INTRODUCTION

Energy crisis and environmental pollution, such as oil shortage and air pollution have created great challenges in the automotive industry. Research interest in recovering vibration loss, during vehicle travel, is growing rapidly. The current cases of energy harvesting, converting otherwise lost energy into a usable form, do not refer to the whole vehicle, but rather to parts of it.1 Vehicle energy loss harvesting technology has a positive significance to the development of the current automotive industry, by improving vehicle energy efficiency, fuel economy, etc.2,3 Energy harvesting technology is widely used in the collection of solar, wind, hydro, thermal, and mechanical energy.4,5 In the automotive industry, emitted heat, braking energy, and vibration energy are the main targets of energy harvesting technology.6-8

As an important part of the vehicle chassis, the suspension system plays a key role in supporting the body and absorbing vibrations, caused by rough road surfaces, while it concentrates the main vibration energy of the vehicle.9 The suspension system energy harvester is a complement module to the onboard alternator, while the harvested vibration energy can charge the vehicle battery and provide power for the respective load. Currently, researchers have conducted...
numerous studies on energy harvesting, based on vehicle suspension systems. In the case of a typical passenger car, driving at a speed of 97 km/h, on a good road surface, the harvested power can reach 100–400 W, which is equivalent to a fuel efficiency increase of 3%.\textsuperscript{10,11} The power harvested by the vibration energy device is closely related to the vibration intensity level of the suspension system. Vehicles with large sprung mass, high driving velocity under poor driving conditions show a high level of vibration intensity.\textsuperscript{12} As a result, the energy harvesting of the suspension system has broader prospects, in cases such as, heavy trucks and off-road vehicles.\textsuperscript{13} For example, in the case of an off-road vehicle traveling on a road of Class D, at a speed of 80 km/h, the generated electric power can reach 2048 W.\textsuperscript{6}

Among the current energy harvesters, electromagnetic energy harvesters are widely used in vehicle suspension systems, for vibration energy harvesting, because of higher energy conversion efficiency, more compact structure, faster response velocity, and higher controllability.\textsuperscript{14–16} The electromagnetic energy harvesters can be divided into two types: linear electromagnetic energy harvesters and rotary electromagnetic energy harvesters.\textsuperscript{17} The rotary electromagnetic energy harvester, of higher energy conversion efficiency and more compact structure than the linear electromagnetic energy harvester,\textsuperscript{18} realizes the conversion of linear motion to rotary motion, by mechanical or hydraulic transmission\textsuperscript{19,20}

In the last few decades, piezoelectric materials have played a vital role, as mechanism of energy harvesting, having the tendency to absorb energy from the environment and transform it into electrical energy, used to drive electronic devices directly or indirectly.\textsuperscript{21} The piezoelectric materials exhibit better electromechanical coupling effects and higher energy conversion efficiency.\textsuperscript{22} The most well-known piezoelectric materials are the piezoelectric ceramics and the piezoelectric polymers.\textsuperscript{23} Prior art literature shows that piezoelectric materials harvest three times as much energy density as electrostatic and electromagnetics do.\textsuperscript{24} Specifically, piezoelectric materials are used to harvest the vibration energy that a person transmits to the ground while walking,\textsuperscript{25} the energy generated by the differential force, between the wearer and the backpack,\textsuperscript{26} the vibration energy generated by high-rise buildings,\textsuperscript{27} the longitudinal or transverse wave motion of seawater,\textsuperscript{28} and the wind.\textsuperscript{29}

Piezoelectric materials have wide range of applications in the field of aerospace and automobile industry.\textsuperscript{30} In Lafarge et al.\textsuperscript{31} the piezoelectric energy harvester is used to collect the vibration energy, inside the vehicle suspension system and tires, where the harvested energy can be used to power the embedded wireless sensor. A new dual-mass piezoelectric energy harvester is designed, to collect effectively and practically vibration energy from vehicle tires, reaching a maximum power of 42.08 W.\textsuperscript{32} The maximum power harvested by the vehicle suspension system can reach 738 W, depending on the road roughness.\textsuperscript{33} The factors affecting the power harvested by the dual-mass piezoelectric energy harvesters, are theoretically discussed, according to an established piezoelectric power generation model. However, the accuracy and rationality of the aforementioned research models are not verified.

