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Magnetostrictive Dynamic Vibration Absorber (DVA) for Passive and Active Damping

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Several partners within the consortium of the European 5th Framework project MESA (Magnetostrictive Equipment and Systems for More Electric Aircraft) are working together to tackle the problem of vibration and noise disturbance in turboprop aircraft. The vibration spectrum is characterised by pronounced peaks at approximately 100 Hz, 200 Hz and 300 Hz corresponding to the first, second and third blade pass frequencies (BPFs). Implementing a novel actuator design with mechanical transformation to achieve specified acceleration forces the resulting DVA passively absorbs vibrations at its resonance frequency tuned to the fundamental frequency of disturbance while operating actively over the range of 50-400 Hz. Optimised with respect to weight and size, the DVA design is presented, illustrating the displacement amplification feature based on elastic members. An electromechanical model of device behaviour is developed followed by a description of a closed-loop control algorithm for effective implementation of the active DVA. Experimentally obtained vibration absorbing characteristics are presented. Laboratory measurements indicate fulfilment of specifications for reducing vibrations in turboprop aircraft, while the DVA design could be applied to vibration problems in other vehicles of transportation such as trains and ships or for improving manufacturing quality in machine tools.

1. INTRODUCTION

The active dynamic vibration absorbers (DVAs) described in this paper were developed within the European 5th Framework R&D project MESA [1] to solve problems of vibration and noise in turboprop aircraft. The reduction of structural vibrations by applying DVAs on frame members is an effective means of reducing the cabin sound pressure level and was the selected approach defining the scope of the present development [2]. The vibration disturbance contains pronounced tonal components corresponding to the first three blade pass frequencies (BPFs) close to 100 Hz, 200 Hz and 300 Hz. Alenia Aeronautica, producer of the ATR42 turboprop aircraft and MESA project partner, defined the specifications for the active DVAs based on measurements on an experimental mock-up. A DVA pair, which should weigh less than 500 g, should fulfil in particular the dynamic force requirements of 29 N, 18 N and 10 N at the three BPFs, respectively [3]. In order to permit its mounting on the structural frame members in the aircraft, each DVA should fit in the rectangular volume defined by the dimensions 30 x 50 x 50 mm^3. By implementing active magnetostrictive materials in the DVA, the device should be capable of generating forces over the entire frequency band 60-360 Hz.

2. DESIGN

Early in the concept phase of development, it was decided to combine passive and active functionality resulting in a so-called hybrid DVA. With a resonance frequency corresponding to the first BPF, the hybrid DVA can passively absorb the main vibration disturbance while actively operating as an effective force generator for all other frequencies within the operating band.

In order to achieve the low weight and size requirements, it was necessary to develop a design in which most of the total DVA mass is effective in generating dynamic forces, i.e. the ratio of seismic mass to static mass should be as great as possible. This design consideration demanded a solution in which the coil(s) used to excite the magnetostrictive material should be part of the seismic mass. Furthermore, the force and mass requirements demand DVA displacements on the order of 150 µm as determined by Newton's law $F = m\cdot a = m\cdot(2\pi f)^2\cdot \hat{x}$. This displacement is necessary in resonance at about 100 Hz. Due to the size limitations of the DVA and a typical achievable magnetostrictive strain of about 1000 ppm, the design would have to incorporate a mechanical transformation to amplify the displacement produced by the active rod. A 40 mm long rod placed along one of the two 50 mm axes of the device could produce about 40 µm unloaded displacement. To achieve the required 150 µm DVA mass displacement under load conditions, a mechanical transformation with a displacement amplification $\ddot{u} \approx 5.5\ldots6$ is necessary. An additional important design requirement is to achieve a robust DVA structure that exhibits the desired directional resonance at the first BPF while exhibiting no other structural vibration modes within the intended frequency range of operation.

