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A NOVEL METHOD FOR DYNAMIC CHARACTERIZATION OF POLYMERIC VIBRATION DAMPERS

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Polymeric vibration dampers are very important devices used in many fields for vibration isolation and damping. These devices are based upon a low stiffness and high damping connection between a vibrating support and the item to be fastened. Whenever the connection is assured by a polymeric element, it is mandatory to be aware that stiffness and damping change with the excitation frequency as well as with temperature. Characterization of stiffness and damping of the polymeric element in terms of storage modulus and loss modulus is commonly carried out at low frequency by means of a Dynamic Mechanical Analysis (DMA), nonetheless this approach cannot be applied at higher frequency. In the present paper, a novel experimental approach for estimating the frequency dependent storage modulus and loss modulus in a polymeric vibration damper is presented. The proposed method is based on a direct measurement of the energy loss in hysteretic cycles, and it is suitable for simple implementation using common instruments for vibration measurement.

Keywords: polymeric vibration damper, loss modulus.

1. Introduction

Vibration control is a relevant design requirement for several industrial applications, often achieved by means of passive damping technologies involving viscoelastic materials. Among them, elastomeric vibration isolators are widely adopted, consisting of a low stiffness and high damping connection between a vibrating support and the item to be fastened. Due to polymeric elements, to characterize the dynamic behavior of a damper it is of paramount importance estimating stiffness and damping changes with excitation frequency and temperature.

Most commonly, experimental results presented in the literature deal with measurements of the viscoelastic properties of beam–like specimens. Although being dealt with several papers, there is still a lack of well–designed experimental work dealing with the measurement of damping properties of polymeric vibration isolators. In this case the characterization is commonly carried out at low frequency by means of a Dynamic Mechanical Analysis (DMA), nonetheless this approach cannot be applied at higher frequency

In the present study, the polymeric vibration damper is modelled as a single degree of freedom system (SDOF), using a shaker as source of vibration. The viscoelastic properties of the system are described by its dynamic stiffness, a complex function of the frequency, whose real (storage modulus) and imaginary (loss modulus) parts are experimentally estimated. The proposed method is based on a direct measurement of the energy loss in hysteretic cycles, and it is suitable for simple implementation using common instruments for vibration measurement.

2. Testing equipment

The test rig used for carrying out damping measurement is shown in Figure 2. It is composed of a large steel mas and a Bruel & Kjear 4808 electrodynamic shaker. Both the mass and the shaker are suspended with low stiffness supports. The shaker exerts a force on an aluminium disk which is connected to the large mass by means of the test specimen (*i.e.* the polymeric damper shown in Figure 1). Instrumentation consists of three accelerometers and a dynamic load cell. Measured points are:

- a_0: mono-axial accelerometer on the shaker;
- a_1: mono-axial micro-accelerometer on the right side of the damper;
- F_1: dynamic load cell on the right side of the damper;
- a_2: mono-axial accelerometer on the steel mass.

The shaker is connected to the aluminium disk by means of a stinger, in order to avoid lateral loading of the load cell and of the test specimen.

Figure 1: Polymeric vibration damper.

