A NEW, HIGH EFFICIENCY, BIDIRECTIONAL, ELECTROMAGNETIC VIBRATION ENERGY HARVESTER FOR AERONAUTICAL APPLICATIONS

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Abstract: The present paper describes the design and realization of a high efficiency, electromagnetic vibration energy harvester with some new and extraordinary properties. In particular this means, that the harvester in discussion can be constructed bidirectional, has a very low mechanical damping, can serve a wide range of frequencies, offers the possibility for auto-tuning of the resonance frequency, possesses a magnetic spring with a long operational life span and is very compact. The principle design of the resonantly working harvester is shown and the design parameters are discussed. Two different prototypes for an aeronautical application were realized and characterized. The measured data are interpreted on basis of a classical spring mass system.

Keywords: vibration energy harvester, electromagnetic, bidirectional, magnetic spring

INTRODUCTION

Up to now a lot of devices were presented and are even available on the market, which convert useless vibration energy into valuable electrical energy to run e. g. an autarkic sensor system [1, 2, 3]. Nevertheless to compete with batteries, such so called vibration energy harvesters have to fulfill some basic demands. So their life time and power output per volume or mass, respectively, should exceed those of batteries. This is especially true for aeronautical applications, where the demands on reliability and weight reduction are even stricter. The presented new design has the potential to overcome some present limits of existing vibration harvesting devices and is therefore especially appropriate for aeronautical applications.

BASIC EQUATIONS

Our design is based on a rotational oscillator with a magnetic, i.e. progressive spring. Nevertheless as the rotation angle during operation of the harvester is very small, equation (1), describing a linear oscillator, is supposed to be a good approximation. Also the spring stiffness shows only a slight nonlinearity and as the input displacement is also small, we expect no jump-up and jump-down behavior in the frequency response curves [4].

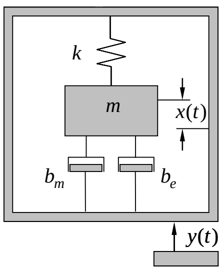

Fig. 1: Schematic sketch of a spring mass system described by equation 1.

The well known differential equation of a forced oscillation of a spring mass system shown in Fig. 1 is:

$$m\ddot{x}(t) + b\dot{x}(t) + kx(t) = -m\ddot{y}(t) \tag{1}$$

Herein m is the oscillating seismic mass, k the spring stiffness, $y(t)$ the displacement of the vibrating base and $x(t)$ the displacement of the seismic mass relative to the housing. $b=b_m+b_e$ is the damping coefficient, b_m the mechanical and b_e the electrical one.

Solving this equation analytically leads to a quite significant relation for the maximum achievable electrical power P_e at the resonance [5], which is:

$$P_e = \frac{m\varsigma_e A^2}{4\omega(\varsigma_m + \varsigma_e)^2} \tag{2}$$

Here A is amplitude of the vibration acceleration, ω the radian frequency of the vibration and ζ_m the mechanical damping ratio. ζ_e is the electrical damping ratio, caused both by the resistance of the coil and the resistance of the load.

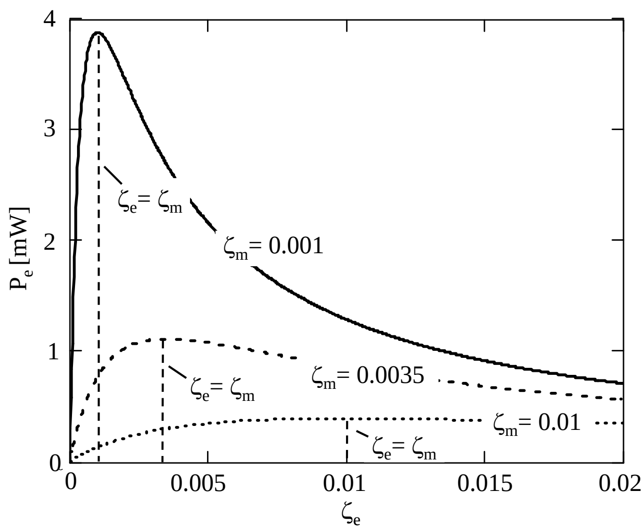

Fig. 2: Electrical output power of a resonantly working energy harvester calculated by equation (2).

From equation (2) two decisive points can be derived. Firstly the maximum output power is directly proportional to mass and secondly, it is the higher the smaller the mechanical damping is. For a given mechanical damping it is highest, when ζ_e is equal to ζ_m, as illustrated in Fig. 2.

