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Piezo stack energy harvesters with protection components for railway applications

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A R T I C L E I N F O

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A B S T R A C T

Piezo stack energy harvesting from railway vibrations shows substantial potential for powering wireless sensor networks responsible for monitoring railway infrastructure. Nevertheless, assuring the safety and durability of these energy harvesters in the condition of the dynamically fluctuating railway vibrations remains a considerable challenge. In this paper, we present two innovative protection methods designed for piezo stack energy harvesters with a frequency up-conversion mechanism. These methods involve the incorporation of ring-type and circular stoppers within the resonant system and the utilisation of proposed impact protection components within the impact system. Two distinct harvesters incorporating the proposed protection components, Harvester Types I and II, are designed to align with their respective operating acceleration targets, which are determined based on the amplitudes of railway vibration acceleration. Finite element modelling is used to guide the design process, involving the evaluation and selection of various design parameters aligned with the acceleration target and stress limits. The protection mechanisms' effectiveness and impact on energy harvesting performance have been validated through experimental testing under measured rail vibration signals, and their performance has been further assessed through stress analyses. Harvester Type I operates seamlessly within a 5 g threshold, effectively mitigating the increase in maximum output acceleration and stress beyond this limit. In contrast, Harvester Type II efficiently dissipates excess energy, enabling the harvester to operate at a 15 g acceleration while reducing the rate of acceleration and stress growth. The results demonstrate that both proposed harvesters provide effective protection from unexpected railway vibration overloads. This research makes a significant impact on the practical, real-world implementation of piezo stack energy harvesters operating within the dynamic environment of railway vibrations.

1. Introduction

Railroads outpace other modes of transportation in terms of energy efficiency, making them the most environmentally responsible choice for passenger travel and freight transportation [1]. With an extensive network spanning over 854,000 kilometres of railway lines in annual operation [2], their impact on sustainable transport is undeniable. Monitoring railway infrastructure plays a critical role in enhancing the efficiency, safety, and reliability of railway operations, and wireless sensor networks (WSNs) are instrumental in this regard. Nevertheless, the persistent challenge in this domain revolves around the availability of a dependable power source. Conventional power sources, including batteries and cables, come with inherent limitations. Batteries suffer from a restricted lifespan, necessitate intricate replacement procedures, and pose adverse environmental consequences. On the other hand, the installation of cables can be quite arduous, particularly in remote locations or specific underground tunnels. These constraints have spurred research endeavours into energy harvesting technologies, which harness ambient energy from the railway environment to generate electrical power. One promising energy source is the energy derived from vibrations induced by passing trains. Researchers are exploring track-side vibration energy harvesters, employing both electromagnetic [3–5] and piezoelectric [6–8] methods. These innovations are poised to address the power source challenge and further advance the effectiveness of railway infrastructure monitoring systems.

However, a primary hurdle in track-side vibration energy harvesters lies in ensuring the safety of the energy harvester amid the dynamically fluctuating and occasionally substantial railway vibrations. The magnitude of these vibrations depends on several factors, including train speed, track condition, and the type of rolling stock, resulting in a

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spectrum of vibration intensities ranging from minor disturbances to significant ground motion. Without adequate protection components, even if the harvester is designed to withstand specific levels of acceleration, the occasional occurrence of substantial accelerations could lead to mechanical overloads, posing a significant risk of damage to the harvester. As a result, the placement of energy harvesters along railway tracks or sleepers accentuates the importance of protection components, as any malfunction or failure could potentially jeopardize not only the harvester itself but also the overall integrity of the railway system. Moreover, protection components play a vital role in minimizing maintenance requirements and associated costs. By reducing the likelihood of damage or malfunction due to high acceleration level, these components extend the operational lifespan of energy harvesters, thereby decreasing the need for frequent repairs or replacements. This not only reduces downtime and operational disruptions but also translates into significant cost savings in the long run.

Overall, the incorporation of protection components into energy harvesters for railway applications is essential not only for ensuring the safety and reliability of infrastructure but also for reducing maintenance costs. This integration yields benefits such as enhancing operational efficiency, bolstering safety measures, and promoting sustainability within the railway sector. Therefore, protection components emerge as indispensable prerequisites for the successful deployment of vibration energy harvesters in railway applications.

Protection techniques have been suggested for electromagnetic energy harvesters tailored to railway applications. These methods encompass the utilisation of flywheels for rotary electromagnetic harvesters and the implementation of stoppers for translational electromagnetic harvesters. For example, a half-wave flywheel was designed to enhance damping forces during the downward track vibration while reducing the damping force in the upward reset process [9]. This method improves the stability of the system and reduces the impact on the vibration source during the reset process. In another research by Kim et al. [10], it was revealed that the primary factor leading to failure in vibration energy harvesters is the occurrence of abrupt overloads. To address this issue, a mechanical stopper was introduced to an electromagnetic energy harvester, aimed at safeguarding the spring from potential damage caused by vibrations originating from the railway track. The stopper's design was engineered to confine the spring's movement to a maximum displacement of 2 mm when subjected to vibrations exceeding 2 g acceleration [11], thus providing a robust solution to prevent failures.

Integrating electromagnetic energy harvesters into the railway metallic environment presents challenges due to the involvement of moving magnets [12] or their relatively large size, which can impede regular track maintenance [1]. In contrast, piezoelectric vibration energy harvesting has emerged as a promising alternative for extracting power from railway track vibrations, owing to its simple design and remarkable energy and power density [13–15]. Various piezoelectric vibration energy harvesting designs have been explored, including piezoelectric cantilevers [16–19], simply supported beams [20], and cantilever arrays [21]. Nonetheless, these designs tend to yield relatively low power outputs. As a result, the utilisation of frequency up-conversion mechanisms involving mechanical impact has recently gained attraction for harnessing railway vibrations [22,23]. This approach enables the piezo stack harvester, characterised by its high resonant frequency, to effectively harness the low-frequency vibrations encountered in railway applications, thus addressing the frequency discrepancy and generating high power output.

Nevertheless, the operation of these harvesters necessitates mechanical impact within the harvester, introducing the potential risk of mechanical overloads that could lead to damage due to the substantial impact forces involved. Despite this critical concern, there is currently a dearth of research on protection methods for piezoelectric harvesters in railway applications. Without adequate protection methods, these harvesters are vulnerable to the harsh vibrations encountered in railway environments, rendering them unsuitable for real-world applications. Existing research concerning protection components for harvesters with frequency up-conversion mechanism tends to prioritise expanding the frequency domain rather than focusing on protection methods. For instance, one innovative design involves the use of a rope to limit the displacement of the cantilever, resulting in a remarkable 4.2-fold increase in the frequency bandwidth [24]. In another approach, Liu et al. [25] employed a straight cantilever to serve dual roles as both a stopper and an impact system generating power, effectively broadening the operational bandwidth by an additional 14 Hz. Additionally, Wang et al. [26] introduced an arc-shaped contact surface to prevent excessive displacements of the cantilever, resulting in improved power output and broader operating bandwidth. As a result, there is an urgent need for the investigation of protection methods, mechanisms, and designs for piezoelectric energy harvesters in railway applications.

