



Article

Design of a Vibration Energy Harvester Powered by Machine Vibrations for Variable Frequencies and Accelerations

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Abstract

A vibration energy harvester (VEH) based on the principle of variable magnetic reluctance has been developed to enable wireless and maintenance-free power supply for condition monitoring sensors in vibrating machinery. Conventional battery or wired solutions are often impractical due to limited lifetime and high installation costs, motivating the use of vibration-based energy harvesting. The proposed VEH converts mechanical vibrations into electrical energy through the relative motion of a movable ferromagnetic core within a magnetic circuit. Unlike conventional VEH designs, where the magnet is the moving element, this concept utilizes a movable ferromagnetic core in combination with a stationary pole piece for voltage induction. This configuration enables a compact and easily adjustable proof mass, as neither the coil nor the magnet needs to be moved. The VEH is designed to operate effectively under excitation frequencies between 16 Hz and 50 Hz and acceleration levels from $9.81 \frac{\text{m}}{\text{s}^2}$ (equivalent to 1 g) up to $98.1 \frac{\text{m}}{\text{s}^2}$ (equivalent to 10 g). To ensure a reliable power supply, the VEH must deliver a minimum electrical output of 0.1 mW at the lowest excitation (1 g) while maintaining structural integrity. Additionally, the maximum permissible displacement amplitude of the movable core is limited to 1.15 mm to avoid mechanical damage and ensure durability over long-term operation. Coupled magnetic-transient and mechanical finite element method (FEM) simulations were conducted to analyze the system's dynamic behavior and electrical power output across varying excitation frequencies and accelerations. A laboratory prototype was developed and tested under controlled vibration conditions to validate the simulation results. The experimental measurements confirm that the VEH achieves an electrical output of 0.166 mW at $9.81 \frac{\text{m}}{\text{s}^2}$ and 16 Hz, while maintaining the maximum allowable displacement amplitude of 1.15 mm, even at $98.1 \frac{\text{m}}{\text{s}^2}$ (10 g) and 50 Hz. The strong agreement between simulation and experimental data demonstrates the reliability of the coupled FEM approach. Overall, the proposed VEH design meets the defined performance targets and provides a robust solution for powering wireless sensor systems under a wide range of vibration conditions.



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Keywords: vibration energy harvesting; finite element simulation; condition monitoring; variable reluctance; wireless sensor systems

1. Introduction

Wireless sensor networks (WSNs) and the Internet of Things (IoT) have created a growing demand for sensor systems capable of real-time monitoring of machines and structural components [1–3]. Conventional power supplies for these sensors, such as batteries or wired connections, pose several challenges. Batteries often suffer from limited lifetimes, large physical dimensions, and the need for frequent replacement, which can lead to costly machine downtime [4,5]. Wired power supply systems, on the other hand, are susceptible to failures and make retrofitting sensors on existing machinery both complex and expensive.

An alternative approach is to use VEHs as autonomous power sources for sensor nodes [6]. For condition monitoring applications, VEHs typically need to provide an average electrical power between 0.1 and 1 mW [7,8]. The wireless data transmission of sensor nodes accounts for the largest portion of this power consumption [6].

Energy harvesting refers to the conversion of ambient energy into usable electrical energy. Possible ambient energy sources include light, wind, temperature gradients, motion, and machine vibrations. This enables the autonomous operation of small wireless devices such as those found in distributed sensor networks [7,8].

Most VEHs can be categorized into two main functional principles: piezoelectric [4,9,10] and electromagnetic [4,9,11–16]. In electromagnetic VEHs, electrical energy is generated either by the relative motion between a coil and a magnet or by changes in the magnetic flux density. Permanent magnets are typically used to create the magnetic field in such systems [6]. Energy Harvesters using the principle of variable magnetic reluctance, to change the flux density, are called variable reluctance energy harvesters. They operate with rotating or periodically moving ferromagnetic parts [17]. It was shown that reducing the air gap between the moving parts relative to each other results in improving the power output [18]. The evaluation of the structure of a rotating Energy Harvester showed the positive effects on performance by guiding the flux through a ferromagnetic return path [19].

Electromagnetic VEHs generally provide higher power outputs compared to piezoelectric VEHs [20]. Commercially available VEHs are typically designed to operate near their resonance frequency, where the conversion efficiency from mechanical to electrical energy is maximized. Consequently, they must be specifically tuned to match the excitation frequency of their intended application [7,12,13,21,22]. However, in most real-world applications, excitation vibrations exhibit non-constant and time-varying frequencies. Once the excitation frequency deviates from the VEH's resonance, the harvested power decreases drastically. Therefore, broadening the operational frequency bandwidth of VEHs is one of the key challenges in current VEH research [8]. Increasing damping can mitigate frequency dependence to some extent [6], while electromagnetic VEHs inherently offer a wider frequency range than piezoelectric ones [20].

To enable a maintenance-free, energy-autonomous sensor system suitable for retrofitting on existing machinery, electromagnetic VEHs represent a particularly promising solution.

1.1. Objective of This Study

This work presents a novel VEH concept designed to supply wireless sensor nodes with electrical energy harvested from machine vibrations. These machine vibrations occur over a wide excitation frequency range from 16 Hz to 50 Hz, which reflects the characteristic operating conditions of industrial screening machines. Such machines are typically driven by two- to six-pole induction motors, whose rotational speeds and therefore the dominant excitation frequencies introduced into the machine structure fall within this frequency interval. Consequently, the VEH is specifically designed to operate efficiently within this

technologically relevant excitation band [23]. The associated acceleration amplitudes cover a wide range from $9.81 \frac{\text{m}}{\text{s}^2}$ up to $98.1 \frac{\text{m}}{\text{s}^2}$ corresponding to approximately 1 g to 10 g. To ensure sufficient energy output even outside resonance conditions, the VEH must exhibit a broad operational frequency bandwidth. In contrast to conventional VEH designs, where the magnet is the moving element [20], the proposed design utilizes a movable ferromagnetic core in combination with a pole piece for voltage induction. This configuration enables a compact and easily adjustable proof-mass, as neither the coil nor the magnet needs to be moved. The objective of this study is to investigate the vibrational and electromagnetic behavior of the system, including its coupled interactions, using FEM simulations.

1.2. Limitations of Existing VEH Solutions

A review of commercially available vibration energy harvesters shows that current solutions cover only a limited range of application scenarios. Most VEHs are designed either for very low excitation accelerations below 1 g or for highly specialized operating points near a narrow resonance frequency. Systems optimized for low acceleration levels tend to be compact and cost-efficient, but they are typically not robust enough to withstand the high vibration amplitudes commonly found in vibrating machines like screening machines and vibratory conveyors. Conversely, devices capable of tolerating higher accelerations often exhibit reduced sensitivity at low amplitudes or provide only limited output power outside a narrow operational frequency range [6,24].