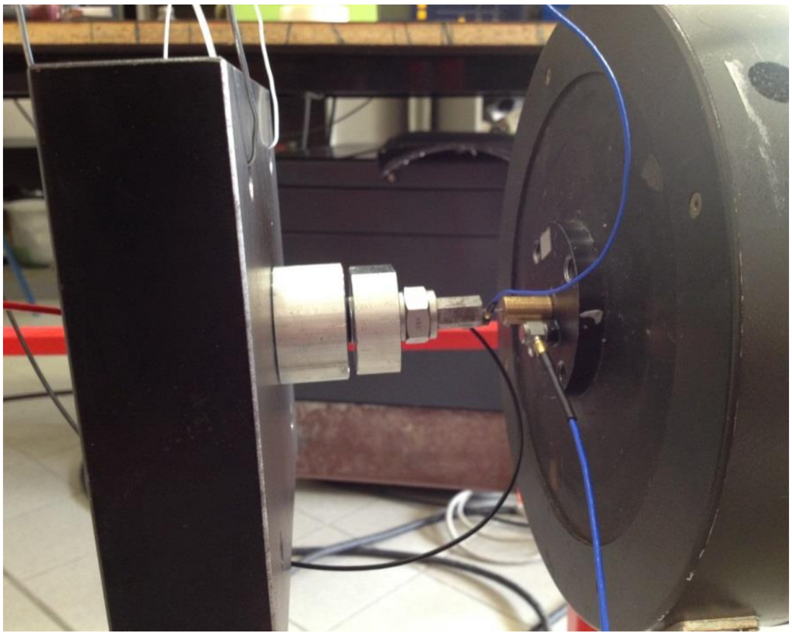

Figure 2: Electrodynamic shaker experimental setup.

3. Measurement technique

In order to characterize damping in the rubber damper, a cyclic loading is applied by means of the shaker, and the values of forces and accelerations are measured during 50 hysteresis loops. In the following, characterization in terms of viscous damping and of complex modulus will be shown.

3.1 Hysteresis loop method

The excitation signal is a stepped sine with varying frequency, from 150 to 1050 Hz (logarithmically spaced). For each frequency, a sufficient amount of periods are neglected in order to avoid transient results and 50 periods are stored. The acceleration signal is integrated numerically in order to obtain a velocity value; the relative velocity v is measured as the difference between mass velocity v_2 and disk velocity v_1. Thanks to measured force F_1, the work lost in a cycle can be estimated as in (1), and the value of the equivalent viscous damping is computed by eq. (2).

$$\mathcal{W}_d = \int_0^{\frac{2\pi}{\omega}} F(t)v(t)\,dt \tag{1}$$

$$c_{eq} = \frac{\int_0^{\frac{2\pi}{\omega}} F(t)v(t)\,dt}{\int_0^{\frac{2\pi}{\omega}} v^2(t)\,dt} \tag{2}$$

It is worthwhile noting that eq. (2) involves measured signals only; integrals are performed by Simpson's rule.

3.2 Analysis of experimental data

Figures 3 and 4 show point out that the frequency range in which the proposed method is reliable. At low frequency, there is interaction between the stinger and the system, so that the response in nonlinear. Conversely, at frequencies over 500 Hz, the response is linear, and a clean hysteresis curve can be observed in Figure 3(a).

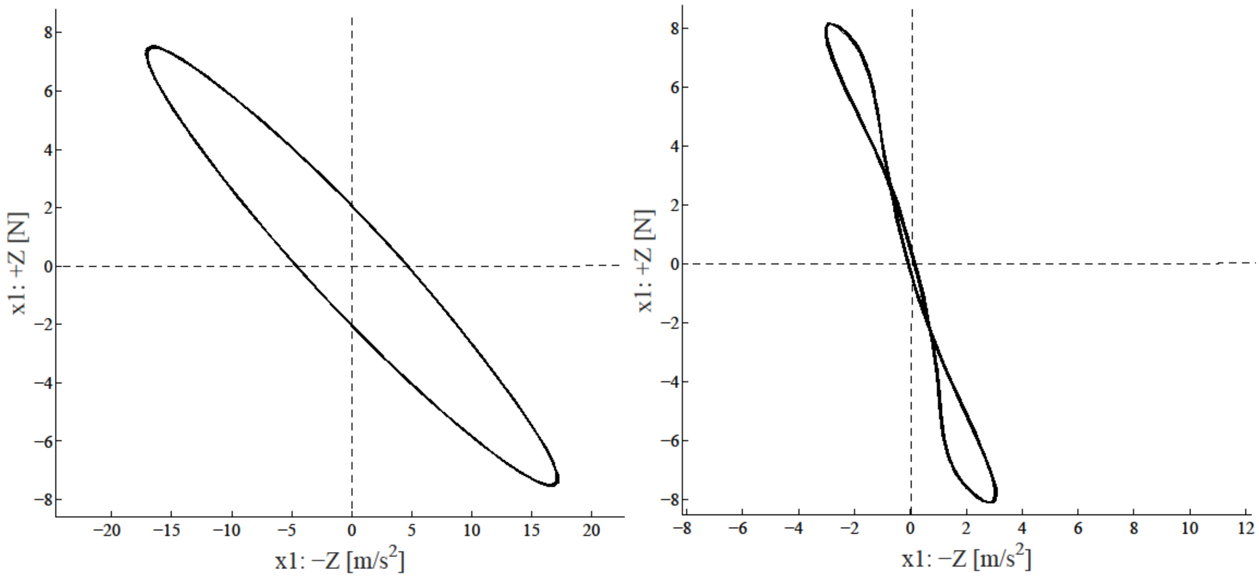

Figure 3: Hysteresis loops, left 583.0 Hz, right 320.4 Hz.

