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VIBRATION ABSORBERS AND METALLIC DAMPING LAYERS

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Alternating forces on an engine induce structure vibrations, and secondary noise is emitted. The means for damping this type of noise - structure-borne noise - are summarised, for instance, in VDI Standard 3727: "Schallschutz durch Körperschalldämpfung". The basic means comprise a damping layer, a constrained layer, and a vibration absorber. The set-up, effect and dimensioning of the vibration absorber and the "metallic" damping layer, both of which were developed at the Aeroacoustic Laboratory in Ottobrunn, are described in the following. Vibration absorbers have been used in rail vehicles for over a decade. Wheel-vibration absorbers on the ICE train reduce noise emission at 200 km/h by 8 dB(A).

1. INTRODUCTION

This paper aims to investigate secondary and passive means for damping structural vibrations and the structure-borne noise emitted. A restriction to secondary means is made as it is assumed that the primary measures which can be implemented directly at the source have already been exploited. Following this restriction to passive means - in contrast to the active anti-noise and anti-vibration techniques - the vibration absorber and the damping layer remain. The set-up, effect, dimensioning and application of the vibration absorber and the damping layer are summarised in the following. In order to take the more stringent requirements into consideration with respect to thermal stability, ageing, resistance to oil and recycling, the "metallic" damping layer is included in the study.

2. THE VIBRATION ABSORBER [1, 2]

2.1 SET-UP AND EFFECT

Basically, a vibration absorber is a bar-shaped waveguide designed, for instance, for flexural waves. It is fixed at one end to the component to be damped. An interfering vibration in the component induces a wave in the vibration absorber that is propagated to the other end and is absorbed there by a damping segment. This damping process, which is based on wave abduction, is illustrated schematically in Fig. 1a. A vibration absorber is defined by its resistance R. A waveguide with wave velocity c, cross-sectional area A and density ρ, features resistance R

$$R = \rho\, c\, A \qquad (1)$$

and represents, according to Fig. 1b, a velocity-proportional damping device, whereby the ratio of the damping force F and the vibration velocity v corresponds to resistance R

$$R = F / v \qquad (2).$$

Assuming that the component to be damped vibrates at vibration velocity v, according to (2), an absorber with resistance R yields a damping force $F = v\,R$ and absorbs the vibration energy N

$$N = v\; F = v^2\, R \qquad (3).$$

The damping force F is supported on the absorber mass. The cross-sectional tapering of the waveguide, as a transformer, amplifies the displacements, thus generating a virtual, seismic mass. With exponential tapering according to

$$A = A_o \exp(-4\, \pi\, f_c\, x / c) \qquad (4),$$

the resistance R as a function of frequency f is

$$R = \rho\; c\; A_o\, \mathrm{sqr}(1\text{-}(\,f_c\,/f\,)^2\,) = R_o\, \mathrm{sqr}(1\text{-}(\,f_c\,/f)^2\,) \qquad (5).$$

Here, the so-called cut-off frequency f_c marks the boundary as of which the damping resistance R is effective. According to Fig. 1a, for frequencies $f >> f_c$, the asymptotic resistance $R_o = \rho c A_o$ for a cylindrical waveguide occurs with cross section A_o. [1*]
Even if every taper profile has its own resistance spectrum, the ratios are comparable to the exponential profile. - A new category of vibration absorbers is opened up if homogeneous waveguides with a constant wave speed c are replaced by gradient waveguides [3].

[1*] Integrated via the horn length x --> ∞ , the mass M of the waveguide $M = R_o/4\, \pi\, f_c$ is yielded as a function of cut-off frequency f_c and asymptotic resistance $R_0 = \rho\; c\; A_o$.

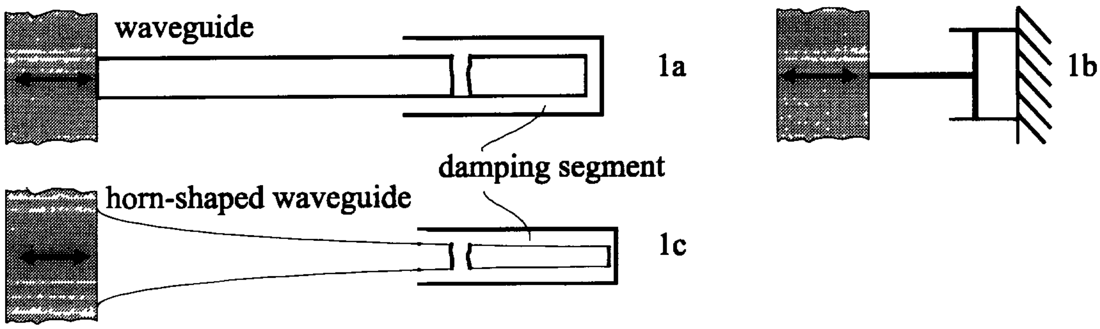

Fig. 1a : Vibration absorber with cylindrical waveguide.
Fig. 1b : Equivalent circuit for the vibration absorber according to 1a.
Fig. 1c : Vibration absorber with tapered waveguide.

