TUNABLE MAGNETOSTRICTIVE DYNAMIC VIBRATION ABSORBER

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Abstract
We present a tunable Magnetostrictive Dynamic Vibration Absorber (DVA) developed within the European project MESA to tackle the problem of vibration and noise in turboprop aircraft. The tunability of the apparent mechanical stiffness and damping coefficients of the DVA is achieved through the proper feedback of the force generated in the magnetostrictive rods to adapt the resonant frequency and damping of the passive DVA to the time-variant first blade pass frequency. Experimentally obtained vibration absorbing characteristics with and without hysteresis compensation are presented. The resonant frequency of the tunable DVA can be varied electronically by about 15%. Implementation of the hysteresis compensation improves the range and stability of DVA tuning.

Keywords: Vibration absorber, magnetostriction, hysteresis compensation, tunability, magnetostrictive actuator

Introduction

Several partners within the consortium of the recently completed European 5th Framework project MESA (Magnetostrictive Equipment and Systems for More Electric Aircraft) worked 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 resonant 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 presented DVA illustrates a displacement amplification feature based on elastic members [1].

This article deals with the extension of this hybrid device shown in Fig. 1 to a semi-active DVA whose apparent mechanical stiffness and damping coefficients are tunable via electrical signals to adapt the resonant frequency and damping of the passive DVA to the time-variant first blade pass frequency. The tunability of the apparent mechanical stiffness and damping coefficients is achieved through the proper feedback of the force generated in the magnetostrictive rods. The force is measured by two thin piezoelectric discs which act as sensors and are located between the two magnetostrictive material rods. This actuator-sensor pair is collocated and this enables the implementation of a so-called "intrinsically passive control" law (IPC) [2]. The IPC law used in our device is a first order transfer function between the measured force and the actuated displacement. This corresponds to the behaviour of a stiffness and damper in parallel.

For large electrical signal amplitudes the complex hysteretic characteristic of the magnetostrictive rods becomes significant thereby limiting the DVA tunability. The consequences are first a nonlinear damping mechanism which predominates the damping actively introduced into the device and second a destabilisation of the feedback loop which decreases the range of tunability. To avoid these problems the hysteretic actuator characteristic is compensated by a feedforward controller which implements the inverse hysteretic characteristic of the magnetostrictive rods and is designed with the modified Prandtl-Ishlinskii approach [3].

The paper will first show the development approach for this type of vibration absorber. Thereafter experimentally obtained vibration absorbing characteristics with and without hysteresis compensation are presented.

Design of the tunable DVA

The design of the DVA is described in detail in [1]. As shown in Fig. 1, the seismic mass of the tunable DVA is comprised of the coils, the backing plates and the magnetostrictive rods connected to the mechanical fixation beams via elastic suspension arms. The elastic suspension achieves a 90° mechanical transformation of the rods elongation to the mass displacement with a displacement amplification u 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 influenced strongly by the contribution of the magnetostrictive rod stiffness [1]. Two thin piezoelectric discs which act as a force sensor are located between the two magnetostrictive rods. In this way the force sensor and the magnetostrictive rods can be considered a collocated actuator-sensor pair.

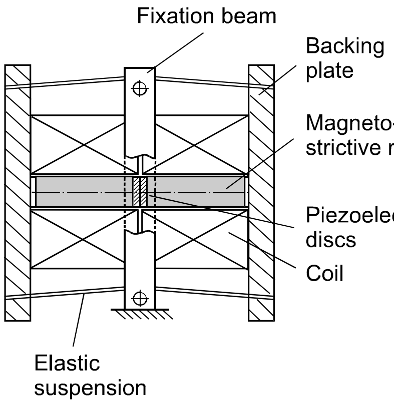

Fig. 1: *Construction of tunable DVA*

Collocated control

A collocated actuator-sensor pair enables the implementation of a so-called intrinsically passive control law. One of the possible collocation concepts is based on position actuation and collocated force measurement.