For a given electromagnetic induction unit, consisting out of a coil and permanent magnets delivering a static magnetic field, ζ_e can be calculated as follows [6]:

$$\varsigma_e = \frac{(NlB)^2}{2m\omega(R_{load} + R_{coil})} \tag{3}$$

Here N is the number of turns of the coil and l the coil diameter across the magnetic flux, where we use double the medium coil diameter, when a so called four magnet arrangement for creating the magnetic field is used [7]. B is the magnetic flux density of one magnet pair the coil moves in. R_{load} is the resistance of the load and R_{coil} the resistance of the coil. The inductivity of the coil is neglected. With equation (3) the electrical power P_e in equation (2) becomes a function of R_{load} and the electrical power at the load resistor $P_{e,load}$ can be given as follows:

$$P_{e,load}(R_{load}) = P_e(R_{load})\left(\frac{R_{load}}{R_{load} + R_{coil}}\right) \tag{4}$$

Referring to N.G. Stephen [8], the first derivative of equation (4) delivers a value for that R_{load}, where the output power becomes maximal. That is

$$R_{load} = R_{coil} + \frac{(NlB)^2}{b_m} \tag{5}$$

Inserting equation (5) into equation (4) one gets the maximum output power at the load, which is

$$P_{e,load,\max} = \frac{mA^2}{16\varsigma_m\omega_0}\left(1 - \frac{R_{coil}}{R_{load}}\right) \tag{6}$$

$\omega_0 = 2\pi f_0$ is the resonance radian frequency, f_0 the resonance frequency of the spring mass system.

For dimensioning the electromagnetic induction unit, we have to consider, that the coil resistance R_{coil} depends, for a given geometry and wire material, only on the number of turns N and the coil wire radius r.

$$R_{coil} = \frac{\rho l_c N}{r^2\pi} \tag{7}$$

ρ is the specific electrical resistance of the coil wire and $l_c = \pi l$ the wire length of one turn.

Inserting equation (5) and equation (7) into equation (6) one gets the maximum output power in dependence on the number of coil turns N and the coil wire radius r. This calculation was exemplarily done with $m=91.5g$, $f_0=39.5Hz$, $A=0.04g$, $\zeta_m=0.0035$, $l=2\cdot 9.08mm$, $B=0.107T$ and $\rho=0.0178\Omega mm^2/m$. The result is summarized in Fig. 3, showing the maximum output power $P_{e,load,max}$ at the load in dependence on the number of coil turns N and the coil wire radius r.

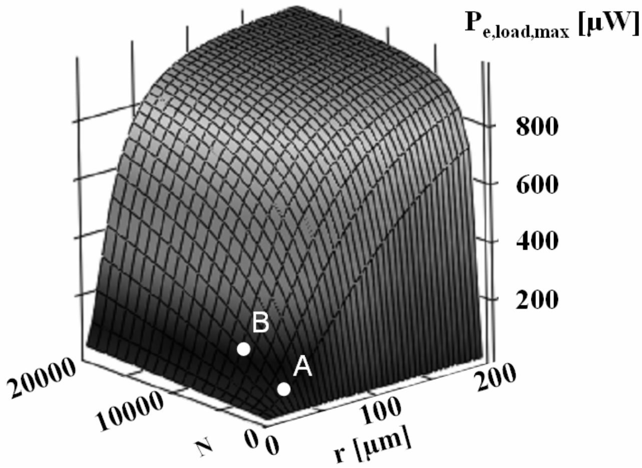

Fig. 3: Calculated maximum electrical output power in dependence on coil turns and coil wire thickness.

Furthermore the generated power in dependence on the frequency is given in [5] and [9] as follows:

$$P_e(\omega) = \frac{m\varsigma_e \dfrac{\omega^2}{\omega_0^3} A^2}{\left(1 - \left(\dfrac{\omega}{\omega_0}\right)^2\right)^2 + \left(2(\varsigma_m + \varsigma_e)\dfrac{\omega}{\omega_0}\right)^2} \tag{8}$$

GENERATOR DESIGN

Design principle

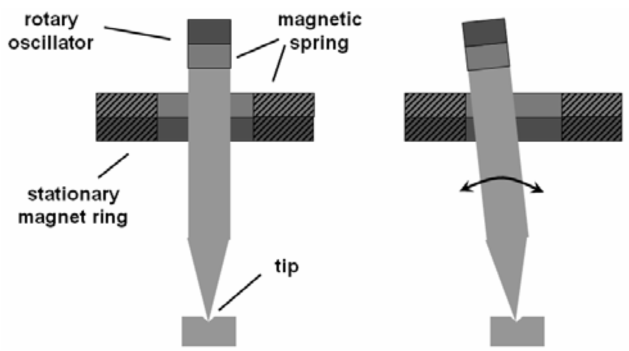

Fig. 4: Illustration of the new design principle of the harvester presented in this paper.

