force acts perpendicular to the cantilever surface. While magnetic tuning methods provide flexible and reversible frequency control, the induced nonlinear dynamics can complicate analytical modeling and may adversely affect stability under variable excitation conditions. In contrast, mass-based tuning strategies focus on modifying the effective inertia of the cantilever system. Alameh et al. [23] demonstrated that adjusting the shape and magnitude of the tip mass can effectively shift the resonance frequency and influence the energy harvesting performance. Although this approach is straightforward and mechanically simple, it typically requires manual intervention and may alter strain distribution along the beam, potentially affecting durability and long-term reliability. Geometry-based tuning approaches achieve frequency adjustment by modifying structural parameters that govern cantilever dynamics. Wang et al. [24] developed a self-frequency-tuning harvester employing a rotating wheel mechanism to vary the effective length of the cantilever beam, thereby enabling adaptive frequency control.

Although extensive studies have investigated individual frequency-tuning mechanisms for PEHs, systematic experimental comparisons across different tuning strategies within a unified application context remain limited. In bridge applications, for instance, Peigney et al. [25] achieved a power output of 0.03 mW by frequency tuning a piezoelectric cantilever based on measured data from a concrete highway bridge.

Karimi [26] et al. employed a vehicle–bridge coupling model and used vibration signals from a shaking table bridge test as the initial excitation, confirming the feasibility of such harvesters for bridge vibration energy harvesting. In terms of frequency matching strategies, Cahill et al. [27] examined the performance of energy harvesters under different train excitations on an international train model bridge, offering theoretical support for tuning the natural frequency of harvesters across varying bridge vibration environments. Zhang et al. [28] compared two PEHs—one matched to the bridge’s natural frequency (PEH-1) and the other to the vehicle–bridge coupled vibration frequency (PEH-2)—and concluded that the harvester tuned to the coupled frequency delivers better real-world energy harvesting performance. Tianchen et al. [29] proposed a drum-type PEH and, through numerical simulation, showed that tuning its natural frequency to track vibration frequencies can yield a peak voltage of approximately 50–70 V and an output power of about 100 mW. Zhang et al. [8], based on in situ vibration measurements of a high-speed railway bridge under train operation, identified the dominant vibration frequency bands of the bridge. Accordingly, they designed a PEH tuned to match these vibration frequency ranges, which achieved a maximum energy harvesting of 2.79 mJ during a single train passage. However, although these studies uniformly highlight the importance of frequency matching, there is still a lack of systematic comparative analysis of different tuning strategies—such as tuning accuracy, voltage output, and bandwidth characteristics—under unified experimental conditions. This gap significantly constrains the selection and design optimization of optimal energy harvesting solutions tailored to specific bridge operating conditions.

Therefore, this study presents a systematic experimental comparison of four representative frequency tuning strategies, i.e., auxiliary spring tuning, magnetic force-based tuning, tip mass adjustment, and effective beam length modification, implemented on a common cantilever PE-VEH. Using an experimental platform that replicates dominant bridge deck vibration characteristics (as shown in Figure 1), the study evaluates each method’s ability to tune the harvester to the target frequency (≈ 38.8 Hz) [30] and examines the corresponding impact on electrical output performance.

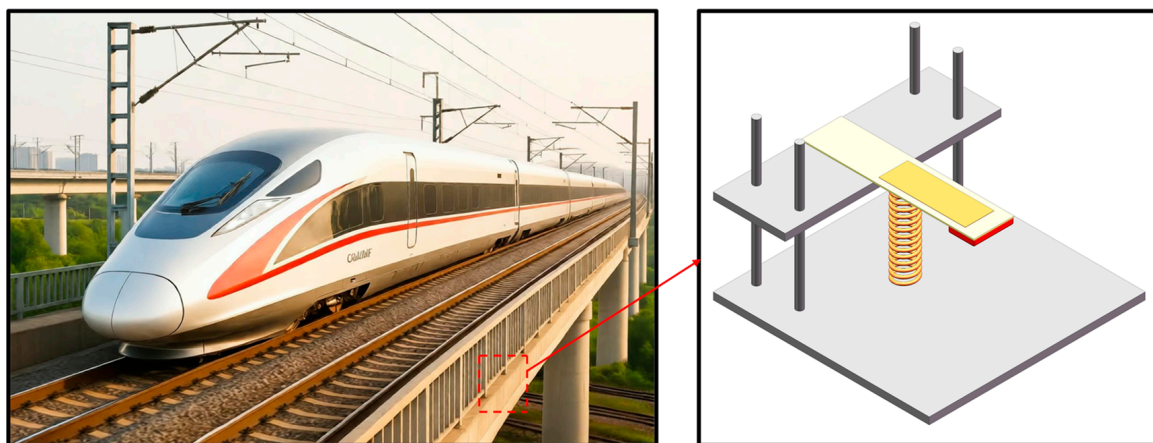


Figure 1. Schematic diagram of installation location.