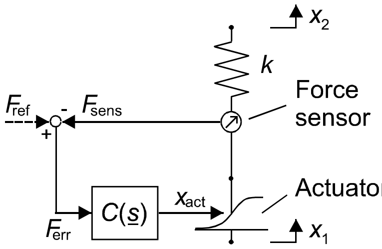

Fig. 2: *Collocation concept [2]*

The stiffness indicated in Fig. 2 as k represents the stiffness of the passive mechanical structure.

The signal that is available for control is given by the difference between a reference force and the measured force: $F_{\mathrm{err}}(t) = F_{\mathrm{ref}}(t) - F_{\mathrm{sens}}(t)$. As the reference signal in vibration minimisation problems is typically zero, the input of the controller $C(\underline{s})$ (Laplace variable $\underline{s} = \sigma + \mathrm{j}\cdot\omega$) is given by $F_{\mathrm{err}}(t) = -F_{\mathrm{sens}}(t)$. If we impose a control law such that:

$$F_{\mathrm{err}}(t) = d_{\mathrm{avc}} \cdot \dot{x}_{\mathrm{act}}(t) + k_{\mathrm{avc}} \cdot x_{\mathrm{act}}(t) \qquad (1)$$

we obtain a first-order transfer function between the measured force and the actuated position:

$$C(\underline{s}) = \frac{x_{\mathrm{act}}(\underline{s})}{F_{\mathrm{err}}(\underline{s})} = \frac{1}{d_{\mathrm{avc}} \cdot \underline{s} + k_{\mathrm{avc}}} = \frac{K_{\mathrm{f}}}{\tau_{\mathrm{f}} \cdot \underline{s} + 1} \qquad (2)$$

with $K_{\mathrm{f}} = \dfrac{1}{k_{\mathrm{avc}}}$ and $\tau_{\mathrm{f}} = \dfrac{d_{\mathrm{avc}}}{k_{\mathrm{avc}}}$.

This corresponds to the behaviour of a parallel connection of a damper and a stiffness (see Fig. 3).

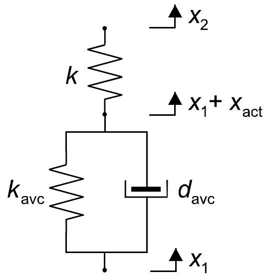

Fig. 3: *Physical interpretation of the control law [2]*

The active stiffness k_{avc} and the stiffness k of the passive mechanical structure are connected in series. Therefore the total stiffness is lower than the original stiffness and the resonant frequency of the DVA decreases. The controller $C(\underline{s})$ has a low-pass filter behaviour, whereby K_{f} is the gain and $\omega_{\mathrm{f}} = 1/\tau_{\mathrm{f}}$ is the cut-off frequency. Tuning of the DVA stiffness and damping is enabled through variation of the gain and the cut-off frequency of the controller.

Modelling and test rig

The schema of the test rig that we have used in our research is shown in Fig. 4. The DVA was attached to the base structure with mass m_1. The structure was excited using an electrodynamic shaker between 90 and 150 Hz. The acceleration a_1 of the base structure and the disturbing force F_{u} were measured.

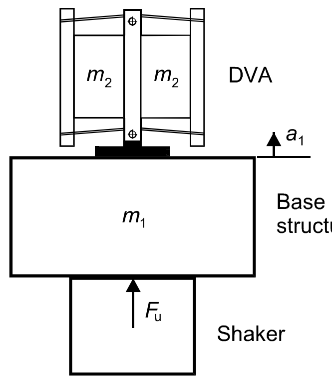

Fig. 4: *Schema of test rig*

The acceleration a_1 of the base structure is a measure for the vibration compensation effect. The transfer function between the disturbing force F_{u} and the acceleration a_1 can be described by

