A NOVEL CONCEPT FOR SELF-ALIGNING GENERATORS

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Abstract: This paper presents a new concept for maximizing the efficiency of the power output of vibration driven inertial generators, if the excitation acceleration vector is not in line with the axis of the proof mass motion. The implementation of this concept by means of a self-alignment mechanism does not require any actuators or electronic circuits and thus works without any additional energy. The self-alignment mechanism is realized by pivot-mounting the generator at its center of gravity. For a generator of 8 cm³ with a proof mass 100 gram and a package mass of 100 gram it takes about 5 s to align to the excitation axis if the misalignment was 50°.

Keywords: energy harvesting, vibration generator, self-aligning mechanism,

INTRODUCTION

Inertial vibration energy harvesting devices typically include one or more mechanical resonators that amplify the external vibration around its fundamental resonance frequency in a single direction only [1, 2]. For optimal harvesting, the axis of proof mass motion has to be aligned with the external acceleration vector. A misalignment of the generator or a varying acceleration vector will reduce the energy conversion efficiency significantly. In order to overcome this limitation Bartsch et al proposed a 2 DOF resonator design incorporating two natural vibration modes that are orthogonal to each other [3]. A fundamental disadvantage of this principle is that the proof mass is able to vibrate only in one of the two directions at any one time. For instance, if the misalignment is 45° only one of the two modes will be active and the generator will be excited with an excitation amplitude reduced by a factor of cos(45°). Therefore, the device operates optimal only for misalignments around 0° and around 90°. Also, the implementation of the mechanical-to-electrical conversion mechanism will be rather complex, because there are two directions of motion. This will make it rather difficult to maximize the power output of the generator with respect to the generator volume.

In this paper we present an innovative concept that allows the generator to self-align with the excitation vector. As a result, the generator is always excited with the maximum acceleration amplitude. A major advantage of this concept is its simplicity in implementation and that the generator can be optimized in the conventional manner. Moreover, this self-alignment mechanism does not require any additional energy for successful operation.

CONCEPT

Fig. 1 shows a schematic view of the working principle of the self-alignment mechanism. The generator package is pivot-mounted in such a manner that the axis of proof mass motion y is orthogonal to the rotation axis z. When a misalignment ($\varphi - \alpha \neq 0$) between the axes of proof mass motion and excitation is induced, an angular momentum is generated, which causes the generator to rotate until the momentum vanishes ($\varphi - \alpha = 0$). The only requirement for optimal operation of the self-alignment mechanism is that the acceleration vector x is approximately orthogonal to the rotation axis z. This means that the mechanism described in this paper does not align when misalignments occur, which are out of plane.
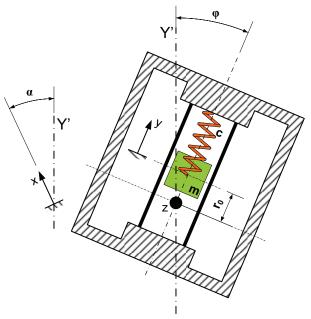

Fig. 1: Concept of the self-alignment mechanism.

The angles α and φ are the angle between the excitation axis x and the reference axis Y', and the angle between the axis of motion y of the proof mass and the reference axis Y'. Y' is a fixed reference axis. The black filled circle is the rotary axis z. The parameter r_0 describes the offset between the center of mass of the generator and the pivot point z.

MODELING

A numerical model was implemented in MATLAB / Simulink to evaluate the potential of the proposed self-aligning concept and to simulate the dynamic behavior of a self-aligning generator. The model is based on a set of two differential equations, which are coupled together by the dependent variables y and φ:

$$J\ddot{\varphi} = -c_r\dot{\varphi} + m_p \cdot \ddot{x} \cdot (r_0 + y) \cdot \sin(\varphi - \alpha)$$
$$+ m_p \cdot g \cdot (r_0 + y) \cdot \sin(\varphi), \tag{1}$$

$$m_p\ddot{y} = -\left(c_m + c_{eopt}\right)\dot{y} - ky + m_p \cdot \ddot{x} \cdot \cos(\varphi - \alpha)$$
$$+ m_p \cdot g \cdot \cos(\varphi), \tag{2}$$

where J, m_p, k, c_r, c_m, c_{eopt} and g are the moment of inertia, the mass of the proof mass, the spring constant, the mechanical damping coefficient of the rotational motion, the mechanical coefficient of the translational motion, the optimal electrical damping coefficient and the Earth's gravity. Equation 1 describes the rotational motion of the generator where a rotational moment is induced when: $\varphi - \alpha \neq 0$. The influence of the gravity is also considered generating a rotational moment if $\varphi \neq 0$. Equation 2 describes the translational motion of the proof mass. Only the y-component of the excitation force vector contributes to the excitation of the proof mass, i.e. the excitation force is a function of $\cos(\varphi - \alpha)$.

