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Procedures for damping properties determination in metal foams to improve FEM modeling

Massimo Goletti^{a,*}, Valerio Mussi^a, Andrea Rossi^a, Michele Monno^b^a*Consorzio MUSP, via Tirotti, 9, I-29122, ITALY*^b*Dipartimento di Meccanica, Politecnico di Milano, Via La Masa 1, 20156 Milano, ITALY*

Abstract

The aim of this work is to review the available procedures suitable for the determination of damping parameters in literature and standards for porous and dense materials and to evaluate their applicability to metal foams.

Preliminary experimental setup and characterization data obtained with the selected procedures are presented and compared with representative FEM models. The experimental dynamic measurement on a test structure is compared with the simulated ones in order to validate the chosen procedure.

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1. Introduction

Aluminum foams are used as filling materials in several applications that take advantages from one or more characteristics of the material. If the machine tool field is taken into account, the vibrational behavior of the structure is of great importance. Generally, Finite Elements Modeling (FEM) software are used to design the whole machine (basement, ram, spindle, etc.) with the aim to obtain the best vibrational behavior for the selected application, the desired material savings, etc. To take the virtual model as a starting point for the design phase, the dynamic material properties (i.e. Young's modulus, damping ratio, etc.) must be known.

* Corresponding author. Tel.: 0039-0523-623190; fax: 0039-0523-645268.
E-mail address: massimo.goletti@musp.it

While well-known standards have been developed for the measurement of the Young's modulus of metal foams, the determination of damping properties for such materials lacks a commonly accepted procedure suitable for a direct application in FEM software. In such software, there are several ways to take into account for damping. The most commonly used are: structural damping, composite modal damping and Rayleigh damping. These coefficients have different significance so a proper selection must be done.

The determination of damping of aluminum foam has been analyzed in few studies with quite different results. In Banhart (1996) a specific experimental setup was prepared in order to minimize unwanted contribution to the overall measured damping. In particular the experimental apparatus was placed in a vacuum chamber and contactless excitation and sensors were used. The specimen geometry was also optimized and the first eigenmode discarded as it produces high levels of friction in the mount. A mean result of 0.3% was found for the loss factor of the sample over frequency.

In Tu and Wang (2010), a different setup was used and a resulting loss factor greater than 100% was found. The samples were mounted on a vibrating table that allowed applying a displacement to the constraint. The acceleration transmissibility between free-end and constraint was used to determine damping for several eigenmodes with a least square method.

In Golovin and Sinning (2003), different mechanisms of internal losses (like thermoelastic effects, dislocations, microcracks, etc.) were considered in the analysis of damping of mechanical vibrations. In particular, the aerodynamic losses can double the intrinsic damping of the beam when the pressure increases from 10^{-4} to 10^5 Pa. This is true at low amplitudes of deformation ($\epsilon = 10^{-5}$). At higher vibration amplitudes, the intrinsic losses are more important which make the relative contribution of the aerodynamic losses much smaller and sometimes negligible.

The ASTM Standard (2010) called "Standard Test Method for Measuring Vibration-Damping Properties of Materials" describe a procedure for the determination of damping and Young's modulus of specimens with the same geometry described in (Banhart (1996)). In this standard it is suggested to use the half power bandwidth method (also described in Ewins (2000)) to compute the loss factor for a specific eigenmode

2. Sources of Variability in Damping Estimation

For sake of simplicity here are reported the relationships between Q-factor Q , loss factor η , damping ratio ζ , exponential decay rate α and the specific loss capacity ψ . The last is related with the energy absorption ΔW and the maximum elastic stored energy W , parameters that can be found in the theory of *internal friction* reported in Blanter et al. (2007) (ω_n is the natural frequency):

$$\zeta = \frac{\eta}{2} = \frac{1}{2Q} = \frac{\alpha}{\omega_n} \quad (1)$$

$$\frac{1}{Q} = \frac{\Delta W}{2\pi W} = \frac{\psi}{2\pi} \quad (2)$$

It is clear from literature that both the experimental setup and the identification techniques are of great importance in the determination of the results as the estimated loss factor ranges from 0.3% to 100%. In most cases, the damping identification is performed with the so-called half power bandwidth technique that can lead to not negligible errors in the damping estimation if, as an example, the frequency resolution is poor or other modes influence is important.

