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EXPERIMENTAL STUDY ON A LORENTZ FORCE ACTUATOR USED AS TUNEABLE PASSIVE VISCOUS DAMPER

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Abstract: *This paper presents an experimental study concerning the possibility to generate tuneable synthetic damping in a mechanical structure, using an electro-dynamic voice-coil system (EVCS) with dual functionality: sensor and actuator. If the mechanical structure vibrates, the EVCS generates a voltage; it works as a velocity sensor. If this voltage is applied on a passive shunt (electrical resistance), a current is generate inside the EVCS. This current produces a viscous (Lorentz) force inside EVCS and in the mechanical structure as well. A negative velocity-force passive feedback loop is generated. We must mention that the damping ratio depends by the resistance value*
Key words: *vibration, passive, synthetic damper, tuning*

1. INTRODUCTION

To begin with, the way in which the danger of resonation amplification can be reduced is well known in the dynamics of mechanical systems. It is useful to add natural or synthetic damping. The synthetic damping is generated with active or passive mechatronic devices. Most of these devices are based on active velocity-force negative feedback loop (Preumont, 2002) with collocated velocity sensor and force generator (actuator). A less expensive solution is to use passive damping devices (de Marneffe, 2007; Huang & Tian, 2009) with dual behaviour, as sensor and actuator (e. g. an electro-dynamic voice-coil system -EVCS- or a piezoelectric device).

We intend to perform an experimental study on a tunable synthetic damper based on an EVCS with an electric resistance placed on the coil terminals as passive shunt. The damping coefficient is tuned by changing the resistance value. The synthetic damper is placed on a single degree of freedom vibratory system (cantilever beam). The work intends to find out the efficiency of this synthetic damper using the computer assisted analysis of the free bending vibrations of the cantilever beam. We claim that the synthetic damper works as a vibration suppression system with negative passive modal power supply. The damping ratio decreases if the modal mass increases. That is why in the future we plan to develop a higher efficiency vibration suppression system based on EVCS supplied with a negative active modal power supply.

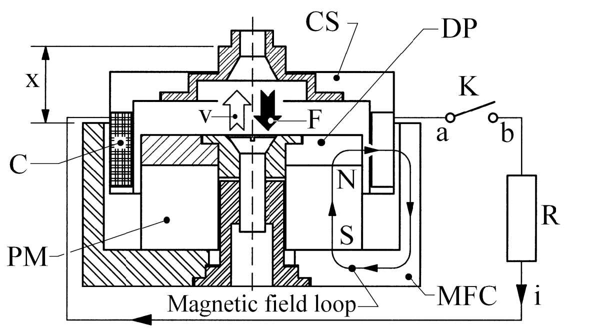

Fig. 1. The negative velocity-force passive feedback loop principle with EVCS used for synthetic damping.

2. SYNTHETIC DAMPER. THE PASSIVE NEGATIVE VELOCITY-FORCE FEEDBACK BASED ON EVCS AND RESISTANCE.

Figure 1 presents the synthetic damper. A coil C is wound on a cylindrical support CS, (made from non-electrical conductive material). It is placed inside the gap of a magnetic field system (MFS) built with a rare-earth magnet (NdFeB), a magnetic field concentrator (MFC), and a deflection plate (DP), both made from mild steel. If the switch K is switched-on, a series electrical closed circuit with the coil C and the electric resistance ***R*** inside occur. If a relative motion ***x*** between CS and MFS is generated, then a current ***i*** appears in the circuit and a relative Lorentz (viscous) negative force ***F*** is generated in the damper. The force ***F*** is roughly given by:

$$F = -\frac{T^2}{(R+R_c)} \cdot \frac{dx}{dt} = -g \cdot \frac{dx}{dt} = -g \cdot v \qquad (1)$$

In Eq. (1) ***T=B·l*** is the EVCS constant (1.4 Newton per Ampere), ***B*** is the average magnetic flux density, ***l*** is the length of coil wire, R_c is the coil resistance (7.1 Ω), ***v =dx/dt*** is the relative velocity. The structure described in Figure 1 works as a synthetic damper with passive negative velocity-force feedback. EVCS works as sensor and actuator. The force ***F*** (and the gain ***g*** as well) are tuned by adjusting the resistance ***R***.

3. THE EXPERIMENTAL SET-UP

An easy way to find out the synthetic passive damper (SPD) efficiency is to use the set-up described in Figure 2.

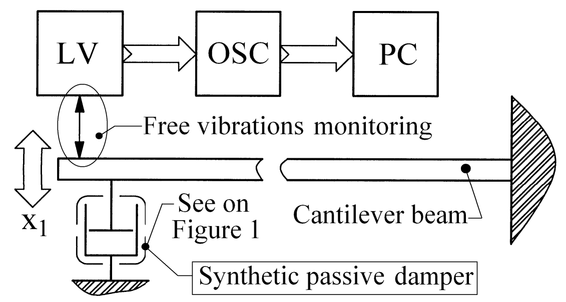

Fig. 2. Experimental set-up used to test the SPD.