Fig. 2 shows vibration absorbers to damp ICE train wheels. The waveguide here is designed for flexural waves and consists of 2 steel plates bonded together by means of a damping material. The broadside of the absorber is fixed to the wheel rim so as to be force-locking. Wheel vibrations are introduced into the waveguide in this way, are propagated as flexural waves to the tapered end, and are damped along this segment. Consequently, vibration energy is withdrawn irreversibly from the wheel. According to DB Acoustic Report 6.93 [5], these vibration absorbers, developed by Dasa and manufactured by GHH, Sterkrade under licence, reduce wheel noise by 8 dB(A) (Fig. 3).

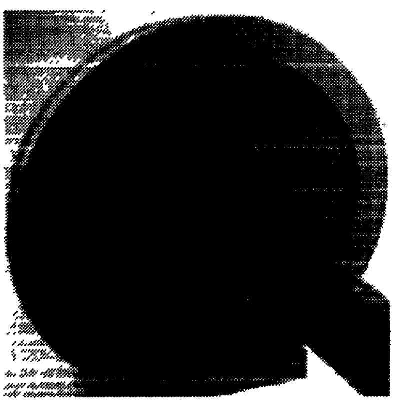

Fig. 2 : Vibration absorber for train wheels (ICE wheels), developed by Dasa of Ottobrunn and manufactured by GHH of Sterkrade.

Fig. 4 : Vibration absorber of sandwich construction for train wheels [3].

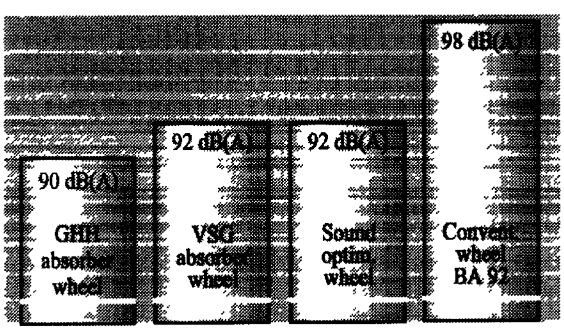

Fig. 3 : Noise immission of different ICE wheels when travelling past at 200 km/h, at a distance of 7.5 m from the track centre and 1.2 m above the rail surface [5].

Fig. 4 illustrates a vibration absorber - again in use to damp wheel-on-rail noise - featuring a waveguide made by layering steel and plastic plates. Despite the discretization, such an arrangement in the frequency range that is of interest can be considered as being a homogeneous waveguide. An arrangement of this type can transmit longitudinal and transverse waves simultaneously. With its broadside fixed to the wheel rim, this absorber type can damp axial as well as tangential wheel vibrations [3].

2.2 DIMENSIONING

The effect of vibration absorbers is based mainly on their resistance. The following section serves to assess which resistance is required to achieve a specific damping effect. A distinction must be made between two borderline cases to which differing design procedures apply: In the first case, an absorber is coupled directly to the exciter point so that, due to the increased impedance, the vibration-energy share introduced into the adjacent structure is smaller. In the second case, changing and diffuse structure-borne noise exciters are present, and absorber elements serve to increase the internal damping of the transmitting and emitting structure. The effort associated with the absorber elements decreases, the nearer they can be attached to the respective exciter point.

Case 1: Vibration absorbers as isolating elements

This application serves to reduce the vibration energy transmitted to the noise-emitting structure by means of vibration absorbers attached directly at an exciter point. This method can be applied when a defined, locally fixed structure-borne noise exciter is the case. Such exciters comprise impacts, imbalances, roller-bearing pairs and, in particular, machinery and engine bearings. If a structure features impedance R_E at an exciter point and if an absorber with resistance R is attached there, less structure-borne noise energy is introduced into the structure even if the exciter power remains the same. The decrease in level amounts to

$$\Delta L = 20 \lg R_E / (R_E + R) \qquad (6).$$

Case 2: Vibration absorbers as damping elements

When changing, diffuse structure-borne noise exciters are present, the internal damping of the transmitting and emitting structure must be increased. For this purpose, vibration absorbers should be distributed over the entire structure, preferably at points with a high vibration level. With respect to performance adjustment, the resistance of an absorber must be the same as or less than the local resistance at the respective attachment point.