Broadband VEH concepts exist, including both electromagnetic and piezoelectric approaches, but they frequently achieve their bandwidth through strong mechanical damping, which reduces peak power output. Piezoelectric broadband systems additionally suffer from material fatigue and reduced long-term stability. Overall, existing VEHs involve fundamental trade-offs between sensitivity, usable frequency bandwidth, maximum acceleration tolerance, and volumetric power density [25].

1.3. Structure of the Paper

Section 2 describes the design and working principle of the proposed VEH. Section 3 presents the coupled FEM simulations that analyze both the mechanical and electromagnetic subsystems. Section 4 details the laboratory experiments conducted to validate the simulation results, while Section 5 compares the numerical and experimental data. Finally, Section 6 discusses the findings and presents the conclusions of this study.

In this work, the term frequency refers to the mechanical excitation frequency unless explicitly stated otherwise. Electrical frequencies are always described as electrical frequency to ensure a clear distinction between mechanical and electromagnetic phenomena. All electrical and magnetic parameters are explicitly identified as such (e.g., magnetic flux density Φ , induced voltage U_{ind} , or electromagnetic force F_{em}).

2. Presentation of the VEH Concept

This chapter introduces the concept, structural design, and operating behavior of the proposed VEH. The VEH converts mechanical vibrations from its environment into electrical energy by means of electromagnetic induction.

To achieve reliable and efficient energy conversion over a wide frequency range, the mechanical and electromagnetic subsystems are analyzed in detail. The following sections describe the mechanical structure, the functional behavior under axial and radial excitation, and the fundamental physical relationships governing the VEH's performance.

2.1. Spring Types

Figure 1 shows a simplified schematic of the two-disk spring types employed in the VEH.

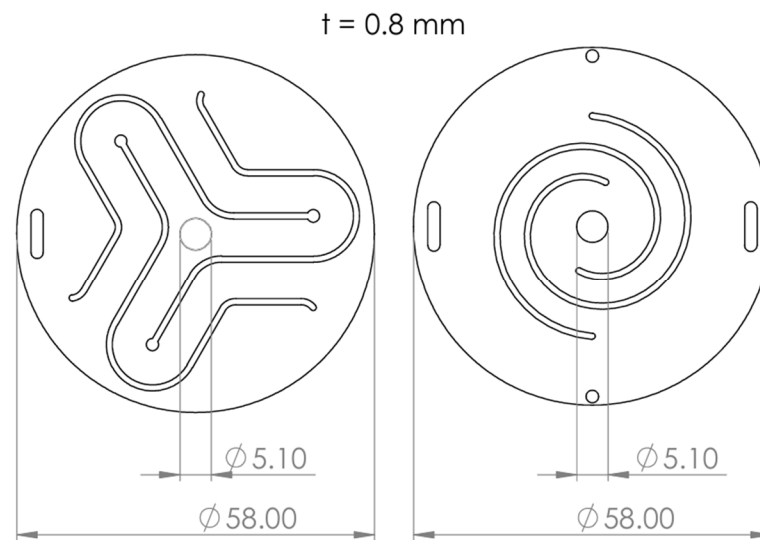


Figure 1. Overview of spring types (left: spring 1; right: spring 2).

To protect proprietary design details, the exact geometry is omitted. The figure highlights only the essential dimensions and mechanical characteristics required for understanding the subsequent analysis.

2.2. Structure of the VEH Model

Figure 2 shows the schematic structure of the proposed rotationally symmetric VEH, highlighting its main components and their spatial configuration.

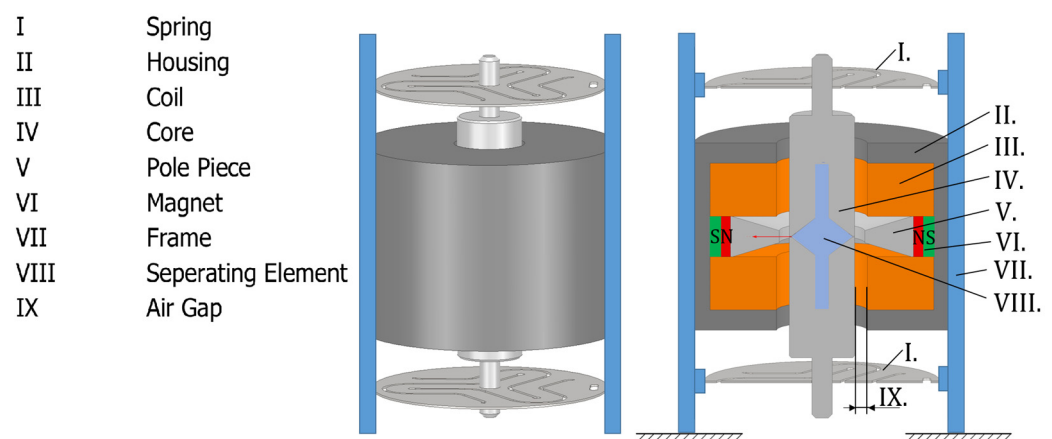


Figure 2. Configuration and primary components of the proposed VEH (left: full view; right: sectional view).

Two disk springs (I) are used to provide axial and radial support for the movable core (IV). The disk springs are made of cold-rolled strip steel grade 1.4310. Two different spring geometries are available, differing in their stiffness. These geometries were developed in preliminary investigations to achieve the highest possible radial stiffness while remaining functional for large vibration amplitudes. FEM pre-studies determined the axial spring stiffness $k_{fa} = 11,353.76 \frac{\text{N}}{\text{m}}$ for spring type 1 and $k_{fa} = 23,710.56 \frac{\text{N}}{\text{m}}$ for spring type 2. The maximum permissible amplitude $\hat{x}_{zul}$ for which the springs remain functional is 1.15 mm. Spring type 1 has a mass of 14.88 g, and spring type 2 weighs

15.28 g. The outer diameter of both springs is 58 mm, and the thickness is 0.8 mm. The vibrating mass of the VEH consists of the two-part core (IV) together with the connecting separation element (VIII). By selecting the spring stiffness and the vibrating mass, the natural frequency of the VEH can be adjusted accordingly. The coils (III) are dimensioned such that the ratio of their height to the total axial assembly height satisfies $\frac{V}{H+V} \approx 0.4$. VEHs with these proportions have been shown to exhibit a high energy density [20]. An aluminum connecting element (VIII) links the two halves of the core. The aluminum has a relative magnetic permeability of $\mu_r = 1.00000065$ [26]. The air gap (IX) between the core (IV) and the pole piece (V) is kept constant at 0.8 mm for all cases. This parameter remains unchanged in both the FEM simulations and the experimental setup to ensure consistency between model and measurement.