The principle of the new design of our harvester is shown in Fig. 4. It is based on a magnetic spring build up by a stationary ring magnet and cylindrical magnet on top of a rotary oscillator. The pivot bearing of the oscillator is simply realized by a sharp steel tip placed on top of a steel screw with a small deepening. The shape of the magnetic spring and of the pivot bearing enables oscillations in two directions. In combination with an appropriate induction unit, the harvester becomes bidirectional. As the oscillator is stabilized in its position only by magnetic forces and the only contact to the housing is the steel tip, it is possible to tune the resonance frequency by adjusting the tip position relatively to the stationary ring magnet.

Design parameter

Several geometrical parameters influence the behavior of the described energy harvester, as there are inner and outer diameter and height of the ring magnet, diameter and height of the cylindrical magnet, strength of the magnets, length of the rotary oscillator, distance of the ring magnet and the cylindrical magnet etc. We have chosen an experimental approach to get the correlation of the different quantities on basis of commercially available magnets. Mainly we regarded the forces in vertical and horizontal direction, F_x and F_y. F_x delivers an approximate linear value for the spring stiffness. Together with the moment of inertia we get the resonance frequency of the oscillator. F_y decisively influences the mechanical damping of the tip bearing. Generally spoken, the mechanical damping is the smaller, the smaller F_y is. By carefully adjusting F_y, mechanical damping ratios ζ_m down to 0.001 could be achieved.

As already mentioned, the presented energy harvester principally can use vibration in two spatial directions. Of course this calls for an induction unit working in two directions, too. Such an induction unit is principally possible. As we still have a patent application pending on this topic, the prototypes reported here are all still realized with a conventional one directional induction unit. It uses the already mentioned four magnet arrangement. Although our rotary oscillator is well approximated by a linear spring-mass-system, the placement of the induction unit on the rotary oscillator offers a new degree of freedom. It makes a difference, whether the induction is placed far away from the centre of rotation or nearer to it. In the first case the average oscillation speed of the induction unit is bigger than in the second and so is the electrical damping. Therefore by placing the induction unit as far as possible away from the center of rotation, the maximum power output approaches more and more the plateau of Fig. 3.

PROTOTYPES

Application scenario

The described energy harvester shall drive a monitoring system for controlling a strut of the high lift system of an Airbus airplane. For running the harvester a vibration at 39.5Hz with up to 0.1g acceleration amplitude can be used. The energy demand of the monitoring system was estimated to be about 0.511mW inclusive power management losses. A continuously measurement of the strain at the strut is supposed. The communication works in the so called polling modus, where every second is checked whether somebody asks for data from the system. Sending the collected information to a maintenance engineer is supposed to be necessary every week.

Construction details

The main features of the present prototype are shown in the photograph of Fig. 5. It has a length of about *115mm* and an outer diameter of about *25mm*. As the rotary oscillator has a weight of about *91.5g* one needs three magnetic springs, built up by NdFeB magnets, to hold this mass even in a horizontal position of the harvester. The magnitude of the mass, made by tungsten, was chosen to meet the power requirements of the application. The tip bearing was realized by a steel tip and a steel screw with a small deepening. The screw can be used to adjust the resonance frequency by changing the relative distance of the ring magnets and the magnet on the oscillator and therefore by changing the spring stiffness. Again not to overstrain the springs the induction unit is placed in the middle of the rotary oscillator, although this is not the optimum position.

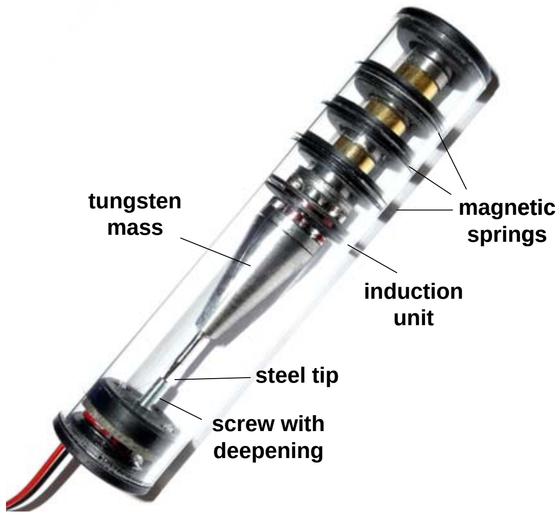

Fig. 5: Photograph of a prototype of the harvester with proposed design. The main features are indicated.

Two different prototypes were realized mainly differing in the layout of the induction unit. Both had the named four magnet arrangement. In typ1 it is realized by eight circular NdFeB magnets, not filling out the whole available area, whereas in typ2 by four semicircular NdFeB magnets fitting perfectly in the circular cross-section of the harvester. The coil of typ1 had 1280 turns of a 50µm wire, that of typ2 about 4760 turns of a 40µm wire. The points A and B in Fig. 3 indicate the expected output power, respectively, with the parameters given there.