$$G_{\mathrm{F}}(\underline{s}) = \frac{a_1(\underline{s})}{F_{\mathrm{u}}(\underline{s})} = K_{\mathrm{F}} \cdot \frac{\dfrac{1}{\omega_2^{\,2}} \cdot \underline{s}^2 + \dfrac{2 \cdot D_2}{\omega_2} \cdot \underline{s} + 1}{\dfrac{1}{\omega_0^{\,2}} \cdot \underline{s}^2 + \dfrac{2 \cdot D_0}{\omega_0} \cdot \underline{s} + 1} \qquad (3)$$

with

$$K_F = \frac{1}{m_1 + 2 \cdot m_2}; \quad \omega_2 = \sqrt{\frac{2 \cdot k_e}{(4 \cdot u^2 + 1) \cdot m_2}};$$

$$D_0 = \frac{1}{2} \cdot \sqrt{\frac{2 \cdot d^2 \cdot (m_1 + 2 \cdot m_2)}{k_e \cdot ((4 \cdot u^2 + 1) \cdot m_1 \cdot m_2 + 2 \cdot m_2^2)}};$$

$$D_2 = \frac{1}{2} \cdot \sqrt{\frac{2 \cdot d^2}{k_e \cdot ((4 \cdot u^2 + 1) \cdot m_2)}};$$

$$\omega_0 = \sqrt{\frac{2 \cdot k_e \cdot (m_1 + 2 \cdot m_2)}{(4 \cdot u^2 + 1) \cdot m_1 \cdot m_2 + 2 \cdot m_2^2}}.$$

The parameters ω_2 and ω_0 are respectively the anti-resonant and the resonant frequency of the system, where $\omega_2 < \omega_0$. D_2 and D_0 are the damping ratios and K_F is the proportional coefficient. The total mechanical stiffness is approximated with the parameter k_e. This parameter depends on the stiffness of the magnetostrictive rod, the stiffness of the elastic suspension and the stiffness k_{avc} introduced through the force feedback.

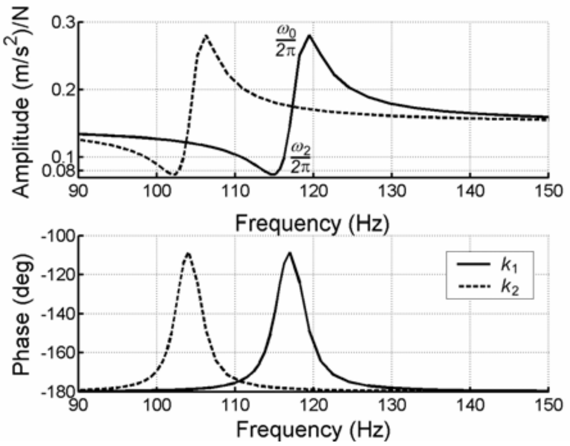

Fig. 5: *Tuning of the stiffness ($k_e = k_i$, $k_1 > k_2$)*

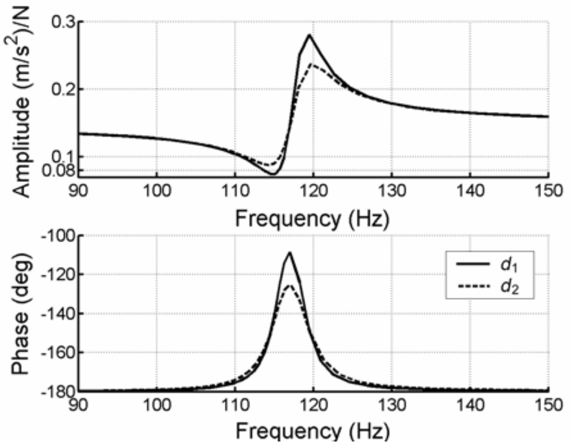

Fig. 6: *Tuning of the damping ($d = d_i$, $d_1 < d_2$; $k_e = k_1$)*

A linear damper with the damping constant d is a first approximation of the system's natural damping. This parameter depends on the damping of the magnetostrictive rods and the elastic suspension as well as the damping d_{avc} introduced by the force feedback. By changing the parameters k_e and d in the model (3) we obtain the ideal curves (Bode diagrams of $G_F(\underline{s})$) shown in Figs. 5 and 6.