In this paper, the SDOF (Single Degree Of Freedom, it is the simplest vibratory system that is described by a single mass connected to the ground by means of a spring and a dashpot) methods used are three: half power bandwidth, circle fit and line fit (all described in Ewins (2000)) depending on the shape of the studied functions. These methods are applied in a recursive procedure to optimize curve fitting with a MDOF (Multiple Degree Of Freedom) equation:

$$y(\omega) = \sum_{i=1}^N \frac{1}{m_i} \frac{1}{(j\omega)^2 + 2\zeta_i \omega_{n,i} j\omega + \omega_{n,i}^2} \quad (3)$$

With m_i modal mass, ζ_i damping ratio and $\omega_{n,i}$ natural frequency for the i^{th} eigenmode and j the imaginary unit.

A correct estimation of the damping properties of aluminum foams has to take into account several sources of uncertainty other than the two just exposed: the variability between specimens and the fixturing procedure.

In order to analyze the former, three specimens are realized as depicted in Figure 1. The shape of the specimen is designed considering the guidelines in Banhart (1996) and ASTM Standard (2010) and thus with a mounting head. The whole specimen is cut from one block of aluminum foam with a square cross-section of 26 mm side, an overall length of 540 mm and a head of 40 mm length and 80 mm height. Each specimen is composed of two beams that are both considered in the measures (i.e. three specimens x two beams = six beams).

3. Experimental Setup

In this paper, the attention is focused on two industrial-level experimental setups and a damping identification technique based on different SDOF methods to compute modal parameters. The first experimental setup is similar to the one proposed in Tu and Wang (2010): a PCB 2075E modal shaker is used to apply a vibration to a constrained aluminum foam beam and the accelerance transmissibility in Ewins (2000) between free end and constraint is calculated as the mean of ten repetition of the same test (top of Figure 1).

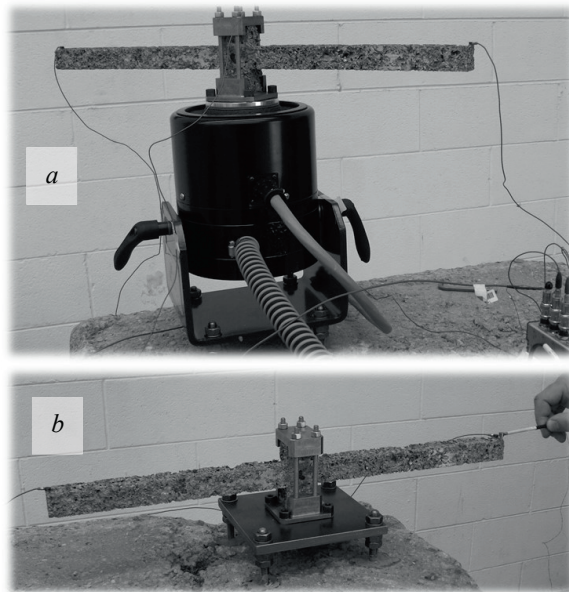


Figure 1: The two experimental setups used. (a) Modal shaker and (b) Instrumented hammer.

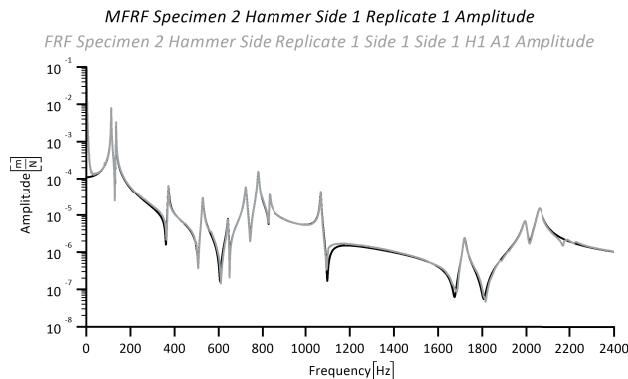


Figure 2: Comparison between experimental (Gray) and identified (Black) FRFs for specimen 2, beam 1 and replicate 1 (hammer test).