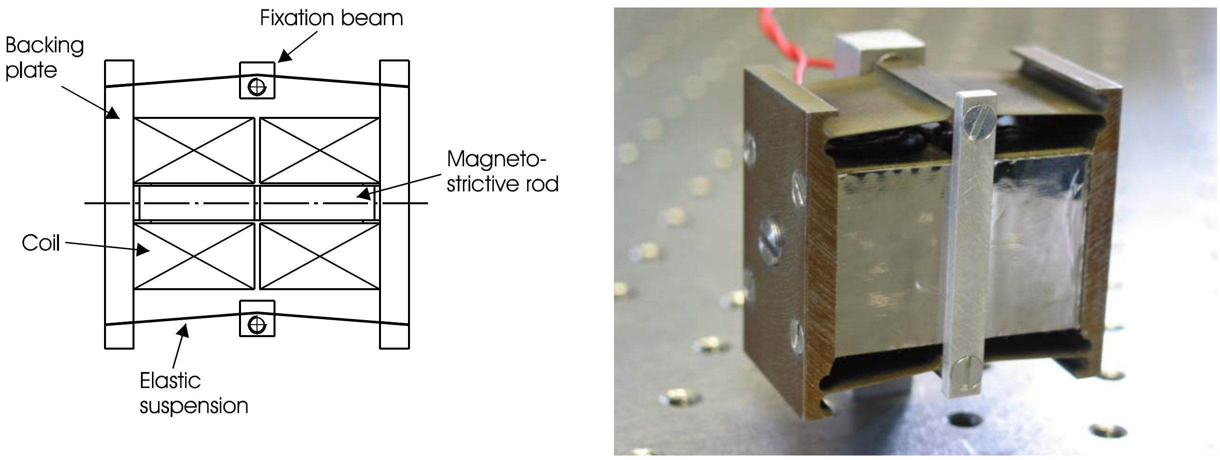


Figure 1: *Cross-sectional view of discretely built DVA (left) and photo of assembled DVA with wire-eroded frame (right)*

These and other design criteria resulted in the DVA structure depicted in Figure 1. The seismic mass is comprised of the coils, the backing plates and the magnetostrictive rod connected to the mechanical fixation beams via elastic suspension arms. The elastic suspension achieves a 90° mechanical transformation of the rod elongation with a displacement amplification of about 6. The elastic suspension also fulfils the function of preload spring, is longitudinally stiff and laterally soft. As a result the overall DVA stiffness is dominated by the contribution of the magnetostrictive rod. Two different DVA versions were produced. The cross-sectional view in the left of Figure 1 depicts a DVA whose structure is comprised of discrete mechanical components, while the photo on the right shows the DVA achieved by wire erosion manufacturing. The overall DVA mass is about 325 g of which the seismic mass is about 90%.

3. MATHEMATICAL MODEL

The electromechanical equivalent circuit diagram of the active magnetostrictive vibration absorber is shown in Figure 2.

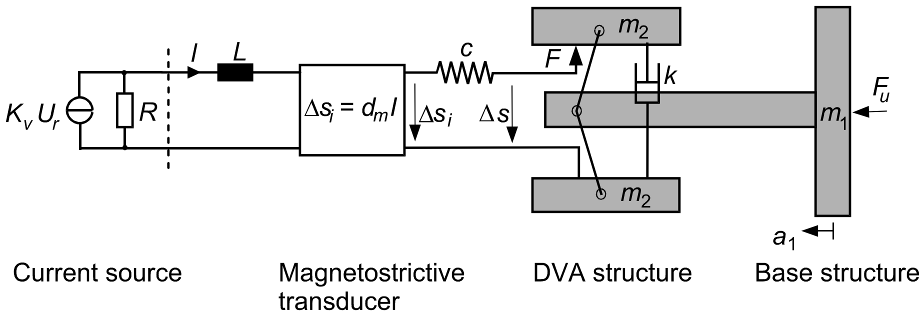


Figure 2: *Electromechanical model of the active magnetostrictive DVA*

As a first approximation the relationship between the force F, the current I and the displacement Δs of the magnetostrictive material can be described by the actuator equation

$$\Delta s = -\frac{1}{c} \cdot F + d_m \cdot I$$

where c represents the mechanical stiffness and d_m the effective magnetostrictive constant of the active material. The DVA structure has a displacement amplification factor of *ü*. The linear damper with the damping constant k is a first approximation of the system's natural damping. The magnetostrictive transducer with electrical inductance L is driven by a voltage-controlled current source with gain K_v and internal resistance R.