2. Methodology

To ensure a consistent comparison of frequency tuning strategies, a dedicated experimental system was designed and implemented. The methodology integrates a controlled vibration test platform, a standardized cantilever PE-VEH prototype, modular tuning components, and unified experimental and data processing procedures, which enable all tuning methods to be evaluated under identical excitation conditions and measurement protocols.

2.1. Experimental Setup

The experiments were performed using a controlled vibration test platform, the main components of which are illustrated in Figure 2a. The platform comprises an electro-magnetic shaker (SA-JZ070; manufacturer: Wuxi Shiao, Wuxi, China) to generate stable harmonic excitations, a power amplifier (SA-PA080; manufacturer: Wuxi Shiao, Wuxi, China) to amplify the excitation signal, and a high-precision signal generator (SA-SG030; manufacturer: Wuxi Shiao, Wuxi, China) to produce sinusoidal waveforms with adjustable frequency and amplitude, thereby replicating steady-state bridge deck vibrations. A digital oscilloscope (DS1104Z-Plus; manufacturer: Rigol, Beijing, China) was employed to acquire and record the output voltage of the energy harvester in the time domain. The recorded signals were subsequently processed to obtain root mean square (RMS) voltage values, which served as the primary electrical performance metric. The shaker was rigidly connected to the base of the energy harvester using a mechanical clamp to ensure efficient and repeatable transmission of vibrational excitation.

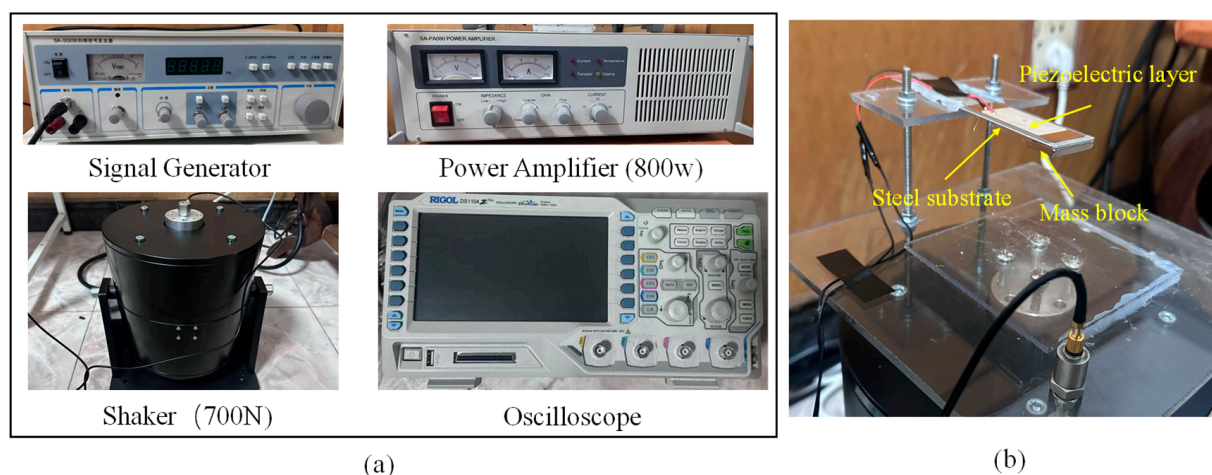


Figure 2. (a) Schematic of the experimental setup, and (b) device schematic.

The baseline cantilever PE-VEH used throughout this study is shown schematically in Figure 2b. The harvester consists of a stainless-steel cantilever beam substrate with a length of 70 mm, a width of 28 mm, and a thickness of 1 mm. These dimensions were selected through preliminary testing to position the first bending-mode natural frequency slightly below the target bridge deck frequency, thereby allowing sufficient tuning range toward higher frequencies. A piezoelectric ceramic patch made of PZT-5H (60 mm × 20 mm × 0.2 mm) was bonded to the upper surface of the beam near the clamped end using high-strength epoxy. This placement coincides with the maximum strain region of the first bending mode, ensuring effective electromechanical coupling and maximizing energy conversion efficiency. At the free end of the cantilever beam, a block-shaped NdFeB permanent magnet with a mass of 16.50 g was attached as the baseline tip mass. The magnet used in the magnetic tuning configuration was an N52-grade NdFeB magnet with a surface remanence of approximately 1.4 T. In addition to providing inertial loading, this magnet also functions as a key component in the magnetic frequency tuning configurations examined in the subsequent experiments.