The cited prior art shows that vibration energy harvesters have been widely used to harvest vibration energy in various environments. In vehicles, they are primarily used to harvest vibration energy from suspension systems, where hundreds of watts of electrical energy can be collected, using electromagnetic or piezoelectric energy harvesters. Both types of harvester have their own associated challenges and disadvantages. Electromagnetic harvesters have the disadvantage of low energy density. Piezoelectric harvesters suffer from various problems, including the harvester not having been fitted to the actual vehicle’s suspension system for testing, so the accuracy of the model could not be verified, and the modeling method could not be evaluated.

In order to address the limitations of the current research on energy harvesting from suspension system, a novel piezoelectric energy harvester is proposed. Taking light electric logistics vehicle as the research object, the piezoelectric energy harvester is installed in the left-front suspension. A dual-mass suspension system vibration model is established and studied in simulation, while also compared to test results. Some key considerations for the developed harvester are hence discussed, for achieving higher power generation. The results show that the RMS of the harvested power is proportional to the driving speed, to the ratio of the moment arms of the lever to the cargo state, and inversely proportional to the piezoelectric material cross-sectional area. This research provides a new design method of a high efficiency energy harvesting system, from vehicle vibrations.

2 | DESIGN AND METHODS

2.1 | Structural design and modeling

A novel piezoelectric energy harvesting method is designed, while a sketch of the proposed dual-mass piezoelectric energy harvester of a quarter vehicle is depicted in Figure 1(A–C). The piezoelectric energy harvester model, as illustrated in Figure 1A, consists of a piezoelectric ceramic, mounted between the left end of the lever and the housing; a rod with a diameter of 4 mm, on the upper surface of the lever, inserted into the inner hole of the piezoelectric ceramic, to guide and fix the piezoelectric ceramic; a connecting support of the leaf spring, hinged to the right end of the lever; a moving bracket, used to support the under surface of the lever and a guide rail, installed between the moving bracket and the housing, realizing the
adjustment of the lever arm \( L_1 \) and \( L_2 \) by regulating the adjustment screw that moves the bracket to the left and right, along the guide rail.

The piezoelectric material selected is PZT-4 (lead zirconate titanate), with stiffness expressed as

\[
k_1 = \frac{E_1 S}{hn^2},
\]

where \( n = L_2 / L_1 \) is a ratio of the moment arms of the lever, \( S = \pi (R^2 - r^2) \), \( h \) and \( E_1 \) denote the cross-sectional area, thickness and Yong's modulus of the piezoelectric ceramic, respectively.

According to the principle that the dissipation energy of damper is equal to the electric energy, generated by the piezoelectric energy harvester, the damping coefficient \( c_1 \) can be derived as follows:

\[
c_1 = n^2 \cdot d_{33} \cdot k_2 \cdot \frac{k_3}{\pi^2} \cdot C \cdot f
\]

where \( d_{33} \) is the piezoelectric constant in the polling direction; \( C \) is the electrical capacity of the piezoelectric ceramic; \( k \) is the equivalent stiffness of the suspension system; and \( f \) is the first natural vibration frequency of the vehicle suspension system.

Since the lever is supposed to be fixed throughout the piezoelectric ceramic and moving bracket, while the force \( F \) is applied in the vertical direction at the right end of the lever, the entire lever can be equivalent to a cantilever beam of length \( L_2 \). The lever stiffness, quality, and damping coefficient can be expressed as

\[
k_2 = \frac{E_2 \cdot ab^3}{4L_2^3}, \quad m_2 = \rho S_l L_2^2, \quad c_2 = 2\sqrt{k_2 m_2 \zeta},
\]

respectively. Where \( E_2, \rho, S_l, \) and \( \zeta \) are Yong's modulus, material density, cross-sectional area, and damping ratio of lever beam, respectively.

