

Research Article

Energy Harvesting Using One-Way Bearing for Vehicle Suspension Applications

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Abstract:

Road degradation amplifies suspension energy dissipation due to factors like vehicle mass, speed, and surface roughness. Harvesting energy from suspension motion is challenging, requiring the conversion of irregular mechanical energy into electrical power. The energy harvesting device must operate effectively under ISO 8608 Class C and D conditions. An indirect-drive system converts linear motion into rotational motion to power a generator, with the 'two-leg mechanism' and one-way bearing rectifying motion direction. This research combines simulation and experimentation to assess power generation under sinusoidal wave conditions (10–20 mm amplitude, 1–3 Hz frequency). Maximum power occurs at 10–20 Ω resistance, increasing with frequency and amplitude. With 88.15% accuracy, the simulation model provides valuable insights for optimizing real-world performance.

Keywords: Energy Harvesting, Vehicle suspension, Road roughness, Indirect-drive mechanism, One-way bearing

1. Introduction

Electricity powers zero-emission vehicles, with a focus on reducing fossil fuel usage and improving energy efficiency. In internal combustion engine vehicles (ICEVs), approximately 75% of energy is lost in the engine, and another 23% is dissipated through the wheels. Electric vehicles (EVs) eliminate engine energy loss, significantly increasing overall efficiency. However, kinetic energy loss from suspension movement and rolling resistance accounts for 3-12% of fuel usage in ICEVs, and this varies significantly based on the road surface, vehicle speed, and mass. For passenger cars at 60 mph, this represents 100-400W of energy loss [2], with higher losses in larger vehicles like SUVs or trucks [3].

Vehicle suspension systems enhance driving performance by minimizing vibrations while ensuring stability and comfort. They isolate wheel motion from the chassis to maintain road contact and prevent excessive vibrations. Road surface roughness (classified from Class A to H) significantly influences suspension design and energy dissipation, with rougher surfaces (H) increasing energy loss [4]

Vibrations are analyzed based on unsprung (wheels) and sprung (body) masses. The 4-8 Hz frequency range is particularly critical for passenger comfort [5, 6]. For optimal ride quality, suspension and chassis frequencies typically fall below 4 Hz and above 8 Hz, respectively.

Suspension energy harvesting can be achieved through direct methods such as electromagnetic systems [7-10] or indirect methods such as mechanical transmission systems [11-17]. Research by Zhang et al.,[18] suggests that indirect conversion offers higher efficiency than direct methods.

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The primary challenge in suspension energy harvesting is effectively converting the bidirectional, irregular, and low-frequency linear motion of the suspension into unidirectional, high-speed rotational motion suitable for electricity generation. Direct-drive systems often suffer from low efficiency due to the high starting torque required to overcome bidirectional damping. Consequently, indirect-drive mechanisms, utilizing motion rectifiers, are favored for maximizing conversion efficiency. This study specifically investigates the use of a one-way bearing to implement motion rectification. The one-way bearing is crucial as it mechanically ensures that the generator shaft rotates only in one direction (unidirectionally), regardless of whether the suspension is compressing or rebounding. This rectification increases the total rotational time and speed of the generator, thereby maximizing power output efficiency.

This research focuses on an indirect-drive system utilizing a two-leg conversion mechanism integrated with the one-way bearing. The proposed device, mounted to the vehicle's suspension, is optimized and tested under simulated road conditions corresponding to moderately rough to rough roads (ISO 8608 Class C and D), highlighting its potential for application. The primary contribution of this research is the detailed development and rigorous experimental validation of the augmented dynamic model for a suspension energy harvesting system that integrates the two-leg motion conversion mechanism and the one-way bearing rectifier. Specifically, we utilize the Lagrangian formulation with constraints to accurately model the complex switching dynamics of the one-way bearing, including the impulse effects during stage transitions. The model's predictive capability is then rigorously validated against experimental data under controlled sinusoidal excitation conditions, achieving an accuracy of 88.15%. This validated dynamic model provides a robust predictive tool for optimizing the design and assessing real-world performance based on simulated testing in the laboratory.

2. Vehicle and road modeling

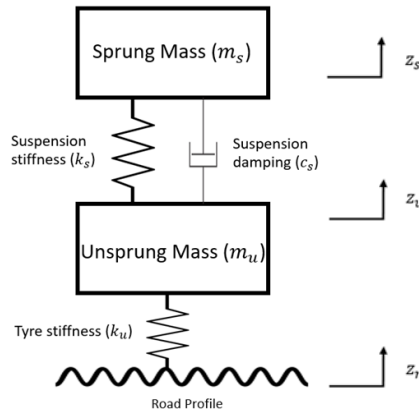


Fig. 1. Schematic diagram of quarter-car model.

As the vehicle moves, the suspension system transfers road roughness motion to the Reciprocating-to-Rotary Motion Mechanism for energy harvesting. While suspension systems can be modeled as full-car, half-car, or quarter-car, this research uses a 2-DOF quarter-car model. The road surface serves as the primary excitation source, generating vertical vibrations in the unsprung mass as the vehicle travels. These vibrations are modeled in a 2-DOF system, as shown in Figure 1, with dynamic equations of motion presented in Equations (1) and (2).

$$m_s \ddot{z}_s + k_s(z_s - z_u) + c_s(\dot{z}_s - \dot{z}_u) = 0 \quad (1)$$

$$m_u \ddot{z}_u + k_u(z_u - z_r) - k_s(z_s - z_u) - c_s(\dot{z}_s - \dot{z}_u) = 0 \quad (2)$$

Here, m_s represents the sprung mass and m_u represents the unsprung mass. k_s and c_s denote suspension stiffness and damping, while k_u and c_u correspond to tire stiffness and damping. z_u represents unsprung mass vertical displacement, and z_s represents sprung mass vertical displacement. Figure 1 shows the road surface profile (z_r), which forms the foundation for this 2-DOF base-excitation system. ISO 8608 classifies road profiles into categories A-H, with Class A being smoothest and Class H roughest. Table 1 displays displacement power spectral density for each class.

Among various approximation methods, this study employs a sinusoidal approach to generate road surface displacement. As outlined by Goenaga et al. [19], this is expressed through a differential equation for Simulink simulation. The simulation starts with white random noise for initial road roughness, which is then canceled during simulation as shown in equation (3).

Table 1: Geometric refers to the value of displacement power spectral density in road classification (ISO 8608:2016).