The acceleration a_1 of the base structure is a measure for the vibration compensation effect. It can be described in the Laplace domain in terms of the electrical and mechanical excitations and the electromechanical transfer functions.

$$\underline{a}_1 = \underline{G}_V \cdot \underline{G}_I \cdot \underline{U}_r + \underline{G}_F \cdot \underline{F}_u \quad .$$

In terms of the complex Laplace variable $\underline{p} = \sigma + \mathrm{j}\omega$, the current source transfer function is

$$\underline{G}_V = \frac{\underline{I}}{\underline{U}_r} = K_V \cdot \frac{1}{\frac{1}{\omega_V} \cdot \underline{p} + 1}, \quad \text{where } \omega_V = \frac{R}{L} \quad .$$

The hybrid DVA can be described by the guiding and disturbance transfer functions

$$\underline{G}_I = \frac{\underline{a}_1}{\underline{I}} = -K_I \cdot \frac{\underline{p}^2}{\frac{1}{\omega_0^2} \cdot \underline{p}^2 + 2 \cdot \frac{D_0}{\omega_0} \cdot \underline{p} + 1} \quad \text{and} \quad \underline{G}_F = \frac{\underline{a}_1}{\underline{F}_u} = K_F \cdot \frac{\frac{1}{\omega_2^2} \cdot \underline{p}^2 + \frac{2 \cdot D_2}{\omega_2} \cdot \underline{p} + 1}{\frac{1}{\omega_0^2} \cdot \underline{p}^2 + 2 \cdot \frac{D_0}{\omega_0} \cdot \underline{p} + 1},$$

where the single parameters are:

$$K_F = \frac{1}{m_1 + 2 \cdot m_2} \quad , \quad D_0 = \frac{1}{2} \cdot \sqrt{\frac{2 \cdot k^2 \cdot (m_1 + 2 \cdot m_2)}{c \cdot ((4 \cdot \ddot{u}^2 + 1) \cdot m_1 \cdot m_2 + 2 \cdot m_2^{\;2})}} \quad , \quad K_I = \frac{2 \cdot m_2 \cdot d_m \cdot \ddot{u}}{m_1 + 2 \cdot m_2}$$

$$D_2 = \frac{1}{2} \cdot \sqrt{\frac{2 \cdot k^2}{c \cdot ((4 \cdot \ddot{u}^2 + 1) \cdot m_2)}} \quad , \quad \omega_0 = \sqrt{\frac{2 \cdot c \cdot (m_1 + 2 \cdot m_2)}{(4 \cdot \ddot{u}^2 + 1) \cdot m_1 \cdot m_2 + 2 \cdot m_2^{\;2}}} \quad , \quad \omega_2 = \sqrt{\frac{2 \cdot c}{(4 \cdot \ddot{u}^2 + 1) \cdot m_2}}$$

4. CLOSED-LOOP FORCE COMPENSATION

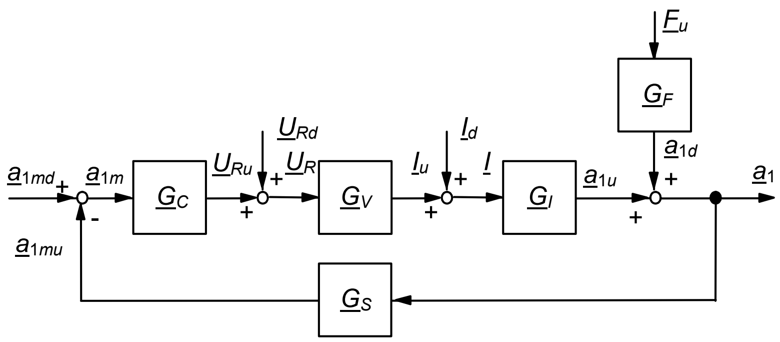
The active compensation of the disturbing forces F_u can now be achieved by a suitable feedback of the measured base acceleration a_1 to the input of the current source. The corresponding acceleration feedback loop is represented in Figure 3a with $\underline{G}_S$ as the sensor transfer function and with $\underline{G}_C$ as the controller transfer function. The acceleration sensor has a low-pass characteristic which is assumed here as a second order filter