The quarter vehicle model equipped with a piezoelectric energy harvester, mounted between the front suspension leaf spring and the front axle, is shown in Figure 1B. The equivalent stiffness and damping coefficient of a piezoelectric energy harvesting suspension system can be expressed as

\[
k = k_1 k_2 k_3 / (k_1 k_2 + k_2 k_3 + k_3 k_1) \quad \text{and} \quad c = c_1 + c_2 + c_3,
\]

respectively. Where, \( k_3 \) and \( c_3 \) are stiffness and damping coefficient of the original suspension system, respectively. The vibration model of the dual-mass piezoelectric energy harvesting suspension system is established and shown in Figure 1C. It consists of sprung mass \( m \), unsprung mass \( m' \), suspension system stiffness \( k \), suspension system damping coefficient \( c \), tire stiffness \( k' \), and tire damping coefficient \( c' \). The parameters of the piezoelectric energy harvesting suspension system are shown in Table 1.

The electric capacity of the piezoelectric material can be computed by Equation (2) as follows:

\[
C = C_v \times \pi (R^2 - r^2) \times 0.01 / (\pi (0.013^2 - 0.006^2) \times h)
\]

Vibrations of a vehicle suspension system are subjected to roughness variations of the uneven road surface \( z(t) \), which can be obtained by the equation:
where $n_{00} = 0.011 \text{ m}^{-1}$ is a minimal boundary frequency; $n_0 = 0.1 \text{ m}^{-1}$ is the reference spatial frequency; $G_q(n_0)$ is the roughness coefficient of the road surface in $m^2$; $W(t)$ is the Gaussian white noise with a zero mean; and $u$ is the vehicle speed in $m/s$.

According to the Lagrange equation, the differential equation of 2 degree of freedom (DOF), of the dual-mass piezoelectric energy harvesting system, is derived, as follows:

$$\ddot{z}(t) = -2\pi n_{00} u z(t) + 2\pi n_0 \sqrt{G_q(n_0)} u W(t) \tag{3}$$

Substituting damping matrix into Equation (7) leads to:

$$\dot{m}\ddot{s} + \dot{c}s + ks = F \tag{7}$$

The damping force, as generated by the system vibration, is proportional to the velocity value and opposite to the direction of the velocity vector. So Equation (4) can be rearranged as:

$$\ddot{s} + \frac{c}{m}\dot{s} + \frac{k}{m}s = \frac{F}{m} \tag{7}$$

or simply written as:

$$\ddot{q}_i(t) + (\alpha + \omega_i^2 \beta)\dot{q}_i(t) + \omega_i^2 q_i(t) = Q_i(t), i = 1, 2 \tag{12}$$

where $\omega_i$ is the $i$-th order natural frequency of the undamped system and $Q(t)$ is equal to $S^TF$.

Assuming that $\alpha + \omega_i^2 \beta = 2\xi_i \omega_i$, Equation (12) can be rewritten as:

$$\ddot{q}_i(t) + 2\xi_i \omega_i \dot{q}_i(t) + \omega_i^2 q_i(t) = Q_i(t), i = 1, 2 \tag{13}$$

where $\xi_i$ is the modal damping ratio of the $i$-th order natural mode. It is evident that the above two equations are uncoupled, while the response of the $i$-th order natural mode can be obtained similarly to the response of a single-degree-of-freedom viscous damping system. When $\xi_i < 1$, the solution of the above equation can be expressed as:

$$q_i(t) = e^{-\xi_i \omega_i t} \left\{ \cos \omega_i t + \frac{\xi_i}{\sqrt{1 - \xi_i^2}} \sin \omega_i t \right\} q_i(0) + \left\{ \frac{1}{\omega_i} e^{-\xi_i \omega_i t} \sin \omega_i t \right\} \dot{q}_i(0) + \frac{1}{\omega_i} \int_0^t Q_i(\tau) e^{-\xi_i \omega_i(t - \tau)} \sin \omega_i(t - \tau) d\tau, i = 1, 2 \tag{14}$$

where $0 < \tau < t$, $\omega_i = \omega_i \sqrt{1 - \xi_i^2}$. The parameters of piezoelectric energy harvesting suspension system

| $m$ (kg) | $m'$ (kg) | $k$ (N/m) | $c$ (N s/m) | $k'$ (N/m) | $c'$ (N s/m) | $h$ (mm) |
|---------|-----------|-----------|-------------|------------|-------------|---------|
| $690$ (unladen) | $970$ (laden) | $123$ | $119$ | $1100$ | $69000$ | $1600$ | $100$ |
| $d_{13}$ (C/N) | $r$ (mm) | $R$ (mm) | $a$ (mm) | $b$ (mm) | $n(L_1/L_2)$ | $L_1 + L_2$ (mm) | $\zeta$ |
| $7.1e-10$ | $2$ | $6-15$ | $30$ | $50$ | $5-10$ | $120$ | $0.0017$ |