In Figure 4, results of three different runs are shown: the three runs differ for the controlled variable (x_0 or x_1) and for the length of the stinger. Despite being characterized by three very different levels of excitation (see Figure 4(a)), the measured value of the viscous damping is the same for the three measurements above 500 Hz (Figure 5(b)). Obviously, the measured c for run 3, is more noisy due to the worse signal to noise ratio.

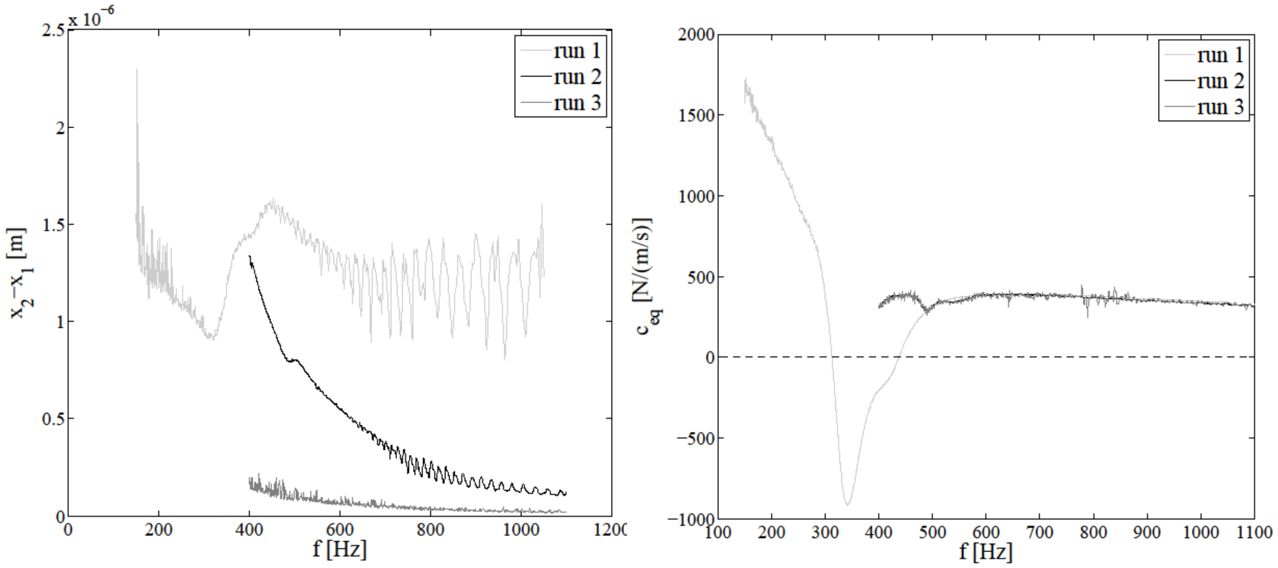

Figure 4: Relative displacement amplitude (left) and Equivalent viscous damping vs frequency (right).

Thanks to performed measurements, it is possible to compute the frequency dependence of the real stiffness k and of the loss factor η, which are needed to characterize the polymeric damper. Figure 5(a) shows that real stiffness is increased from 500 Hz to 1000 Hz by 40 %. The average error between run 1 and run 2 is 2% (note that the stinger length has been changed). The value of the loss factor η is shown in Figure 5(b). Above 500 Hz, the measured values are in agreement with the expected behaviour of a polymer where real stiffness is increasing. At 1000 Hz the loss factor is 0.26, which is equivalent to a dimensionless damping of 0.13.