2.2. Configuration of Different Frequency-Tuning Methods

To enable a systematic and unbiased comparison of frequency tuning strategies, five experimental configurations were designed, comprising one baseline case and four representative tuning methods. Each configuration was implemented using modular, interchangeable components to ensure consistency across tests.

Specifically, a baseline configuration was first established, using a standard Cantilever PE-VEH without any tuning elements. This configuration served as the reference benchmark for evaluating the performance of all tuning strategies, with the positions of the tip mass and the piezoelectric patch in all subsequent tuning configurations remaining as illustrated in Figure 2b. For the spring tuning configuration, auxiliary helical springs were introduced to modify the effective stiffness of the cantilever system. Springs with four wire diameters ($d = 1.4, 1.6, 1.8,$ and 2.0 mm) were used to provide a range of stiffness values. Their corresponding stiffnesses, measured using a manual tensile–compression testing machine, were $k = 5.95, 3.24, 1.53,$ and 1.34 N/mm, respectively. Each spring was installed sequentially at three predefined locations along the cantilever beam: near the clamped root, at mid-span, and near the free end, corresponding to distances of 0, 15, and 30 mm from the clamped end, respectively. For the magnetic tuning configuration, a repulsive magnetic arrangement was selected in this study because it provides an adjustable equivalent stiffness through distance-dependent magnetic interaction, thereby enabling continuous resonance tuning by varying the spacing between the magnets. An identical NdFeB permanent magnet was positioned beneath the baseline tip magnet with like poles facing each other in a parallel orientation. The vertical separation between the two magnets was varied (3.5, 5.5, 7.5, and 8.5 mm) to adjust the magnitude of the repulsive force acting on the cantilever. It is, however, worth mentioning that, unlike the spring-based tuning strategies, the magnetic configuration modifies the harvester response through displacement-dependent magnetic interaction while also contributing to the tip mass. Therefore, its tuning effect reflects coupled changes in effective stiffness and mass. However, since direct force measurement was not conducted, the relative magnitude of the magnetic force and the force associated with the elastic elements was not quantified in the present study. In the mass tuning configuration, the effective inertia of the system was altered by replacing the tip mass while maintaining a constant beam length and piezoelectric patch position. Four tip masses (6.25, 11.03, 14.38, and 16.50 g) were tested to assess the influence of mass variation on resonance tuning and energy harvesting performance. Finally, beam length tuning was investigated by varying the effective length of the cantilever beam. Beams with lengths of 68, 69, and 70 mm were employed, while the piezoelectric patch location and tip mass were held constant. This configuration enabled direct evaluation of geometric tuning through modification of the cantilever's effective stiffness and modal characteristics.

2.3. Experimental Workflow and Data Processing

A standardized experimental workflow was adopted for all test configurations to ensure consistency and reproducibility. For each configuration, an initial frequency sweep was conducted over a predefined range to identify the fundamental resonant frequency of the energy harvester. Once the resonance peak was located, a refined sweep was performed within a narrow frequency band centered on this peak to capture the detailed dynamic and electrical response.

At each excitation frequency, the steady-state output voltage of the harvester was recorded in the time domain and converted to its root mean square (RMS) value. These RMS voltages were then used to construct frequency response curves for each tuning configuration. Based on the frequency response, the operational bandwidth was quantified using the -3 dB criterion, defined as the frequency range over which the RMS output voltage remains above 70.7% of the peak value. This metric provides a consistent basis for evaluating the influence of different frequency tuning methods on both resonance behavior and effective energy harvesting bandwidth.

3. Results and Discussion

In the following section, the effectiveness of different tuning strategies is examined and discussed independently. It is to be noted that the present analysis is limited to the specific harvester configurations and controlled test conditions, and thus the reported differences among the tuning strategies should be interpreted as a case-dependent assessment for the selected bridge-relevant low-frequency scenario.

3.1. Spring-Based Frequency Tuning

Spring tuning is a mechanical resonance adjustment strategy in which an auxiliary elastic element is introduced to modify the effective stiffness of the harvester and, consequently, shifts its resonant frequency toward the target excitation range. For cantilever piezoelectric harvesters, this approach can reduce the effective natural frequency without substantially increasing the proof mass.