In order to keep the model simple we assume the generator casing to be a hollow cylinder with a specific wall thickness. The moment of inertia becomes:

$$J = \frac{1}{2}m_c \cdot (r_a + r_i), \tag{3}$$

where m_c, r_a and r_i are the mass of the generator casing, the outer radius of the casing and the inner radius of the casing, respectively.

The generated power was calculated by means of the general electrical damping coefficient:

$$P = \frac{1}{2}c_{eopt} \cdot \left(\frac{dy}{dt}\right)^2. \tag{4}$$

The electrical damping coefficient c_{eopt} was optimized in such a manner that the proof mass oscillates with the maximum possible displacement amplitude y_{max}. The optimal damping coefficient can be calculated as follows:

$$c_{eopt} = D_{eopt} \cdot 2 \cdot m_p \cdot \omega_n, \tag{5}$$

where D_{eopt} and ω_n are the optimal electrical damping ratio and the natural frequency of the translational system. The optimal electrical damping ratio can be calculated by Equation 6:

$$D_{eopt} = \frac{A}{2 \cdot y_{max} \cdot \omega_n^2} - D_m, \tag{6}$$

where A and D_m are the excitation amplitude and the mechanical damping ratio. The mechanical damping ratio is calculated from the mechanical damping coefficient as described in Equation 7:

$$D_m = \frac{c_m}{2 \cdot m_p \cdot \omega_n}. \tag{7}$$

In Fig. 2 a schematic diagram of the signal-flow model is presented showing the input variables and the coupling of the two mechanical systems (translational and rotational) by the dependent variables y and φ. The excitation force F is described as $m_p \cdot \ddot{x}$, with $\ddot{x} = A \cdot \sin(2\pi f \cdot t)$.
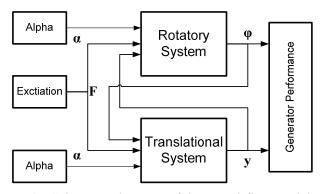

Fig. 2: Schematic diagram of the signal-flow model.

Table 1: Simulation parameters.

Parameter	Value	Unit
Excitation frequency f	100	Hz
Max. displacement amplitude y_{max}	2	mm
Mechanical damping coeff. c_m	0.1	1
Mechanical damping coeff. c_r	0.0001	1
Offset r_0	0	mm
Mass m_p & m_c	100	gram
Excitation amplitude A	2	g

The simulation parameters used in this study are summarized in Table 1. For most of the parameters (e.g. excitation frequency, displacement amplitude, etc.) a fixed value is used throughout this study. Only the mass m_p and m_c were varied with respect to a scaling factor.

DISCUSSION

For a generator typically mounted to the vibration source (generator casing is fixed and therefore not able to rotate), the power output decreases with the misalignment as shown in Fig. 3. For misalignments of $\alpha > 10°$ between the axis of proof mass motion y and the axis of excitation x the generated power (normalized) decreases significantly and becomes zero for $\alpha = 90°$. However, if the generator is pivot-mounted as described above, the output power decreases only temporarily. Fig. 4 shows simulation results of the step response of the angle φ for different step values of α. The angle alpha is stepped at time = 4s from 0° to a specific value as shown in the diagram. The angle φ converges towards the angle α until the deviation between the angles vanishes ($\varphi - \alpha = 0$). As a result the generator (y-axis) aligns with the axis of excitation x within a specific time period.