When the vibration excitation is the same, the vibration level in the structure and thus also the noise emission are reduced by

$$\Delta L = 10 \lg R_0 / (R_0 + R) \qquad (7);$$

whereby R_0 is the inherent resistance of the structure and R the sum of the resistances of the attached vibration absorbers. The inherent resistance R_0 of a structure with mass M can be estimated approximately via its reverberation time T_0 in accordance with the numerical-value relation [1,2]

$$R_0 = 13{,}8\ M / T_0 \qquad (8).$$

3. DAMPING LAYER

3.1 ORGANIC DAMPING LAYER [1,8]

The conventional damping layer involves an organic damping material, e.g. bitumen or plastic. It is most suited to damping sheet metal with a wall thickness of up to 5 mm. For this purpose a whole-area coating 1 to 3 times the sheet-metal thickness is applied. The characteristics of a damping layer are its complex Young's modulus E, comprising the elongation modulus E´ and the loss modulus E"

$$E = E' + i\,E'' \qquad (9).$$

A further characteristic of the damping layer is its loss factor η

$$\eta = E'' / E' \qquad (10).$$

3.2 METALLIC DAMPING LAYER [9,10]

Organic damping materials have very much smaller Young's moduluses than the metals to be damped. Consequently, comparatively thick damping coatings are required and only flexural, but not longitudinal or transversal, vibrations can be damped. Further disadvantages are the great dependence on temperature of the elongation and loss modulus and, from the operational point of view, ageing, embrittlement, oil solubility, toxicity and, being a composite material, separation work during recycling. In contrast, Plunkett and Lee [9] have proposed a "metallic" damping layer consisting of layered metal platelets arranged above each other staggered like tiles and connected by means of a thin bonding and damping coating. If a coating of this type is subjected to stress, the elastic force is transmitted from one platelet to the next through shear via the damping coating. The version shown in Fig. 6 exhibits the same mechanism. The linear cross-section distribution of the metal platelets yields a homogeneous stress distribution and - as could be shown in [10] - the maximum possible loss modulus E".

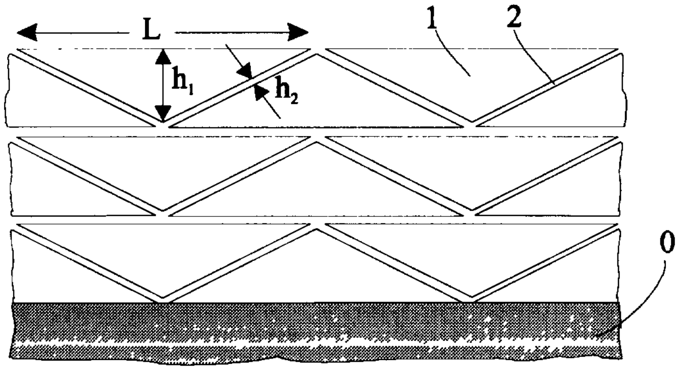

Fig. 5 : Metallic damping layer with a maximum possible loss modulus [10]. Cross section.

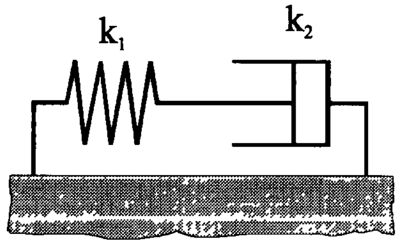

Fig. 6 : Equivalent circuit for the metallic damping layer shown in Fig. 5.