2.3. Functional Behavior

Figure 3 shows a sectional view of the VEH in its resting position (left) and with a displacement of the vibrating part (right). The physical parameters required to describe the system behavior are indicated in the illustration and defined as follows.

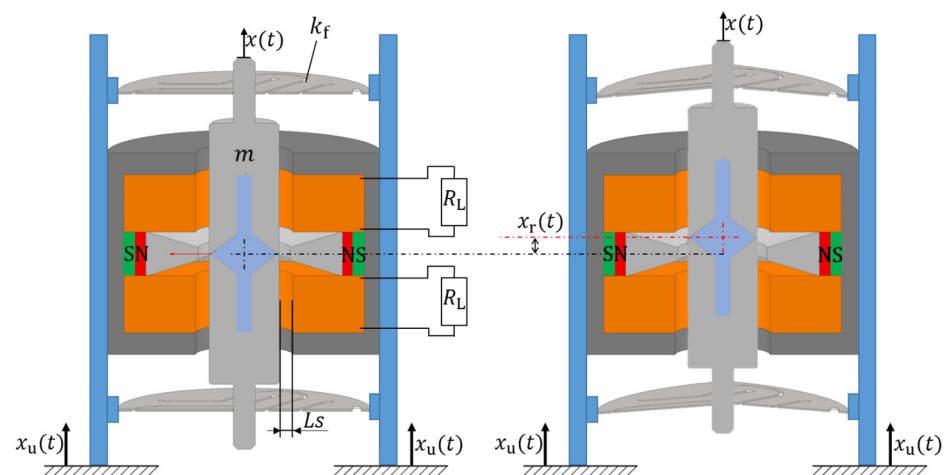


Figure 3. Sectional view of the functional prototype with indicated physical quantities.

The total displacement of the system is denoted by $x(t)$, while $x_u(t)$ represents the displacement of the foundation and $x_r(t)$ the relative displacement between the vibrating part and the housing.

The symbol m denotes the combined mass of components IV and VIII, and k_f represents the spring constant of the disk springs (I). The electrical load connected to the coils is characterized by the load resistance R_L , and L_s denotes the air gap between the core and the pole piece. Figure 4 shows a sectional view of the core with all axial forces caused by axial vibrations and defined as follows.

The forces acting on the core during axial motion, as illustrated in Figure 4, are defined as follows.

F_t represents the inertial force, F_d the damping force, and F_{fa} the axial spring force.

The parameters m , d , and k denote the mass, damping constant, and spring constant of the system, respectively. Considering the forces acting on the movable steel core (IV) as illustrated in Figure 3, the differential equation of motion of the VEH can be expressed as follows:

$$F_t + F_d + F_{fa} = 0 \quad (1)$$

$$m(\ddot{x}_r + \ddot{x}_u) + d\dot{x}_r + kx_r = 0 \quad (2)$$

$$m\ddot{x} + d\dot{x} + kx = 0 \quad (3)$$

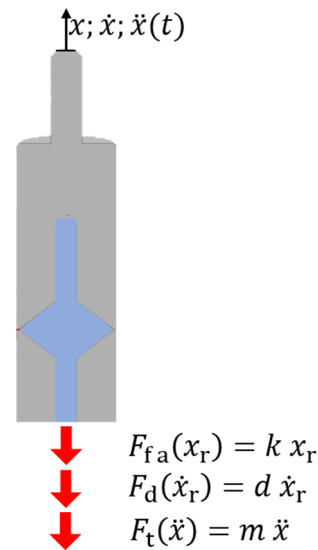


Figure 4. Sectional view of the core and the forces considered during axial motion.

The base excitation is defined as the displacement of the housing (IX), is expressed as $x_u(t) = \hat{x}_u \sin(\Omega t)$ where $\hat{x}_u$ is the excitation amplitude and Ω is the excitation angular frequency. x_r represents the axial displacement of the core (IV) and the separating element (VIII) relative to components (II), (III), (V), (VI) and (IX), and is given by $x_r = x - x_u$ [26].

The second-order term $m\ddot{x}$ in Equation (1) represents the inertial force F_t , which can be expressed as $F_t = m \ddot{x}$. Here, m denotes the combined mass of the core (IV) and the separating element (VIII), and $\ddot{x}$ is their acceleration.

The first-order term $d\dot{x}_r$ in Equation (1) represents the damping force F_d .

This damping force consists of a mechanical component $F_{d,m}$ and an electromagnetic component $F_{d,em}$.

The variable $\dot{x}_r$ denotes the relative velocity, and d is the total damping constant, which includes both a mechanical part d_m and an electromagnetic part d_{em} .

$$d = d_{em} + d_m \quad (4)$$

The mechanical damping force F_d results from the internal damping of the steel, the damping at the joints, and air resistance.

A literature value for steel fits is assumed, corresponding to a damping ratio of $D = 0.02$ [27].

Using the relationship $D = \frac{d_m}{2m\omega_0}$, it follows that:

$$d_m \approx 0.02 * 2 * m * \omega_0 \quad (5)$$

The electromagnetic damping force $F_{d,em}$ results from the voltage induced in the coils (III) and is defined by the following equation: $d_{em} = \frac{1}{R_L + R_c + j\omega L_c} \left(\frac{d\Phi}{dx_r} \right)^2$ where L_c is the coil inductance, R_c the coil resistance, R_L the load resistance, and Φ the magnetic flux. Given the maximum vibration frequency of 50 Hz, the simplification $j\omega L_c \approx 0$ is applied.

This approximation is valid for excitation frequencies up to 1000 Hz [20].

Accordingly, the electromagnetic damping constant can be expressed as:

$$d_{em} = \frac{1}{R_L + R_c} \left(\frac{d\Phi}{dx_r} \right)^2 \quad (6)$$

The term kx_r in Equation (1) represents the restoring force. It consists of the axial spring force F_{fa} and the axial reluctance force $F_{rel,a}$. x_r denotes the relative displacement and k is the total spring constant of the system.

It is composed of the spring constant of the mechanical spring (I), denoted as k_{fa} , and the coefficient k_m of the axial magnetic force–displacement characteristic.

$$k = k_{fa} + k_m \quad (7)$$

k_{fa} represents the axial spring constant of the mechanical spring. Its value is $k_{fa} = 11,353.76 \frac{N}{m}$ for spring type 1 and $k_{fa} = 23,710.56 \frac{N}{m}$ for spring type 2. These values were determined in preliminary studies using mechanical FEM simulations.