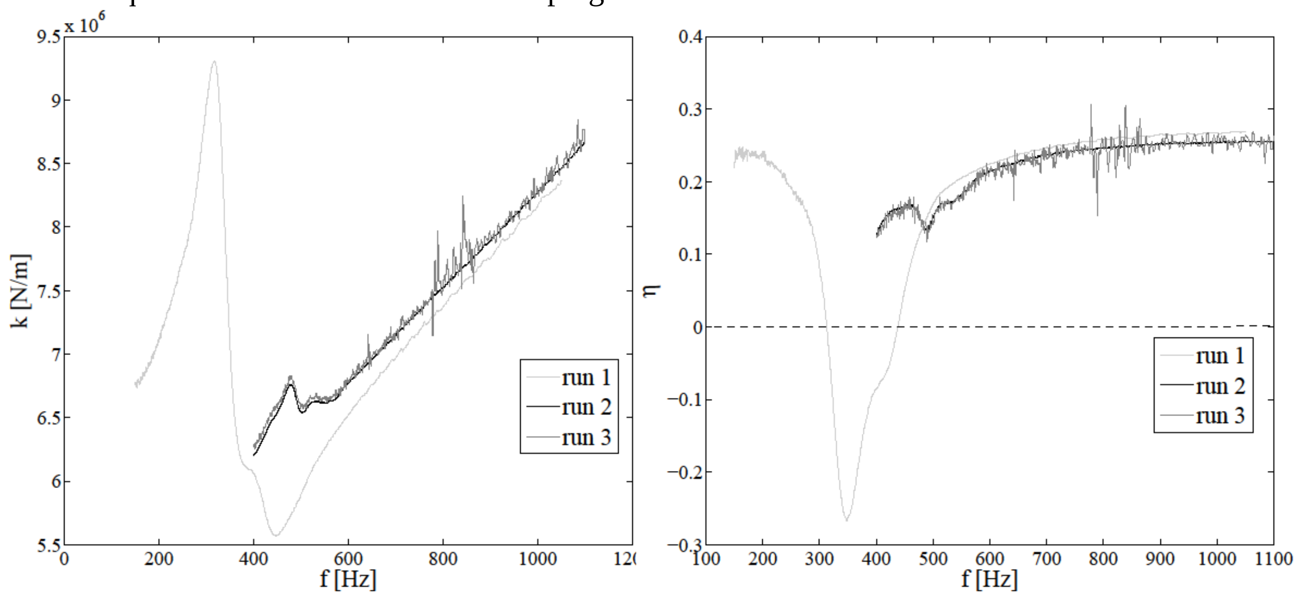

Figure 5: Real (left) and Imaginary (right) parts of dynamic stiffness vs frequency.

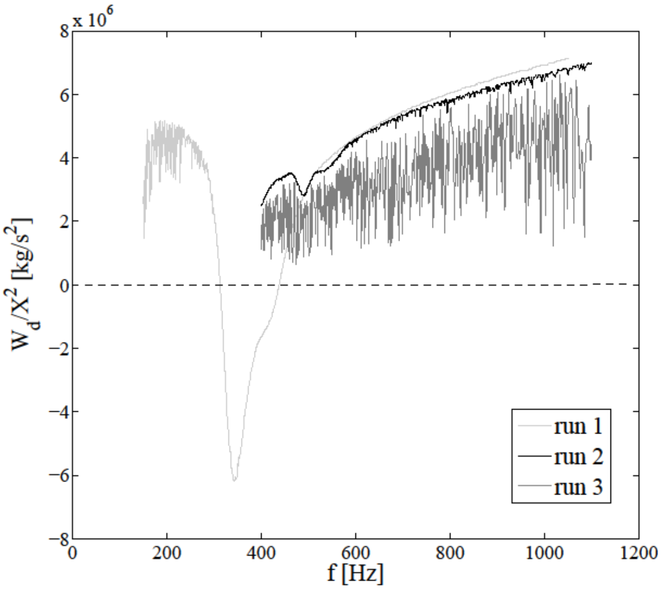

Figure 6: Normalized work per cycle (W_d/x^2) vs frequency.

An additional investigation has been aimed to clarify the work/frequency dependency. Figure 6 shows that lost work per cycle normalized over the square oscillation amplitude is increasing less than linearly with frequency. Obviously, values below 500 Hz are not reliable and are to be neglected.

