Oberst Beam Excitation Using Piezo Electric Actuators

Jean-Luc Wojtowicki
Acoustics and Vibration Group (G.A.U.S.), Université de Sherbrooke, Sherbrooke, Quebec J1K 2R1

1. Introduction

The «Oberst beam» is a classical method for the characterization of damping material based on a multilayer cantilever beam (base beam + one or two layers of other materials). As the base beam is made of a rigid and lightly damped material (steel, aluminum), the most critical aspect of this method is to properly excite the beam without adding weight or damping. So, exciting the beam with a shaker is not recommended because of the added mass. Alternative solutions are suggested in [1], electro-magnetic non-contacting transducer (tachometer pick-up, for example) can provide a good excitation but it is limited to ferro-magnetic materials. As aluminum and stainless-steel are widely used for the base beam, a small bits of magnetic material must be fastened adhesively to achieve specimen excitation. This method creates two other problems. The first one is the difficulty to properly measure the excitation force and the second one is the added damping due to the magnetic materials bits in the case of non-magnetic base beam.

However, the measurement of the motion of the beam can be easily made using a non-contact transducer (a laser vibrometer for example).

2. Presentation of the method

2.1. Piezo electric actuators excitation

When two piezo-electric actuators are placed facing each other on both sides of a structure and cabled out of phase, they create a bending moment which is proportional to the applied voltage. This applied voltage can directly be used to calculate the velocity *vs* force transfer functions (FRF) used in the determination of modal parameters.

In the case of the Oberst beam, two piezo electric actuators have been glued near the root of the beam, where the displacements are small to lower energy loss due to added damping (Figure 1).

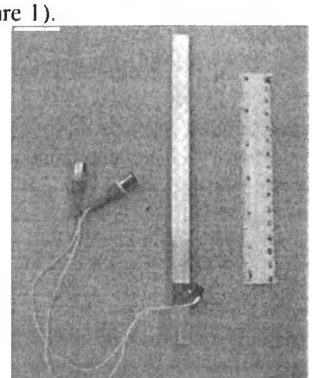

Fig. 1. Oberst beam excited using two piezo electric actuators

The beam used in this study has the following dimensions: length (l), 8''; width (b), ½'', thickness (h), 0.100'' and the material is aluminum. The piezo electric actuators are ¾'' long (l_{act}).

2.2. Added damping

The first problem was the matter of the added damping due to the gluing of the piezo electric actuators. As they are made out of ceramics, the intrinsic damping may be high. So, the overall damping of the beam has been measured before and after the gluing of the actuators (Table 1).

Mode order	Frequency (Hz)	Damping without actuators	Damping with actuators
1	47.1	0.09%	0.11%
2	295.8	0.14%	0.13%
3	827.2	0.18%	0.19%
4	1616.3	0.06%	0.13%
5	2668.5	0.07%	0.21%

Table 1. Base beam damping with and without piezo-electric actuators

Some damping has been added to the base beam for the 4th and 5th modes, but for the first three modes, there is no significant added damping. However, the overall damping remains low for all modes and should not interfere with the measurement of highly damped material.

2.3. Non covered length

A set of theoretical equations has been developed to determine the damping of each layer of a composite cantilever beam [2] under the assumption that the beam is entirely covered. In the present case it is not possible to cover the area occupied by the actuators with the material under test.

The structural damping is included in the complex part of the Young modulus and can be expressed as follow.

$$\hat{E} = E \cdot (1 + j \cdot \eta) \qquad (1)$$

The *Euler-Bernouilli* equation (2) for thin beam shows that the dissipated energy is included in the first term of and is proportional with both the fourth derivative of the displacement and the frequency. The fourth derivative of the displacement is directly equal to the displacement multiplied by a modal constant.

$$E.I \frac{\partial^4 w}{\partial x^4}(x,t) + \rho.A \frac{\partial^2 w}{\partial t^2}(x,t) = f(x,t) \qquad (2)$$

An estimation of the error on the measurement of the damping may be done using two indicators. The first one (ε_f) (3.1) is the ratio of the frequency shift of modes between the entirely covered beam and the partially covered beam. The second one (ε_w) (3.2) is the ratio of the area under modal shape curve between 0 and l for the entirely covered beam and between l_{act} and l for the entirely covered beam.

$$\varepsilon_f = \left| 1 - (f_n^{covered} / f_n^{partial})^2 \right| \qquad (3.1)$$

$$\varepsilon_w = \left| 1 - \int_{l_{act}}^{l} w(x).dx \Big/ \int_{0}^{l} w(x).dx \right| \qquad (3.2)$$

Table 2 gives the estimated error using the beam described above and a damping material stuck on one side of the beam (density: 1740 kg/m³, thickness: 1mm).