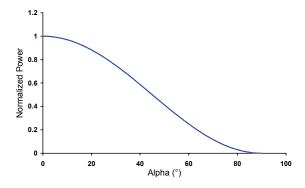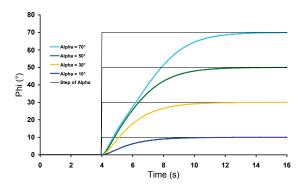*Fig. 3: Power output as a function of α for a fixed generator casing (not able to rotate); the mass mp was 100 gram.*

Fig. 4: Step response of the angle φ; the angle α was stepped at time = 4 s from 0° to a specific value; the initial value of φ at time = 0 s is 0°

The corresponding normalized power output is shown in Fig. 5 (the angle α is stepped according to Fig. 4). Since the generator is pivot-mounted the axis of proof mass motion y aligns with the axis of excitation x and the power regains its maximum value It clearly shows that the output power regains its maximum value after successful self-alignment of the generator device.

The time for the generator to align to the excitation axis increases for larger misalignment angles. For a misalignment of $\alpha = 70°$ the alignment process takes about 8 s.

In order to evaluate this concept for small scale devices (mm-range) we investigated the step response for different masses based on the scaling factors 1, 0.8, 0.65 and 0.5 respectively. The results are shown in Fig. 6. When scaling the device by a factor of 0.5 the mass scales by a factor of approximately 0.1, which means that the masses of the proof mass and of the casing decrease to 10 gram. In this case it takes about 60 s for the generator to align to 98 % of the target angle. For masses of 50 gram and 30 gram it takes 10 s and 16 s, respectively, for 98 % alignment.
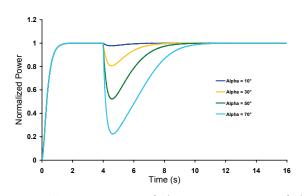

Fig. 5: Step response of the output power of the generator; the angle α was stepped in the same manner as shown in Fig. 4.
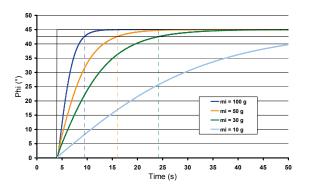
Fig. 6: Step response of the angle φ for different masses; the angle α was stepped at time = 4 s from 0° to 45°; in this study the masses m_p and m_c were varied at the same time in accordance to a specific scaling factor; the displacement amplitude was kept at 2 mm.

We also varied the maximum displacement amplitude, which decreases for small scale devices (assuming constant excitation amplitude). In Fig. 7 the results are shown for three different displacement amplitudes. It is noticeable that the time for alignment is significantly dependent on the displacement amplitude. For a displacement amplitude of 0.5 mm it takes approximately 90 s to align to 98 % of the target angle. When performing a simulation with a mass of 10 gram for m_p and m_c and with a displacement amplitude y_{max} of 0.5 mm it takes ca. 1 hour for the generator to align to the axis of excitation ($α = 45°$). As a consequence, the concept of self-alignment by means of a pivot-mounted generator casing works efficiently (i.e. in a reasonable short time period) for larger masses and displacement amplitudes. If we consider micro-generators, in particular chip-based devices, the concept proposed in this paper seems not practical for implementation, especially if the misalignment α is time-variant.
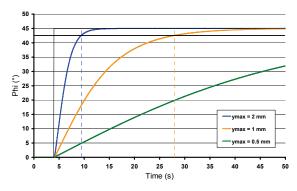

Fig. 7: Step response of the angle φ for different displacement amplitudes; the angle α was stepped at time = 4 s from 0° to 45°; the masses m_p and m_c were kept at 100 gram.

CONCLUSION

A novel concept was demonstrated, which enables to increase the efficiency of a linear 1 DOF vibration generator if the excitation acceleration vector is not in line with the axis of the proof mass motion. This self-alignment mechanism works without additional energy and is therefore suitable for energy harvesting applications. We found that this concept is applicable for vibration generators with masses larger than 30 gram and displacement amplitudes larger than 1 mm. For a generator of 8 cm³ with a proof mass 100 gram and a package mass of 100 gram it takes about 5 s to align to the excitation axis if the initial misalignment between the axis of excitation and the axis of proof mass motion was 50°. The alignment mechanism is based on an angular momentum that is generated when a misalignment occurs. The value of the angular momentum is primarily proportionally dependent on the mass of the proof mass and the displacement amplitude. Therefore, the proposed concept seems not practical for generators with a device size at bottom of the mm-scale.

A major advantage of this concept is its simplicity of implementation and its efficiency. Future work will involve a prototype and experimental validation of the theoretical data presented in this paper.

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