In summary, there is a notable research gap concerning the development of protection methods, mechanisms, and designs for piezoelectric energy harvesters incorporating frequency up-conversion mechanisms when subjected to railway vibrations. Addressing this gap is imperative for advancing the practical implementation of these energy harvesters within railway applications. Consequently, this study introduces two protection methods tailored for piezo stack harvesters with frequency up-conversion mechanisms and integrates them into harvesters for railway applications. Subsequently, two distinct harvester types, referred to as Harvester Types I and II, are proposed and designed in alignment with their specific design criteria. FEM (Finite Element Method) modelling is performed to guide the design of the energy harvesters. The efficacy of their protection mechanisms and energy harvesting performance is validated through experimental tests. The results demonstrate that these novel harvesters effectively shield the energy harvesters from mechanical overload and moderate stress levels within the demanding railway environment.

2. Design of the harvester with protection components

2.1. Design targets

This work focuses on developing innovative protection strategies for energy harvesters and incorporating them smoothly into a piezo stack energy harvester. The objective is to reliably protect the harvester from overloads that arise unpredictably due to the highly variable and occasionally substantial vibrations experienced during railway operations. By doing so, a durable and robust harvester could be designed and manufactured to withstand the challenging conditions of railway environments.

Fig. 1 depicts the schematic of vibrations induced by a passing train. As a train passes the track, both the rail track and sleepers experience vibrations at a specific acceleration. These vibrations caused by railways are broadly divided into two categories based on their acceleration

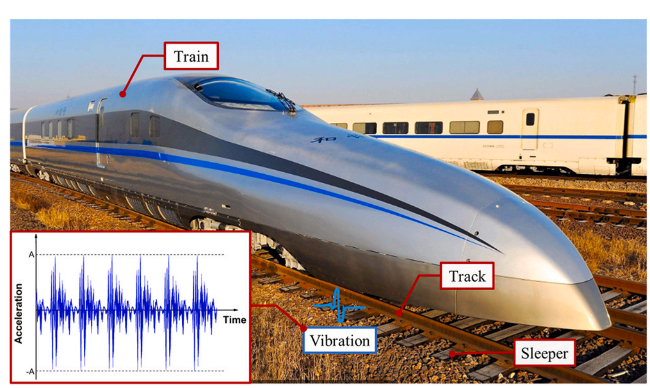

Fig. 1. The schematic of train-induced vibrations.

amplitudes. The first category encompasses vibrations with relatively low acceleration amplitudes, usually between 1 g and 5 g. Examples of this include vibrations from low-speed rail tracks, such as those caused by China's A-type metro trains travelling at speeds between 5 km/h and 25 km/h [27], and vibrations generated by locomotives VL80 with a speed limit of 110 km/h [28]. Additionally, vibrations from sleepers on high-speed inter-city train routes fall into this category, with the Great Western Main Line at Steventon in the UK, where trains travel at speeds up to 195 km/h [29–31], serving as a case in point. In these situations, the location for installing the harvester, whether on the rail or the sleeper, can be selected based on where these lower levels of acceleration are most prevalent.

In contrast, the second category is marked by substantially higher acceleration amplitudes, generally falling within the range of 60 g to 70 in cases of rail track vibrations, observed in track structures like the Lotschbergbasis railway tunnel in Europe [20] and the Qinghai–Tibet railway in China [32]. These intense track accelerations pose a significant challenge to the harvester's durability and survival. Despite these high vibration acceleration levels in the tracks, it's worth noting that the accelerations at the sleeper are relatively lower. Studies indicate that while rail accelerations might soar up to 70 g, the accelerations recorded at the sleepers often average below 15 g [20], a parameter that can be harnessed for the harvester's operation, suggesting that its installation should be limited to the sleepers for optimal performance and durability.

Consequently, 5 g and 15 g acceleration targets are employed as the designated design criteria for our case studies in these two distinct categories, respectively. This work first proposes two innovative protection methods and then introduces two distinct energy harvester designs with these protection components (referred to as PC), denoted as Harvester Types I and II, each tailored to suit the respective scenarios. In cases featuring relatively low acceleration magnitudes, the primary aim is to ensure that the harvester functions effectively under a 5 g threshold while safeguarding the device against abrupt overloads. Conversely, for cases confronting relatively high acceleration magnitudes, the goal is to enable the harvester to efficiently dissipate excess energy and operate effectively under a 15 g threshold while shielding the device from unexpected overloads.

2.2. Design and working principle of the protection methods

Fig. 2(a) illustrates the proposed methods. This includes the stopper design within the resonant system and the Impact Protection Components (IPC) design within the impact system. Fig. 2(b) and 2(c) illustrate the working principle of the stopper and IPC design, respectively.

In the stopper design, two stoppers are positioned both above and below the spring at a distance denoted as d_1. The top stopper takes the form of a solid circular plate, serving to limit the upward displacement of the spring. In contrast, unlike conventional designs found in previously reported resonant energy harvesters that typically employ a solid plate or rod, the bottom stopper introduces a novel ring-type structure with a central aperture. This design innovation allows the inertial mass to oscillate and interact with the impact system beneath the resonant system while curtailing spring deformation upon reaching its limiting state.

In terms of its working principle, during the operating state, the spring and the inertial mass undergo deformation while maintaining a distance from the stopper. This ensures that the stoppers remain inactive, allowing the energy harvester to perform as intended. However, when a large input excitation is applied, which could potentially jeopardise the harvester, the stoppers come into play, transitioning into its limiting state. In this mode, the stoppers effectively curtail the spring's displacement, thus preventing the excessive impact forces between the two systems. This protection method serves as a reliable shield, safeguarding the energy harvester from abrupt mechanical overloads.