Mode order	Error due to frequency shift ε_f	Error due to non covered area ε_w
1	0.001%	0.09%
2	0.004%	0.47%
3	0.002%	1.19%
4	0.001%	2.12%
5	0.010%	3.20%

Table 2. Estimated error on damping

As expected, the error increases as the mode order increases only for the estimation of the non-dissipated energy in the root area, but the effect of frequency shift is negligible in the case of the tested material.
The maximum estimated error is about 3% on the fifth mode which is reasonable in the case of damping measurement or the modal parameter estimation.

3. Measurement of damping

3.1. Composite damping

Some methods are available to measure the modal damping, the most used is the half-power bandwidth which can be very imprecise when experimental curves are directly used, curve fitting methods are preferred. In this study, a semi-direct algorithm is proposed. The calculation of $F(\omega)$, the beam displacement divided by the bending moment can be written as in equation (4).

$$F(\omega) = \sum_{n=1}^{\infty} K_n . \frac{e^{(-\omega_n / \omega)}}{\omega - \omega_n} \qquad (4)$$

K_n is the modal amplitude and is a function of force amplitude and the measurement point position. Near the ith mode, the contribution of other modes can be neglected, the function $F(\omega)$ and its first derivative can be written as in equations (5.1) and (5.2).

$$F(\omega) \approx K_i . \frac{e^{(-\omega_i / \omega)}}{\omega - \omega_i} \qquad (5.1)$$

$$\frac{\partial F(\omega)}{\partial \omega} \approx F(\omega).(\frac{1}{\omega} - \frac{1}{\omega - \omega_i}) \qquad (5.2)$$

If A_m and $\tilde{A}_m$ are the experimental values of FRF and first derivative respectively near the ith mode, equations (5.1) and (5.2) can be combined and give equation (6.1) and (6.2).

$$\omega_i = \frac{\tilde{A}_m . \omega^2}{\tilde{A}_m . \omega - 1} \qquad (6.1)$$

$$K_i = A_m . \frac{\omega - \omega_i}{e^{-\omega_i / \omega}} \qquad (6.2)$$

Equation (6.1) and (6.2) give values than can be calculated from a certain number of frequency points near the resonance and averaged for a more precise estimation of modal parameters. Figure 2 represents the experimental and optimized curves for the two first modes of the damped beam.

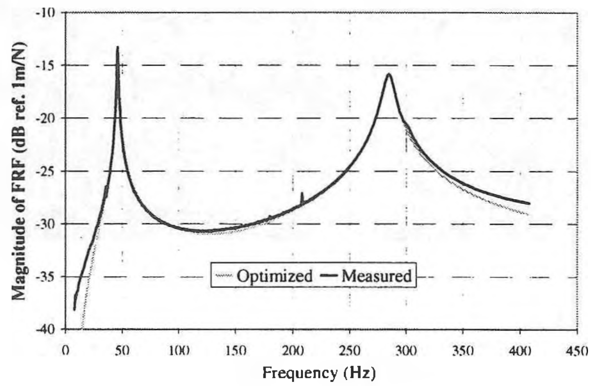

Fig. 2. Experimental and optimized FRF for the damped beam

This method always gives a good agreement as long as the experimental datas are properly measured.

3.2 Damping of tested material

Finally, the damping properties of the tested material have been directly calculated using formulas given in [1]. Table 3 gives the results for the Young modulus and damping ratio calculated using composite damping and modal frequencies estimated using the method developed in this study.

Mode order	Damping (%)	Young Modulus (MPa)
1	83.3	660
2	97.9	930
3	91.3	1520
4	ASTM E756 validity criteria not passed	
5	$(f_c/f_n)^2.(1+D.T)<1.01$	

Table 3. Estimated error on damping

4. Conclusion

The excitation of a cantilever beam using piezo electric actuator provides a good alternative to other excitation methods. The main advantage is its ease of settlement and of use, because it requires few adjustments and precaution. Moreover, the errors in the estimation of damping are small. The proposed algorithm for the calculation of modal parameters has been specially developed to use a more precise estimation of modal parameters than a simple half-power bandwidth method without using an advanced modal testing software.

5. References

[1]: ASTM E756 – 98, *Standard Test Method for Measuring Vibration-Damping Properties of Materials*, American Society for Testing and Materials

[2]: A. D. Nashif, D.I.G. Jones, J.P. Henderson, *Vibration Damping*, John Wiley & Sons, 1985