Vibration Reduction of a Car Component by a Semi-passive Piezoelectric Concept

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Introduction

Lightweight structural design is often accompanied with a reduction in structural stiffness. This can result in an increase of structural vibrations. Viscoelastic damping layers may be efficiently applied to reduce unwanted vibrations. However, such a measure adds significant mass per area to a structure. Alternatively, a semi-passive concept, realized by piezoceramics, can be employed.

Piezoelectric ceramics, e.g. lead zirconate titanate (PZT), have the ability to transform electrical energy into mechanical energy and vice versa. PZT can be used as a semi-passive device by connecting it to an electrical impedance. Fig. 1 shows a simplified illustration of this concept. As PZT is bonded to a transversely vibrating structure, stresses σ are generated within the piezomaterial. Owing to the direct piezoelectric effect, electrical energy is generated. The stored electric energy is dissipated as Joule heat using resistors as part of the passive electrical network (PEN) with a characteristic admittance Y^{PEN}.

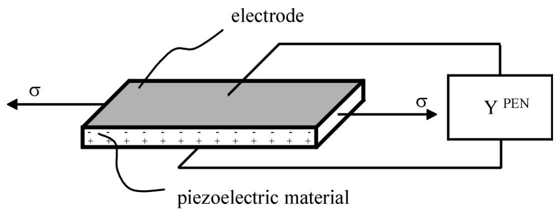

Figure 1: Loaded PZT-element with an electrical network

The model proposed by Hagood and von Flotow [5] is based on a change in stiffness of the piezoelectric material as a result of a purely resistive PEN connected to it. This characteristic of a shunted piezo-element is modeled by means of a complex modulus, similar to that of a viscoelastic material. The optimal frequency dependent loss-factor η^{max} for a resistively shunted piezoelectric element is given in [5] as

$$\eta^{\max} = k_{31}^2 \Big/ 2\sqrt{1-k_{31}^2} \quad at \quad \omega = \frac{1}{RC^S}\sqrt{1-k_{31}^2}, \qquad (1)$$

where k_{31} denotes the characteristic coupling coefficient, R the shunt resistor, C^S is the capacitance of the PZT under constant strain, and ω is the driving frequency. The loss factor in 31-mode can be as high as 6.2 %

The paper at hand investigates the implementation of the semi-passive technique to a hood of a car structure. PZT-elements are applied at positions subjected to large strains. A resistive PEN is implemented and connected to the electrodes of all PZT-elements. By shifting the resistance of the PEN, equation (1), the dissipated energy can be maximized for a given excitation frequency. The modal parameters of the test structure are determined from experimental modal analysis. Cubic splines are used to interpolate the resulting eigenforms. The interpolated eigenforms can than be differentiated to determine the strain distribution and energy ratio. Finally, experiments are conducted to investigate the implemented concept.

Investigated Structure

Semi-passive damping augmentation is employed on a car hood. The investigated structure basically consists of two components: (1) a surface plate with a thickness of 1.5 mm, which is assumed to be constant and (2) a frame, which is attached to the top plate in order to stiffen the hood. Although the hood's geometry is slightly curved, a two dimensional plane structure is a close approximation of the top component. Its dimensions are 1380 mm *x* 1300 mm. Two hinges at the back allow for opening the hood. A lock mechanism is located at the front to fasten the hood. The boundary conditions of the assembled hood are difficult to ascertain. For this reason, an experimental modal analysis is employed to extract modal parameters of the structure under investigation. The experimentally extracted eigenfrequencies, f_i, and modal damping ratios, ζ_i are summarized in Table 1.