The parameter k_m denotes the linearized coefficient of the magnetic force–displacement characteristic. The axial magnetic reluctance force $F_{rel,a}$ acts on the core components (IV) and (VIII) when they are displaced by x_r from their equilibrium position [14–16,20,21].

The relationship between the magnetic force and the displacement of a movable armature within an iron magnetic circuit is referred to as the magnetic force–displacement characteristic [28].

From the axial magnetic reluctance force $F_{rel,a}$ and the relative displacement x_r of the core, the linear coefficient k_m of the magnetic force–displacement characteristic is determined.

The value of k_m was obtained from FEM-based preliminary investigations.

Within the displacement range of $x_r = -0.9 \text{ mm}$ to $x_r = 0.9 \text{ mm}$, the coefficient amounts to $k_m = 9296.32 \frac{N}{m}$.

Substituting the expression for the damping constant from Equation (4), $d = d_m + d_{em}$, and the expression for the spring constant from Equation (7), $k = k_{fa} + k_m$, into the equation of motion (Equation (1)) yields

$$m \ddot{x} + \left(\frac{1}{R_L + R_c} \left(\frac{d\Phi}{dx_r} \right)^2 + d_m \right) \dot{x}_r + (k_{fa} + k_m) x_r = 0 \quad (8)$$

The natural angular frequency is derived from Equation (8) using the following relationship [26]:

$$\omega_0 = \sqrt{\frac{k_{fa} + k_m}{m}} \quad (9)$$

The relative amplitude is determined from Equation (8) using $\vartheta = \frac{d}{2}m$ [26]:

$$\hat{x}_r = \frac{m}{k_{fa} + k_m} \hat{\hat{x}}_u \frac{1}{\sqrt{\left[1 - \left(\frac{\Omega}{\omega_0} \right)^2 \right]^2 + \left[2\vartheta \left(\frac{\Omega}{\omega_0} \right) \right]^2}} \quad (10)$$

The term of the magnification function of the forced damped vibration can be derived from Equation (10) as

$$\hat{x}_r = \frac{m}{k_{fa} + k_m} \hat{\hat{x}}_u V_1 \quad (11)$$

where

$$V_1 = \frac{1}{\sqrt{\left[1 - \left(\frac{\Omega}{\omega_0} \right)^2 \right]^2 + \left[2\vartheta \left(\frac{\Omega}{\omega_0} \right) \right]^2}} \quad (12)$$

The electrical power P_e is determined as follows [20]:

$$P_e = F_{em}(t) \dot{x}_r(t) \quad (13)$$

Since the VEH is also operated outside its natural frequency, the output power depends primarily on the frequency ratio. By substituting $\dot{x}_r = x_r \Omega$ and x_r from the magnification function in Equation (11) into Equation (13), the expression for the power as a function of frequency ratio is obtained.

$$\hat{P}_e = \frac{m}{k} \hat{\dot{x}}_u V_1 \Omega F_{em}(t) \quad (14)$$

Figure 5 displays a sectional view of the core (IV) with the radial forces acting on it caused by the radial eccentricity.

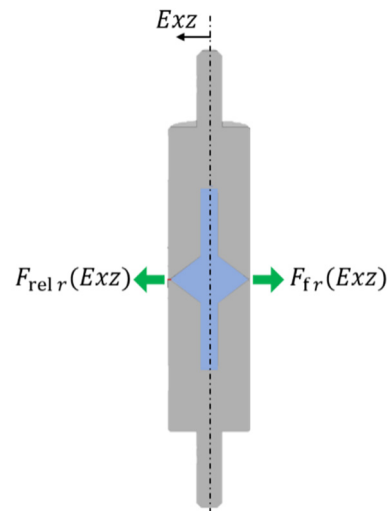


Figure 5. Sectional view of the core and the forces caused by eccentric displacement from its equilibrium position.

The parameters illustrated in Figure 5 are defined as follows. Exz denotes the radial eccentricity of the core, $F_{rel,r}$ represents the radial reluctance force, and $F_{f,r}$ the radial spring force. The air gap L_s represents the radial distance between the core (IV), the housing (II), the coil (III), and the pole piece (V). As L_s decreases, the magnetic flux Φ in the magnetic circuit increases, resulting in higher electromagnetic forces F_{em} [29]. A small air gap L_s therefore leads, according to Equation (6), to a high electromagnetic damping d_{em} , and according to Equation (14), to a high maximum electrical power $\hat{P}_e$.

However, a smaller L_s also causes a higher radial reluctance force $F_{rel,r}$, which tends to displace the core from its concentric equilibrium position.

The force $F_{rel,r}$ results from the radial displacement of the core, denoted by Exz .

This radial reluctance force forms an unstable equilibrium of the core around its concentric rest position at $Exz = 0$ [14–16,20,21].

The radial spring force of the disk spring $F_{f,r}$ must therefore be greater than the radial reluctance force $F_{rel,r}$ to ensure proper operation of the VEH.

It follows that:

$$F_{rel,r}(Exz) < F_{f,r}(Exz) \quad (15)$$

Preliminary FEM analyses determined that the minimum air gap L_s at which the condition in Equation (15) is satisfied is $L_s = 0.8$ mm.

3. Analyses of the Dynamic and Electrical Performance of the VEH Using FEM

This chapter focuses on the detailed analysis of the dynamic and electrical behavior of the proposed VEH through coupled FEM simulations. By employing coupled magnetic transient and mechanical FEM simulations, interactions between mechanical motion and electromagnetic induction are captured accurately. These simulations form the basis for

understanding system responses and guiding design optimizations. The following sections present the methodology and results of the simulation analysis, demonstrating the VEH's capability to deliver reliable power over the specified excitation range.

3.1. Coupled Magnetic Transient and Mechanical Simulation of Relative Deflection

In this section, the maximum relative amplitudes $\hat{x}_r$ of the movable core under maximum excitation are investigated using magnetic transient and mechanical simulations. Based on $\hat{x}_r$ and the permissible amplitude $\hat{x}_{zul}$, the functionality of the different mass–spring combinations is determined.

3.1.1. Modeling

The simulation setup consists of a magnetically transient FEM simulation and were performed in Ansys Electronics Desktop 2023 R1 (Ansys, Inc., Canonsburg, PA, USA) using the Ansys Maxwell solver. It is coupled with mechanical and electrical co-simulations. The mechanical co-simulations (a–d) are implemented in Ansys Simplorer. Their structure, shown in Figure 6, follows the axial forces noted in Figure 4 and numerically solves the equation of motion (8). Specifically, (a) applies the base excitation x_u , (b) implements the mechanical damping force $F_{d,m}$ using the damping coefficient d_m , (c) incorporates the axial spring force F_{fa} with spring constant k_{fa} and (d) includes the mass m of core (IV) and the connecting element (VIII), enabling the calculation of the inertial force F_t . Within the Ansys Maxwell co-simulation, the axial reluctance force $F_{rel,a}$ and the electromagnetic force F_{em} are determined.