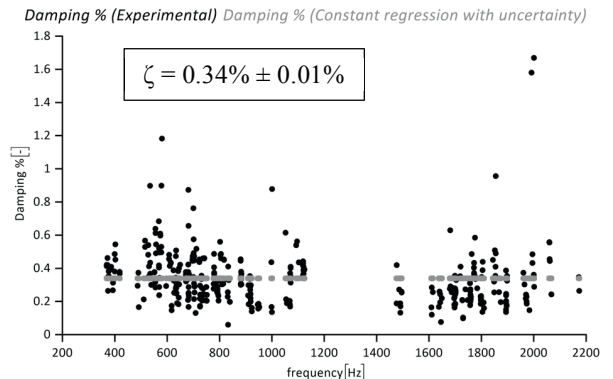


Figure 3: Damping as a function of eigen-frequency for all the specimens, sides, replicates and solicitation types.

Table 1. Specimens data.

Specimen	Density	Mass	E (Static)	E (ASTM)
Unit	Kg/m ³	g	MPa	MPa
1	318	131.89	938	981
2	342	140.57	1187	1376
3	327	134.87	938	957

The second setup is realized with the same clamping device but ground constrained in order to determine the stiffness of the various eigenmodes. An instrumented hammer (PCB 086E80) is used to excite the free end of the specimen (bottom of Figure 1) allowing to determine all the modal parameters with suitable identification techniques.

All the accelerometers used in the free end of the structure are PCB 352C23 whereas the one placed at the constraint in setup *a*. is a PCB 352A24 with higher sensitivity.

The clamping system consists of two steel plates properly designed to hold the head of the specimen. In the experimental trials, the specimen is mounted with a deformation of the 0.5% to ensure a working point in the linear response region.

4. FEM Model

In order to replicate the damping behavior of metal foam in FEM analysis, a virtual Abaqus V6.10 model is prepared. This contains the global geometry of the specimen and of the clamping solution. The constraint was imposed fixing the degrees of freedom on the top and bottom surfaces of the beam head root section as in the experimental setup (see Figure 1 b).

The virtual analysis is composed of two steps: a first frequency extraction one and a subsequent modal dynamic time dependent one. In the former the eigenmodes are extracted (no damping information needed). In the latter, a time-dependent force excitation is applied with the same boundary condition as in the former but with the aim to replicate hammer impact. The numerical values and shape of the virtual force are selected replicating a measured hammer impact one.

Mechanical material properties are applied to the virtual specimen with an estimated Young's modulus and a Poisson's ratio of 0.33 as in Ashby et al (2000). Density properties are applied with a mean value of 329 Kg/m³ from experimental data (see Table 1). For the Modal Dynamic step, damping ratio ζ from experimental values is applied as a composite modal damping over the modes.

The mesh of the specimen is composed of 1168 linear hexahedral elements of Abaqus type C3D8R with reduced integration in order to have a good hourglass deformation control. For this aim, a mesh of 4 elements through thickness is applied.

Acceleration and displacement values are extracted in the centered point of the free end section in order to minimize section rotation effects.

Table 2. Identified modal parameters for specimen 2, beam 1 and replicate 1 (hammer test).

Freq.	Damping	Stiffness	Freq.	Damping	Stiffness
Hz	Ratio (%)	MN/m	Hz	Ratio (%)	MN/m
113	0.56	0.011	779.5	0.36	0.88
135	0.33	0.046	831.5	0.26	6.3
372	0.46	1.74	1068.5	0.18	6.22
527.5	0.48	3.69	1719	0.35	60.66
642	0.38	17.92	1995	0.43	19.57
723	0.44	1.93	2060	0.56	6.09

Table 3. Experimental identified FRF with FEM comparison for both types of Young's modulus estimation techniques.

Mode	Frequency Error		Amplitude Error	
	Static	ASTM	Static	ASTM
1	-2.21%	5.31%	40.05%	17.70%
2	-15.93%	-9.26%	224.70%	140.66%
3	-13.79%	-7.18%	384.46%	317.64%
4	-12.19%	-5.45%	68.75%	43.75%
5	-11.69%	-4.91%	272.94%	213.24%
6	-12.44%	-5.72%	39.22%	28.20%

4.1. Young's modulus estimation

It is of great importance for FE Models to have a correct estimate of the Young's modulus. To achieve this, two techniques are used for the estimation. First, it is calculated using a static test with a constant load and measuring the displacement of the beam free end by means of a centesimal indicator:

$$E = \frac{1}{3} \frac{PL^3}{uI} \quad (4)$$

Where P is the constant weight of 2.6 N used as a load; u is the measured free end displacement; I is the inertia of the beam cross section and L is the beam length.

