RELUCTANCE SPRINGS FOR NONLINEAR ENERGY HARVESTING GENERATORS

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Abstract: This work reports the design of a nonlinear magnetic spring using a reluctance circuit for vibrational energy harvesting applications. Introducing nonlinear spring elements into a harmonic oscillator can lead to a strong broadening of the frequency response while maintaining a high spectral power density. Here, the generator's seismic mass is part of a magnetic reluctance circuit which leads to a nonlinear spring restoring force. Different geometries of the magnetic spring are evaluated experimentally in terms of their static and dynamic characteristics. The results demonstrate the suitability of this concept at a sufficiently small mechanical damping.

Keywords: magnetic spring, reluctance circuit, Duffing oscillator, vibration harvesting, nonlinear oscillator

INTRODUCTION

Vibrational energy harvesting using nonlinear mechanical systems has gained growing interest recently [1-3]. Such systems comprise a mechanical oscillator using a spring with displacement-depending stiffness. Compared to linear systems these devices show an increased bandwidth at high displacements. Therefore, they provide several advantages over a linear system, especially environments with a dominant, but variable vibration frequency.

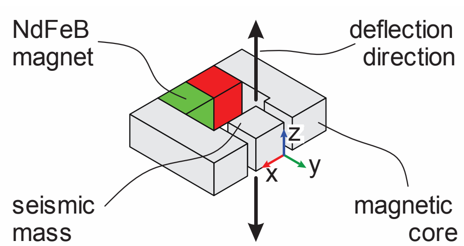

Fig. 1: Basic design of the springs with a cubic mass. The mass is pulled back into the gap upon a deflection along the z-axis. The picture shows the position of zero deflection.

However, concerning the design of such a nonlinear oscillatore, several challenges arise which have to be solved to guarantee a stable operation. In a certain frequency band close to the linear resonance frequency, a nonlinear resonant system has two stable oscillation states of which one delivers only very low power. Transitions between these two states lead to a hysteresis in the frequency response. A second problem arises from the fact that the displacement-dependency, hence the nonlinearity, of the spring stiffness depends on the excitation amplitude. Therefore, a high bandwidth is only found for large excitations. To make up for this effect, tunable springs can be used to shift the resonance frequency and the spring stiffness and to thereby guarantee a high energy oscillation. Proposed systems mostly use the direct interaction between magnets to realize the nonlinear restoring force. Actual literature verifies the advantages of a nonlinear spring but does also emphasize the unsolved problem of a compensation of the hysteresis [3]. Only few of today's approaches provide the possibility to tune the spring constant by altering the distance between the magnets. This often requires either high actuation forces or large travel ranges, hence large stroke actuators.

THEORY

The Duffing equation describes the behavior of a mechanical oscillator with nonlinear spring

$$m \cdot \ddot{x} + c \cdot \dot{x} + k_1 \cdot x + k_3 \cdot x^3 = F_{ext} \ , \qquad (1)$$

with the oscillating mass m, the damping constant c, the linear spring constant k_1 and the nonlinear spring constant k_3, respectively. To get a high bandwidth, the nonlinear spring constant has to be large compared to the linear spring constant. Also, the damping constant is crucial to realize a high bandwidth since a large damping can suppress the nonlinear effects [4].

We propose a nonlinear magnetic spring based on a reluctance circuit as shown in figure 1. The soft-magnetic core guides the magnetic flux of a permanent magnet through a gap. The system's seismic mass oscillating in the gap is made of soft-magnet material as well, thus changing the total reluctance of the circuit. Therefore, the total magnetic energy in the circuit changes depending on the displacement of the seismic mass leading to a force on it. The shape of the seismic mass determines how the total reluctance depends on the displacement and it can thus be used to tune the spring characteristics.

Although such reluctance circuits can be easily described by an electrical equivalent model, which gives the possibility to obtain an analytical solution of the force-displacement relation, this approach cannot be used for this special system. NdFeB magnets are used to generate a strong magnetic flux in the system. This leads to flux saturation in the seismic mass at very small penetration depth. Additionally, field fringing is not considered in analytical models. Both effects have a strong influence on the total reluctance of this circuit and, therefore, have to be considered by means of the material's B(H)-curve. For this reason, FEM simulation is used as a tool for a system model.