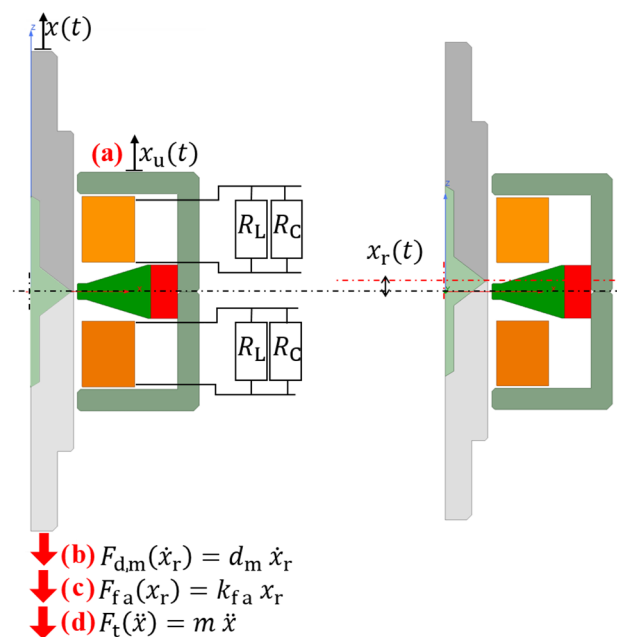


Figure 6. Setup for Ansys Maxwell co-simulations.

The geometry used in the magnetically transient simulation corresponds to the rotationally symmetric 2D model of the functional schematic shown in Figure 2, however without the housing and springs. In the magnetically transient simulation itself, the axial reluctance force $F_{rel,a}$ as well as the electromagnetic force F_{em} are calculated. The effects of the co-simulations (a) to (d) are visualized in Figure 7.

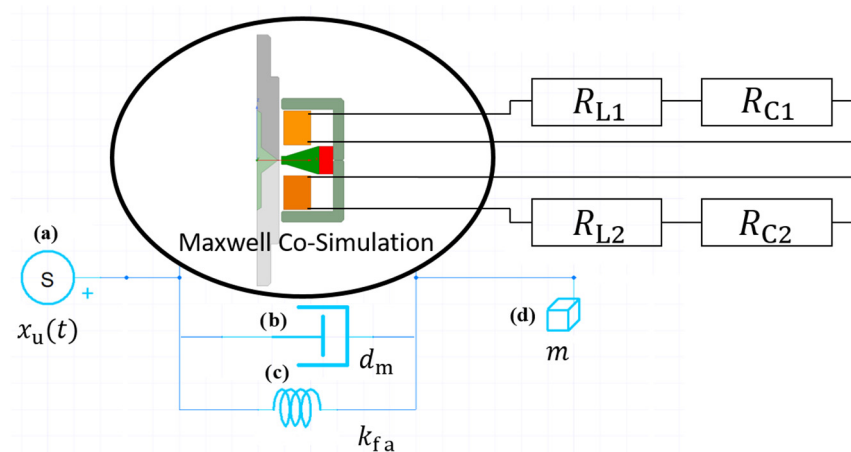


Figure 7. Geometry of the magnetic transient simulation (co-simulation influences in red).

Housing (II), core (IV), and pole piece (V) are made of unalloyed structural steel 1.0034, with permeability defined by the B-H curve given in Ansys Maxwell. The coil (III) is modeled as stranded copper wire ($\mu_r = 0.9999936$ [20]). The magnet (VI) is radially magnetized (NdFe35B35; see Table 1):

Table 1. Parameters of NdFe35B35.

Parameter	Description	Value
B_r	Remanence	1.17 T
H_c	Coercive field	-9.39×10^5 A/m
μ	Permeability	991.54×10^{-3}

The core is axially movable. Each coil is connected to two load resistors. R_{C1} and R_{C2} represent the internal resistance of the coils. They were measured as $R_{C1} = 2000 \Omega$ and $R_{C2} = 2100 \Omega$. Load resistors are set to $R_{L1} = R_{L2} = 1000 \Omega$, maximizing electromagnetic damping (Equation (8)).

Mesh convergence is achieved when the reluctance force changes by less than 1%.

3.1.2. Procedure

The system is excited using $x_u(t) = \hat{x}_u \sin(\Omega t)$. Mechanical frequency varies from 16 Hz to 50 Hz with a step size of 4.25 Hz. The defined frequency range and amplitude reflect the characteristics of the intended application. Amplitude $\hat{x}_u$ corresponds to 10 g acceleration at each frequency (Table 2):

Table 2. Input parameters for excitation at $98.1 \frac{m}{s^2}$ (10 g).

Frequency (Hz)	Amplitude (mm)
16	9.704
20.25	6.058
24.5	4.139
28.75	3.005
33	2.281
37.25	1.790
41.5	1.442
45.75	1.187
50	0.994

Spring constant k_{fa} and mass m are chosen so that the expected relative amplitude $\hat{x}_r$ (Equation (11)) remains below the amplification function's maximum (16–50 Hz). The time step is defined as 0.3 ms. The period under consideration is defined as 900 ms.

3.1.3. Result

Figure 8 shows the maximum relative amplitude $\hat{x}_r$ versus excitation frequency. The data points are connected using linear interpolation. The limit $\hat{x}_{zul}$ is represented by the horizontal line. Only combinations with $\hat{x}_r < \hat{x}_{zul}$ are considered functional.

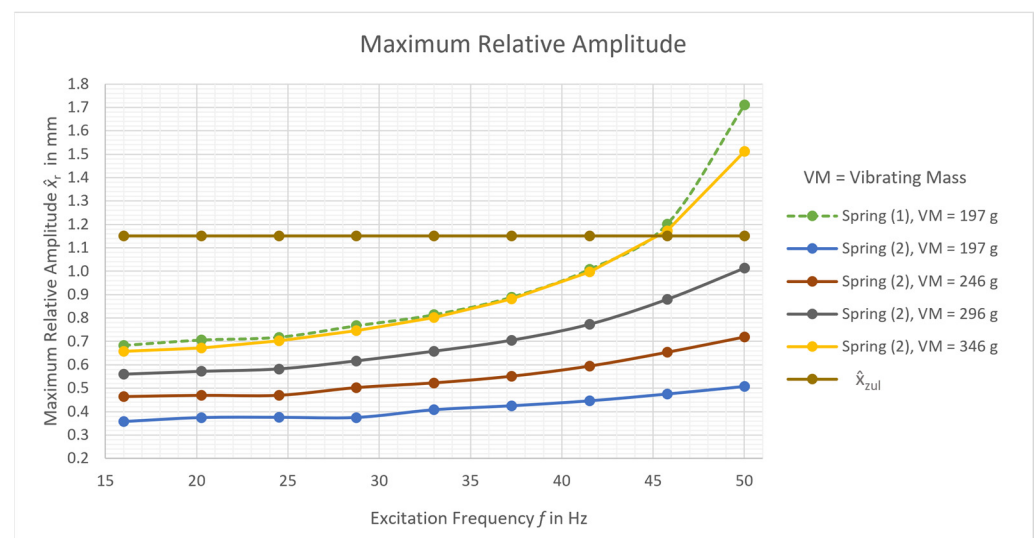


Figure 8. Maximum relative amplitude of different vibrating mass–spring combinations; $\hat{x}_{zul}$: maximum permissible amplitude.