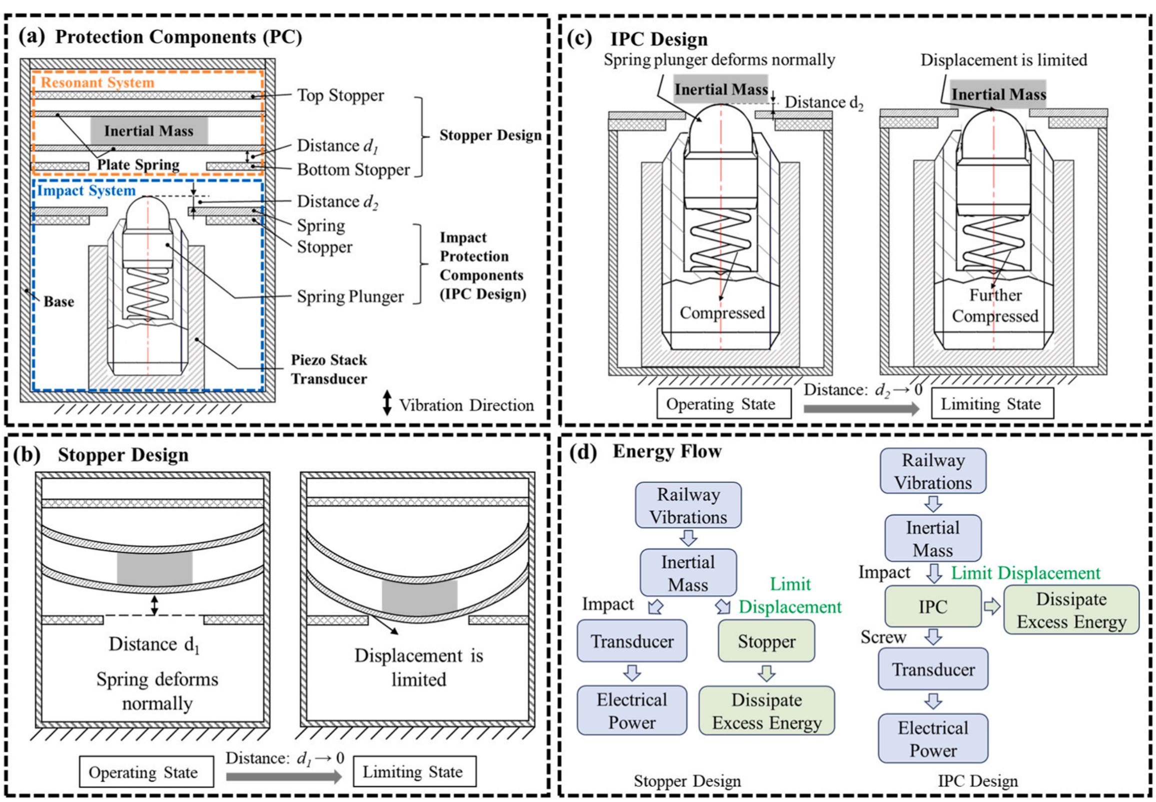

Fig. 2. The schematic of (a) the proposed protection components and their relative position, including the stopper design within the resonant system and the impact protection components (IPC) design within the impact system, (b) the working principle of the stopper design, (c) the working principle of the IPC design, and (d) the energy flow of the harvesters with stopper and IPC design.

Regarding the IPC design, it is situated above the piezo stack transducer within the impact system, comprising a spring, a stopper, and a spring plunger. The spring plunger, designed for applying a controlled force or pressure when engaged, comprises a hollow screw or pressing sleeve with an integrated spring and a ball or a pin. This component is directly connected to the piezo stack transducer. It serves as the conduit for transmitting impact forces to the transducer. Simultaneously, the spring and stopper are situated on the base. Notably, the spring features a central aperture, allowing the head of the spring plunger to protrude. A distance of d_2 separates the head of the spring plunger from the spring.

At the operating state, the compression spring within the spring plunger undergoes compression as a result of the impact force exerted by the inertial mass, and then the force is transferred to the transducer via the spring plunger. In this state, the head of the spring plunger is pushed down while still being above the spring. However, when a large excitation is applied, the spring plunger experiences further compression by the inertial mass until the inertial mass reaches the surface of the spring, where both the displacement of the inertial mass and the head of the spring plunger are constrained by the spring. In this limiting state, a portion of the energy is transmitted to the transducer, while the remaining energy is absorbed by the spring, stopper and base. In situations where d_2 is at a short distance, excess energy can be efficiently dissipated, allowing the IPC to not only serve as a vital safeguard protecting the harvester against potential overload conditions, but also facilitate the harvester's operation at the intended high excitation level.

Fig. 2(d) illustrates the energy flow of the harvesters with stopper and IPC designs at their limiting states. During railway vibrations, the inertial mass oscillates. In the harvester with the stopper design, the inertial mass collides with the transducer, thereby generating electrical power. However, the displacement of the mass is restricted by the stopper, which dissipates excess energy and safeguards the device from damage. Conversely, in the harvester employing the IPC design, the inertial mass impacts the IPC embedded within the transducer for power generation. Meanwhile, the IPC also plays a crucial role in limiting the inertial mass displacement, ensuring it remains within safe parameters. In doing so, surplus energy is effectively dissipated by the IPC, and the device is shielded from potential harm.

2.3. Design of Harvester Type I

Fig. 3(a) illustrates the design for Harvester Type I. The working principle of the harvester has been previously detailed in our previous work [23]. In a nutshell, the inertial mass oscillates in response to vibrations induced by railway motion. It subsequently impacts the piezo stack transducer, which has an added mass attached to it. This interaction allows the piezo stack to generate voltage and power when connected to a resistive load. However, the present study primarily focuses on the design and integration of the protection components. The objective of Harvester Type I is to operate effectively under a 5 g acceleration while ensuring robust overload protection.

Under 5 g excitation, the stopper design alone is sufficient enough to provide ample protection. Consequently, Harvester Type I only employs the stopper design. The essential protection components are a circular stopper situated above the inertial mass, and a ring-type stopper beneath it, both integrated at a distance of d_1 from the inertial mass. These stoppers are made from a relatively soft material to minimise the impact force generated upon contact. The circular stopper constrains the upward movement of the inertial mass. Meanwhile, the ring-type stopper, featuring a central hole, allows the inertial mass to oscillate and interact with the piezo stack transducer below while preventing its downward displacement. The circular stopper is covered by a top lid, and the ring-type stopper is secured by a stopper holder, both composed of a more rigid material, to support and shield them from substantial displacements.

2.4. Design of Harvester Type II

Fig. 3(b) provides an overview of the Harvester Type II design. The aim of Harvester Type II is to operate effectively under a 15 g acceleration while maintaining robust overload protection. When subjected to 15 g excitation, which poses a substantial risk of harvester damage even with the stopper design in place, the inclusion of an IPC design becomes imperative. The IPC design serves to further reduce impact forces, dissipate surplus energy, and provide device protection. It acts as a supplementary safeguard, particularly vital when confronted with high acceleration environments where the stopper design alone cannot suffice. Thus, Harvester Type II integrates both stopper and IPC designs.

