OPTIMISATION OF A MAGNETOSTRICTIVE AUXILIARY MASS DAMPER

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Abstract

A patented magnetostrictive auxiliary mass damper design has been optimised within the ongoing European project MESEMA (Magnetoelastic Energy Systems for Even More Electric Aircraft) using a wide variety of engineering tools including finite element and lumped parameter modelling. The result of this mechatronic development process is a new, light-weight magnetostrictive auxiliary mass damper, which will be replicated 30 to 50 times for structural vibration reduction experiments in a mock-up of an aircraft. The optimised device weighs only 150 g, of which 90% is involved in generating dynamic forces. The resonant frequency can be tuned within the range of 82…107 Hz by adjusting the preload. At frequencies up to 1000 Hz the magnetostrictive device generates dynamic forces in the range of 2.5…13 N. This paper begins with a discussion on auxiliary mass damping pointing out and comparing numerous operating modes including semi-active, active, hybrid and adaptive which become possible through the use of active materials. The optimisation resulted in a lower device weight, greater bandwidth, but also in increased control authority, which improves device efficiency and adaptability.

Keywords:
auxiliary mass damper, magnetostriction, hysteresis compensation, tunability, magnetostrictive actuator

Introduction

Unwanted externally excited or self-induced vibrations occurring in many mechanical structures of our engineered environment need to be reduced in the interest of lower environmental impact and increased structural durability. Reducing vibrations can be achieved with the help of passive vibration absorbers which extract kinetic energy from the vibrating host system or active ones which introduce opposing forces into the structure to affect compensation. With an auxiliary mass damper (AMD) both classes of absorber can be unified in a single, compact unit through the use of active materials and an adaptronic system approach. The optimisation of a magnetostrictive AMD originating from the European project MESA is reported following a review of five possible operating modes of active AMD.

Passive auxiliary mass damper

Considering an effective mass m_1 which defines an equivalent excitation force $F_1(t) = m_1 \cdot a_1(t)$ leading to a local acceleration amplitude $a_1(t) = (\mathrm{d}^2/\mathrm{d}t^2)s_1(t)$, the purpose of an auxiliary mass damper, as apparent in **Figure 1**, is to generate a secondary force $F_2 = F_c + F_d$ which compensates the primary force F_1 thereby counteracting the excitation of mass m_1. In its most simple form, an AMD consists of an auxiliary mass m_2 which is coupled to the primary system via a spring element with the stiffness c_2 and a damping element with the damping constant d_2. Beginning with the equations of motion for the two

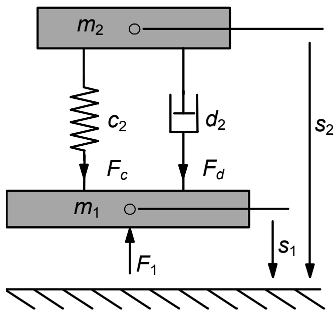

Figure 1: *Effective mass with primary excitation and passive auxiliary mass damper*

masses m_1 and m_2 the equation for the disturbance transfer function of the effective mass becomes

$$G_D(p) = \frac{a_1(p)}{F_1(p)} = +K_D \frac{\frac{1}{\omega_a^2} p^2 + \frac{2D_a}{\omega_a} p + 1}{\frac{1}{\omega_r^2} p^2 + \frac{2D_r}{\omega_r} p + 1} \qquad (1)$$

where

$$\omega_r = \sqrt{c_2 \frac{m_1 + m_2}{m_1 m_2}}\,,\ D_r = \frac{1}{2} d_2 \sqrt{\frac{m_1 + m_2}{c_2 m_1 m_2}}\,,$$

$$\omega_a = \sqrt{\frac{c_2}{m_2}}\,,\ D_a = \frac{1}{2} d_2 \sqrt{\frac{1}{c_2 m_2}}\,,\ K_D = \frac{1}{m_1 + m_2}\,.$$

In the undamped case ($d_2 = 0$) the denominator of transfer function (1) has a zero at $p = \mathrm{j}\omega_r$ for the frequency ω_r. In the field of structural dynamics this special frequency is defined as the resonant frequency. The numerator of transfer function (1) on the contrary exhibits a zero at $p = j\omega_a$ for the resonant frequency ω_a of the passive AMD. At this particular frequency, the so-called anti-resonant frequency of the overall system (effective mass with damper), the effective mass m_1 cannot experience any motion independent of the strength of the excitation force F_1. For this reason, the parameters auxiliary mass m_2 and spring stiffness c_2 are typically selected such that the anti-resonant frequency ω_a of the passive AMD corresponds with the dominating frequency component of the primary force F_1 [1-3].

Semi-active auxiliary mass damper

One possible implementation of a semi-active AMD involves coupling the auxiliary mass to the host structure via an electrically controllable material such as piezoelectric ceramic or magnetostrictive alloy [4]. The mechanical parameters stiffness and damping of the passive AMD can then be influenced with help of the actuator and sensory properties of the active material and feedback of the actuator force [1, 5, 6].