The mass–spring combinations using spring 2 with masses of 197 g, 246 g and 296 g are suitable for the VEH. To achieve the maximum power output, spring 2 with a vibrating mass of 296 g is used for further investigation.

3.2. Coupled Magnetic Transient and Mechanical Simulation of Electrical Power

This section analyzes the electrical power output of the VEH via coupled magnetic transient and mechanical FEM simulations under a constant excitation of 1 g across frequencies from 16 Hz to 50 Hz. By varying the load resistance, the study identifies optimal conditions for power generation, providing insights into the VEH's performance and guiding design optimization.

3.2.1. Modeling

The simulation setup corresponds to that of the simulation of the maximum amplitude occurrence. In this case, the excitation corresponds to the minimum value of 1 g.

3.2.2. Procedure

The system is excited using $x_u(t) = \hat{x}_u \sin(\Omega t)$. The frequency varies between 16 Hz and 50 Hz. The amplitude $\hat{x}_u$ is adjusted at each frequency to correspond to an excitation of 1 g. The corresponding input parameters are shown in Table 3.

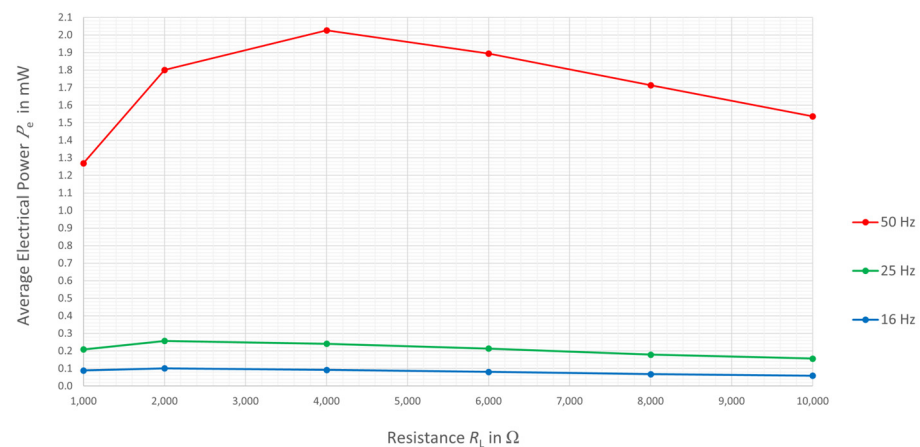
The load resistances R_{L1} and R_{L2} are varied between 1000 Ω , 2000 Ω , 4000 Ω , 6000 Ω , 8000 Ω , and 10,000 Ω to optimize power output.

Table 3. Input parameters for excitation at $9.81 \frac{\text{m}}{\text{s}^2}$ (1 g).

Frequency (Hz)	Amplitude (mm)
16	0.970
25	0.397
50	0.099

3.2.3. Result

Figure 9 shows the average electrical power $\bar{P}_e$, which is plotted as a function of load resistance for excitation frequencies of 16 Hz, 25 Hz, and 50 Hz. Linear interpolation is performed between the six data points.

**Figure 9.** Power output under varying electrical load and frequency.

Of the six discrete load resistance values, the maximum average power $\bar{P}_e$ at 16 Hz excitation is 0.101 mW for 2000 Ω . At 25 Hz, $\bar{P}_e$ reaches its maximum of 0.257 mW for 2000 Ω . At 50 Hz, the maximum is 2.026 mW for 4000 Ω .

The FEM simulations indicate that the mass–spring combination (spring 2, 296 g) is both suitable for the specified maximum amplitude and delivers at least 0.1 mW of power under minimum excitation.

4. Laboratory Tests

This chapter presents the experimental validation of the maximum relative amplitude of the VEH's movable core for selected mass–spring combinations previously identified through FEM simulations. Using a laboratory prototype that replicates the simulation model, vibration tests are conducted under controlled sinusoidal excitations. The setup includes measurement of both the base excitation and the core's acceleration using precision sensors, allowing determination of the relative displacement amplitudes. The experimental results are compared with simulation predictions to verify the accuracy of the modeling approach and to confirm the functionality of the investigated configurations.

4.1. Experimental Validation of Maximum Amplitude

The mass–spring combinations that exhibit a maximum amplitude below $x_r = 1.15$ mm, as determined in the previous FEM analysis, are investigated.

4.1.1. Setup

The coils are connected to load resistors with resistance values of $R_{L1} = R_{L2} = 1000 \, \Omega$, as in the simulation. The laboratory prototype shown in Figure 10 consists of the same components as the simulation model in Figure 3. The Core (IV) is equipped with additional weights, while the aluminum frame (VII) includes extra feet for mounting the prototype on the shaker.

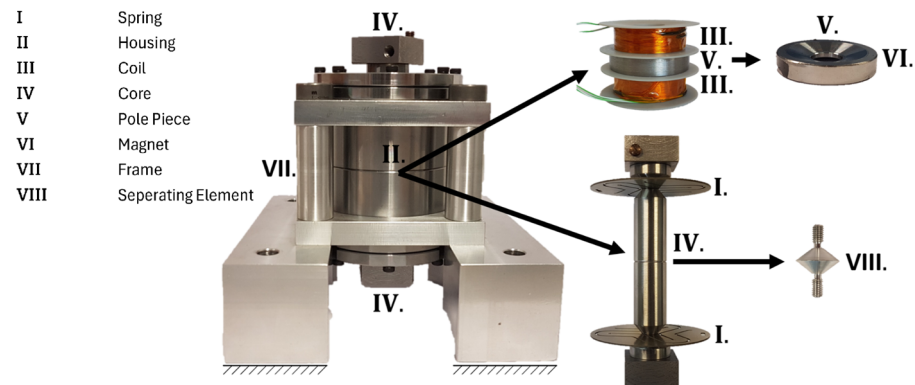


Figure 10. Laboratory model assembly with exploded view.