$C_v$ (pF) 485 for the piezoelectric ceramic cylinder with geometry of 6 mm, 13 mm, 10 mm.
Consequently, we can obtain the displacements $s_2, s_1$ and the velocities $\dot{s}_2, \dot{s}_1$ of the sprung mass and unsprung mass, with respect to their respective equilibrium positions at any time by substituting Equation (14) into $s(t) = \dot{S}q(t)$. The relative displacements and velocities of the sprung mass and unsprung mass can be expressed as $s_{21} = s_2 - s_1$ and $\dot{s}_{21} = \dot{s}_2 - \dot{s}_1$, respectively. Following, the generated charge $Q(t)$, voltage $V(t)$, and current $I(t)$, from the piezoelectric ceramic, at time $t$, can be solved according to the force acting on it. The equations are shown as follows:

$$Q(t) = nd_{33} \cdot k \cdot s_{21}(t)$$  \hspace{1cm} (15)$$

$$V(t) = Q(t)/C = nd_{33} \cdot k \cdot s_{21}(t)/C $$  \hspace{1cm} (16)$$

$$I(t) = \dot{Q}(t) = nd_{33} \cdot k \cdot \dot{s}_{21}(t)$$  \hspace{1cm} (17)$$

The force applied to the piezoelectric ceramic, in its poling direction, at time $t$, is a random excitation at the end of the lever from the driving vehicle. Hence, the RMS value of the power generated by piezoelectric ceramic, from 0 to $t$, can be obtained as:

$$P_{\text{rms}} = \sqrt{\frac{1}{t} \int_0^t [P_e(\tau)]^2 d\tau}$$  \hspace{1cm} (19)$$

where $P_e(\tau) = V(\tau) \cdot I(\tau) = F(\tau) \cdot \dot{S}^2 \cdot F'(\tau)/C$ is the harvested power of the piezoelectric ceramic, at time $\tau (0 < \tau < t)$. In order to estimate the RMS of the generated power, the period $t$ can be divided into $j$ time steps, with a sufficiently short time interval $\Delta t$. As a result, the expression in Equation (18) can be rewritten in a discrete form as:

$$P_{\text{rms}} = \sqrt{\frac{1}{j} \sum_{i=1}^{j} P_e^2(t_i)}$$  \hspace{1cm} (19)$$

### 2.2 Model verification

In order to verify the accuracy of the model established in the previous section, a comparison between the actual vehicle test data and the simulation analysis data is performed. The field test diagram of Class B road surface is shown in Figure 2(A,B), where Figure 2A illustrates a random road test and Figure 2B shows a pulse road test. The acceleration value of the sprung mass of the vehicle, traveling on the Class B road surface, at speeds ranging within 10-80 km/h, can be measured.

The ensemble empirical mode decomposition (EEMD) method is used to filter the acquired acceleration signal, to improve the signal-to-noise ratio. The partial decomposition results of EEMD, shown in Figure 3, can be obtained by noise reduction processing, on the acceleration signal of the sprung
mass, when the unladen vehicle moves at a driving speed of 30 km/h on a Class B road.

According to the correlation coefficient between the standard deviation of each order intrinsic mode function (IMF) and the original signal, the relevant IMF component is selected for signal reconstruction. As the mean square error (MSE) of the reconstructed signal decreases, the noise reduction effect improves. The correlation coefficient reflects the degree of similarity between the reconstructed signal and the original signal, as the higher its value is, the better the fitting effect is. The comparison between the reconstructed and the original acceleration signal of the sprung mass, when

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**FIGURE 3** The partial decomposition results of EEMD at a driving speed of 30 km/h, Class B road, unladen

**FIGURE 4** Reconstructed signal and original acceleration signal at a driving speed of 30 km/h, Class B road, unladen
the unladen vehicle moves at a driving speed of 30 km/h on the Class B road, as shown in Figure 4.