Figure 7 shows the hysteresis loop on the force/displacement diagram. This picture is obtained by numerically integrating the measured accelerations twice. The shown measurements are clean, and the 50 cycles are almost overlapping, therefore the whole procedure of measurement and data processing is robust and repeatable.

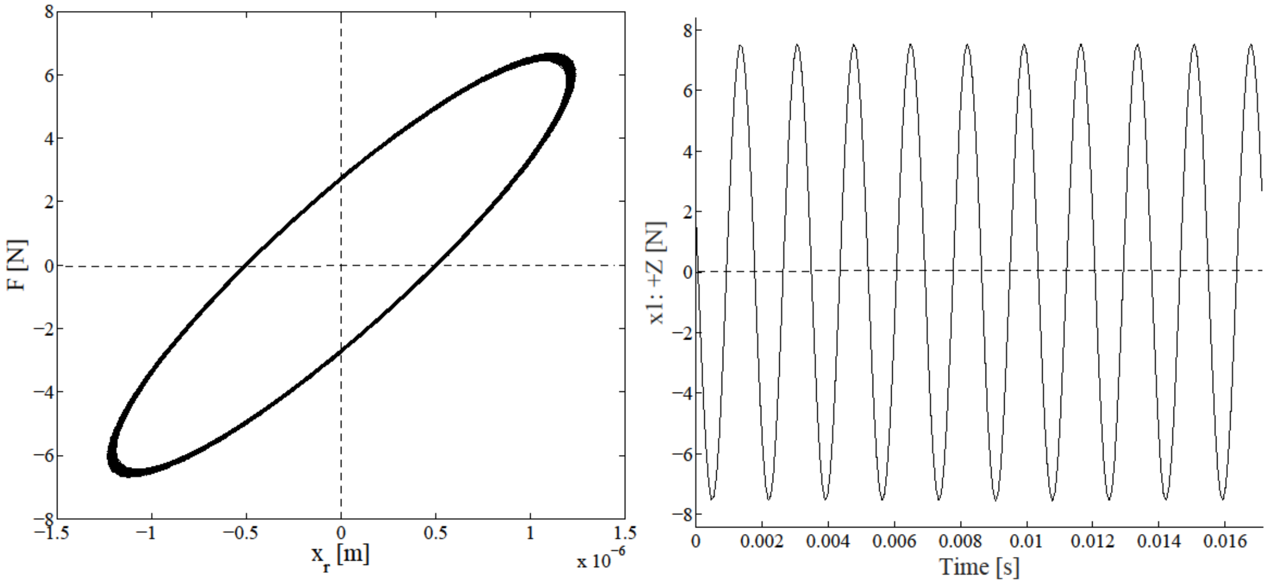

Figure 7: Work lost per cycle at 1050 Hz (left) and related time history (right).

4. Identification technique

A proper choice of the material constitutive model plays a fundamental role in the identification of a mechanical structural system exhibiting internal dissipation.

4.1 Constitutive models

With reference to Fig. 8, the Fractional Zener constitutive model is considered, which generalizes the standard Zener model with substitution of a Newton element (damping coefficient c) with a Scott–Blair element (fractional damping coefficient c_f). The time–domain constitutive equation of the Fractional Zener model reads:

$$\left[1+\frac{c_f}{k_1+k_2}\frac{d^\alpha}{dt^\alpha}\right]f(t)=\frac{k_2}{k_1+k_2}\left[k_1+c_f\frac{d^\alpha}{dt^\alpha}\right]x(t) \tag{4}$$

yielding the following expression of the dynamic stiffness:

$$k(\omega)=k_0\frac{1+(i\omega\tau_\varepsilon)^\alpha}{1+(i\omega\tau_\sigma)^\alpha} \tag{5}$$

where $\alpha \in [0, 1]$ is the non–integer (fractional) derivative order, k_0 is the static stiffness, τ_ε and τ_σ are the creep retardation time and the relaxation time respectively, defined according to:

$$k_0=\frac{k_1k_2}{k_1+k_2},\quad \tau_\varepsilon^\alpha=\frac{c_f}{k_1},\quad \tau_\sigma^\alpha=\frac{c_f}{k_1+k_2} \tag{6}$$