The individual platelet segments 1 feature a length L, a maximum thickness h_1, and consist of a metal, e.g. steel, with Young's modulus $E_1 \approx E'_1$. (The loss modulus E''_1 of steel is negligible). The bonding and damping coating 2 between the segments exhibit a constant thickness h_2 and a purely viscous shear modulus $G_2 = iG''_2$, with a negligible elongation shear modulus G'_2. Materials featuring Newton's or Coulomb's friction have this property. The effect of such a coating combination can be represented by means of the equivalent circuit according to Fig. 6 as the series arrangement of a - formulated mathematically - real and imaginary spring k_1 and k_2. The "real" suspension is formed by the purely elastic metal platelets and amounts to

$$k_1 = E_1 h_1 / 2L \qquad (11).$$

The "imaginary" suspension results from the purely plastic behaviour of intermediate coating 2

$$k_2 = G_2 L / h_2 \qquad (12).$$

It can be shown [10] that the design $k_1 = k_2$ yields maximum damping. Despite the segmenting, the metallic damping layer can be described macroscopically by an integral Young's modulus E

$$E = (E_1 + i E_1) / 2 \qquad \text{for } k_1 = k_2 \; ; \quad L >> h_1, h_2 \qquad (13).$$

In particular, the loss modulus E" determining the damping is

$$E'' = E_1 / 2 \qquad (14).$$

When using steel with $E_1 = E_{steel} = 2.1 \cdot 10^{11}$ Pa, the metallic damping layer has a loss modulus $E'' = 1.05 \cdot 10^{11}$ Pa. It is surprising that this loss modulus depends on the segment material only and not on the properties of the intermediate damping coating. This finding can be demonstrated on the basis of the $k_1 = k_2$ condition. The viscous spring $k_2 \sim L$ can be made to assume the size of the segment spring $k_1 \sim 1 / L$ through the segment length L alone. Consequently, Young's modulus E_1 of the metallic segment material is the "weakest", the limiting parameter.

Segments with a triangular cross section, as shown in Fig. 5, require more manufacturing effort than the plane-parallel platelets. However, versions also exist that combine the advantages of both designs, namely cheap manufacture and a high loss modulus. This is achieved, for instance, by means of a gradient with a thickness of h_2 and the shear modulus G_2.

It is possible to design the metallic damping layer to additionally damp shear stresses. If segment material 1 features the shear modulus $G_1 = G'_1$, the integral shear modulus

$$G = (G_1 + i G_1) / 2 \qquad (15)$$

is obtained with an analogous optimisation condition as in (13). This integral shear modulus has the maximum possible loss shear modulus G"

$$G'' = G_1 / 2 \qquad (16).$$

With steel as the platelet material, $G'' = 0.4 \cdot 10^{11}$ Pa is obtained.

3.3 Damping Flexural Vibrations

Since an integral loss modulus has been determined, respectively, for the segmented damping layer, it can be treated formally like a homogeneous damping layer. It need merely be taken into account that the segment length should be shorter than the wavelengths to be damped. Based on Oberst's derivation [1], the loss factor η_B for flexural vibrations is yielded as

$$\eta_B = (E''d\,(d_0 + d)^2) \,/\, (E_0 d_0^3/3 + E'd\,(d_0 + d)^2) \approx 3\,E''d \,/\, E_0 d_0 \qquad \text{for } d \ll d_0 \quad (17),$$

whereby d_o and E_o represent the thickness, respectively Young's modulus of the plate to be damped, and d resp. E" the corresponding sizes for the coating layer.

Work cycle	Sound level **without** structure-borne noise damping dB(A)	Sound level **with** structure-borne noise damping dB(A)	Level reduction through damping
Riveting	106 - 116	100 - 108	6 - 8
Hand-sawing (skin plate)	89 - 94	88 - 92	1 - 2
Hand-sawing (MP 2)	87 - 93	78 - 82	9 - 11

Fig. 7 : Noise reduction in aircraft construction through damping by means of a "metallic" damping layer [11].

3.4 Damping Torsional Vibrations

Let us take, as a model case, a shaft having the radius r_0 and the shear modulus G_0. A coating layer with thickness d and the complex shear modulus G are applied in full to the wave. The shaft features the polar moment

$$\Theta_0 = \pi\, r_0^4\, G_0 / 4 \qquad (18).$$

The moment Θ of the layer is yielded as

$$\Theta = \pi\,((r_0 + d)^4 - r_0^4)\,(G' + iG'')/4 = \Theta' + i\Theta'' \qquad (19).$$

The loss factor η_T of torsional vibrations in the wave is determined by the relation

$$\eta_T = \Theta'' / (\Theta_0 + \Theta') \qquad (20).$$

On using a damping layer with the loss modulus G", the loss factor η_T of the wave is yielded as

$$\eta_T \approx 4\,G''d \,/\, G_0 r_0 \qquad \text{for } d \ll r_0 \qquad (21).$$

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