The transverse motion function \(z(t)\) of a Class B road, at different speeds, can be calculated by Equation (3), where \(G_q(\nu_0) = 64 \times 10^{-6} \text{ m}^3\). When the unladen vehicle moves on a road of Class B, at different speeds, the acceleration value of the sprung mass, at any time, can be calculated by the vibration model of dual-mass piezoelectric energy harvesting suspension system. The comparison between the test results and the simulation results of the sprung mass acceleration RMS of the piezoelectric energy harvesting suspension system, at different speeds on a Class B road is shown in Table 2.

Table 2 shows that the acceleration RMS of the two groups have the same trend, which increases as the vehicle speed rises. Specifically, when the speed was at 60 km/h, the maximum deviation between the simulation and test value was 13.36%.

The power spectral density curve of the simulation and test values, of the sprung mass acceleration, is shown in Figure 5, in the case where the unladen vehicle is driven at a speed of 30 km/h. According to the goodness of fit theory, the accuracy of the model is verified by the coefficient of determination \(R^2\), while the feasibility and effectiveness of the model establishment method are further illustrated. The coefficient of determination can be expressed as:

\[
R^2 = \frac{\sum_{i=1}^{n} (\hat{\alpha}_i - \bar{\alpha}_i)^2}{\sum_{i=1}^{n} (\alpha_i - \bar{\alpha}_i)^2}
\]

where \(n\) is the number of test data; \(\hat{\alpha}_i\) is the simulation value of the \(i\)-th model response; and \(\alpha_i\) is the test value or real value of the \(i\)-th model and \(\bar{\alpha}_i\) is the average value.

The simulation results were experimentally verified, to ensure modal accuracy, as seen in Figure 5, where 40 sample points were selected for the verification. The coefficient of determination \(R^2\), calculated in the Equation (20), is 0.9361. The model has higher accuracy when the coefficient of determination is close to 1. Considering the simplification and establishment of the model, the calculation accuracy of the simulation, the smoothness of the actual road surface, the actual test position and driving skills, as well as other factors, the deviation between the simulation result and test result does satisfy the accuracy requirements and the actual project needs. In summary, the model exhibits high precision and the method of the model establishment is feasible and effective.

3 | RESULTS AND DISCUSSIONS

In the following tests, the urban electric logistics vehicle is driven on a Class B road, at a test site, as shown in Figure 2. The relative velocity \(\dot{s}_{21} = \dot{s}_2 - \dot{s}_1\) and relative displacement \(s_{21} = s_2 - s_1\) of the sprung mass and unsprung mass, of the piezoelectric energy harvesting suspension system, can be detected, as the vehicle is moving under various working conditions. According to the test values and Equations (14)-(19), the RMS of the harvested power, by the piezoelectric energy harvesting suspension system, is also obtained. Various parameters, such as driving speed, road type, and cargo state are included in the tests, for basic harvesting energy.

During the random road test, the vehicle was driving, at different speeds, on a Class B road. At this time, \(n(L_2/L_1) = 5\), the effect of the driving speed on the relative velocity, relative
displacement and the RMS, of the harvested power, of the sprung mass and unsprung mass of the piezoelectric energy harvesting suspension system, under unladen and laden conditions, is studied and illustrated in Figure 6(A-C). At a vehicle speed of 60 km/h, the curves of $\dot{s}_{21}$ and $s_{21}$, as a function of time, are shown in Figure 6A,B. As the vehicle was driving under laden and unladen conditions, the maximum amplitude of the relative velocity was 0.4548 and 0.2807 m/s, respectively. When the speed increased from 20 to 60 km/h, as shown in Figure 6C, the RMS of the harvested power, of the piezoelectric energy harvesting suspension system, increased from 0.32 to 5.07 W at unladen conditions, while it increased from 0.35 to 7.23 W, at laden conditions. The RMS of the harvested power, under the two loading states, was approximately equal to the one at a speed of 20 km/h. As the speed increased, the RMS of the harvested power, under the laden state, was greater than the one, under the unladen state, at the same speed. It can be seen that both curves have

\textbf{FIGURE 6} Relative velocity, relative displacement, and RMS of the harvested power based on random road. A, Relative velocity of sprung mass and unsprung mass, B, Relative displacement of sprung mass and unsprung mass, and C, RMS of the harvested power versus speed of vehicle.
consistent trend, where the RMS value and the growth rate increase along the speed.