The stopper design within the inertial mass system of Harvester Type II remains identical to that of Harvester Type I. However, the key difference between these two types lies in the integration of the IPC design within the piezo stack transducer system of Harvester Type II. This strategic integration effectively manages the dissipation of surplus energy when subjected to a 15 g high-intensity excitation. When such conditions arise, the spring and stopper components constrict the displacement of the inertial mass, subsequently limiting the displacement of the head of the spring plunger. This synchronised effort serves to efficiently absorb and dissipate a portion of the energy, thereby guaranteeing that only a fraction of the impact force is transmitted to the transducer. This ratio of force transmission, when compared to using the spring plunger alone, is defined in this paper as the coefficient of

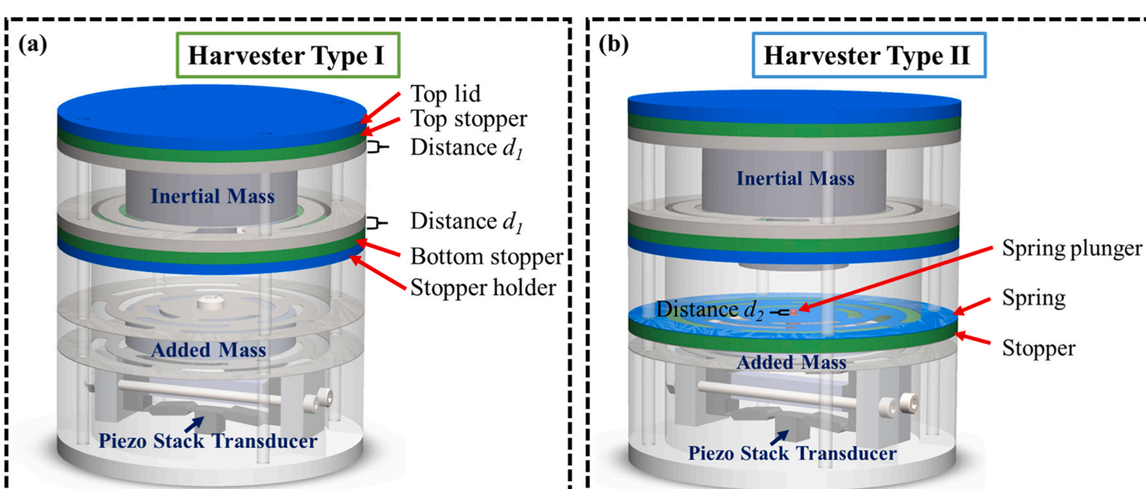

Fig. 3. The schematic of (a) the design of Harvester Type I, and (b) the design of Harvester Type II.

damping effects C_r.

3. Modelling and design of the parameters

3.1. Durability considerations

To ensure the robust safety and prolonged durability of the energy harvester, stress analysis is conducted to pinpoint the weakest component within the system. A full model of the energy harvester is then built incorporating the physical mass and spring elements, with the resulting stress distribution visually represented in Fig. 4(a). Notably, the piezo stack transducer emerges as the critical focal point, displaying the highest stress concentration. In Fig. 4(b), an enlarged view of the stress distribution within the piezo stack transducer is presented. It becomes evident that the maximum stress is consistently localised at the flexure hinges of the mechanical transformer due to substantial bending motion. Consequently, the structural integrity of the entire energy harvester is predominantly contingent on the resilience of this mechanical transformer. Thus, the stress limit of the mechanical transformer becomes the cornerstone of the harvester's design.

When subjected to railway vibration, the mechanical transformer experiences two types of forces. The first arises from the impact force between the inertial mass and the piezo stack transducer. The second is the inertial force induced by the track's random vibrations. To gauge the structural robustness under these dual forces, our developed model [33,34] is employed to conduct a stress analysis for each force. The most demanding scenario unfolds when both sets of forces generate their maximum stresses concurrently. Consequently, we compute the total combined maximum stress by summing the peak stresses from each force, ensuring that the harvester can withstand these conditions effectively.

In this work, the mechanical transformer is crafted from spring steel 60Si2CrVA, with a fatigue limit of approximately 750 MPa [35]. In the case of Harvester Type I, to ensure the longevity of the harvester, a safety factor of 1.5 is applied, which means the combined maximum stress would be kept at less than 500 MPa. Regarding Harvester Type II, ensuring the device's safety under such substantial excitation calls for a heightened safety factor. As a result, a safety factor of 2 is implemented in this case, meaning that the combined maximum stress does not exceed 375 MPa.

3.2. FEM modelling methods

FEM modelling is then developed in COMSOL to guide the design of the energy harvesters. The FEM in this study adopts the method established in a prior investigation focused on protective strategies for harvesters featuring frequency up-conversion mechanisms [34]. In this approach, the resonant system's velocity is initially analysed to calculate the impact force, which is subsequently employed as the excitation force for the impact system.

In the configuration of the resonant system, a plate spring structure is modelled, as depicted in Fig. 5(a). An added mass node is introduced to the top of the plate spring to represent the inertial mass. A spring foundation node is positioned on top to match the harvester's resonant frequency in experimental conditions. Fixed constraints are then applied to the four holes of the outer circle. A boundary load, representing the excitation, is introduced. To incorporate the effect of the stopper, an additional spring foundation node is included. The velocity of the resonant system is obtained and then employed to calculate the impact force, utilising the impulse-momentum principle [22,36]. This impact force is subsequently utilised as a boundary load F_{impact} for the impact system.

For the impact system, a piezo stack transducer is modelled as shown in Fig. 5(b). An added mass node is incorporated into the lower layer above the transducer to represent the added mass. Fixed constraints are imposed on the bottom of the transducer. In representing the spring plunger within the IPC design, a thin elastic layer node is applied to the central layer atop the transducer. A periodic triangular force F_{impact} is applied to the upper layer to account for the impact force. The impact force is further adjusted by a coefficient C_r, capturing the influence of the spring and stopper within the IPC design.

3.3. Design of the parameters

The impact force is a critical factor affecting the strength of the harvester, and the impact force transmitted to the transducer can be expressed as the following equation according to the Impulse-Momentum Theorem Formula [22,36]:

$$F = C_r \frac{\sum_{i=1,2} M_i v_i - \sum_{i=1,2} M_i u_i}{\Delta t} \tag{1}$$

where M_1 represents the total mass of the inertial mass; M_2 is the total mass of the transducer and added mass; u_i and v_i are the velocities of the mass before and after the collision; Δt denotes the change in time. Of these parameters, the added mass is variable. The velocity u_1 is influenced by the distance d_1, while the coefficient of damping effects C_r is affected by the distance d_2. As a result, added mass, distance d_1 and distance d_2 are discussed in this section.