As can be seen from Figure 3 and Table 1, spring-based tuning effectively shifted the resonance frequency of the cantilever PE-VEH. However, the electrical performance depended strongly on the spring installation position and stiffness. The baseline harvester resonated at 34.6 Hz with a maximum voltage of 8.97 V. Introducing helical springs at the root, mid-span, or tip ($s = 0$ mm, 15 mm, and 30 mm, respectively, as shown in Figure 3) increased the resonant frequency in all cases. In contrast, voltage and bandwidth responses varied systematically with installation position. Root-mounted springs produced a monotonic increase in both resonant frequency and maximum voltage with increasing stiffness. Mid-span mounting elevated the resonant frequency but yielded no consistent voltage trend. Tip-mounted springs increased the resonant frequency while causing a monotonic reduction in maximum voltage, particularly at higher stiffness levels. Across all configurations, the -3 dB bandwidth remained limited (approximately 1.5–2.0 Hz), with only modest variation among cases (Table 1). Among the tested configurations, a 2.0 mm wire-diameter spring installed at the root achieved the most favorable performance, shifting the resonance to 39.2 Hz, increasing the maximum voltage to 9.01 V, and maintaining a bandwidth of 1.5 Hz. In contrast, high-stiffness springs installed at the tip consistently degraded voltage output.

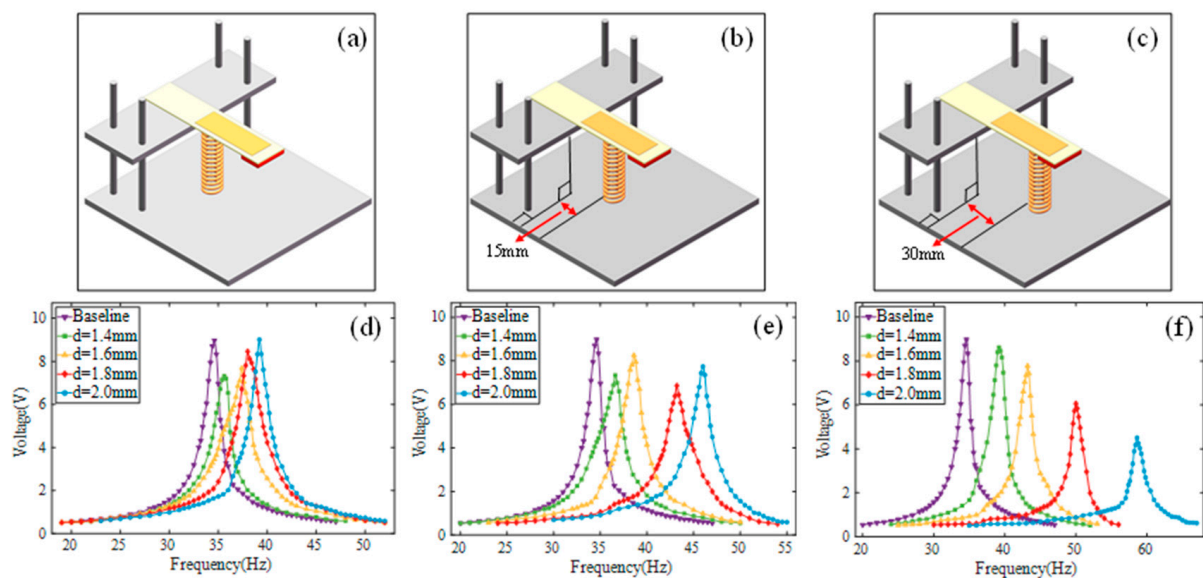


Figure 3. Schematic diagrams of different spring installation positions: (a) $s = 0$ mm, (b) $s = 15$ mm, and (c) $s = 30$ mm. Frequency–voltage response at the corresponding positions: (d–f).

Table 1. Spring-based frequency tuning output performance.