The RMS of the harvested power was 1.61 W, under unladen state, and 4.47 W, under laden state, while the vehicle smoothly accelerated from 0 to 80 km/h. The relative velocity and displacement, as a function of time, at unladen and laden state, are illustrated in Figure 7(A,B). During the vehicle acceleration process, the maximum amplitudes of the relative velocity and displacement under the unladen state, are less than those under the laden state, where the maximum amplitudes of the maximum are 0.5198 m/s and 0.01251 m, respectively. It can be inferred that, as the speed of the vehicle increases, the relative velocity and displacement increase. Comparison of the curves in Figure 7(A,B) and Figure 6(A,B) leads to the conclusion that the maximum amplitudes of relative velocity and displacement, during acceleration, are lower than those in the stable driving condition, under the same cargo state and driving speed. The main reason for this is that the vertical load, on the front axle, during acceleration, is lighter than that, at constant speed, due to the acceleration.

Urban electric logistics vehicles often encounter deceleration strip during driving, as shown in Figure 2B. During the pulse road test, the vehicle passed through two sets of deceleration strip, with a distance of 20 m at 10-40 km/h, respectively, while the spam of analysis time was 10 seconds.

The effect of vehicle speed and cargo state, on the RMS of the harvested power of the piezoelectric energy harvesting suspension system, during driving on a pulse road, is given in Figure 8(A-E). The changes in the relative velocity and displacement of the sprung mass and unsprung mass, as the vehicle passed the deceleration strip, at a speed of 30 km/h, under unladen and laden states, are shown in Figure 8(A-D), respectively. The maximum amplitudes of relative velocity and displacement, under unladen and laden states, are 2.493 m/s, 0.05858 m, 2.473 m/s, and 0.0536 m, respectively. The variation trend of the RMS of harvested power by the piezoelectric energy harvesting suspension system, under unladen and laden states, as illustrated in Figure 8E, is basically the same. It is clear-cut that the RMS of the harvested power that corresponds to pulse road is much higher than that of random road. This is due to the higher instantaneous relative velocity and displacement between the sprung and unsprung mass of pulse road. The RMS of the harvested power increases, as the speed increases from 10 to 30 km/h, while it decreases, as the speed increases from 30 to 40 km/h; and the maximum values are 102.24 and 88.09 W, when the vehicle passes through the deceleration
strip, at a speed of 30 km/h, under unladen and laden states, respectively. The main reason is that, as the vehicle passed the deceleration strip of pulse road, at a speed of 30 km/h, the excitation frequency was close to the natural frequency of the vehicle suspension system, while the vibration intensity of the vehicle suspension system was the largest at that time.

**FIGURE 8** Relative velocity, relative displacement, and RMS of the harvested power based on pulse road. A, Relative velocity of sprung mass and unsprung mass at unladen, B, Relative displacement of sprung mass and unsprung mass at unladen, C, Relative velocity of sprung mass and unsprung mass at laden, D, Relative displacement of sprung mass and unsprung mass at laden, and E, RMS of the harvested power versus speed of vehicle.
In addition, in Figure 8E, at different vehicle speeds, the RMS of the harvested power, under unladen state, is greater than the RMS of the harvested power, under laden state. This occurs due to the pulse road excitation frequency being closer to the natural frequency of the suspension system, as the vehicle is under unladen state. When the vehicle passes through the deceleration strip, the vibration frequency and amplitude of the suspension system, under unladen state, are greater than those under laden state, which directly leads to the enhanced power generation, under unladen state.