3.3.1. Added mass

First, the added masses required for Harvester Types I and II are

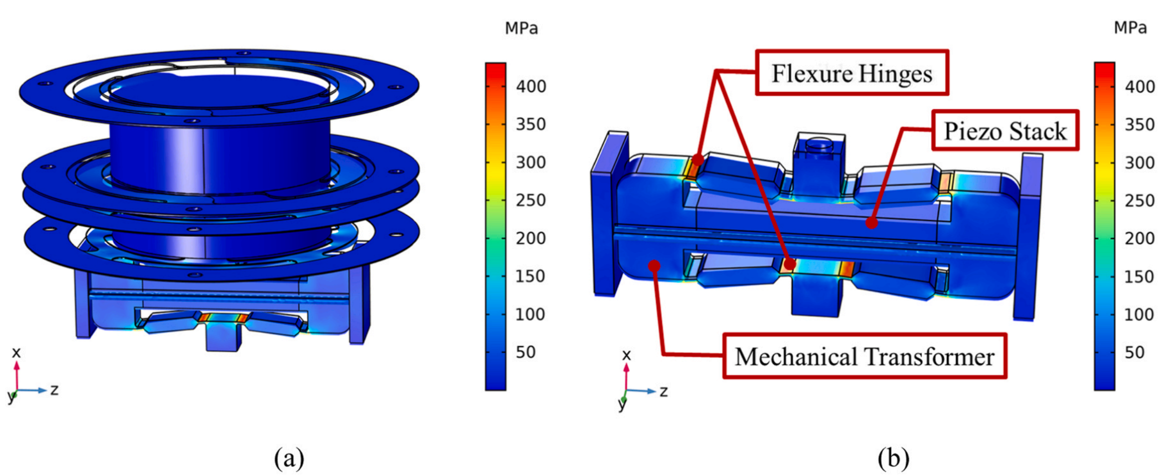

Fig. 4. (a) Full model of the energy harvester illustrating the stress distribution, and (b) an enlarged view of the stress distribution within the piezo stack transducer.

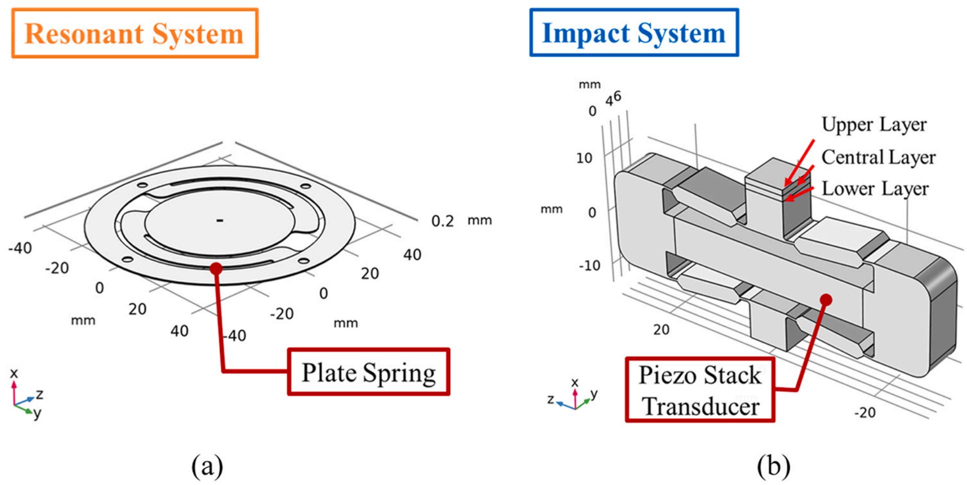

Fig. 5. Schematic of the finite element models: (a) plate spring structure modelling the resonant system, and (b) piezo stack transducer modelling the impact system.

chosen based on their individual stress thresholds. Fig. 6 presents the combined maximum stress levels of the energy harvester without any PC under various added masses subjected to 5 g and 15 g excitations. As the added mass is augmented, both the impact force and inertial force experience a corresponding increase. A larger impact force yields higher output, yet meanwhile leads to a heightened maximum stress level. Notably, under a 15 g excitation, the maximum stress is three times greater than that observed under a 5 g excitation.

In the case of Harvester Type I, the stress levels beneath the stopper's limiting threshold remain consistent with those in the absence of the PC. Consequently, an added mass of 150 g is selected, resulting in a combined maximum stress of 480 MPa, comfortably below the 500 MPa threshold.

Regarding Harvester Type II, under a 15 g excitation, the maximum stress levels in the harvesters without the PC significantly surpass the desired threshold. To mitigate this issue, measures are taken to reduce the stress, including a reduction in the added mass and the incorporation of the IPC design. Consequently, an added mass of 50 g is chosen.

3.3.2. Distance d_1

To determine the appropriate distance d_1 for the stopper design, the 5 g input acceleration threshold is taken into consideration. Fig. 7 illustrates the relationship between the input acceleration threshold and the corresponding distance d_1. As the input acceleration threshold rises, the displacement of the inertial mass correspondingly escalates, leading to an increase in the distance d_1. When the excitation acceleration remains below the input acceleration threshold, distance d_1 provides ample space for the inertial mass to vibrate freely without constraints.

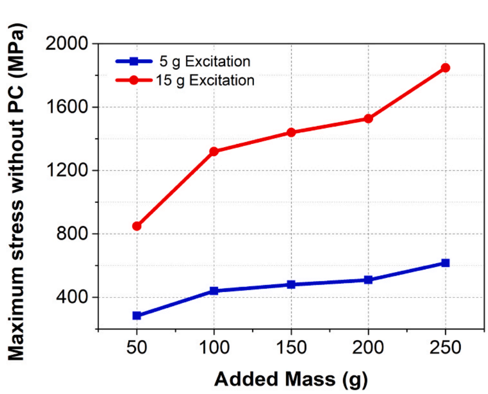

Fig. 6. The combined maximum stress levels of the energy harvester without any PC under various added masses subjected to 5 g and 15 g excitations.

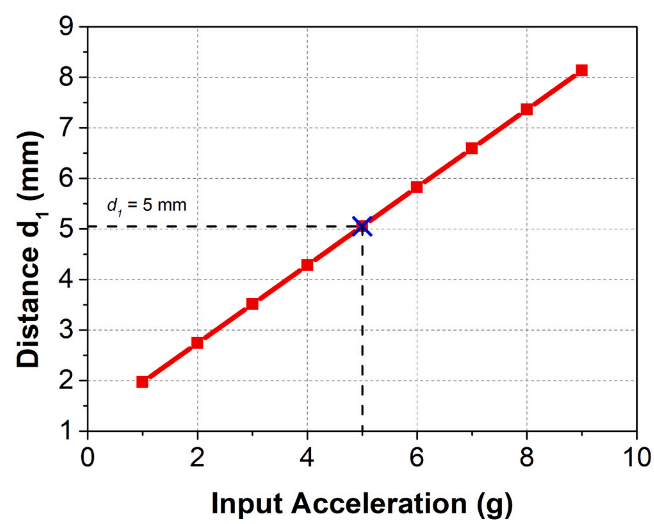

Fig. 7. The relationship between the input acceleration threshold and the corresponding distance d_1.

Conversely, if the excitation acceleration reaches or surpasses the input acceleration threshold, the displacement of the inertial mass exceeds the designated d_1, triggering the stopper mechanism. This promptly halts any further displacement of the inertial mass, effectively safeguarding the device from sudden overload. It is noted that when the input acceleration threshold is set at 5 g, the corresponding distance d_1 is approximately 5 mm. Consequently, a 5 mm distance for d_1 has been chosen.