Road class	Geometric mean $\Phi(\Omega_0)$ (Unit: $10^{-6} \text{ m}^3/\text{rad}$)
A	1
B	4
C	16
D	64
E	256
F	1,024
G	4,096
H	16,384

$$\dot{z}_r(t) = -\alpha v z_r(t) + \delta(t) \quad (3)$$

where $\alpha = 0.15 \text{ m}^{-1}$ for asphalt concrete roads.

v = velocity (m/s^2)

$\delta(t)$ = White random noise with a PSD noise power of $2\alpha v \sigma^2$

σ^2 = road roughness variance which define as $4\Phi(\Omega_0)$

$\Phi(\Omega_0)$ = value defined by ISO 8608

(Class C and D were selected from Table 1).

The precomputed results from Equation (3) define the amplitude range (A), while the selected frequencies (f) are 1, 2, and 3 Hz. The road surface profile, approximated using a sinusoidal function, is presented in Equation (4).

$$z_r = A \sin(2\pi f) \quad (4)$$

3. Reciprocating-to-Rotary Motion Mechanism

The Reciprocating-to-Rotary Motion Mechanism is the core component responsible for converting the vehicle suspension's low-frequency, bidirectional linear (reciprocating) motion into the high-speed, unidirectional rotation suitable for a generator. Its purpose is to efficiently bridge the kinetic energy from road excitation to the electrical power generation unit. In this study, we utilize a two-leg mechanism (as shown in Figure 2) to achieve this conversion. The following section details the geometric and dynamic modeling of this mechanism, which is essential for predicting the overall performance and power output of the harvesting system.

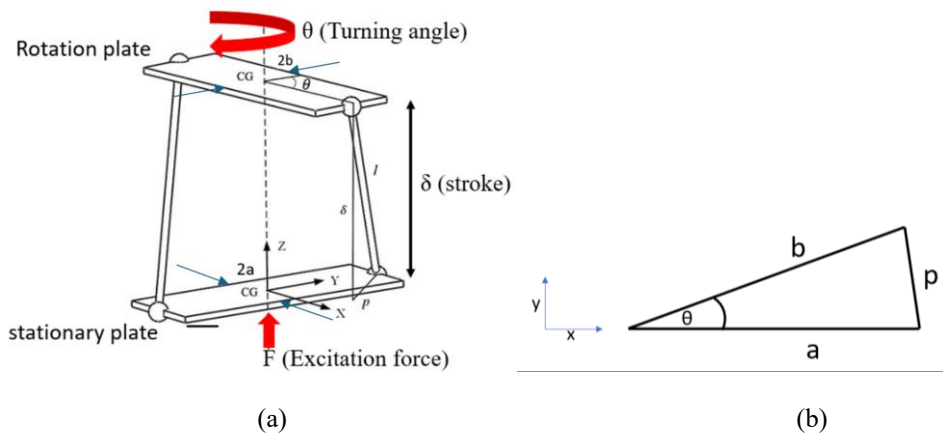


Fig. 2. (a) Schematic of 2-leg mechanism [14] and (b) Projection of rotation in XY-plane.

Maravandi et al. [14] present a mechanism that converts linear motion into rotation via two connected plates: one stationary and one rotating. The stationary plate moves linearly with the vehicle suspension, transferring force through rods to rotate the other plate (Figure 2a). A simplified two-leg design, utilizing fewer components, also enables efficient energy conversion. The rods link both plates with the stationary plate's displacement driving rotation as shown in equation (5).

$$\delta^2 = l^2 - p^2 \quad (5)$$

Let l represent the connecting rod length, δ the plate stroke, and p the projection length of rotation. In the two-leg mechanism, the centers of the stationary and rotating plates align in a vertical plane. To determine the projection length of rotation (p), the center width of the plates is important, as shown in Figure 2b. In this setup, θ is the rotating plate's angle, a is the stationary plate's center width, and b is the rotating plate's center width. The length p is not perpendicular to a and b , but it can be calculated using p_x and p_y , as shown in equation (6).

$$p^2 = p_x^2 + p_y^2 \quad (6)$$

$$\text{where } p_x^2 = a - b\cos(\theta) \text{ and } p_y^2 = b\sin(\theta)$$

The relationship between the stroke of the 2-leg mechanism and the rotating plate's angle is determined by substituting equation (6) into equation (5), as shown in equation (7).

$$\delta = \sqrt{l^2 - a^2 - b^2 + 2ab\cos\theta} \quad (7)$$

The dimensions of the two-leg mechanism impact both the stroke range and the rotating plate's turning angle (θ). From Equation (7), the turning angle can be expressed in terms of the sprung and unsprung mass responses using Equation (8), which defines the relationship between the stroke (δ) and the responses of the sprung mass (z_s) and unsprung mass (z_u).

$$\delta = s + (z_u - z_s) \quad (8)$$

where s is the equilibrium of stroke when the overall system is in place before operation. Transmission of rotation operates from the rotating plate's angle (θ) to the angular displacement (γ) of an electrical generator, as discussed in the following section.

The two-leg mechanism converts the linear movement (piston displacement) into angular rotation of the connecting links, which drives the rotational component. The rotating plate (Figure 2(b), upper part) is connected to the outer race of the integrated one-way bearing, which, in turn, transmits the unidirectional rotation to the generator shaft for power generation.

4. Analysis of Generator Dynamics in a Vibration-Based Energy Harvesting Suspension System

The voltage derived from the generator, as depicted in Figure 3b. It is connected to the circuit for the purpose of energy measurement. The dynamics equation of the generator circuit can be articulated in equation (9).

$$L \frac{di}{dt} + iR_i = k_e \dot{\psi} - iR_e \quad (9)$$

where $\dot{\psi} = n\dot{\gamma}$, $i = U/R_e$, U represents the load voltage, while R_e denotes the resistance of external load. n is gear ratio, R_i is internal coil resistance, L is the coil inductance, and k_e represents the electro-mechanical coupling constant. $\dot{\gamma}$ denotes the angular velocity of the electrical machine's shaft. The dynamic equation governing the generator circuit is derived in equation (10).

$$L \frac{\ddot{U}}{R_e} + (R_i + R_e) \frac{\dot{U}}{R_e} - nk_e \dot{\gamma} = 0 \quad (10)$$

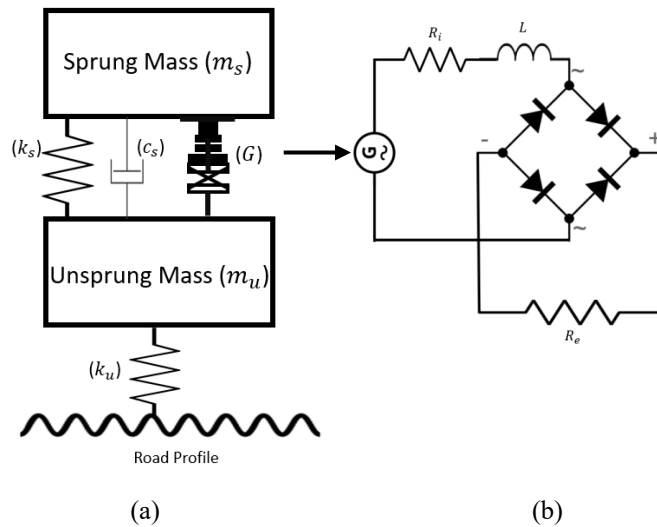


Fig. 3. (a) Schematic diagram of overall system. and (b) Circuit of the generator.