Active auxiliary mass damper

Passive and semi-active AMD can only be used to achieve a narrowband damping of the primary force field. Active AMD achieve broadband damping of the primary force excitation by coupling the auxiliary mass m_2 to the host structure with active materials. Based on the dynamic equilibrium of forces acting on the effective mass m_1 shown in **Figure 1**, its acceleration a_1 is a measure of the compensating effect of the secondary force F_2 originating from the coupled auxiliary mass damper. The goal of active damping is to control the motion of the auxiliary mass m_2 with an appropriate electrical driving signal to the active material such that the resulting force $F_2 = m_2 \cdot a_2$ compensates the immeasurable primary force F_1 and extinguishes the acceleration a_1 [1, 7].

Hybrid auxiliary mass damper

The characteristic damping behaviour of the actively driving auxiliary mass damper indicates that a combination of the active and semi-active operating modes can be useful when the dominating primary force component undergoes frequency shifts. This hybrid approach is based on the idea of shifting the resonant frequency of the passive AMD so as to affect an electrically controllable variation of the damping behaviour of the active AMD [1].

Adaptive auxiliary mass damper

In combination with suitable signal processing methods and learning algorithms, a hybrid AMD can be brought to adapt its damping behaviour optimally and automatically to time-variant vibration conditions [1].

The damping behaviour of the five AMD implementation examples described above are summarised in the following table.

***Table 1**: Functionality of five AMD operating modes*

	AMD operating mode	Narrow-band	Wide-band	Controllable stiffness and damping	Self-adapting properties
	Passive	X			
Active materials	Active		X		
	Semi-active	X		X	
	Hybrid		X	X	
	Adaptive		X	X	X

Device optimisation

Relative to the patented device design developed for the preceding European project MESA [1, 6-9], the auxiliary mass damper used to control noise and vibration in turbofan aircraft within the MESEMA project should fulfil a new set of requirements:

- reduced device weight to 100…150 g
- increased operating band to 120…600 Hz
- reduced force requirements in the range 1…8 N

This new set of requirements could only be achieved through a radical optimisation process involving 1) a drastic reduction of active material volume, 2) improvement of the magnetic circuit effectiveness including measures to lower eddy current losses and 3) optimisation of the structural components with respect to weight and stiffness. Structural 3D modelling (see **Figure 2**) and simulation tools supported the optimisation process by providing insight in to modal behaviour, mechanical stresses and strains as well as magnetic flux distribution. The 3D views of the components also supported the process of programming the CNC production machinery.

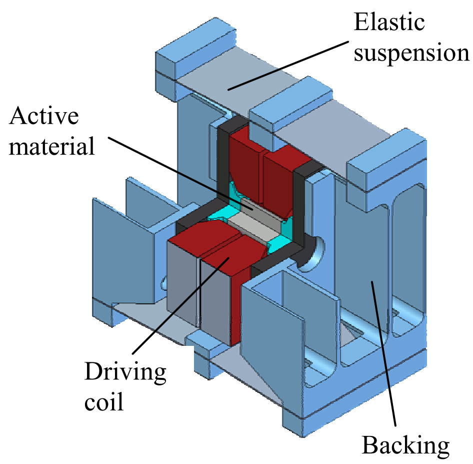

***Figure 2:** 3D model of the optimised magnetostrictive auxiliary mass damper*

Selection of a high-quality magnetostrictive material permitted the mass of the active core to be reduced from 5.2 g to 0.9 g, which by the way has a considerable effect on the device costs. The associated reduction in driving coil length made a valuable contribution to reducing the overall device weight. The active element with its small cross-section combined with magnetic circuit components made of mid-frequency soft-magnetic materials such as Permedyn® and ferrite reduce eddy current losses, thereby increasing the operating frequency band by a factor of 5 to 10 from roughly 500 Hz into the kilohertz range. Structural device components were reduced in weight by replacing steel with aluminium. With the aid of finite-element modelling (FEM) a ribbed construction was developed for the backings increasing the bending stiffness while reducing the weight of the structural components by about two thirds. Consequently, the overall weight of the resulting device shown in **Figure 3** was halved from 325 g to approx. 150 g.

Of great significance for improving device performance but also for extending the adaptability of the magnetostrictive auxiliary mass damper in its semi-active, hybrid and adaptive operating modes lies in the optimisation of the preload and suspension stiffnesses. In the MESA design, the elastic suspension elements simultaneously fulfilled the function of preload springs with the drawback that the preload stiffness was quite high and consequently also the bending stiffness of the elastic suspension. The associated reactive load detriments the device performance and lowers the authority of the active magnetostrictive element. In the new MESEMA design these two functions have been decoupled with the result that both the preload stiffness as well as the bending stiffness of the elastic suspension is substantially reduced. Consequently, the authority of the active element is increased, which according to the mathematical modelling, should extend the range of device adaptability.