Tuning Methods	Tuning Component Specs	Resonant Frequency (Hz)	Voltage (v)	Bandwidth (Hz)	Quality Factor
Baseline	Baseline	34.6	8.97	1.2	28.83
spring at root	d = 2.0 mm	39.2	9.01	1.5	26.13
	d = 1.8 mm	38.0	8.46	1.9	20.00
	d = 1.6 mm	37.4	7.68	2.0	18.70
	d = 1.4 mm	35.6	7.31	1.5	23.73
spring at mid-span	d = 2.0 mm	46.0	7.74	1.6	28.75
	d = 1.8 mm	43.2	6.86	1.9	22.74
	d = 1.6 mm	38.6	8.25	1.8	21.44
	d = 1.4 mm	36.6	7.34	1.9	19.26
spring at the tip	d = 2.0 mm	58.6	4.51	1.5	39.07
	d = 1.8 mm	50.0	6.06	1.5	33.33
	d = 1.6 mm	43.2	7.78	1.5	28.80
	d = 1.4 mm	39.2	8.62	1.7	23.06

These results indicate that the effectiveness of spring-based tuning is governed not only by the magnitude of stiffness augmentation but also by its interaction with the modal strain distribution of the cantilever. Root-mounted springs introduce additional stiffness within the region of maximum curvature for the fundamental bending mode. This placement increases the global natural frequency while preserving, and in some cases slightly enhancing, the strain energy density within the piezoelectric patch region. Consequently, resonance alignment is achieved without compromising electromechanical coupling. When springs are mounted at the mid-span, their influence on modal behavior becomes more complex. Although the added stiffness still raises the natural frequency, it may alter the spatial distribution of curvature and shift strain nodes along the beam. Depending on stiffness level and exact placement, this redistribution can either enhance or suppress strain in the piezoelectric region. In contrast, tip-mounted springs primarily constrain free-end motion, where displacement rather than curvature dominates the dynamic response. Increasing stiffness at the tip restricts overall bending deformation and reduces curvature near the clamped end, thereby diminishing strain in the piezoelectric layer.

The variation in output voltage under different spring configurations can be interpreted in terms of electromechanical energy transfer. In a cantilever piezoelectric harvester, the conversion efficiency is mainly governed by the mechanical response of the structure, the electrical load condition, and the distribution of strain energy within the piezoelectric layer. For the root-mounted configuration, the added spring increases the local stiffness near the clamped end, which promotes larger bending curvature in the vicinity of the piezoelectric patch. Since this region is electromechanically active, the strain energy is concentrated more effectively where electrical conversion occurs, thereby helping to maintain, or in some cases increase, the output voltage. By contrast, when the spring is installed near the free end, the overall bending deformation of the cantilever is increasingly constrained. This reduces the strain transferred to the piezoelectric layer and weakens the electromechanical coupling, leading to a progressive decline in output voltage as spring stiffness increases.

The corresponding trends in bandwidth and quality factor are consistent with this interpretation. For example, the root-mounted case with $d = 2.0$ mm produced a relatively high output voltage while exhibiting a moderate quality factor of 26.13, whereas the tip-mounted case with the same spring diameter showed a higher quality factor of

39.07 but a marked reduction in voltage. This contrast suggests that the lower output in the tip-mounted configuration cannot be attributed primarily to increased energy dissipation. Rather, it is more consistent with a reduction in effective strain-energy input to the piezoelectric layer.

3.2. Magnetic Tuning

Magnetic tuning modifies the dynamic response of a piezoelectric cantilever through a distance-dependent magnetic interaction that changes the effective stiffness of the system. Because this interaction varies nonlinearly with magnet spacing, it can shift the resonant frequency, alter the shape of the frequency response curve, and, in some cases, broaden the effective operating range. Compared with conventional elastic tuning, magnetic tuning offers several practical advantages, including non-contact stiffness adjustment, continuous tunability through simple spacing control, and compact implementation without substantial modification of the cantilever structure.

As observed in Figure 4 and Table 2, magnetic-based tuning achieved effective and continuous resonance adjustment through repulsive forces between like magnetic poles. Reducing the inter-magnet spacing from 8.5 mm to 3.5 mm increased the resonant frequency from 38.0 to 41.4 Hz, indicating that magnetic interaction provides a sensitive and controllable stiffness modulation mechanism. Notably, a spacing of 7.5 mm resulted in a resonant frequency of 38.8 Hz, which exactly matches the dominant vibration frequency of the bridge deck. In contrast to its frequency tunability, magnetic tuning consistently reduced electrical output. Across all tested spacings, maximum RMS voltages were lower than those of the baseline configuration and decreased monotonically as the magnetic gap narrowed. The maximum voltage declined from 7.54 V at 8.5 mm to 6.70 V at 3.5 mm, corresponding to an attenuation of approximately 15–25% relative to the baseline. Despite this reduction, the -3 dB bandwidth remained relatively stable, and exhibited a slight increase compared with the baseline case.