The electromagnetic generator converts the rotational kinetic energy into electrical power output. Due to the nature of the fluctuating rotational speed, the generator produces Alternating Current (AC). As illustrated in the electrical schematic (Figure 3(b)), a full-bridge rectifier circuit is employed downstream to effectively convert this AC power into usable Direct Current (DC) power. It is specified that the voltage U shown in the electrical schematic (Figure 3(b)) refers to the load voltage (DC voltage) measured across the load resistance R_e after rectification.

5. Dynamics of an Energy Harvesting System Utilizing a One-Way Bearing

The harvesting device is placed between the sprung and unsprung masses, with the tire transmitting excitation force through the unsprung mass and sharing energy with the shock absorber (Figure 3(a)). The indirect-drive mechanism operates in two stages: the **lock stage**, where the one-way bearing transfers rotational force to the electrical machine, and the **free stage**, where no force is transmitted. The device alternates between these stages. A prototype was designed and engineered for experimental validation (Figure 4a). System dynamics are analyzed using a transformed mechanism as a constraint, with the Lagrange multiplier (λ), as shown in equation (11).

$$\mathcal{L}(\mathbf{q}, \dot{\mathbf{q}}, \lambda) = f(\mathbf{q}, \dot{\mathbf{q}}) + \lambda g(\mathbf{q}) \quad (11)$$

Where λ is Lagrange multiplier

$\mathcal{L}(\mathbf{q}, \dot{\mathbf{q}}, \lambda)$ is Lagrangian function of variable $\mathbf{q}$

$f(\mathbf{q}, \dot{\mathbf{q}})$ is the function to determine local maxima and minima

$g(\mathbf{q})$ is the equality constrain

The overall system is formulated by combining the vehicle's quarter-car suspension and the electrical machine using the Lagrangian approach. The key variables are coordinate variables, which form the column matrix $\mathbf{q}$. The system's coordinate variables include those of both subsystems: the vehicle's suspension and the electrical machine. The vehicle suspension system, shown in Figure 1, is described by equations (1) and (2), with coordinate variables representing the responses of the sprung mass (z_s) and unsprung mass (z_u), reflecting two degrees of freedom.

The electrical machine system, shown in Figure 3b and described in equation (10), uses a coordinate variable that represents the angular position of the machine's shaft during operation. Therefore, the column matrix $\mathbf{q}$ consists of the responses of the unsprung and sprung masses, as well as the electric current of the generator, as expressed in equation (12)

$$\mathbf{q} = \begin{bmatrix} z_u \\ z_s \\ \gamma \end{bmatrix} \quad (12)$$

To analyze the dynamics of the augmented system, both the vehicle's suspension system and the electrical machine system are formulated using Lagrangian equations. Initially, the dynamics of the vehicle's suspension system, considered as **the first subsystem**, are described by equations (1)-(2) and can be expressed in the form of equations (13) and (14)

$$\begin{bmatrix} m_u & 0 \\ 0 & m_s \end{bmatrix} \begin{bmatrix} \ddot{z}_u \\ \ddot{z}_s \end{bmatrix} + \begin{bmatrix} c_u \dot{z}_u - c_u \dot{z}_s + (k_u + k_s)z_u - k_s z_s \\ -c_u \dot{z}_u + c_u \dot{z}_s - k_s z_u + k_s z_s \end{bmatrix} = \begin{bmatrix} k_u z_r \\ 0 \end{bmatrix} \quad (13)$$

$$\mathbf{M}_{s1} \ddot{\mathbf{q}}_s + \mathbf{h}_{s1}(\mathbf{q}_s, \dot{\mathbf{q}}_s) = \mathbf{g}_{f1} \quad (14)$$

The matrix containing the masses of the sprung and unsprung masses is referred to as the matrix $\mathbf{M}_{s1}$ of the first subsystem. The matrix that incorporates all the damping and spring constants is called the matrix $\mathbf{h}_{s1}$ while $\mathbf{g}_{f1}$ generalized forces associated with the corresponding coordinate $\mathbf{q}_s$. The vehicle's suspension system, modeled with two degrees of freedom in the quarter-car model, has its primary input as the force generated by the vehicle traveling on the road surface. This force serves as the primary input for the overall system.

The second subsystem, the electrical machine, is derived from both the mechanical and electrical components of a dynamo, as schematically illustrated in Figure 3b.

The conversion of rotational kinetic energy from the two-leg mechanism into usable electrical power is achieved using an electromagnetic generator. The overall electrical system (e.g., the power conditioning and load circuit) is initially treated as a separate subsystem, as conceptually described in Equation (10).

However, the electromechanical coupling—specifically the resistive torque exerted by the generator—is critically incorporated into the mechanism's dynamic model. This resistive torque, derived from the fundamental electrical equation and essential for accurately calculating the power output, is explicitly formulated within the generalized equations of motion for the mechanical subsystem as presented in Equation (15).

$$I_{eq} \ddot{\gamma} + c_e \dot{\gamma} = -k_t i \quad (15)$$

Where I_{eq} is equivalent moment of inertia of electrical machine ($I_l + I_{fw} + I_g + n \cdot I_r$), with the following parameters:

- I_l is moment of inertia of rotating plate
- I_{fw} is moment of inertia of flywheel
- I_g is moment of inertia of gearhead
- I_r is moment of inertia of generator's rotor
- n is gear ratio
- c_e is damping coefficient of generator,
- k_t is generator torque constant

Equation (15) describes the dynamics of the electrical machine in the absence of a transmission force, utilizing a two-leg mechanism. It also characterizes the generator's behavior during the free stage, which serves as the system's initial phase. The mechanical equations of motion become decoupled due to stage switching by the rectifier motion mechanism.