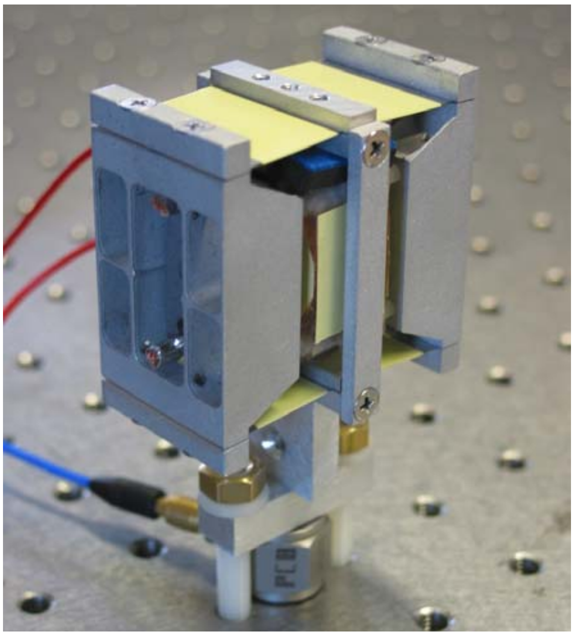

***Figure 3:** Arrangement for measuring the force generating capability*

Device performance assessment

To test the force generating capability of the magnetostrictive AMD, the device was mounted to a steady table with a force sensor in between (**Figure 3**). The plastic preload bolts running parallel to the force sensor are soft and therefore introduce much less than 1% error to the force measurement. The broadband force characteristic was investigated by applying a sinusoidal current sweeping the frequency from 50 Hz to 1000 Hz. The amplitude response of the force is shown in **Figure 4** for small, medium and large driving current signals. The minor perturbations seen on the right-hand flank of the resonant peaks seem to be attributable to vibration behaviour associated with the sensor mounting arrangement. The extraneous peak appearing in the vicinity of 200 Hz for large driving currents is associated with an undesired vibration mode in the device itself. This behaviour is not expected in broadband control applications, but nonetheless demands attention to try to eliminate it. All in all, the magnetostrictive AMD demonstrates high-quality broadband force generating characteristics up to 1000 Hz.

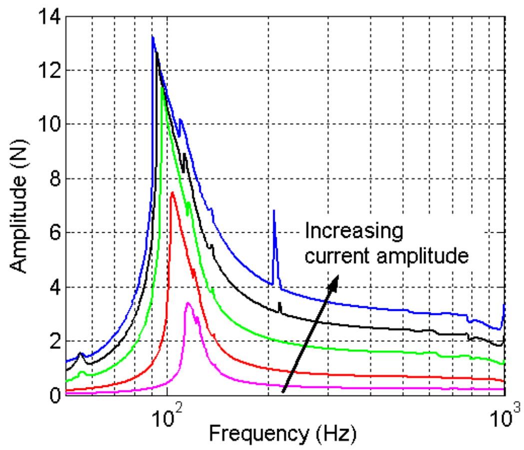

***Figure 4:** Amplitude response of force*
(I = (2.1 ± a) A, *where* a = 0.2; 0.5; 1.0; 1.5; 2.0)

The device was also investigated with respect to its sensitivity to the mechanical preload. **Figure 5** shows that the mechanical preload has a substantial influence on the resonant frequency which ranges from 82 Hz to 107 Hz. This dependability can be useful for tuning the magnetostrictive AMD to a particular predominant frequency in the primary force F_1. Additionally, an optimum preload is recognisable with respect to the force amplitude achievable from the device. Increasing the preload to this optimum value increases the force generating capacity of the device.

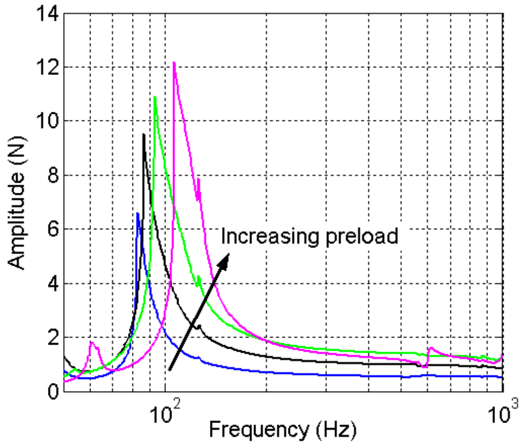

Figure 5: *Amplitude response of force for wide range of mechanical preload* (I = (2.1 ± 1.0) A)

Conclusions and outlook

The MESEMA project activity has permitted far-reaching optimisation of an already robust, patented magnetostrictive auxiliary mass damper design resulting from the preceding European project MESA. Between 30 and 50 of these devices are being produced and will be implemented together with other system components developed within MESEMA to abate structural vibrations and resulting cabin noise in a full-scale aircraft mock-up. Within the scope of continued research at Saarland University's Laboratory of Process Automation (LPA), auxiliary mass dampers based on active materials such as the one reported here are being extended in their functionality through the development of semi-active, hybrid and adaptive control methods. As reported in [6] the compensation of hysteresis improves the tunability of active AMD. Therefore, the new high-speed hysteresis compensation module reported in [10, 11] is expected to make a significant contribution.

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