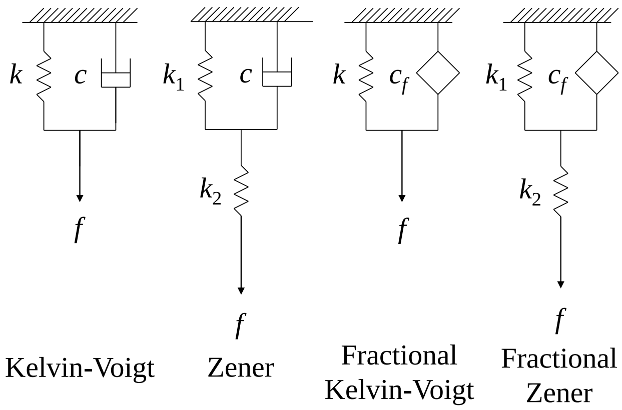

Figure 8: schematic of basic constitutive models.

Consequently, the storage modulus and the loss modulus take the form:

$$\begin{cases}\mathrm{Re}[k(\omega)]=k_0\dfrac{1+c_\alpha\omega^\alpha(\tau_\varepsilon^\alpha+\tau_\sigma^\alpha)+\omega^{2\alpha}\tau_\varepsilon^\alpha\tau_\sigma^\alpha}{1+2c_\alpha\omega^\alpha\tau_\sigma^\alpha+\omega^{2\alpha}\tau_\sigma^{2\alpha}}\\ \mathrm{Im}[k(\omega)]=k_0\dfrac{s_\alpha\omega^\alpha(\tau_\varepsilon^\alpha-\tau_\sigma^\alpha)}{1+2c_\alpha\omega^\alpha\tau_\sigma^\alpha+\omega^{2\alpha}\tau_\sigma^{2\alpha}}\end{cases},\quad c_\alpha=\cos\left(\frac{\pi}{2}\alpha\right),\quad s_\alpha=\sin\left(\frac{\pi}{2}\alpha\right) \tag{7}$$

with:

$$\lim_{\omega\to 0}\mathrm{Re}[k(\omega)]=k_0,\quad \lim_{\omega\to\infty}\mathrm{Re}[k(\omega)]=k_2,\quad \lim_{\omega\to 0}\mathrm{Im}[k(\omega)]=\lim_{\omega\to\infty}\mathrm{Im}[k(\omega)]=0 \tag{8}$$

The energy loss per cycle of oscillation (with maximum strain amplitude x_0 due to a steady–state harmonic loading) is proportional to the loss modulus:

$$W(\omega)=\pi x_0^2\,\mathrm{Im}[k(\omega)] \tag{9}$$

while the maximum strain potential energy is proportional to the storage modulus:

$$V(\omega) = \frac{1}{2} x_0^2 \operatorname{Re}[k(\omega)] \tag{10}$$

hence the loss factor:

$$\eta(\omega) = \frac{\operatorname{Im}[k(\omega)]}{\operatorname{Re}[k(\omega)]} = \frac{W(\omega)}{2\pi V(\omega)} \tag{11}$$

gives a relative measure of the energy loss per cycle of oscillation.

5. Conclusions

In the present work a test rig for characterizing dynamic stiffness and damping of a polymeric damper is presented. While standard quasi-static procedures (DMA) provide information at low frequency, the current method is capable to provide data at frequencies in the range 500-1000 Hz, which are of high interest in the automotive field. Conversely, the method is not reliable at low frequencies, due to the occurrence of instabilities in the connecting stinger.

A great advantage of the proposed method is that it can be implemented using standard vibration testing equipment, such as a small shaker, mono-axial accelerometers and a dynamic load cell.

Measured real stiffness and loss factor curves are in agreement with the literature, and therefore they can used to fit a rheological model of the polymeric material, which will be useful for modelling purposes. In the present work, an overview of constitutive models have been proposed; in the next future, dynamically measured properties will be mixed with standard low frequency measurements in order to characterize the damper in terms of a minimum number of parameters.

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