The gear ratio (n) in Equation (10) used in the experiment is unity, as specified in Table 2. It is important to note that Equations (10) and (15) establish the coupling between the electrical circuit of the generator and its mechanical rotation. A 5-phase generator is utilized for electrical power generation in the experiment. According to Equation (10), the power generated by the 5-phase generator is given by:

$$P = 2.5 k_e^2 \dot{\gamma}^2 / (R_e + R_l) \quad (16)$$

Meanwhile, the power dissipated in the electromechanical system, as described in Equation (15), is:

$$P = c_e \dot{\gamma}^2 \quad (17)$$

By equating these power expressions, the damping coefficient of the generator (c_e) can be expressed as:

$$c_e = 2.5k_e^2/(R_e + R_i) \quad (18)$$

By integrating the generalized equations of motion for the quarter-car model Equation (13) and the two-leg mechanism Equation (15), the two subsystems are combined. Prior to presenting the final augmented system in Equation (19), the necessary terms related to the electrical power and damping coefficient (c_e) are derived in Equations (16)-(18). This step ensures the completeness of all components required for the final system equation. The resulting augmented system equation is then presented in detail in Equation (19).

$$\begin{bmatrix} m_u & 0 & 0 \\ 0 & m_s & 0 \\ 0 & 0 & I_{eq} \end{bmatrix} \begin{bmatrix} \ddot{z}_u \\ \ddot{z}_s \\ \ddot{\gamma} \end{bmatrix} + \begin{bmatrix} c_u \dot{z}_u - c_u \dot{z}_s + (k_u + k_s)z_u - k_s z_s \\ -c_u \dot{z}_u + c_u \dot{z}_s - k_s z_u + k_s z_s \\ c_e \dot{\gamma} \end{bmatrix} = \begin{bmatrix} k_u z_r \\ 0 \\ -k_t i \end{bmatrix} \quad (19)$$

Although Equation (19) describes the augmented system, the unique, non-smooth dynamic behavior introduced by the one-way bearing (OWB)—the mechanism responsible for motion rectification—is modeled through the application of time-varying kinematic constraints. The detailed operation of the OWB, specifically its crucial state-switching mechanism between the free-wheeling stage and the locking stage, is mathematically formulated in the following equations. Calculations involving the generalized constraint force (λ), impulse ($\hat{\lambda}$), and coordinate variable derivatives ($\dot{q}$) (as shown in the following Equations (20)–(32)) are utilized to accurately capture this dynamic behavior within the system.

$$\mathbf{M}\ddot{\mathbf{q}} + \mathbf{h}(\mathbf{q}, \dot{\mathbf{q}}) = \mathbf{g}_f \quad (20)$$

Equation (19) can be rewritten in a compact form, as expressed in Equation (20), where $\mathbf{g}_f$ represents the associated generalized forces. To analyze the system's response, the augmented equation is utilized to resolve the constraint, requiring an examination of the system's dynamics with a rectifier motion mechanism. Initially, the system comprises two uncoupled subsystems, as shown in Equation (19), which are subsequently linked through a constraint equation representing the rectifier motion mechanism (one-way bearing).

For the dynamics of a constrained subsystem, the differential conditions of the constraints must be satisfied through admissible position, velocity, and acceleration, as expressed in Equation (21).

$$g(\mathbf{q}) = 0, \dot{g}(\mathbf{q}) = 0, \text{ and } \ddot{g}(\mathbf{q}) = 0 \quad (21)$$

To derive the combined system equation incorporating the effect of the one-way bearing, the relevant constraint condition is expressed as $\dot{g}(\mathbf{q}) = 0$. The impact of the one-way bearing on the system's dynamics is assessed by deriving the constraint equation at the angular velocity transition point, as explicitly presented in Equation (22).

$$\dot{\theta} = \dot{\gamma} \quad (22)$$

Equation (22) serves as the kinematic constraint equation for the entire system, ensuring the physical coupling between the quarter-car model and the two-leg mechanism. Furthermore, Equation (22) specifically describes the system during the lock stage of the energy harvesting process, where the angular velocity of the rotating plate matches that of the electrical machine shaft. The constraint equation can be derived by incorporating equation (22) into a Jacobian matrix with the variable $\mathbf{q}$, noting that θ is expressed in terms of z_u and z_s . Equation (23) then represents the Jacobian matrix for the permissible velocity condition, which is associated with the constraint equality

$$\begin{bmatrix} \frac{\partial \theta}{\partial z_u} & \frac{\partial \theta}{\partial z_s} \end{bmatrix} \begin{bmatrix} \dot{z}_u \\ \dot{z}_s \end{bmatrix} = (1)(\dot{\gamma}) \quad (23)$$

The permissible velocity condition, associated with the equality constraint $g(\mathbf{q})$ in the Lagrangian formulation, is expressed in Equation (24).

$$\begin{bmatrix} \frac{\partial \theta}{\partial z_u} & \frac{\partial \theta}{\partial z_s} & -1 \end{bmatrix} \begin{bmatrix} \dot{z}_u \\ \dot{z}_s \\ \dot{\gamma} \end{bmatrix} = 0 \quad (24)$$

Based on the definition of the column matrix $\mathbf{q}$, Equation (24) defines the Jacobian matrix as $\mathbf{J}$ and the matrix of coordinate variable derivatives as $\dot{\mathbf{q}}$. This can be rewritten as $\mathbf{J}\dot{\mathbf{q}}=0$.

From equation (7), the turning angle of the rotating plate can be rewritten in terms of the response of sprang mass and unsprang mass by using equation (8) which expresses the relationship of stroke (δ) and response of sprang mass (z_s) and unsprang mass (z_u).

Matrix $\mathbf{J}$ can be derived by applying Equations (7) and (8) and taking the partial derivatives with respect to the coordinate variables. After rearranging the denominator's multiplication, Equation (25) is obtained.

$$\begin{bmatrix} -(s + (z_u - z_s)) & (s + (z_u - z_s)) & ab \sin \left(\cos^{-1} \left(\frac{(s + (z_u - z_s))^2 - l^2 + a^2 + b^2}{2ab} \right) \right) \end{bmatrix} \dot{\mathbf{q}} = 0 \quad (25)$$

In the Lagrangian formulation, the constraint term comprises both the equality constraint and the Lagrange multiplier, which physically represents a generalized force. The system, incorporating the Lagrange multiplier, is derived by combining Equation (20) and Equation (25) into a Lagrangian equation with the constraint, as shown in Equation (26). The complete augmented system is governed by the matrix form presented in Equation (26), where Equation (22) is inherently incorporated as the kinematic constraint, ensuring the physical consistency of the overall dynamic model.

$$\mathbf{M}(\mathbf{q})\ddot{\mathbf{q}} + \mathbf{h}(\mathbf{q}, \dot{\mathbf{q}}) = \mathbf{g}_f(\mathbf{q}) + \mathbf{J}^T(\mathbf{q})\lambda \quad (26)$$

Building on Equations (24) and (25), along with the permissible velocity condition in Equation (21), the permissible velocity is explicitly expressed in Equation (27). Consequently, the permissible acceleration condition is established, as presented in Equation (28).