The weights are used to vary the mass m of the core. The springs (I) are used to vary the axial spring stiffness k_{fa} . The test setup is shown in Figure 11.

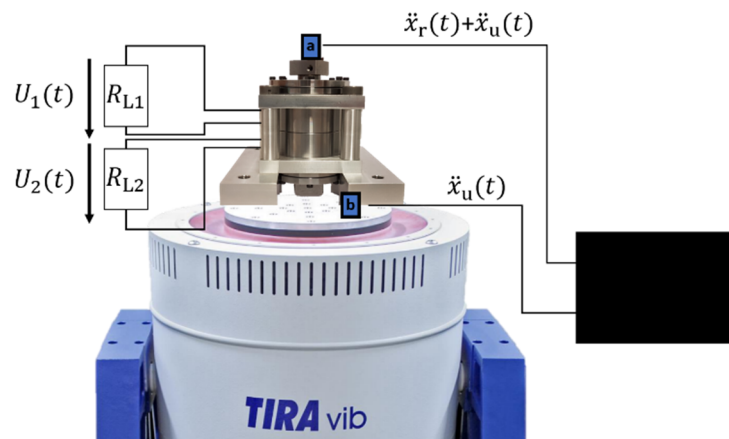


Figure 11. Experimental setup showing the laboratory prototype, vibration exciter, acceleration sensors a and b, load resistors R_{L1} and R_{L2} , and the data acquisition board connected to the sensors a and b.

The excitation x_u is generated by a vibration exciter S 51010/LS-340 (TIRA GmbH, Schalkau, Germany). The time response of the vibrations is recorded using two acceleration sensors 3055D6 (Dytran Instruments, Inc., Chatsworth, CA, USA) and a data acquisition board DT9837A (PCB Synotech GmbH, Hückelhoven, Germany). The acquisition was performed with a resolution of 24 bit, a sampling rate of 52.7 kHz, and a block size of 4096 samples. The sensors exhibit a frequency response within $\pm 10\%$ over the range from 1 Hz to 10,000 Hz. The data acquisition and signal processing are carried out using the software DasyLab (National Instruments, Austin, TX, USA). A high-pass filter with a cutoff frequency of 1 Hz is used. The cutoff frequency of the low-pass filter is set 20 Hz above the applied vibration frequency.

4.1.2. Procedure

Acceleration sensor b is attached to the mounting plate of the vibration exciter and measures the excitation acceleration $\ddot{x}_u$. Acceleration sensor a is mounted to the movable core of the laboratory prototype and measures the total acceleration $\ddot{x}_r + \ddot{x}_u$. Since the VEH provides a threaded mounting point, sensor a could be directly screwed onto the harvester. The vibration exciter does not offer a threaded interface, therefore sensor b was fixed to the mounting plate using an adhesive bond to ensure secure and repeatable attachment. The relative acceleration $\ddot{x}_r$ is obtained from the difference between the time signals of sensors a and b, from which the relative amplitude $\hat{x}_r$ is subsequently determined. The sinusoidal displacement excitations correspond to those listed in Table 2.

4.1.3. Result

The maximum relative amplitude $\hat{x}_r$ is determined under conditions in which the system had fully settled into its periodic response and all ambient parameters, including a constant temperature of 20 °C, remained unchanged during the measurement period, as a function of the excitation frequency, as shown in Figure 12. Between the nine measurement points for each mass–spring combination, linear interpolation is applied. The horizontal line indicates the maximum permissible amplitude $\hat{x}_{zul}$. Only mass–spring combinations below $\hat{x}_{zul}$ are suitable for the VEH.

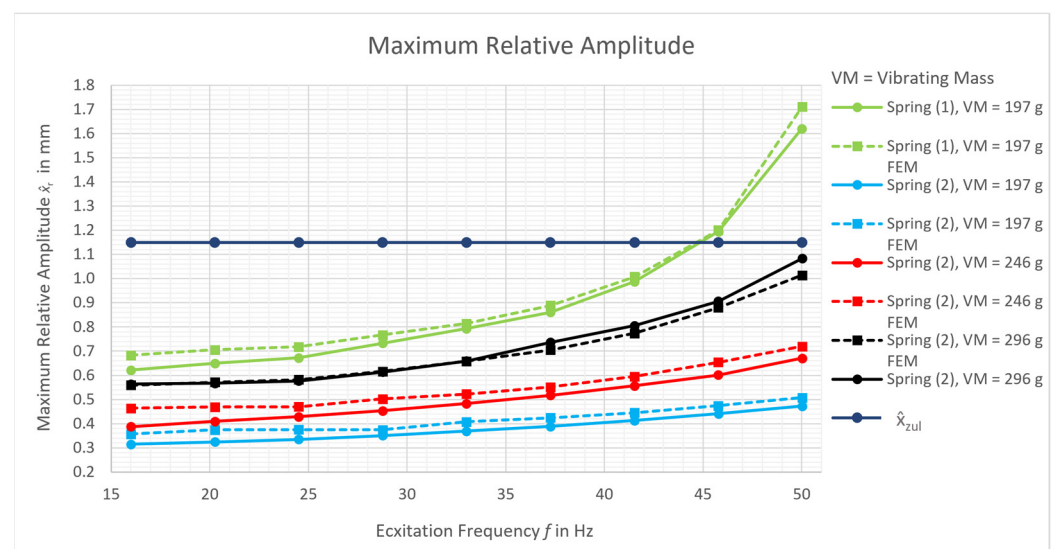


Figure 12. Comparison between experimental and FEM results of the maximum relative amplitude for different mass–spring combinations. Solid line: experiment; dashed line: FEM; $\hat{x}_{zul}$: maximum permissible amplitude.

The FEM results were confirmed. The mass–spring combinations of spring (2) with 197 g, 246 g, and 296 g are suitable for operation.

4.2. Experimental Validation for Power Delivery

In this section, the electrical power output $\bar{P}_e$ of the movable core under minimal excitation is investigated through laboratory experiments. The experiment serves to validate the FEM simulations of the maximum electrical power $\bar{P}_e$. The investigation is conducted using the same mass–spring combination as in the simulation. According to Equation (7), mass–spring combinations with a higher mass-to-axial-stiffness ratio provide greater electrical power output. Therefore, the combination consisting of Spring 2 and a mass of 296 g is examined.

4.2.1. Setup

The laboratory prototype is identical to the one used in Section 4.1. The experimental setup is shown in Figure 13.