Test results without hysteresis compensation

Fig. 7 shows the measurement of the base acceleration a_1 related to the disturbing force F_u for different values of k_{avc}. We can recognise the typical behaviour of the passive dynamic vibration absorber with an anti-resonance and a resonance, similar to the behaviour presented for the transfer function $G_F(\underline{s})$.

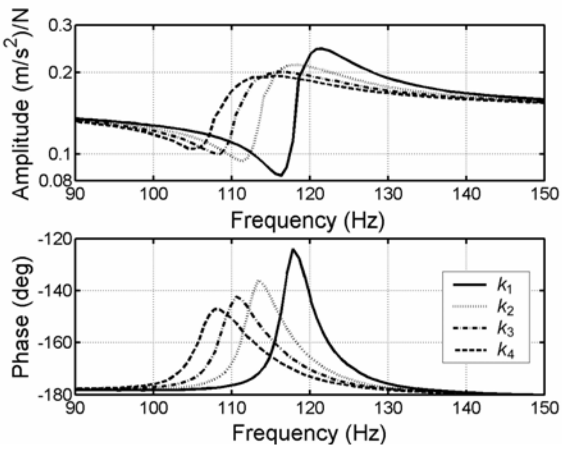

Fig. 7: *Tuning of the stiffness ($k_{avc} = k_i$, $k_1 > k_2 > k_3 > k_4$)*

The deviation of the measurement curves in Fig. 7 from the theoretical transfer functions shown in Fig. 5 is marked by a decrease in amplitude with increased feedback.

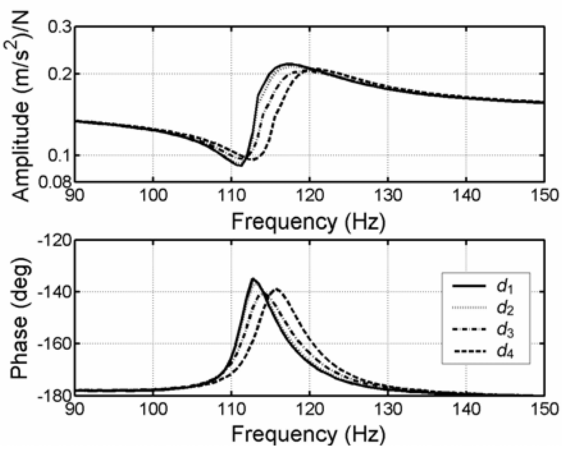

***Fig. 8**: Tuning of the damping ($d_{avc} = d_i$, $d_1 < d_2 < d_3 < d_4$; $k_{avc} = k_2$)*

This behaviour could possibly be explained by increased hysteretic damping resulting from wider hysteresis loops. The curves recorded in Fig. 8 for four different values of d_{avc} also indicate a possible influence of hysteresis. Therefore, it should be important to compensate the hysteretic behaviour of the actuator.

Test results with hysteresis compensation

According to [3,4] the real complex hysteretic actuator characteristic W of the magnetostrictive rods shown in Fig. 9 as curve 1 (in gray) is modelled by a modified Prandtl-Ishlinskii hysteresis operator Γ shown in Fig. 9 by curve 2 (in black) and is compensated by a feedforward controller which implements the inverse hysteretic characteristic Γ^{-1} as shown in Fig. 9 by curve 3.