$$\dot{\mathbf{g}}(\mathbf{q}) = \mathbf{J}(\mathbf{q})\dot{\mathbf{q}} = 0 \quad (27)$$

$$\ddot{\mathbf{g}}(\mathbf{q}) = \frac{\partial \dot{\mathbf{g}}(\mathbf{q})}{\partial t} = \frac{\partial \mathbf{J}(\mathbf{q})\dot{\mathbf{q}}}{\partial t} = \dot{\mathbf{J}}(\mathbf{q})\dot{\mathbf{q}} + \mathbf{J}(\mathbf{q})\ddot{\mathbf{q}} = 0 \quad (28)$$

The combined system in Equation (26), together with the permissible acceleration condition in Equation (28), facilitates the determination of the Lagrange multiplier (λ). The corresponding equation for computing the Lagrange multiplier is given in Equation (29).

$$\lambda = (\mathbf{J}(\mathbf{q})\mathbf{M}^{-1}\mathbf{J}^T(\mathbf{q}))^{-1}(\mathbf{J}(\mathbf{q})\mathbf{M}^{-1}(\mathbf{h}(\mathbf{q}, \dot{\mathbf{q}}) - \mathbf{g}_f(\mathbf{q})) - \dot{\mathbf{J}}(\mathbf{q})\dot{\mathbf{q}}) \quad (29)$$

Equation (29) is the derived expression for the generalized constraint force (λ) that governs the system when the kinematic constraint of the lock stage is assumed. The physical condition for maintaining the lock stage is that this calculated constraint force must be non-negative ($\lambda \geq 0$). If $\lambda < 0$, the constraint force is no longer physically exerted, signaling the precise moment the system exits the lock stage and transitions back into the free-wheeling stage.

The system's operation is characterized by the non-smooth dynamics inherent in the one-way bearing's state-switching mechanism. When the bearing transitions from the free-wheeling stage to the lock stage, an impulse occurs, instantaneously aligning the angular velocity of the electrical machine's shaft with that of the rotating plate. This crucial velocity change during the transition is mathematically analyzed using the Impulse-Momentum Equation (30).

$$\mathbf{M}\dot{\mathbf{q}}^+ - \mathbf{M}\dot{\mathbf{q}}^- = \mathbf{J}^T(\mathbf{q})\hat{\lambda} \quad (30)$$

Equation (30) describes the phenomenon occurring over a short period when the one-way bearing switches stages. During this time, the momentum before impact ($\mathbf{M}\dot{\mathbf{q}}^-$) changes to the momentum after impact ($\mathbf{M}\dot{\mathbf{q}}^+$) due to the impulse inserted into the system. Here, $\dot{\mathbf{q}}^-$ represents the velocity before the impact, while $\dot{\mathbf{q}}^+$ is the velocity after the impact. The impulse torque, denoted as $\hat{\lambda}$, is what causes the velocity change to $\dot{\mathbf{q}}^+$. The values of $\hat{\lambda}$ and $\dot{\mathbf{q}}^+$ can be derived using Equations (30) and (27), as shown in Equations (31) and (32), respectively.

$$\hat{\lambda} = -(J(q)M^{-1})^{-1}J(q)\dot{q}^- \quad (31)$$

$$\dot{q}^+ = (I - M^{-1}J^T(q)(J(q)M^{-1}J^T(q))^{-1}J(q))\dot{q}^- \quad (32)$$

where I is an identity matrix
 $\dot{q}^-$ is the velocity before impact

When the torque λ in equation (29) is greater than zero, an impulse torque occurs as described in Equation (31). From Equation (32), the velocity after impact ($\dot{q}^+$) becomes higher than before impact ($\dot{q}^-$). This happens because, during the lock stage, the one-way bearing briefly transmits an impulse torque ($\hat{\lambda}$) to the electrical machine's shaft. This impulse torque causes the rotational speed to change rapidly until it matches the speed of the rotating plate.

Numerical simulations were performed using parameters aligned with the experimental setup, as shown in Table 2. The prototype used in the experiment is detailed in the following section. The simulations primarily relied on Equations (4), (9), (10), (17), (19), (25), (29), and (32), with MATLAB employed for computation.

6. Development and Testing of a One-Way Rotational Power Harvesting Prototype

The One-Way Rotational Power Harvesting prototype was specifically designed and fabricated with integrated components to facilitate a comprehensive, system-level performance evaluation. Its functional architecture is based on three critical elements: a mechanism for converting the suspension's reciprocating motion into rotation, an integrated one-way bearing for directional motion rectification, and an electromagnetic system for converting the resultant kinetic energy into electrical power. The physical prototype, schematically detailed in Figure 4, was scaled and dimensioned to meet the spatial constraints of the laboratory experimental setup. Although the two-leg geometric configuration is structurally similar to the design presented by Maravandi et al. [14], the core originality of this work is established by the novel integration and subsequent system-level dynamic modeling of the two-leg mechanism coupled with the one-way bearing. This integration, particularly the precise analysis of motion rectification and the associated impulse dynamics, represents a crucial innovation not addressed in the referenced literature.

The conversion of reciprocating to rotational motion is analyzed using Equations (7) and (8) to assess its suitability for the selected road classes. The two-leg mechanism is designed with optimized components for efficient rotation. Additionally, the one-way bearing and power conversion components are evaluated through numerical simulations, as detailed earlier. Table 2 presents the optimal parameter values for the experiment.

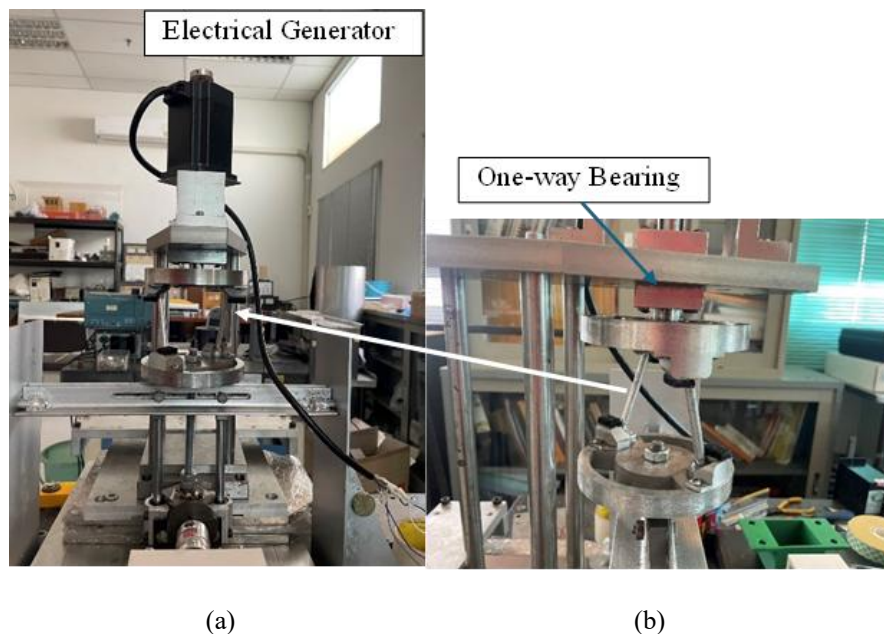


Fig. 4. (a) Prototype of the energy harvesting device. and (b) Prototype of the 2-leg mechanism.