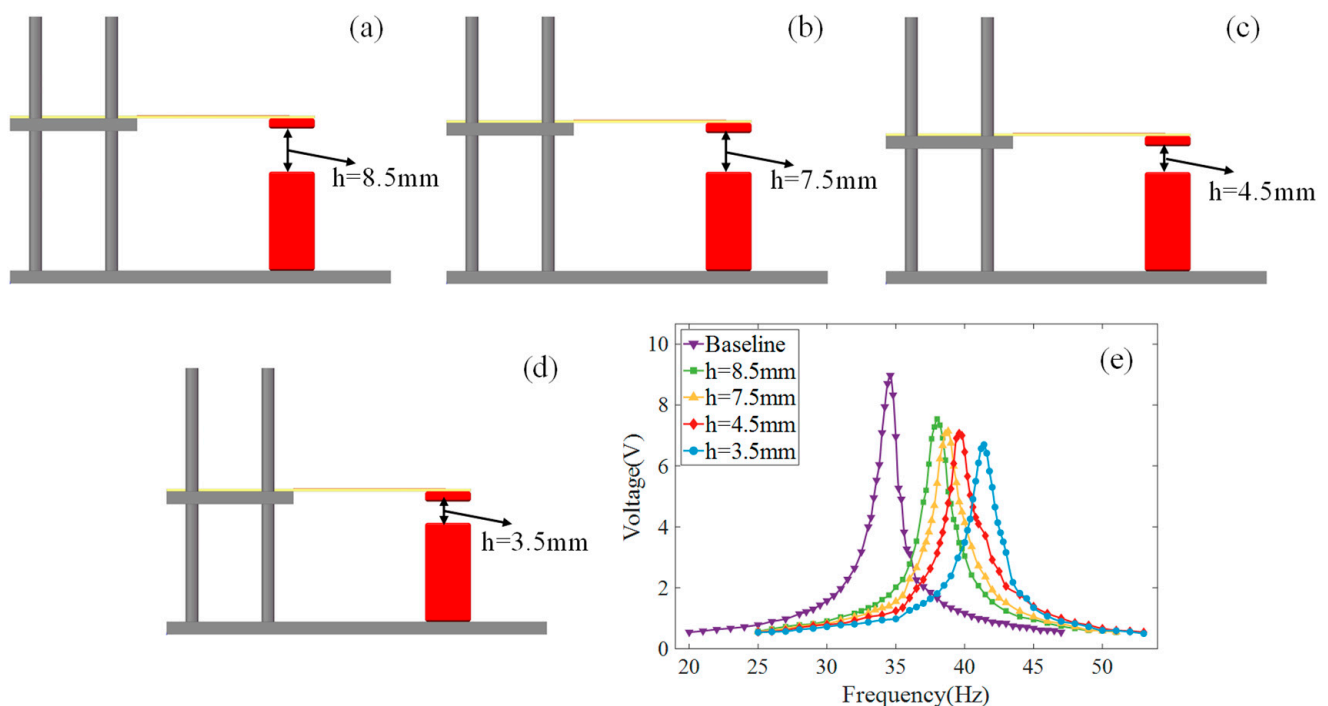


Figure 4. Schematic diagrams of different magnetic tuning configurations (a–e) frequency response curves.

Table 2. Magnetic frequency tuning output performance.

Tuning Methods	Tuning Component Specs	Resonant Frequency (Hz)	Voltage (v)	Bandwidth (Hz)	Quality Factor
Magnetic tuning	Baseline	34.6	8.97	1.2	28.83
	h = 8.5 mm	38.0	7.54	1.5	25.33
	h = 7.5 mm	38.8	7.14	1.6	24.25
	h = 4.5 mm	39.6	7.07	1.5	26.40
	h = 3.5 mm	41.4	6.70	1.6	25.88

The observed trends reflect the dual role of magnetic repulsion in modifying both the dynamic stiffness and strain distribution of the cantilever system. The repulsive magnetic force introduces an effective nonlinear stiffness that elevates the natural frequency in a continuous and contact-free manner. However, this stiffness augmentation is concentrated near the free end of the beam, where it restricts displacement-dominated motion and alters the global bending profile. As the magnetic force intensifies, the curvature near the clamped end is reduced, leading to diminished strain energy within the piezoelectric patch region. This redistribution of strain provides a consistent explanation for the monotonic decrease in the voltage output with decreasing magnet spacing. The relatively stable bandwidth observed under magnetic tuning suggests that the nonlinear stiffness contributes to mild resonance broadening without inducing pronounced damping or instability within the tested parameter range.