$$\underline{G}_S = \frac{1}{(\frac{1}{\omega_{S1}} \cdot \underline{p} + 1) \cdot (\frac{1}{\omega_{S2}} \cdot \underline{p} + 1)}$$

with ω_{S2} , $\omega_{S1} > \omega_V > \omega_0 > \omega_2$. In practice it is mandatory that for each bounded input signal vector $\boldsymbol{X}^{\mathrm{T}} = (a_{1md}\ a_{1d}\ U_{Rd}\ I_d)$ the signal vectors $\boldsymbol{Y}^{\mathrm{T}} = (a_{1m}\ a_1\ U_R\ I)$ and $\boldsymbol{Z}^{\mathrm{T}} = (a_{1mu}\ a_{1u}\ U_{Ru}\ I_u)$, i.e. all signals which occur in a feedback loop, remain bounded and furthermore that the condition

$$\|\boldsymbol{Z}\|_\infty \le L_1 \cdot \|\boldsymbol{X}\|_\infty \quad \text{and} \quad \|\boldsymbol{Y}\|_\infty \le L_2 \cdot \|\boldsymbol{X}\|_\infty$$

is fulfilled with the constants $L_1, L_2 \ge 0$. In such case the feedback loop according to Figure 3a is characterized as internally stable [4].



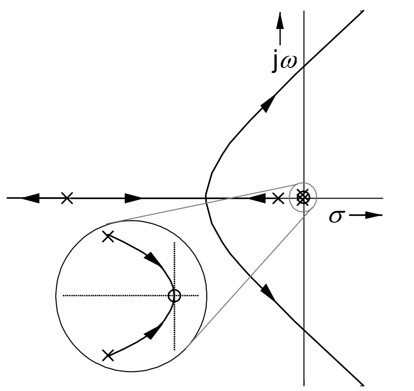


Figure 3: *a) Force compensation within a closed control loop, b) Root locus graph of the characteristic equation*

According to [4] this is equivalent to the fact that the roots of the characteristic equation

$$\underline{P}_C \cdot \underline{P}_V \cdot \underline{P}_I \cdot \underline{P}_S + \underline{Z}_C \cdot \underline{Z}_V \cdot \underline{Z}_I \cdot \underline{Z}_S = 0$$

are in the open left-half $\underline{p}$-plane whereby the denominators of the transfer functions $\underline{G}_C$, $\underline{G}_V$, $\underline{G}_I$ and $\underline{G}_S$ are given by $\underline{P}_C$, $\underline{P}_V$, $\underline{P}_I$ and $\underline{P}_S$ and the numerators by $\underline{Z}_C$, $\underline{Z}_V$, $\underline{Z}_I$ and $\underline{Z}_S$. From this for the control loop according to Figure 3a follows the characteristic equation

$$(\frac{1}{\omega_0^2} \cdot \underline{p}^2 + \frac{2 \cdot D_0}{\omega_0} \cdot \underline{p} + 1) \cdot (\frac{1}{\omega_V} \cdot \underline{p} + 1) \cdot (\frac{1}{\omega_{S1}} \cdot \underline{p} + 1) \cdot (\frac{1}{\omega_{S2}} \cdot \underline{p} + 1) \cdot \underline{P}_C - K_I \cdot K_V \cdot \underline{p}^2 \cdot \underline{Z}_C = 0$$

depending on the controller transfer function $\underline{G}_C$. A more careful consideration of this equation shows that integral parts in the controller transfer function always produce poles at $\underline{p} = 0$ and thus violate the demand for internal stability of the feedback loop. Whereas the use of a simple proportional controller with the controller transfer function

$$\underline{G}_C = -K_C$$

leads to a root locus graph of the characteristic equation as shown in Figure 3b, and thus to an internally stable feedback loop for a sufficiently small gain K_C [5]. The disturbing transfer function of the closed loop describes the relationship between the disturbing force $\underline{F}_u$ and the base acceleration $\underline{a}_1$ in the Laplace domain. Based on the fact that the controller gain K_C can only influence the coefficient of the power $\underline{p}^2$ in the denominator of this transfer function, the feedback has no effect on low and high frequencies. For this reason, the active damping of the disturbance F_u emerging through feedback is only possible in a middle frequency band.