Table 1: Parameters of the overall dynamics system in simulation.

Parameters (Unit)	Road Roughness Profile
Amplitude, A (mm)	10 / 20
Speed while travelling, v (m/s)	22.2
Vehicle travelling distance, d (m)	10
Parameters (Unit)	Vehicle
Unsprung mass, m_u (kg)	85
Sprung mass, m_s (kg)	1,300
Tire stiffness, k_u (N/m)	245,000
Suspension stiffness, k_s (N/m)	28,400
Suspension damping, c_u (N s/m)	1,800
Parameters (Unit)	Energy Harvesting Device
Width of stationary plate, 2a (mm)	80
Width of rotation plate, 2b (mm)	80
Length of connecting rod, l (mm)	80
Moment of inertia of 2-leg mechanism, I_l (kg m ²)	378,058.17 x 10 ⁻⁹
Moment of inertia of flywheel, I_{fw} (kg/m ²)	11,470.09 x 10 ⁻⁹
Gear ratio, n	1
Parameters (Unit)	Electrical Machine
Moment of inertia of generator's rotor, I_r (kg m ²)	56 x 10 ⁻⁷
The electro-mechanical coupling constant, k_e (V s/rad)	0.3072
Coil inductance, L (H)	1 x 10 ⁻³
Internal resistance, R_i (Ω)	5
External resistance, R_e (Ω)	0 - 100

It is important to clarify the relationship between the conceptual dynamic model (Figure 3) and the physical prototype (Figure 4(a)). While the dynamic model conceptually places the mechanism between the sprung and unsprung masses, the physical prototype's configuration in Figure 4 is the actual laboratory testing arrangement, where the generator and its housing are mounted externally (above the main mass location) for ease of assembly, instrumentation, and maintenance. Crucially, this external arrangement does not affect the validity of the dynamic model in Sections 2–5. This is because the modeling relies solely on two factors: first, the mechanism being driven by the relative linear displacement of the two mounting points ($\Delta z = z_u - z_s$, which is z_r in Equation (4)); and second, the generated reaction force acting along the suspension axis.

To ensure high manufacturing accuracy, a GF 5-axis CNC machining center and wire-cut EDM (Electric Discharge Machine) were employed to fabricate the two-leg mechanism and testing station. The One-Way Rotational Power Harvesting prototype is illustrated in Figure 4a, while Figure 4b depicts the two-leg mechanism with the one-way bearing integrated into the rotating plate. Experimental data were acquired using the PicoScope 400 series and meticulously analyzed in MATLAB.

The prototype of the energy harvesting device (as shown in Figure 4) was developed with an overall dimension of approximately 200 mm (length)x100 mm(diameter). Crucially, this prototype is designed solely for laboratory testing and validation of the dynamic model, and it has not been integrated or tested on a real vehicle. Its operation relies on the two-leg mechanism to convert the vertical linear motion into rotation. The integrated one-way bearing then rectifies this input motion, ensuring the generator shaft rotates unidirectionally, thereby maximizing continuous power output from the oscillatory suspension movement.

7. Results and Discussion

This research systematically categorizes results to improve clarity in simulations and experiments. The outcomes cover road roughness, the two-leg mechanism, simulation data, and experimental findings. Key simulation parameters include road roughness, vehicle properties, energy harvesting device characteristics, and electrical machine properties, as detailed in Table 2.

Road roughness, derived from Equation (4) and the power spectral density for Classes C and D, follows an approximate sinusoidal wave. Class C has a 10 mm amplitude, while Class D reaches 20 mm, defining the initial excitation for simulations and experiments. Rougher roads amplify vehicle response, especially in heavier vehicles and at higher speeds.