The continuous decrease in output voltage with decreasing magnet spacing can be interpreted in terms of electromechanical energy-transfer efficiency. Although the repulsive magnetic interaction enables resonance tuning, it also introduces a position-dependent nonlinear force near the free end of the cantilever. This additional force modifies the local effective stiffness and may alter the bending deformation pattern of the beam. As a result, the curvature in the vicinity of the clamped end, where the piezoelectric layer is attached, is reduced, leading to lower strain input to the electromechanically active region and, consequently, a reduction in voltage output. In parallel, the operating bandwidth remains relatively stable across different magnet spacings, suggesting that the observed voltage decrease is unlikely to arise primarily from increased damping. Instead, the results are more consistent with a reduction in power-transfer efficiency caused by a less favorable strain distribution relative to the position of the piezoelectric sheet.

3.3. Frequency Tuning via Proof Mass and Beam Length Modification

In addition, proof mass tuning and cantilever length tuning are among the most direct mechanical approaches for adjusting the resonance of piezoelectric cantilever harvesters. Increasing the tip mass lowers the natural frequency by increasing the effective inertia of the beam–mass system, whereas increasing the beam length reduces the bending stiffness and similarly shifts the resonance toward a lower-frequency range.

Table 3 demonstrate that adjusting the proof mass at the free end and modifying the cantilever beam length both produced substantial shifts in resonant frequency, but were accompanied by pronounced reductions in the electrical output. Specifically, decreasing the proof mass from 16.50 g (baseline model) to 6.25 g increased the resonant frequency from 34.6 to 49.2 Hz, while the maximum voltage decreased monotonically from 8.97 to 4.14 V. A similar trend was observed for beam length modification, where shortening the cantilever from 70 to 68 mm raised the resonant frequency to 42.3 Hz, but reduced the maximum voltage to 4.65 V. Across both tuning approaches, the -3 dB bandwidth exhibited only minor fluctuations, with no consistent broadening observed.

Table 3. Geometric frequency tuning output performance.

Tuning Methods	Tuning Component Specs	Resonant Frequency (Hz)	Voltage (v)	Bandwidth (Hz)	Quality Factor
beam length modification	Baseline	34.6	8.97	1.2	28.83
	L = 69 mm	39.2	5.86	1.0	39.20
	L = 68 mm	42.2	4.66	1.7	24.82
proof mass modification	Baseline	34.6	8.97	1.2	28.83
	m = 14.38 g	35.8	8.73	1.2	29.83
	m = 11.03 g	39.2	6.92	2.1	18.67
	m = 6.25 g	49.2	4.54	1.5	32.80

These trends reflect fundamental changes to the system's inertial and stiffness properties. Reducing the proof mass directly lowers the inertial force generated under harmonic excitation, thereby decreasing bending moment and strain energy along the beam. As a consequence, the electromechanical coupling within the piezoelectric patch is weakened, leading to a substantial drop in voltage output despite effective frequency elevation. Beam length reduction similarly increases the system's natural frequency by increasing the effective stiffness and altering the area moment of inertia. However, it also shortens the lever arm over which bending develops. This geometric constraint suppresses global deformation and curvature near the clamped end, thus resulting in diminished strain-induced electrical output. It is noteworthy that, unlike spring- or magnetic-based tuning, which modify effective stiffness locally or externally, mass and length tuning fundamentally alter the global structural parameters governing modal behavior. The observed lack of bandwidth enhancement further indicates that these approaches primarily shift resonance rather than reshape the frequency response.

From the perspective of electromechanical energy transfer, both proof-mass tuning and beam-length tuning alter the global structural characteristics that govern energy conversion in the harvester. Reducing the proof mass decreases the effective inertia of the cantilever system and, therefore, lowers the inertial force available to drive bending deformation. Consequently, less strain energy is transferred to the piezoelectric layer, and the electrical output decreases even when the resonance frequency is adjusted to the target value. Similarly, shortening the cantilever beam increases the effective stiffness while reducing the bending moment arm. This suppresses curvature development near the clamped end, where the piezoelectric patch is located, and weakens the electromechanical coupling responsible for voltage generation. In both cases, the reduced strain input to the piezoelectric region leads to lower power-transfer efficiency. Since neither method produced a notable increase in bandwidth, the results suggest that mass and length tuning mainly shift the resonance frequency without fundamentally improving the energy-conversion process.