No.	1	2	3	4	5	6	7	8	9	10
f_i[Hz]	24.4	45.6	47.5	53.1	80.0	91.9	94.4	108.7	143.1	162.5
ζ_i [%]	1.12	0.51	0.60	0.50	0.91	0.85	0.75	0.90	0.72	0.60

Table 1: Modal parameters of investigated hood

Strain Energy Method

According to the strain energy method [1], damping augmentation η_{aug} of a composed structure (host structure+PZT) can be expressed as

$$\eta_{\text{aug}}^{(r)} = \frac{\sum_i \eta_i U_{\text{i, eff}}^{(r)}}{U_{\text{tot}}^{(r)}}, \qquad (2)$$

where η_i denotes the loss factor of each PZT-element and U_{tot} the total strain energy of the composed structure for the *r*-th mode. The total strain energy is computed as the strain energy of a rectangular plate augmented by the strain energy of all attached PZT-elements. The energy which can be dissipated is proportional to the electrical energy, stored in all PZT-elements. This energy is termed the effective strain energy U_{eff} [1,3]

$$U_{\text{eff}} = \frac{1}{2} EI^* \left| \int_0^l \left(\frac{\partial^2 w}{\partial x^2} \right) \left| \frac{\partial^2 w}{\partial x^2} \right| \mathrm{d}x \right|. \qquad (3)$$

In this equation, w denotes the transverse deflection and EI^* the modified bending stiffness of a PZT-element with length l which is attached to the host structure.

Spline Interpolation

In order to apply the strain energy method, the scatter plot data of the experimentally obtained mode shapes must be interpolated by an analytical expression. In this work the spline interpolation is used, which approximates the eigenmodes in subintervals by means of cubic polynomials. Compared to traditional methods as Lagrange and Hermite interpolation, the advantage of this method is that for a high number of sampling points, no oscillation due to high order polynomials occurs. However, more numerical effort is required to implement the functions valid in the subintervals. The cubic Ansatz function

$$f_i(x_i) = a_i\left(x - x_i\right)^3 + b_i\left(x - x_i\right)^2 + c_i\left(x - x_i\right) + d_i \qquad (4)$$

for the splines is subjected to the following conditions to fulfil C-2 continuity:

$$f_i(x_i) = f_{i+1}(\mathrm{x_i}),\ f'_i(x_i) = f'_{i+1}(x_i) \text{ and } f''_i(x_i) = f''_{i+1}(x_i). \quad (5)$$

After determining the coefficients a_i, b_i, c_i and d_i, a recurrence formula can be computed for the assumed equality of the second derivative at the point of intersection of the subintervals. Assembling all formulas for the complete curve yields a set of *(n-1)* linear equations with *(n+1)* unknowns, where *n* is the number of sampling points. Accordingly, two boundary conditions have to be defined. In this work, the natural spline interpolation is used, i. e.

$$f''_0(x_0) = f''_n(x_n) = 0 \,. \qquad (6)$$

This corresponds to freely suspended edges of the hood. Finally, the first derivatives can be calculated by solving the set of linear equations. Introducing these results into the Ansatz-function, the one-dimensional spline interpolation is carried out.

For two dimensional interpolation, the 2-D function $X(u,v)$ is interpolated by a tensor product, which requires a rectangular distribution of all sampling points. The bi–cubic spline interpolation can be formulated as:

$$X_{ij}(u,v) = \sum_{k=0}^{3}\sum_{l=0}^{3} A_{ijkl}(u-u_i)^k (v-v_j)^l \qquad (7)$$

and consists of the coefficient matrix A_{ijkl} and the Ansatz-functions in *u* and *v* direction. Compared to the 1-D case, A_{ijkl} in equation (7) is extended by the mixed derivative with respect to *u* and *v*.

The interpolation is implemented in Matlab. Comparing interpolated and real mode shapes reveals consistent results as shown for the eigenmode at 80 Hz in figure 2.

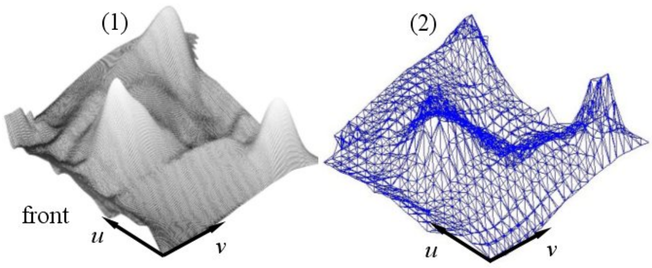

Figure 2: Interpolated (1) and EMA-extracted (2) mode shape at 80 Hz

Using the interpolated mode shape functions, the strain energy and effective strain energy can be calculated. Finally, the positions of the PZT-elements are chosen which maximize the energy ration of mode number 3 and 5 (see Table 1). In this study 19 PZT-elements are bonded onto the structure.