The SPD is placed to the free end of a cantilever beam (CB). The structure works as a spring-mass-damper system. The free vibration x_1 of the beam (first bending mode) is described (Thomas & Laville, 2007) as follows:

$$x_1(t) = a \cdot e^{-\xi \cdot \omega_n \cdot t} \sin(\omega_d \cdot t + \varphi) \qquad (2)$$

The viscous force $\boldsymbol{F}$ given in Eq. (1) has the highest influence on the damping ratio $\xi=(c+g)/(2\cdot m\cdot\omega_n)$, with c - the natural damping coefficient, m - the equivalent mass, ω_n - the natural pulsation (close by ω_d, the pulsation of damped vibration). The experimental data fitting of the CB free vibration can be used to evaluate the $\xi = \xi(R)$ dependence. A laser vibrometer LV (Polytec OFV-5000) converts the motion in electrical signal. An oscilloscope OSC (ADC 212-50) converts this signal in numerical format towards a personal computer PC.

4. SOME EXPERIMENTAL RESULTS

Figure 3 presents a part of the experimental and fitted x_1 free response signal, due to the step mechanical excitation of

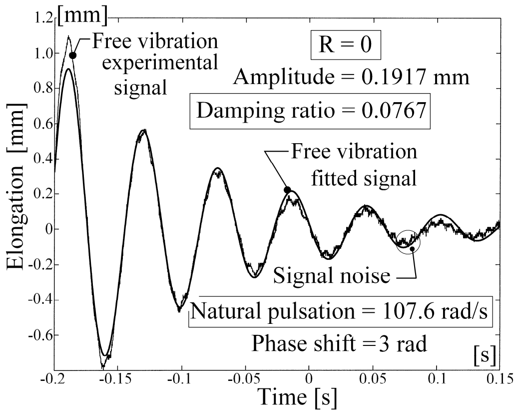

Fig. 3.Experimental and fitted free response signal (***R=0***)

the CB. A shunt resistance ***R=0*** on SPD is used (for maximal synthetic damping). The switch K is switched-on. A computer program was designed for signal fitting (find out a, ξ, ω_n and φ values, if we suppose $\omega_n=\omega_d$), see the results on Figure 3.

If the switch K is switched-off then $R=\infty$ and the synthetic damping is almost nought. The experimental free response and fitting data results are described on Figure 4. There still exists a small viscous force in the system, generated by the eddy-currents in the SPD coil-wire material (Sodano et al., 2006).

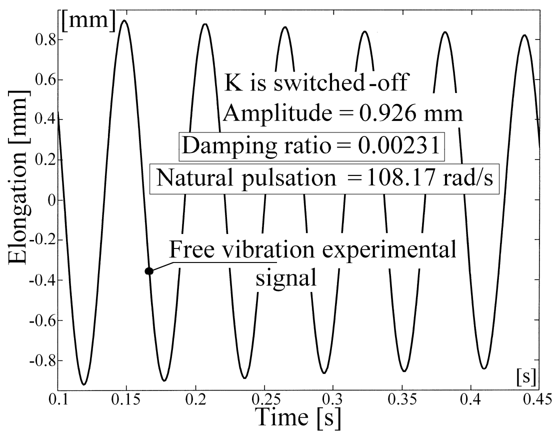

Fig. 4. Free response signal ($R=\infty$) and fitting data results

Here the damping ratio is smaller as before ($\xi = 0.00231$, so 33 times less). The natural pulsation is slightly bigger (ω_n = 108.17 rad/s), according to the theory (Thomas & Laville, 2007). A better way to describe the SPD efficiency is to use the fitting results (ξ and ω_n, for different R values) to plot the evolution of the CB resonance amplification ratio $A(\omega)$, given in Eq. (3) according to (Thomas & Laville, 2007). Here $\eta=\omega/\omega_n$ is the relative pulsation and ω is the excitation pulsation. The $A(\omega)$ evolutions are described on Figure 5 (by computer simulation).

$$A = \frac{1}{\sqrt{(1-\eta^2)+(2\xi\eta)^2}} \tag{3}$$

In Figure 5, the damping ratio (DR), the resistance of the

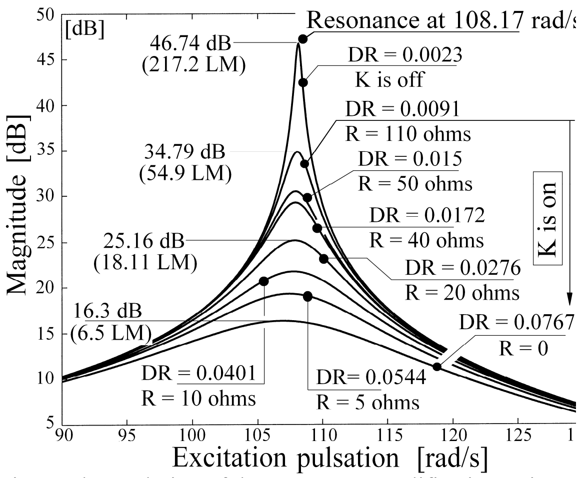

Fig. 5. The evolution of the resonance amplification ratio (in abscissa is the excitation pulsation ω).

external shunt (R) and the amplitude for some resonance peaks (decibels dB and linear magnitude LM) are indicated on each curve. With ***R=0*** the maximum attenuation of the resonating peak is 30.44 dB (33.26 in LM).

5. CONCLUSION

A mechatronic device used as tuneable passive viscous damper was designed, manufactured and evaluated in experimental terms by means of a computer assisted bench-test based on a cantilever beam. An external adjustable electrical resistance was used to tune the damping ratio. According to the theory, the research showed an inverse relationship between the damping ratio and the resistance value. A 30.44 dB maximum attenuation of the resonating amplification was calculated, based on the experimental results (the numerical fitting of the free response signal).

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