3.3.3. Distance d_2

To determine the appropriate distance d_2 for the IPC design, the stress levels under a 15 g input acceleration are considered. It is observed that the maximum stress during random vibrations of the track with an added mass of 50 g is 112 MPa, leaving our goal to reduce the stress level from the impact force to 263 MPa. The ratio of this stress reduction between the device employing an IPC design and one utilising the spring plunger alone, determines the coefficient of damping effects C_r, which is linked to the distance d_2. Fig. 8 presents the relationship between the input acceleration, the coefficient of damping effects C_r, and the distance d_2. As C_r is not a constant, the range between the maximum value of C_r and 80 % of that value is used to partition regions for various distances d_2. With an increasing target input acceleration, a smaller percentage of energy is expected to be transmitted to the

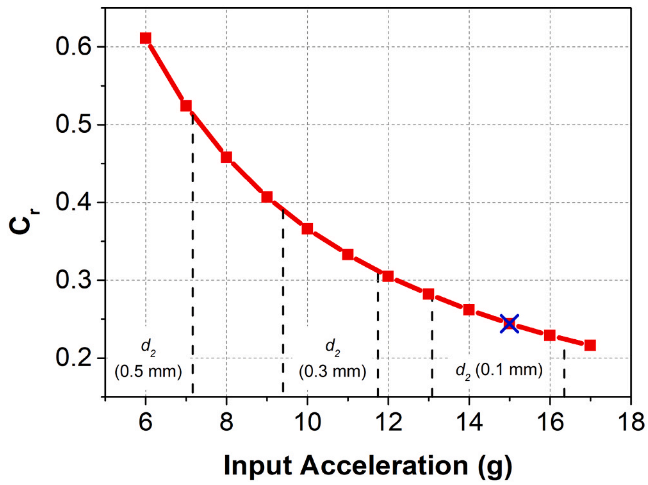

Fig. 8. The relationship between the input acceleration, the coefficient of damping effects C_r, and the distance d_2.

transducer to meet the stress requirements. Consequently, C_r decreases, resulting in a reduction in the distance d_2. Notably, when the input acceleration is set at 15 g, C_r should be approximately 0.24, and thus the distance d_2 of approximately 0.1 mm is selected.

4. Experiments

4.1. Experimental setup

To evaluate the performance of the Harvester Types I and II, experiments have been carried out employing the design as previously specified. Fig. 9(a) illustrates the experimental setup. In this configuration, a computer generates a measured rail vibration signal, which is then channelled into a Tektronix AFG3022C signal generator. This signal is subsequently amplified by an APS 125 power amplifier before being directed to an electromagnetic shaker (APS 113). The energy harvester, along with an accelerometer (Kistler 8762A5), is mounted onto the shaker, while the energy harvester itself is connected to a load resistor. The NI Data Acquisition Card (cDAQ-9174) is used to gather essential data including the output voltage and acceleration, which are then transmitted to the computer and analysed by LabVIEW to calculate the power output. The material properties of the harvesters are documented in Reference [8], whereas the materials used for the protection components (PC) are outlined below.

Fig. 9(b) shows the configuration of the IPC design in Harvester Type II. The IPC design is integrated into the impact system and consists of three key components: a spring plunger, a spring, and a vibration-dampening stopper positioned beneath the spring. The spring plunger and the spring are crafted from stainless steel, while the stopper is constructed from TPU, renowned for its remarkable vibration-dampening attributes. The spring plunger is screwed into the piezo stack transducer, while the spring and stopper are situated on the base.

In Fig. 9(c), the left side illustrates the stopper design employed in both Harvester Types I and II. Meanwhile, the right side offers a closer look at the custom-designed ring-type bottom stopper and its restricted area. Both the top and bottom stoppers are crafted from TPU. To provide structural support and protection against substantial displacements, the stopper holder and top lid are manufactured using PLA. Washers are employed to adjust the distance settings. The hole diameter in the stopper is precisely calibrated to match the inner side of the plate spring. This innovative design allows the central circular plate and the inertial mass to vibrate freely while simultaneously restricting the deformation of the plate spring.

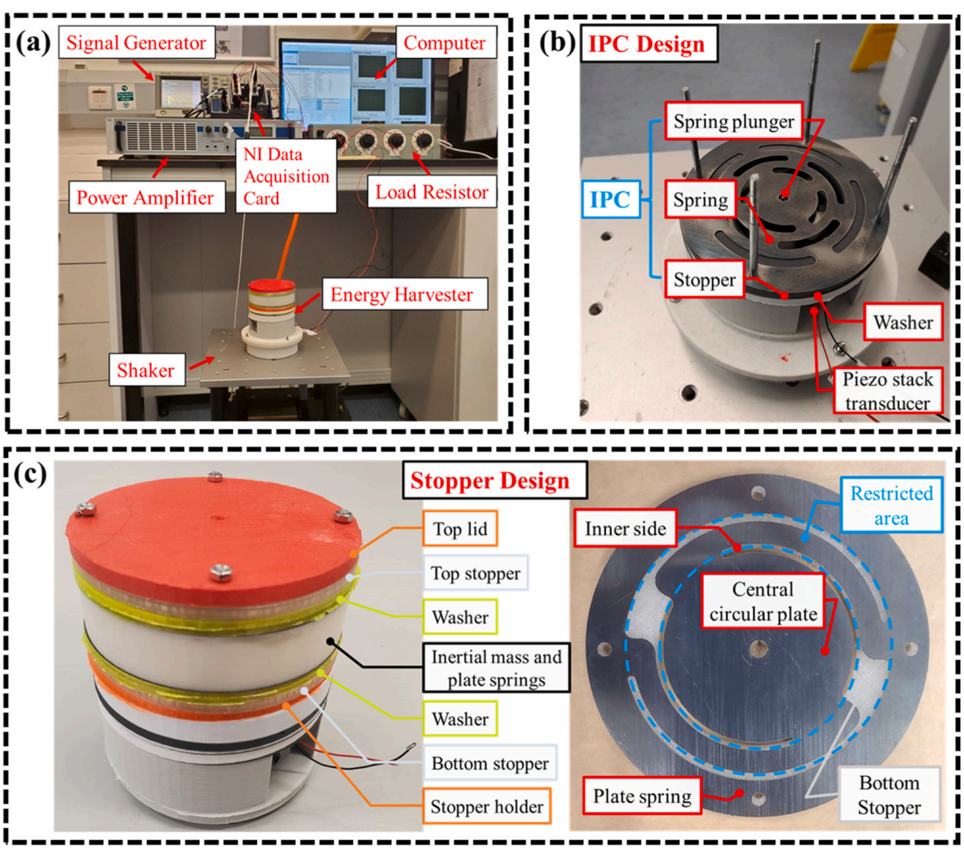

Fig. 9. (a) The experimental setup for testing the performance of the harvesters, (b) the configuration of the IPC design in Harvester Type II, and (c) the configuration of the stopper design employed in both Harvester Types I and II.