In the following simulations, some important factors in designs, such as the ratio of the moment arms of the lever, and piezoelectric material cross-sectional area, that influence the RMS of the generated power are investigated for the proposed harvester. The parameters of piezoelectric energy harvesting suspension system are shown in Table 1. The effect of the ratio of the moment arms of the lever on the RMS of the harvested power of the piezoelectric energy harvesting suspension system, is revealed by Figure 9(A-C). As the laden vehicle, with parameters of \( n = 10 \) and \( v = 60 \text{ km/h} \), is driven on a Class B road, the maximum amplitude, shown in Figure 9A, of the relative velocity, is 0.3111 m/s. The relative displacement, with a maximum amplitude of 0.01615 m, of the sprung mass and unsprung mass, as a function of time is provided in Figure 9B. The analysis in Section 2.1 shows that the stiffness \( k_1 \) decreases with an increase in \( n \), leading to a decrease in stiffness \( k \); the damping coefficient \( c_1 \) increases with an increase in \( n \), leading to an increase in damping coefficient \( c \). An approximate linear fitting of the RMS of the harvested power, under an increased ratio of the lever arm, is shown in Figure 9C. The RMS of the harvested power is increased, from 7.19 to 18.83 W, along an increase in the ratio of the moment arms of the lever, from 5 to 10.

The effect of the cross-sectional area of the piezoelectric ceramic, on the RMS of the harvested power, by piezoelectric energy harvesting suspension system, is demonstrated in Figure 10(A-C). Considering the structure of the piezoelectric energy harvester, the radius of the inner circle of the piezoelectric ceramic is kept constant at 2 mm, while the radius of the outer circle is in the range of 6-15 mm.
Considering $R = 15$ mm, the relative velocity and displacement of the sprung mass and unsprung mass, of the piezoelectric energy harvesting suspension system, as a function of time, are shown in Figure 10(A,B). The maximum amplitudes of the relative velocity and displacement are 0.349 m/s and 0.01724 m, respectively. Figure 10C shows that, when $R$ rises from 6 to 15 mm, the RMS of the harvested power is reduced from 18.83 to 4.49 W. Thus, it is obvious that a slight decrease of the radius of the outer circle will lead to a remarkable augment of the RMS. The observation is interpreted as an increase in the radius of the outer circle of the piezoelectric ceramic would lead to an increase in the electric capacity of $C$, and in turn a decrease in the electric voltage, a decreased generated power.

**FIGURE 9** Relative velocity, relative displacement, and RMS of the harvested power based on random road. A, Relative velocity of sprung mass and unsprung mass, B, Relative displacement of sprung mass and unsprung mass, and C, RMS of the harvested power versus ratio of $n(L_2/L_1)$
4 | CONCLUSIONS

With the purpose of harvesting the traditionally, otherwise dissipated, vibration energy, in vehicle suspension system, a novel piezoelectric energy harvester is designed, to convert the vibration energy, in suspension system, into usable electrical energy. A dual-mass suspension system vibration model, including piezoelectric energy harvester, is developed, while a light electric logistics vehicle, equipped with a piezoelectric energy harvester, for harvesting energy is used for testing. The accuracy and credibility of the model were verified, based on field test results of the vehicle, as well as simulation results of the model. The generated power, voltage, and current of the harvester were all calculated, based on this model. The computation results show that the RMS increases, with an increase in the vehicle speed and the ratio of the moment arms of the lever, while also a decrease in the piezoelectric material cross-sectional area. For an energy harvester structure, with material parameters of $v = 60 \text{ km/h}$,
n = 10, and R = 6 mm, the maximum RMS of the harvested power is 18.83 W. On a pulse road drive, the greatest values of 102.24 and 88.09 W are obtained at 30 km/h, under unladen and laden states, respectively. Each vehicle can be equipped with four or more of these piezoelectric energy harvesters, which can generate nearly one hundred watts of power, when running on urban roads, where deceleration strips are common. This harvested electricity can be used to power some auxiliary systems in electric vehicles, such as lamp, combined meter, and air conditioning compressor and so on. The research provides a new design method for use of piezoelectric technology, to harvest the vibration energy, from suspension system, thereby improving the energy efficiency of the vehicle.

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CONFLICT OF INTEREST

The authors declared no potential conflicts of interest with respect to the research, authorship, and/or publication of this article.

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