DESIGN

The final dimensions of the magnet, soft-magnetic core and seismic mass have been chosen based on simulations of the geometry shown in figure 1. The dimensions are shown in figure 2.

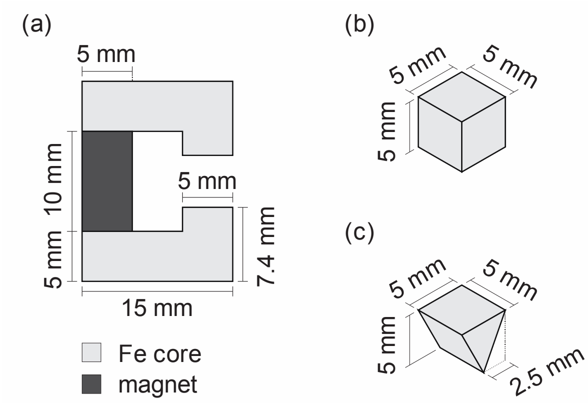

Fig. 2: Dimensions of the fabricated spring design. (a) shows the reluctance circuit with a homogeneous thickness of 5 mm, (b) the cubic mass, and (c) the wedge-shaped mass.

The magnetic cores and the seismic mass are made of high purity iron using a CNC milling machine, with dimensions of the fabricated parts shown in figure 2. Two different seismic mass designs are presented here, a cubic mass and a wedge-shaped mass. A NdFeB N35 permanent magnet with a remanent magnetic flux density of $B_R = 1.17$ T is glued to the cores. Additionally, the gap size can be increased to change the spring characteristics by inserting spacers between the cores and the magnet.

In addition, a multilayer core is fabricated to reduce eddy current losses and therefore reduce damping. A total of 14 layers of 300 µm thin soft-magnetic material (M150-30S, Waasner GmbH) is cut with a laser. The single parts are fixed with a 50 µm thin glue layer (ATG 976, 3M). The final bonding strength is provided by a thin epoxy resin coating of the layer stack (Stycast 2057, Emerson Cuming).

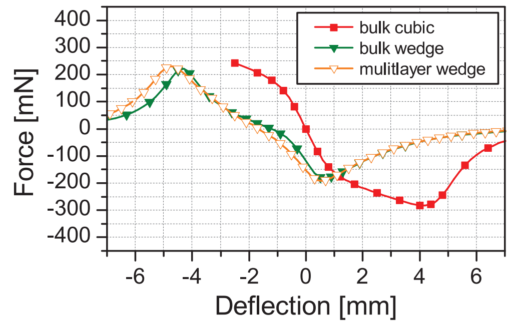

Fig. 3: Measured spring characteristics for different mass geometries and core types.

EXPERIMENT

The fabricated reluctance springs are characterized in terms of their static and dynamic performance. The static force-deflection properties are measured to determine the nonlinear spring constant and, thus, the optimal working range of the spring. Additionally, the damping ratio of the springs is measured as dynamic key figure of the springs. In a last section, the spring performance in a nonlinear oscillator is characterized by measuring the system's frequency response.

Static characterization

The force-deflection measurements are performed with a 5N force sensor (KD40s, ME Meßsysteme) which is mounted on a motorized z-stage. The seismic mass is attached to the sensor while the main part of the reluctance circuit is fixed on an x-y-stage. The zero position when the mass is completely inserted in the gap is taken as reference ($z = 0$ mm, see figure 1).

The results from these measurements are shown in figures 3 and 4. Figure 3 shows a comparison between the two evaluated seismic mass designs. Comparing the bulk and multilayer reluctance springs with a wedge-shaped mass shows that the multilayer version increases the total force because of the enhanced magnetic properties of the material. Although the spring characteristics of the two variants are similar in shape, the bulk core delivers a larger nonlinear spring constant. As a result, the wedge-shaped mass was identified as the optimal design for all other tests. It provides a strong nonlinear spring constant and a nearly symmetric force-deflection characteristic around a deflection of -0.5 mm.

Figure 4 shows the force-deflection characteristic for the wedge-shaped mass with different spacers between core and magnet. Linear and nonlinear spring constants for these measurements are given in table 1.