4.2. Experimental results

4.2.1. Performance of Harvester Type I

Fig. 10 illustrates the energy harvesting performance of Harvester Type I. In Fig. 10(a), the maximum output acceleration across various input voltage amplitudes is presented. To assess the impact of the damping effects of the PC, a reference line (represented as a dashed curve) that models the acceleration of the energy harvester without the stopper is introduced. This reference curve is fitted based on the measured data within the acceleration threshold. According to the simulation, in the absence of any PC, the output acceleration closely aligns with the input acceleration. Consequently, the target excitation threshold is set at the voltage amplitude at which the harvester without the PC reaches its maximum output acceleration of 5 g. When the input is below the target excitation, Harvester Type I exhibits a similar growth rate in maximum acceleration as observed when no PC is present. However, as the input exceeds the target excitation, the growth rate of Harvester Type I's maximum acceleration slows down compared to the situation without a PC. This phenomenon highlights the protective benefits offered by the PC components.

Having confirmed the protective characteristics of Harvester Type I, its power output subjected to the measured rail vibration signal is then assessed. Fig. 10(b) presents the variations in average power and peak power of Harvester Type I across a range of input voltage amplitudes. Notably, both the average power and peak power exhibit a consistent upward trend as the input voltage amplitudes increase. At the target excitation level, the harvester yields an average power output of approximately 4.6 mW, with the peak power reaching around 718 mW.

4.2.2. Performance of Harvester Type II

Fig. 11 provides an insight into the energy harvesting capabilities of Harvester Type II. Fig. 11(a) depicts the maximum output acceleration across a range of input voltage amplitudes. Similarly, to assess the impact of the damping effects of the PC, a reference line (represented as a dashed curve) that models the acceleration of the energy harvester without the stopper and IPC is introduced. This reference curve is constructed using measured data within the acceleration threshold. The target excitation threshold is set at the voltage amplitude where the harvester, operating without the stopper and IPC design, attains its maximum output acceleration of 15 g. Furthermore, the dots represent the measured data under the constraint imposed by the shaker, limiting the voltage amplitude up to 2.5 V, while the solid lines correspond to the fitted curves generated from the data collected. It's noteworthy that Harvester Type II exhibits significantly lower maximum acceleration than the harvester without the stopper and IPC, indicative of its ability to effectively dissipate excess energy. At the target excitation, the output acceleration is efficiently damped to approximately 5 g, providing a safety margin for the harvester. Moreover, with rising input voltage amplitudes, Harvester Type II demonstrates a diminishing growth rate in its maximum acceleration, distinguishing it from the scenario without a PC. Consequently, Harvester Type II plays a dual role: dissipating excess energy to maintain operation at the target excitation level and safeguarding the harvester under overload conditions.

After establishing the protective capabilities of Harvester Type II, its power output is then evaluated. Fig. 11(b) depicts the changes in both average power and peak power of Harvester Type II across various input voltage amplitudes. It is found that both the average power and peak power consistently increase as the input voltage amplitudes rise. Upon reaching the target excitation level, the harvester achieves an average power output of approximately 11 mW, while the peak power surges to around 1365 mW.

4.3. Discussion on stress analysis

Fig. 12 illustrates the combined maximum stresses experienced by Harvester Types I and II across various input accelerations, as determined through a combination of the experimental data and FEM modelling. To provide a comparison, a dashed curve serves as a reference line, representing the combined maximum stress of energy harvesters without PC under their respective operational conditions. In the case of Harvester Type I, as the input acceleration reaches the target of 5 g, the rate of growth in combined maximum stress becomes notably more gradual when compared to the harvester operating without a PC. Consequently, with a safety factor of 1.5 in place, the maximum allowable input acceleration for this harvester increases from 5 g to 7 g. Conversely, Harvester Type II displays a marked reduction in both the maximum stress levels and their corresponding growth rates when compared to the harvester without PC. This outcome results in a remarkable increase in the maximum tolerable input acceleration, from 3 g to 18 g when employing a safety factor of 2.

In summary, Harvester Types I and II demonstrate their effectiveness in accomplishing their respective objectives. Harvester Type I operates seamlessly under a 5 g threshold, effectively moderating the increase in maximum stress beyond this limit, thereby safeguarding against sudden overloads. On the other hand, Harvester Type II efficiently mitigates maximum stress, enabling the harvester to maintain operation at a 15 g acceleration, while also moderating the rate of stress growth to shield the device from unexpected overloads.

It is noteworthy that the maximum tolerable input acceleration of Harvester Type II significantly exceeds that of Harvester Type I. Meanwhile, the power output of Harvester Type II is lower than that of Harvester Type I under a 5 g excitation, as depicted in Fig. 10(b) and Fig. 11(b). This variance is attributed to the fact that the IPC design consumes a portion of the energy to safeguard the device. As a result, the selection of harvester type hinges on the excitation acceleration level, considering both power output and maximum tolerable input

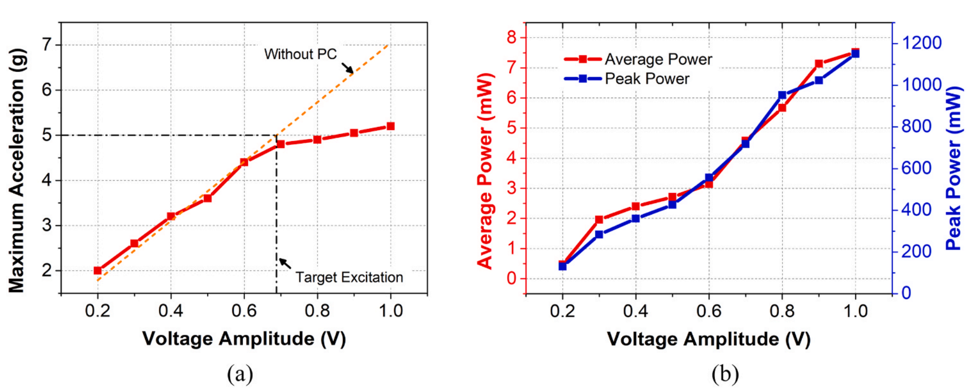

Fig. 10. The measured energy harvesting performance of Harvester Type I subjected to the rail vibration signal: (a) the maximum output acceleration and (b) the average and peak power across various input voltage amplitudes.