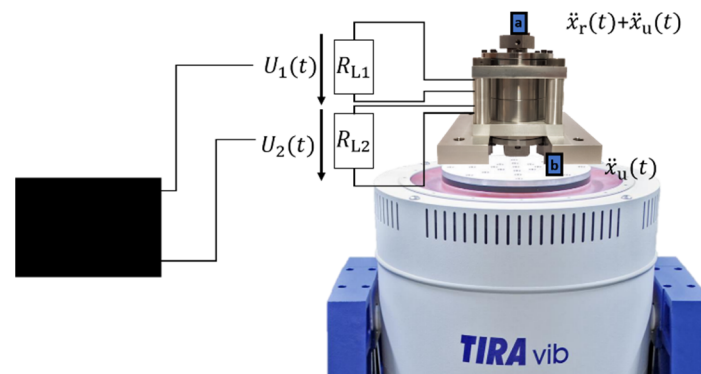


Figure 13. Experimental setup showing the laboratory prototype, vibration exciter, acceleration sensors (a and b), load resistors (R_{L1} and R_{L2}), and the data acquisition board connected to the sensors.

In contrast to the setup in Section 4.1, the voltages U_1 and U_2 across the load resistors R_{L1} and R_{L2} are measured instead of the accelerations $\ddot{x}_u$ and $\ddot{x}_r$. The time signals of U_1 and U_2 are recorded using the USB-6000 data acquisition device from National Instruments at a sampling rate of 10,000 samples per second.

4.2.2. Procedure

The mean electrical power $\bar{P}_e$ is calculated from the time signals of U_1 and U_2 and the corresponding load resistances. The sinusoidal displacement excitations and the resistance values R_{L1} and R_{L2} are identical to those used in Section 3.2.2.

4.2.3. Result

Figure 14 shows the mean electrical power $\bar{P}_e$ as a function of the load resistance for excitation frequencies of 16 Hz, 25 Hz, and 50 Hz. Linear interpolation is applied between the six measurement points for each excitation frequency.

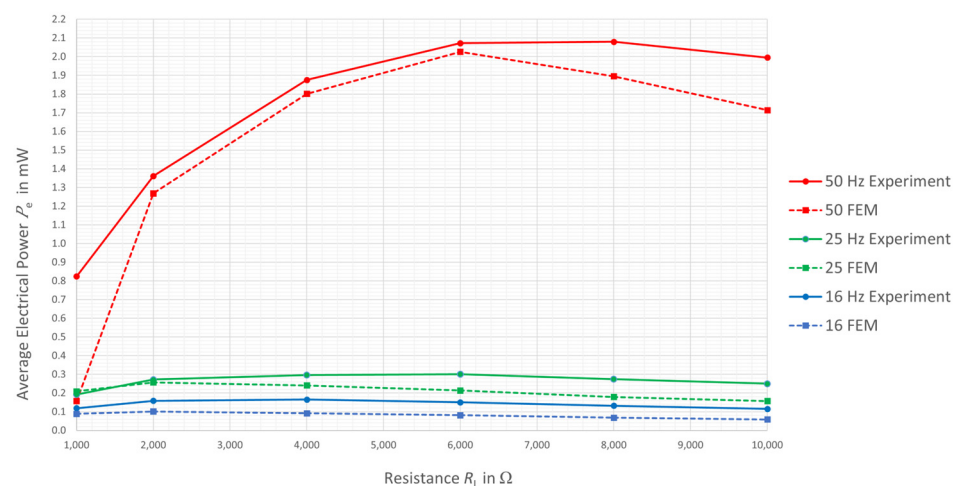


Figure 14. Comparison between experimental and FEM results of the power output of the vibrating mass–spring combination (296 g and Spring 2) as a function of load resistance and excitation frequency at $9.81 \frac{m}{s^2}$. Solid line: experiment; dashed line: FEM.

At an excitation of 16 Hz, the maximum mean power is $\bar{P}_e = 0.166$ mW at a load resistance of $4000\ \Omega$. At 25 Hz, the maximum value is $\bar{P}_e = 0.300$ mW at $6000\ \Omega$, and at 50 Hz, it reaches $\bar{P}_e = 2.080$ mW at $8000\ \Omega$.

Based on the analysis of functional behavior, an increasing load resistance reduces the current and, therefore, electromagnetic damping. It follows that the amplifying effect of the reduced damping, which increases the amplitude, is a significant factor up to the point of optimal output power. The results indicate that the mass–spring combination consisting of Spring 2 and 296 g is suitable for operation under maximum excitation and provides at least 0.1 mW of electrical power under minimal excitation.

5. Discussion

The novel VEH concept successfully met the design requirements regarding maximum amplitude and power output through targeted parameter adjustments. By optimizing the spring stiffness k_f , core mass m , air gap length L_s , and load resistance R_L , both the amplitude limitation and the power generation objectives were achieved. The laboratory experiments confirmed the FEM simulation results, showing consistent behavior between simulation and measurement. The minimum generated power of $\bar{P}_e = 0.166$ mW meets the current design requirements for operation under low excitation conditions. Further optimization will aim to increase this minimum power level. Reducing the air gap L_s increases the magnetic reluctance force and thus the achievable power output; however, a smaller air gap also requires a precise radial guidance of the moving core to avoid instability due to eccentricity. Considering the system's functional behavior, damping plays an increasingly important role at lower excitation levels (see Equations (12) and (14)). To enable reliable operation of the VEH even under very low excitation conditions, it is therefore essential to further investigate the damping characteristics in relation to the resulting power output and frequency response. Future work will focus on optimizing the magnetic flux path, implementing progressive spring characteristics, and evaluating advanced magnetic and structural materials, together with the influence of damping, in order to improve both the efficiency and long-term stability of the VEH system.

6. Conclusions

This study presented the design, simulation, and experimental validation of a VEH based on the principle of variable magnetic reluctance. The aim was to design an autonomous power source for wireless sensor systems in machine condition monitoring, capable of reliable operation under varying excitation conditions. Coupled magnetic–transient and mechanical FEM simulations were used to investigate the interaction between magnetic forces, mechanical displacement, and induced voltage. The simulation results were confirmed by laboratory measurements, showing close agreement in both displacement and electrical power output. The VEH provides a minimum electrical power of 0.166 mW at $9.81\ \frac{\text{m}}{\text{s}^2}$ and 16 Hz, while the maximum permissible amplitude of 1.15 mm is not exceeded even at $98.1\ \frac{\text{m}}{\text{s}^2}$ and 50 Hz. These results demonstrate that the design meets the defined requirements for energy supply and mechanical stability. This study also shows that coupled multiphysics simulation is a suitable tool for predicting the performance of electromagnetic energy harvesters. Further work will focus on increasing the energy output through optimization of the air gap. Therefore, the use of progressive spring elements and alternative magnetic materials, as well as a magnetic bearing, will be considered. In addition, integrating adaptive power management and wireless communication electronics will be key for future applications in fully autonomous sensor systems.

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