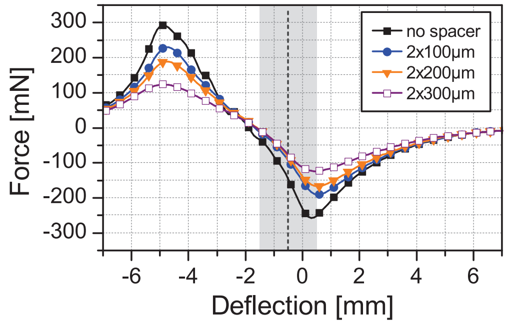

Fig. 4: Measured spring characteristics for different spacer geometries. The indicated range (grey) provides the strongest nonlinearity for the wedge-shaped mass

They have been determined in the indicated range by a least square fit of a third order polynomial. The ratio of linear to nonlinear spring constant changes only slightly so the spring characteristics remain nearly constant.

Table 1: Linear and nonlinear springs constants of the wedge-shaped bulk reluctance spring for different spacer configurations

	k_1 [mN/mm]	k_3 [mN/mm^3]	k_1/k_3 [1/mm^2]
no spacer	109.9	-46.5	-2.36
2×100 µm	107.6	-35.3	-3.05
2×200 µm	88.4	-28.3	-3.12
2×300 µm	52.2	-22.5	-2.32

Dynamic characterization

For the dynamic characterization, the seismic mass is attached to a laser-cut beam spring (1.1274 C-steel) as shown in figure 5. The seismic mass is fixed at the beam tip with a 2-component epoxy glue (UHU plus sofortfest, Henkel). Optionally, a piezo-ceramic sheet (PIC255, PI Ceramic GmbH) is glued to the beam covering approximately the first half of the length.

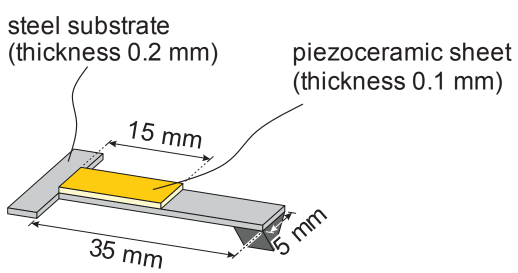

Fig. 5: Piezoelectric unimorph generator with attached wedge-shaped mass.

For the dynamic measurements, the beam is fixed on a shaker (TV51110, TIRA GmbH) together with the second part of the reluctance spring. Spring and core elements are aligned so that the stationary position of the seismic mass are aligned in such a way that the stationary position of the seismic mass is at a deflection of -0.5 mm. Since the magnetic force is not zero at this position, the beam is pre-stressed in its stationary position. The tip displacement and the displacement of the oscillator's base are measured with a laser vibrometer (CLV-2534, Polytec GmbH) and a laser distance sensor (AWL7/2, Welotec GmbH), respectively. The relative tip displacement is calculated as the difference between these two values.

For the measurements of the damping ratio, the system is excited close to the linear resonance frequency to reach an amplitude of the tip displacement of 800 µm. The decay of the oscillation is measured after switching off the excitation to determine the damping. By fitting an exponential decay curve to the measured values using the least squares method, the non-dimensional damping ratio D can be calculated from the decay constant δ and the angular resonance frequency ω_0:

$$A(t) = e^{-\delta \cdot t} \qquad \rightarrow \qquad D = \delta \cdot \omega_0 \ . \qquad (2)$$

The results of the damping measurements are shown in figure 6. In a first test, the damping ratio of a simple steel beam with attached tip mass but without reluctance spring was measured as a reference value. Next, the damping ratio with integrated reluctance circuit was measured.

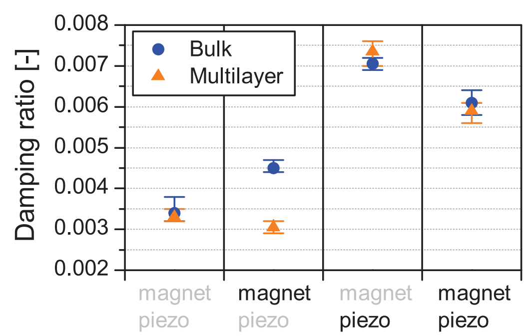

Fig. 6: Measured damping ratio for different combinations of linear and reluctace springs.