5. EXPERIMENTAL RESULTS

The test and measurement set-up for verifying the performance of the hybrid DVA consists of a base mass m_1 to which an electro-dynamic shaker is attached for generating the external disturbance F_u. The solid line in Figure 4 shows the constant amplitude response of the acceleration a_1 relative to the disturbance force F_u over the frequency range of interest. The dashed line shows the amplitude response of the entire system consisting of mass m_1, the electro-dynamic shaker and the passively working DVA. The passive DVA achieves a narrowband damping of about 6 dB at anti-resonance and an acceleration increase of approx. 4 dB in resonance. The dash-dot line shows the amplitude response of the system with the DVA operating actively in the closed control loop. In contrast to passive operation, the active DVA achieves broadband damping of acceleration with a maximum of nearly 12 dB.

After characterizing the DVA devices, six units were sent to the MESA partners Department of Aeronautical Engineering at the University of Naples Federico II and the Department of Computer Science at the Second University of Naples for testing the devices on a rectified frame member together with control hardware produced by Newlands Technologies, Ltd. These tests simulate the vibration disturbance within the ATR42 aircraft and allow a verification of the hybrid DVA performance [2].

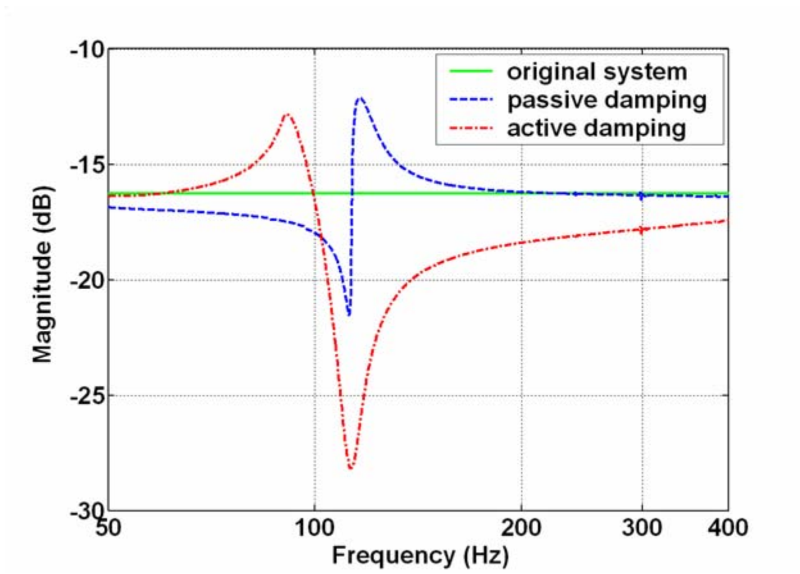


Figure 4: *Amplitude response* $|\underline{a}_1/\underline{F}_u|$ *in the test rig without DVA (original system) as well as with the hybrid DVA acting passively (dashed curve) and actively (dash-dot curve)*

6. CONCLUSIONS

A hybrid magnetostrictive dynamic vibration absorber with a passive resonance tuned to about 100 Hz has been developed that fulfils the functional requirements for combating vibration and noise disturbances in the frequency band 50-400 Hz. This hybrid DVA design is therefore suitable for reducing the sound pressure level in the cabin of turboprop aircraft where tonal disturbances at the first three blade pass frequencies near 100 Hz, 200 Hz and 300 Hz are predominant. As disturbances of this nature can be found in many transportation applications, the achieved DVA design, for which a patent is pending, has large application potential.

ACKNOWLEDGEMENTS

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