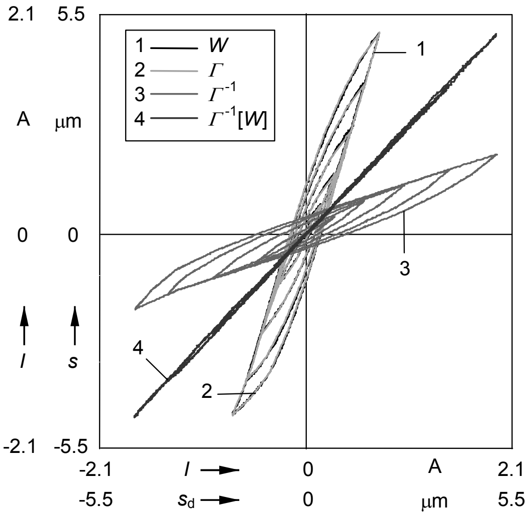

Fig. 9: Hysteresis modelling and compensation

As a result the feedforward hysteresis compensation scheme leads to a strongly linearised actuator characteristic

$$s(t) = W[\Gamma^{-1}[s_{\mathrm{d}}]](t) \tag{4}$$

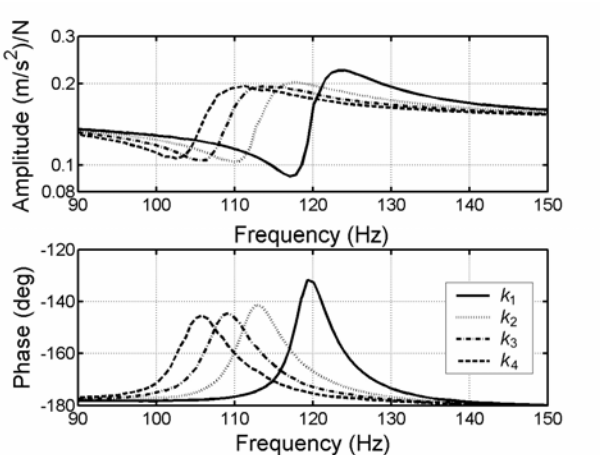

***Fig. 10**: Tuning of the stiffness ($k_{\mathrm{avc}} = k_{\mathrm{i}}$, $k_1 > k_2 > k_3 > k_4$)*

as shown in Fig. 9 by curve 4. In Fig. 9 and Eq. (4) s_{d} is the desired displacement, s the real displacement and I the actuator current. Figures 10 and 11 show similar characteristics as Figures 7 and 8, resulting from the use of hysteresis compensation. One can see that compensating the hysteresis of the active material leads to a more regular behaviour closer to the theoretical behaviour (Figures 5 and 6). Additionally, the range of tunability increases.

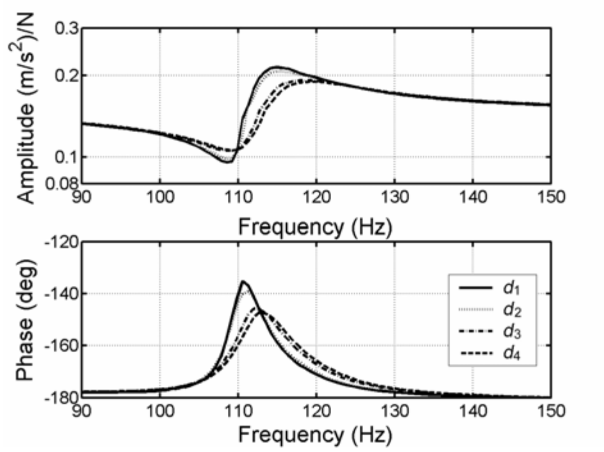

***Fig. 11**: Tuning of the damping ($d_{\mathrm{avc}} = d_{\mathrm{i}}$, $d_1 < d_2 < d_3 < d_4$; $k_{\mathrm{avc}} = k_2$)*

Conclusions

This work demonstrates that the resonant frequency of a magnetostrictive DVA can be reduced electronically by about 15% by applying an appropriate control law with collocated force sensing. Furthermore, the implementation of a feedforward hysteresis compensation algorithm increased the achievable range of tunability and damping. This tunability is beneficial in applications involving tonal disturbances of variable frequency as can be found in turboprop aircraft and helicopters. Further improvements in the tunability of magnetostrictive DVA can be expected by refining the control laws applied.

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