Application onto the Structure

For practical implementation of the PZT-elements onto the structure a special epoxy resin is used. Due to the curved hood structure, a relatively thick epoxy layer (~ 100 μm) is applied. Normal stress is applied to the PZTs during the curing phase thereby forcing total contact between the curved hood surface and the flat PZT surfaces.

Reduction of vibrations

The final measurements are carried out for the most significant eigenmode at 80 Hz. Since the calculated strain energy fraction of the applied PZT is low and limited damping augmentation is expected, the sensors are positioned at the maxima of this mode shape. Before focusing on the increased damping, the change of the mechanical properties is investigated. Although mass augmentation is negligible, a change in stiffness due to the bonded PZTs can be identified by a shift of this eigenfrequency from 80 to 83 Hz.

The performance of the semi-passive concept is evaluated by comparing the open-circuit modal damping ratio to that resulting from shunting the PZTs. Applying the Peak Amplitude Method the results are computed using the accerlerance – frequency plot. Figure 3 shows the plot for sensor 2, positioned at the left maxima in front of the passenger cabin.

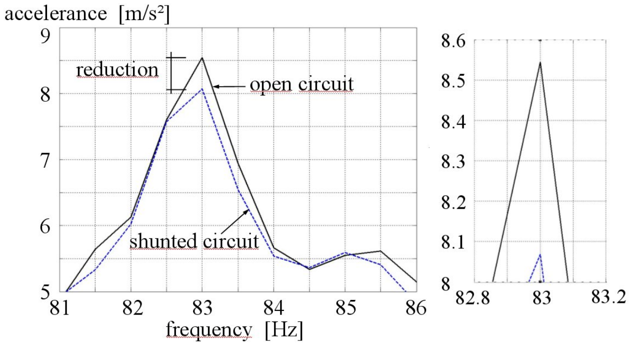

Figure 3: Accerlerance vs. frequency

The local damping ratios at two sensor locations (where vibration is maximum), for open- and shunt-circuit operation, are compared in Table 2. At resonance the relation $\eta = 2\zeta$ is valid. For sensor 2 and 6 an increase in modal damping ratio of 8.9% and 6.8%, respectively, can be observed. The theoretical expected increase $\Delta\zeta_{rel,th.}$ (see Table 2), however, is well overestimated. This is a result of the simple Kirchhoff model applied in equation (2). A finite element model of the hood and the bonding layer should compensate this differences.

Sensor No.	ζ_{open} [%]	$\zeta_{shunted}$ [%]	$\Delta\zeta_{rel}$ [%]	$\Delta\zeta_{rel,\,th.}$ [%]
2 (left)	1.175	1.280	8.936	54.2
6 (right)	1.170	1.250	6.837	54.4

Table 2: Increase in modal damping ratio

Finally, the increase in mass is analyzed. Computing the mass ratio of total PZT-mass to the mass of the hood yields an increase of 1.3%, which is negligible compared to that resulting from methods where viscoelastic material is applied.

Conclusions

A semi-passive concept has been successfully applied to a car structure to reduce structural vibrations. Compared to classical concepts based on viscoelastic material, the increase in mass is insignificant. The applied concept results in a local increase in damping. However, to increase damping of the entire hood a finite element model of the investigated structure is essential to optimize the layout of the PZT-elements. This will also incorporate any stiffening effects due to the applied PZTs. Furthermore, the concept can be more efficient by using PZT-elements to a greater extend. For broadband reduction, this method can be extended by an adaptive PEN [4] or, alternatively, by a combination of PZT-elements and viscoelastic layers applied to the structure.

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