3.4. Comparative Analysis at the Dominant Bridge Deck Frequency

As highlighted hereinabove, a primary objective of this study is to identify frequency tuning strategies that perform effectively within the characteristic vibration band of bridge decks, centered at approximately 38.8 Hz. To this end, six representative experimental configurations were selected and evaluated under resonance-matched conditions. The comparison focuses on maximum RMS voltage and -3 dB operating bandwidth, which together reflect energy conversion capability and tolerance to frequency variation.

As summarized in Table 4, the root-mounted spring configuration with a wire diameter of 2.0 mm achieves the highest overall performance, delivering a maximum voltage of 9.01 V with a bandwidth of 1.5 Hz. It is to be noted that this is the only tuning strategy

that simultaneously raises the resonant frequency to the target value and enhances voltage output relative to the baseline, while other tuning methods incur a noticeable voltage penalty, with reductions ranging from approximately 15% to 35%, despite successful frequency alignment. The differences in the operating bandwidth among the tuning strategies are comparatively modest. Mass tuning produces the widest bandwidth (2.1 Hz), while beam length modification yields the narrowest (1.0 Hz).

Table 4. Output performance of various tuning methods.

	Tuning Methods	Tuning Component Specs	Voltage (v)	Bandwidth (Hz)	Quality Factor
Case 1	spring at root	d = 2.0 mm	9.01	1.5	26.13
Case 2	spring at mid-span	d = 1.6 mm	8.25	1.8	18.70
Case 3	spring at tip	d = 1.4 mm	8.62	1.7	23.06
Case 4	magnetic frequency tuning	h = 7.5 mm	7.14	1.6	24.25
Case 5	mass tuning	m = 11.03 g	6.92	2.1	18.67
Case 6	length tuning	L = 69 mm	5.86	1.0	39.20

4. Concluding Remarks

This study addressed the challenge of resonance mismatch in bridge vibration energy harvesting by experimentally comparing four representative frequency tuning strategies for cantilever PE-VEHs under bridge-relevant excitation conditions (≈ 38.8 Hz). Using a unified experimental framework, the dynamic and electrical performances of spring tuning, magnetic tuning, tip mass adjustment, and beam length modification were systematically evaluated and directly contrasted.

The results demonstrate that stiffness augmentation at the cantilever root via spring tuning provides the most favorable overall performance. Specifically, root-mounted spring tuning increased the resonant frequency from the baseline value to the target bridge frequency while simultaneously enhancing the maximum output voltage, and maintaining a comparable operating bandwidth. This behavior is consistent with the underlying mechanism identified in the discussion, whereby stiffness modification in the region of maximum modal strain preserves and, in some cases, enhances electromechanical coupling. In contrast, magnetic tuning enabled continuous frequency adjustment but incurred a voltage reduction of approximately 15–25%, attributable to strain redistribution caused by a free-end stiffness constraint. Tip mass reduction and beam length shortening produced large frequency shifts but resulted in pronounced voltage degradation (up to $\sim 50\%$), reflecting fundamental losses in inertial excitation and bending deformation.

While the results in this study provide important implications for enhancing the performance of similar cantilever PE-VEHs, several limitations should be acknowledged. It is to be noted that all tuning configurations were evaluated under a fixed electrical load, and configuration-specific load optimization was not carried out. Consequently, the comparison was conducted based on RMS output voltage rather than on maximum output power under optimal impedance matching. Although the voltage-based comparison provides a consistent basis for assessing the relative effectiveness of different frequency tuning strategies under a unified experimental framework, it does not fully represent the maximum power generation capability of each configuration. On the other hand, the performance comparison among selected tuning strategies is confined to the specific harvester designs and controlled experimental conditions. In practice, the relative effectiveness of a given tuning method may depend on multiple case-specific factors, including cantilever dimensions, proof mass distribution, boundary conditions, electromechanical coupling characteristics, and target frequency range. Therefore, the present results should

be interpreted within the scope of the tested configurations and operating conditions, i.e., representing a case-dependent, application-oriented assessment for the bridge-relevant low-frequency conditions.

On this account, future studies should extend the present work to include systematic load resistance sweeping for each tuning configuration to determine the optimal electrical load and maximum output power under matched impedance conditions, which would enable a more complete comparison of energy-harvesting performance and improve consistency with the broader literature. In addition, the analysis should be expanded to a wider range of harvester geometries, proof mass distributions, boundary conditions, and tuning implementations, given that the relative effectiveness of each strategy may depend strongly on configuration-specific factors. Furthermore, more realistic bridge vibration environments should be considered, including those that are time varying and that are characterized by broadband excitation induced by traffic, trains, and wind.

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