MEASUREMENTS AND THEORY FIT

Fig. 6 shows the output power $P_{e,load}$ of the typ1 generator plotted against the load resistor R_{load}. The theoretical curve was determined by equation (4) with $m=91.5g$, $f_0=39.5Hz$, $A=0.04g$, $\zeta_m=0.0035$, $l=2\cdot 9.08mm$, $B=0.107T$ and $\rho=0.0178\Omega mm^2/m$. R_{coil}

was about 331Ω. Fig. 7 shows $P_{e,load}$ in dependence on the frequency. The fit is based on equation (8).

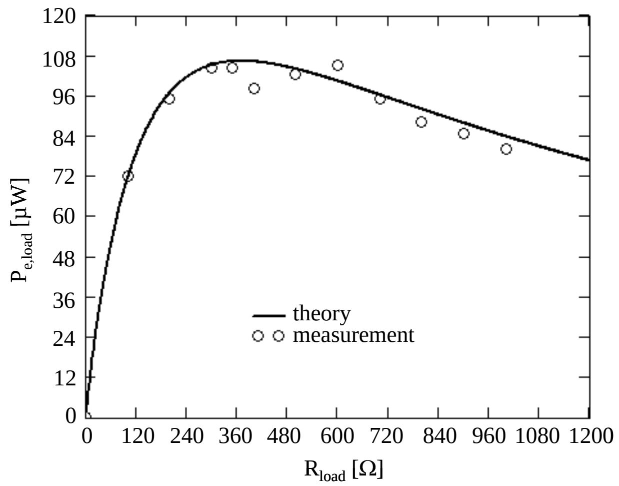

Fig. 6: Electrical output power plotted against the load resistor for the harvester typ1.

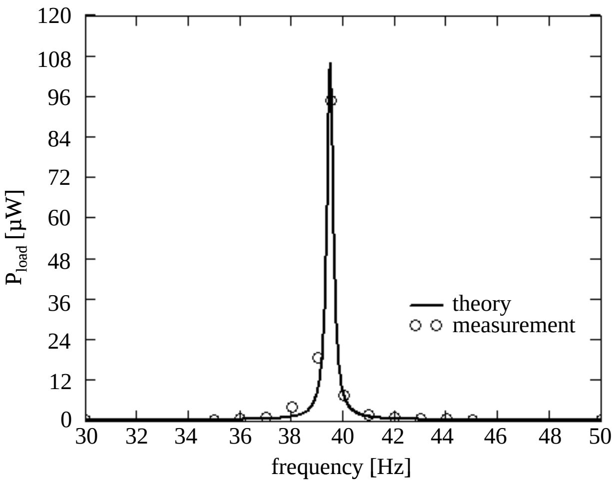

Fig. 7: Electrical output power plotted against the frequency for the harvester typ1.

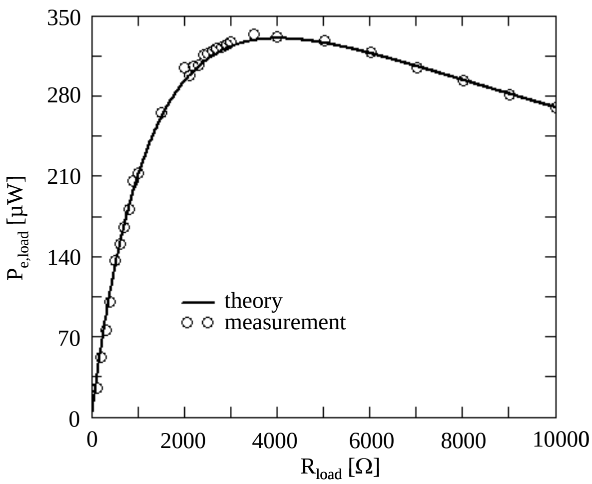

Fig. 8: Electrical output power plotted against the load resistor for the harvester typ2.

Fig. 8 shows the output power of the typ2 generator plotted against the load resistor R_{load}. The theoretical curve was determined by equation (4) with $m=91.5g$, $f_0=39.5Hz$, $A=0.0435g$, $\zeta_m=0.005$, $l=2\cdot 11.5mm$, $B=0.1725T$ and $\rho=0.0178\Omega mm^2/m$. R_{coil} was about 2436Ω.

CONCLUSION

The measurement results can well be fitted by the theoretical curves of the simple linear model. A relatively low mechanical damping could be reached, helping to achieve high output power at relatively low masses. The use of a magnetic spring let expect a long life time. The generator principally offers the possibility of a bidirectional use. Furthermore auto-tuning seems to be possible by moving the receiving point of the rotary oscillator by a small energy saving linear drive.

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