The reluctance circuit made of bulk material shows an approximately 30% larger damping ratio compared to the system with the multilayer spring which is most likely caused by higher eddy current losses in the bulk material. However, adding a piezoceramic sheet increases the damping by 100% due to the increased mechanical damping of the spring material. This effect is so strong that the reduced damping ratio of the multilayer cores does not have a significant impact on the overall damping ratio of a piezoelectric unimorph generator with reluctance springs.

The damping ratio of the piezoelectric unimorph generator is only weakly influenced by the reluctance spring. Additionally, the bulk springs show the stronger nonlinear spring constant. Therefore, frequency response measurements are performed using the piezoelectric unimorph generator in combination with a wedge-shaped seismic mass and the bulk core reluctance springs. To show the possible performance of the proposed reluctance springs in a system with sufficiently small mechanical damping, the same bulk core reluctance spring is characterized in a setup with a simple steel beam spring.

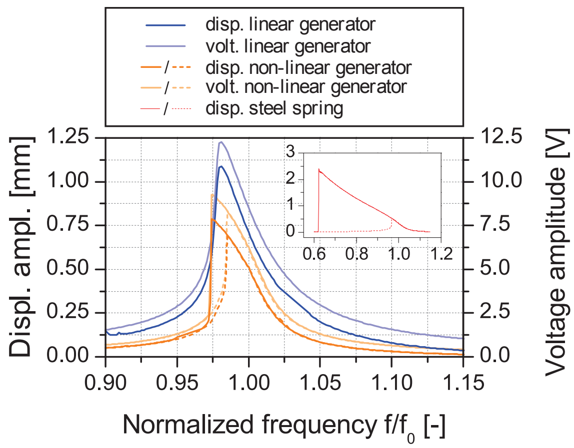

Fig. 7: Measured frequency response (at 5 m/s²) for a piezoelectric generator without reluctance spring and with bulk reluctance spring. Frequencies are normalized to the linear resonance frequency for better comparison. The inset shows the frequency response of a steel spring with reluctance spring.

The generator is mounted onto the shaker described before. The excitation acceleration is measured with an acceleration sensor (Typ8636C5, Kistler AG), also tip displacement and no-load voltage at the piezo-generator are determined. Frequency sweeps in increasing and decreasing direction are performed to determine a complete frequency response.

Results of these measurements are shown in figure 7. Because of the different resonance frequencies of the systems, the frequency axis is normalized to the linear resonance frequency for a better comparison. The single linear resonance frequencies are determined from the decay curves at a low displacement of 50 µm. The resonance frequencies as well as the bandwidths are given in table 2. The bandwidth of a nonlinear system is measured as the value at which the output voltage drops to half its maximal value. It can be seen that the nonlinear systems provide a strongly increased bandwidth compared to the linear system. However, the results with the steel spring show as well that a system with lower mechanical damping can lead to an even larger bandwidth.

Table 2: Linear resonance frequencies (f_{Res}) and absolute and relative 3dB-bandwidth (Δf and $\Delta f/f_{Res}$) of the characterized systems.

	f_{Res} [Hz]	Δf [Hz]	$\Delta f/f_{Res}$ [%]
Linear generator	68.3	1.45	2.12
Nonlinear generator	107.8	2.40	2.23
Nonlin. steel spring	100.3	10.75	10.72

CONCLUSION

We have successfully demonstrated the use of reluctance springs for vibrational energy harvesting purposes. The fabrication using CNC milling of bulk material combined with rapid-prototyping of thin metal layers delivers high-precision parts. As an improvement, water jet cutting of core and seismic mass could enhance the material's magnetic properties due to reduced mechanical impact. Different spring characteristics can be achieved by choosing suitable designs of the seismic mass. Eddy current damping can be reduced by fabrication of multilayer parts. Using small changes in the gap distance changes the stiffness of the springs but the overall characteristics remain the same. This could be used in principle to compensate the hysteresis effects in such a system.

Further research will focus on the evaluation of material properties and the enhancement of the spring design. Exact magnetic properties of the used materials would allow for a better representation of the real force-deflection ratio by simulations. This would make it possible to establish a reliable design process based on FEM simulations.

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