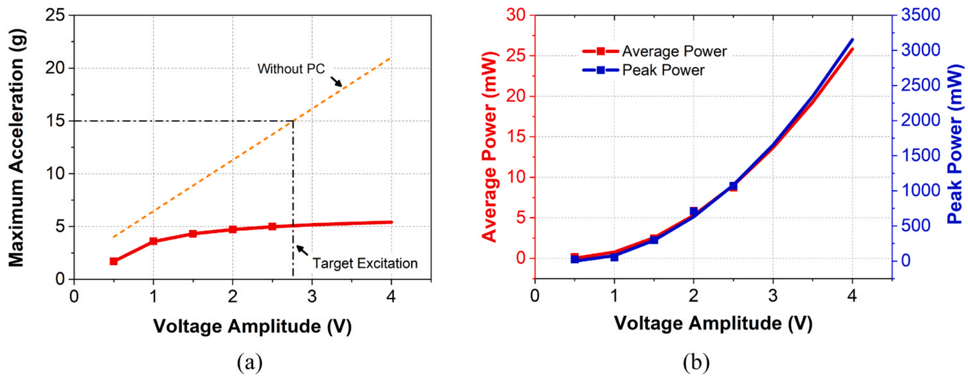

Fig. 11. The measured energy harvesting performance of Harvester Type II subjected to the rail vibration signal: (a) the maximum output acceleration and (b) the average and peak power across various input voltage amplitudes.

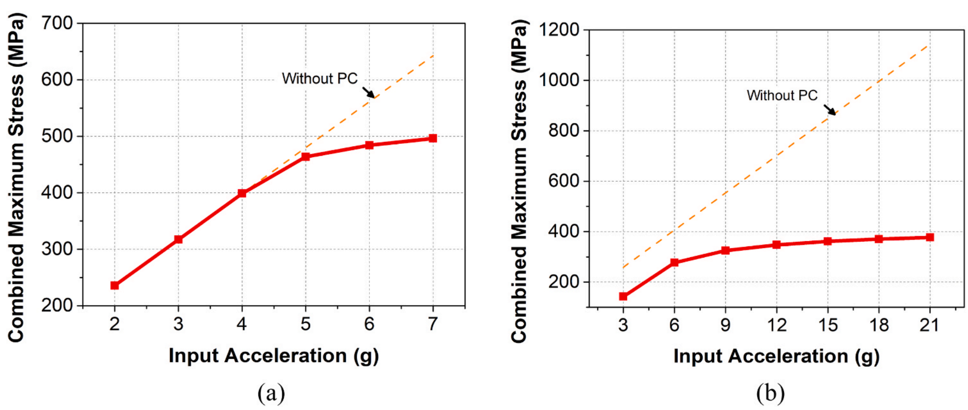

Fig. 12. The combined maximum stresses experienced by (a) Harvester Type I, and (b) Harvester Type II across various input accelerations.

acceleration. When the harvester with stopper design is robust enough to withstand the associated impact forces, under 5 g in this case, Harvester Type I is the preferred choice for a higher power output. In contrast, if the harvester with stopper design alone is inadequate to handle the ensuing impact forces, such as those occurring under a 15 g acceleration, Harvester Type II is in need for protection of the device.

5. Conclusions

The railway environment presents a demanding setting where the safety of energy harvesters installed on railways holds great significance for both the longevity of the harvester and the secure operation of the railway system. The highly variable nature of railway vibrations demands the design and integration of protection components into energy harvesters. To address this need, this study introduces two innovative protection strategies and seamlessly integrates them into piezo stack energy harvesters. Two types of harvesters with the designed protection components, denoted as Harvester Types I and II, are then proposed and engineered to specific design objectives. Their protection mechanism and energy harvesting performance are validated through experimental tests. In conclusion:

1. Two innovative protection methods have been proposed to safeguard energy harvesters from mechanical overload: the stopper design and the IPC design. The stopper is designed to limit the movement of the spring when vibrations exceed a certain acceleration threshold, thereby preventing too much force from being transferred between the resonant system and the impact system. In contrast, the IPC design, which consists of a spring, a stopper, and a spring plunger, allows only a portion of the energy to reach the transducer in its liming state. This dissipation of excess energy ensures that the harvester can operate at the desired large excitation level while protecting it from overload conditions.
2. Harvester Types I and II have been developed to incorporate the proposed protection strategies, aiming to address the specific requirements of 5 g and 15 g acceleration targets that correspond to the amplitudes of vibrations caused by railways. FEM modelling is developed to evaluate and determine the design parameters, such as added mass, distance d_1, and distance d_2 in accordance with the acceleration target and the stress limits.
3. Experiments are carried out to assess the protection mechanism of the harvesters. Results show that Harvester Type I operates seamlessly within a 5 g threshold, effectively moderating the increase in maximum output acceleration and stress beyond this limit, thereby safeguarding against sudden overloads. In contrast, Harvester Type II efficiently mitigates the maximum output acceleration and stress, enabling the harvester to maintain operation at a 15 g acceleration, while also moderating the rate of acceleration and stress growth to shield the device from unexpected overloads.
4. Their energy harvesting capabilities have also been evaluated under measured railway vibration signals. At their respective target excitation level, Harvester Type I delivers an average power output of approximately 4.6 mW, with the peak power peaking at around 718 mW, while Harvester Type II achieves an average power output of approximately 11 mW, with the peak power surging to around 1365 mW.
5. A comparative analysis of stress levels reveals that the inclusion of protection components enhances the harvesters' maximum tolerable input acceleration. Compared to harvesters without these components, Harvester Type I's maximum tolerable input acceleration

improves from 5 g to 7 g, while Harvester Type II experiences a remarkable boost from 3 g to 18 g.

In essence, this work presents a comprehensive solution for enhancing the safety of energy harvesters under challenging railway-induced vibrations. By effectively safeguarding the energy harvesters from mechanical overload and moderating stress levels in the demanding railway environment, it ensures their durable and reliable performance. As a result, this work offers substantial benefits for the practical, real-world implementation of these energy harvesters operating within the dynamic environment of railway vibrations.

Author statement

All authors have seen and approved the final version of the manuscript being submitted. The article is the authors' original work, hasn't received prior publication and isn't under consideration for publication elsewhere.

CRediT authorship contribution statement

Dong Wang: Writing – review & editing. **Guansong Shan:** Writing – review & editing, Writing – original draft, Visualization, Validation, Methodology, Investigation, Formal analysis, Data curation, Conceptualization. **Meiling Zhu:** Writing – review & editing, Supervision, Resources, Funding acquisition.

Declaration of Competing Interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

Data Availability

Data will be made available on request.

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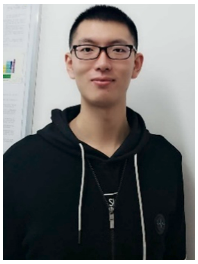

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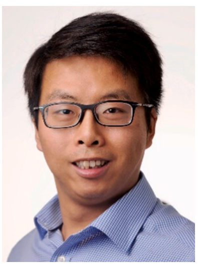

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