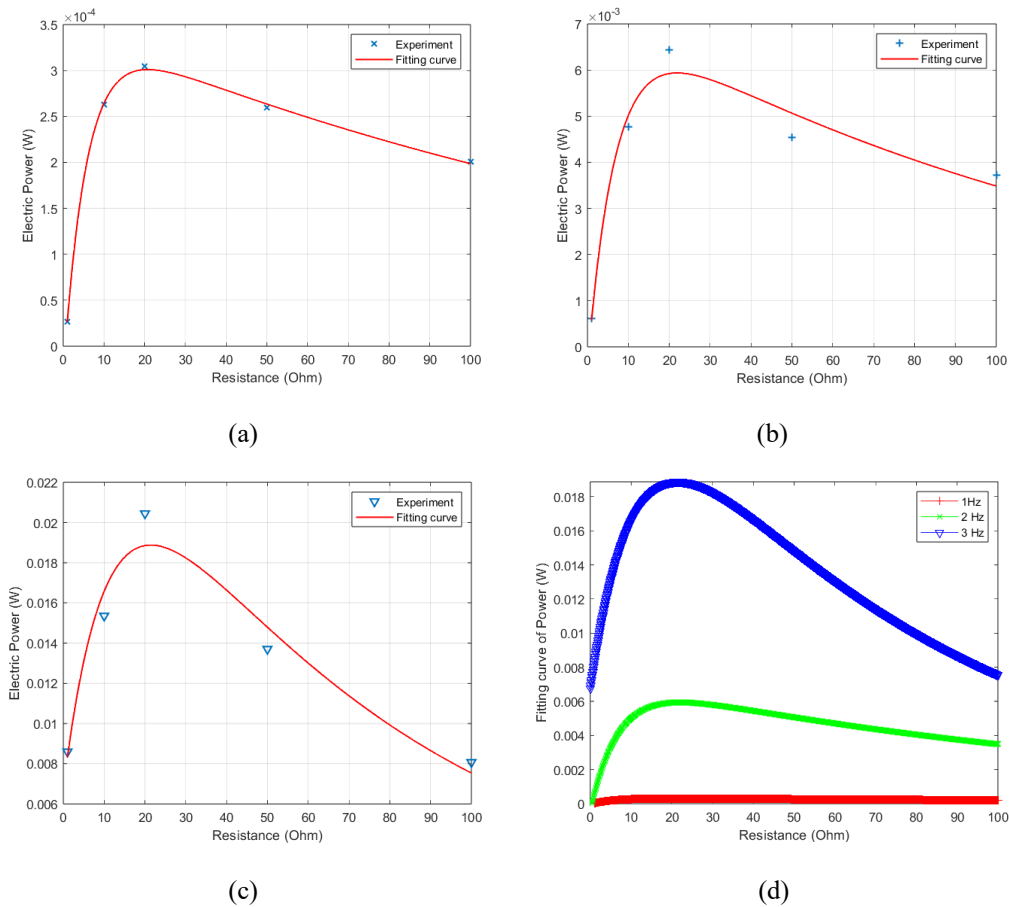


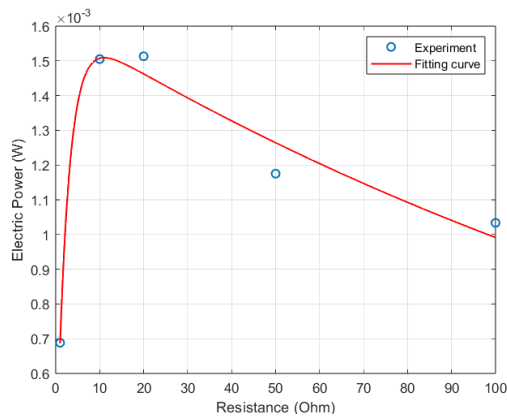
Fig. 5. The trendline of converted power in the experiment with an amplitude of 10 mm at (a) 1 Hz, (b) 2 Hz, (c) 3 Hz, and (d) 1-3 Hz.

The experimental analysis focuses on the relationship between generated power and load resistance to identify the optimal electrical load for maximum power transfer. We explicitly state that the resistance values presented in Figures 5–7 refer to the external load resistance (R_e), which is essential for determining the system's practical harvesting performance.

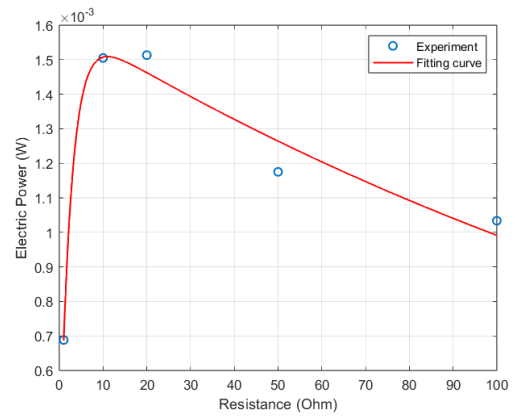
Experimental results show how converted power varies with resistance at 10 mm and 20 mm amplitudes. According to Ohm’s Law, increasing resistance reduces power. The experiment adjusts resistance and plots a trendline to identify the optimal range. Figure 5 shows that at 10 mm amplitude, power rises rapidly from 1 Ω to 10 Ω but declines exponentially beyond 20 Ω, approaching 0 W.

The optimal resistance range for maximum power conversion at 10 mm amplitude is 10–20 Ω. Power also depends on frequency, peaking at 18.8 mW at 3 Hz, 6 mW at 2 Hz, and 0.3 mW at 1 Hz, with sharper declines at higher frequencies.

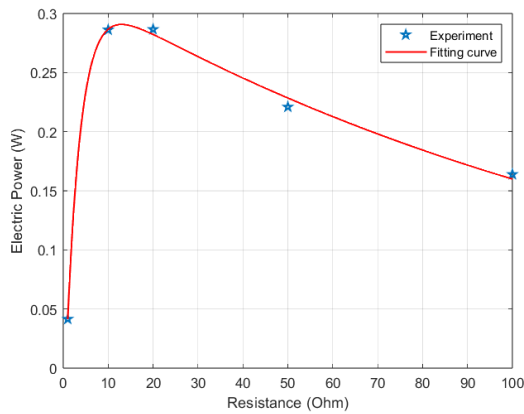
The experiment at 20 mm amplitude, shown in Figure 6, follows a similar approach to the previous test. Results reveal an exponential trend: power rises rapidly from 1–10 Ω before declining beyond 20 Ω. Peak power reaches 1.5 mW at 1 Hz, 0.12 W at 2 Hz, and 0.28 W at 3 Hz, with the steepest decline at higher frequencies.



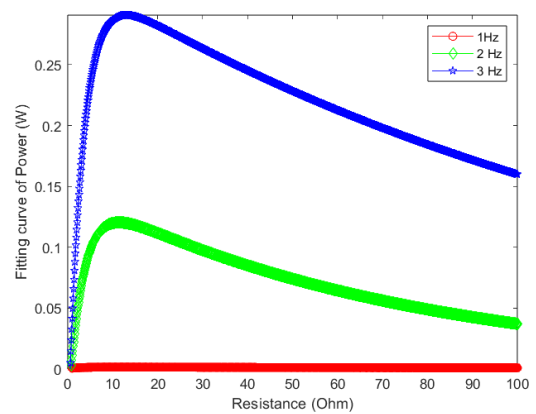
(a)



(b)

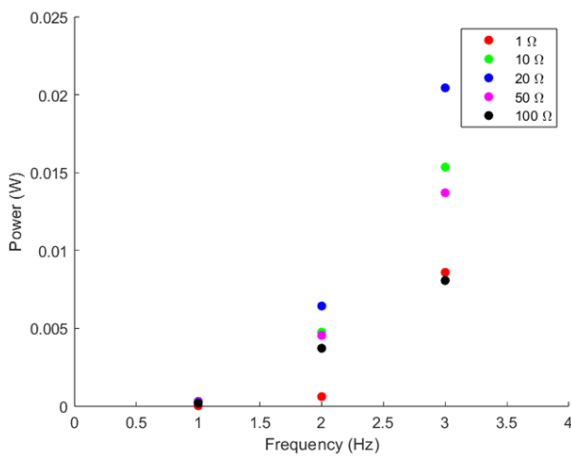


(c)

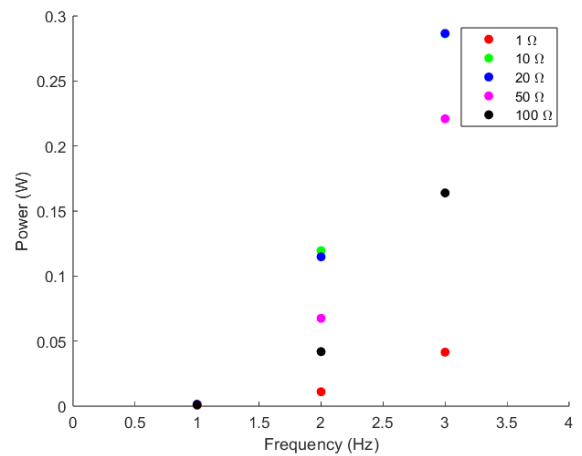


(d)

Fig. 6. The trendline of converted power in the experiment with an amplitude of 20 mm at (a) 1 Hz, (b) 2 Hz, (c) 3 Hz, and (d) 1-3 Hz.



(a)



(b)

Fig. 7. The comparison between the converted power in the different resistance at the experiment and frequency range at (a) amplitude of 10 mm and (b) at amplitude of 20 mm.

Figure 7 compares power distribution across frequencies within a set resistance range, showing maximum power at $20\ \Omega$ for all frequencies. Power is minimal at 1 Hz but increases exponentially up to 3 Hz. Experimental results confirm that resistance influences converted power in two phases: an initial increase (1–10 Ω), followed by a decline, consistent with Ohm's Law. The 20 mm amplitude generates more power than the 10 mm amplitude, with peak output in the 10–20 Ω range for both.

Figure 8 illustrates the numerical simulation results for the power generated as a function of the rotational speed of the electrical machine's shaft. This data, derived from the dynamic model (Equations (4) to (32)), demonstrates the system's performance when the external load resistance is fixed at $20\ \Omega$.

Figure 8 clearly shows the relationship between the generated power and the angular velocity of the rotating plate. Higher excitation frequencies expand the achievable power range for both input amplitudes, with the 20 mm amplitude achieving a broader range than the 10 mm amplitude. This highlights the key role of frequency in maximizing power generation, while larger excitation amplitudes further extend the potential power range.

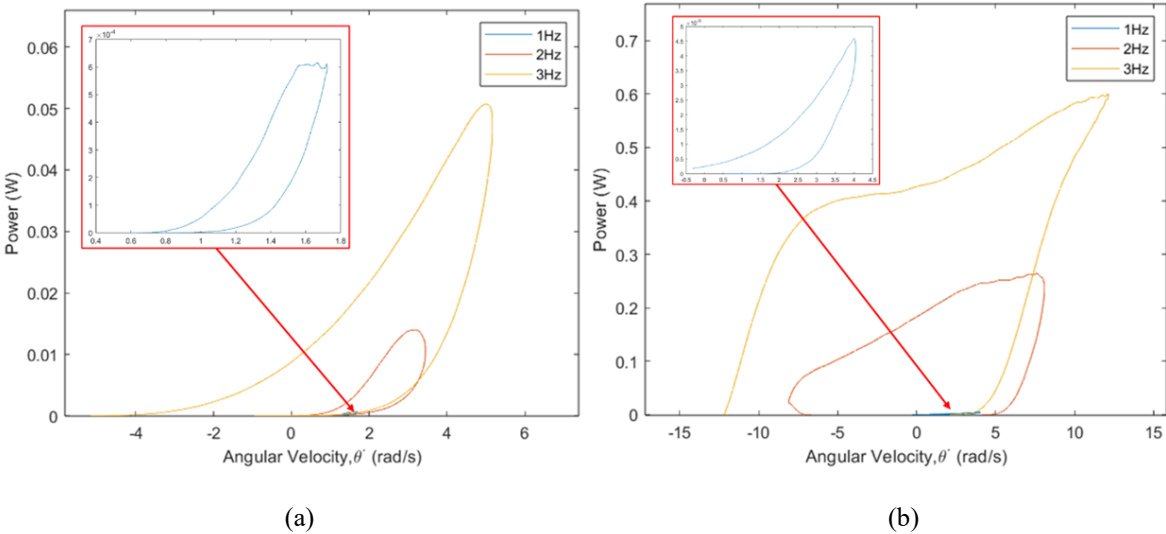


Fig. 8. The relationship between generating power and the angular velocity of the rotation plate of excitation (a) amplitude of 10 mm and (b) amplitude of 20 mm.

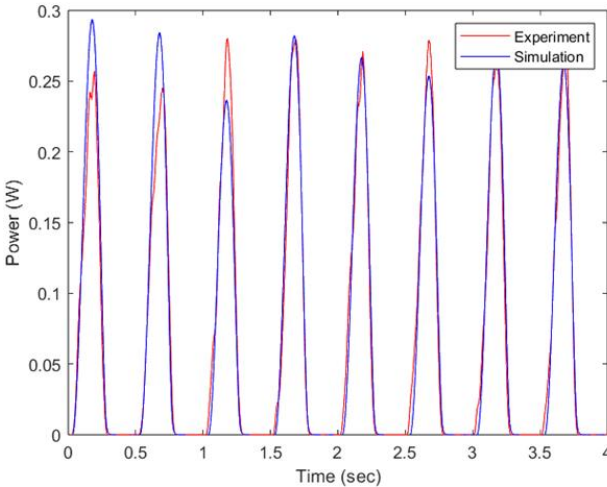


Fig. 9. The comparison graph of the converted power between the experiment and simulation with an amplitude of 20 mm at frequency 2 Hz and external resistance of $10\ \Omega$.

The experiment validates the simulation's accuracy. The primary focus of the experimental validation is the direct measurement of the electrical power generated across the external load (R_e), which serves as the tangible output for comparison with the simulated results. Adjusting excitation and energy harvesting parameters can improve the simulation for better comparison. Figure 9 shows a close match between experimental and simulated power, with minimal discrepancies in peak values. The simulation achieves 88.15% accuracy, confirming its reliability in predicting real-world performance.

8. Conclusions

This research successfully developed and experimentally validated a comprehensive dynamic model for an energy harvesting device utilizing a one-way bearing (OWB) within an indirect-drive mechanism. The study focused on exploiting the kinetic energy available in vehicle suspension systems for conversion into usable electrical power, addressing the inevitable energy losses that occur, particularly under rough road conditions.

The analytical framework employed a quarter-car model to represent the vehicle dynamics, simulating vibrations and energy collection under ISO 8608 Class C and D road profiles (10 mm and 20 mm amplitude). The system's core design—the indirect-drive mechanism utilizing a two-leg linkage—requires the OWB for motion rectification to stabilize rotational motion for optimized power conversion. Crucially, the OWB served as the primary element driving the dynamic model's formulation, necessitating the specialized analysis of its complex switching dynamics and impulse effects (Sections 3–5) to accurately predict system performance.

Experimental results confirmed the effectiveness of the integrated harvesting system, demonstrating that the generator's rotor produces the desired unidirectional rotation. The harvested power output was found to be highly dependent on the external load resistance, with an optimal range identified between 10–20 Ω for maximum power transfer. Furthermore, the numerical simulation model achieved a high validation accuracy of 88.15% against experimental data, with simulation results confirming that power generation significantly enhances with increased excitation frequency and amplitude.

In conclusion, the OWB is confirmed as an effective motion rectifier for the indirect-drive system. The validated dynamic model is established as a reliable and robust predictive tool for optimizing system parameters and assessing real-world performance based on laboratory-simulated testing. This research provides a valuable technical foundation for the future design and efficient optimization of vehicle suspension energy harvesting systems.

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