The other method uses ASTM Standard (2010) estimation technique where E is calculated by the knowledge of the natural frequencies of the specimens (available from the experiments depicted in the next Section):

$$E = \frac{12\rho L^4 f_n^2}{H^2 C_n^2} \quad (5)$$

Where C_n is a coefficient for mode n ; f_n the resonance frequency for mode n in Hz; H is the thickness of the beam in vibration direction in meters; ρ is density of the beam in kg/m^3 .

These two methods for the Young's modulus estimation are under the hypothesis of a uniform density distribution, whereas the real specimens shows lack of uniform density. However, this is a generally accepted hypothesis.

5. Experimental Method

The experimental tests are conducted as follows: a specimen is randomly selected and is mounted into the clamping system that is ground constrained and the two beams are measured exciting the beam itself with the instrumented hammer. The Frequency Response Functions (FRF) between displacement and input force at the free end are computed and the specimen with clamping system is attached to the shaker and measurement is performed with constraint-imposed vibration. The Accelerance FRF (AFRF) is computed between free end and constraint acceleration. From the FRF the modal parameters (i.e. natural frequency, damping ratio and stiffness) are identified with the previously exposed techniques. From the AFRF it is possible to extract the damping ratio and natural frequency but not the stiffness of the eigenmode.

The values of damping have been verified to be independent from specimen number, side and replicate. The variability of the damping ratio with respect to the solicitation type (Hammer or Shaker) has also been verified to be unimportant in the factor.

6. Result and Discussion

The overall result of the identified damping ratio over frequency is reported in Figure 3: the values are distributed over frequency; the best fitting curve is a constant value.

After the execution of the just described experiment, it is possible to estimate the Young's modulus. Starting from the results in Figure 2, the estimated E values are reported in Table 1 for the three specimens along with the static tests results.

The best fitting of FEM computed FRF with the experimental one is obtained using ASTM Standard estimation of the Young's modulus as reported in Figure 4 and Table 3. In the Figure, it is reported a simplified identification of the experimental Frequency Response Function (in black) that consists of the elimination of the local eigenmodes from the identified result.

The three couples of eigenmodes in Figure 4 have similar shapes: the first is antisymmetric and the second is symmetric with respect to the constraint.

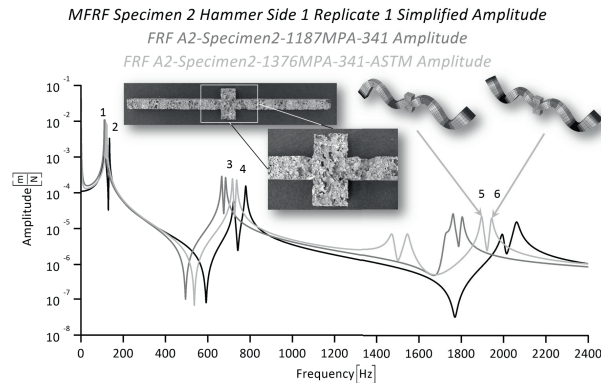


Figure 4: Comparison of the simplified modes identification for specimen 2 and two FEM simulations (the darker one with the static computed E and the lighter one with ASTM estimated E).

7. Conclusion

In this work a series of experimental tests have been conducted with two types of solicitation apparatus. The first is similar to the one proposed in Tu and Wang (2010) with the difference that the specimens are symmetric to ensure an axial resistant force (i.e. its best working conditions). In the second setup the solicitation is provided with an instrumented hammer. In this case, the costs are 10 times lower and allow determining all the modal parameters with a MDOF (Multiple Degree Of Freedom) algorithm with optimal results (Figure 2).

No difference in mean and dispersion can be found if the damping values estimated with the two setups are compared.

The main result is that the damping ratio is constant over frequency with a mean value of 0.34% and a standard deviation of 0.01%. Along with a correct estimation of the Young's modulus, this value is used in a virtual FEM hammer test to predict the dynamic behavior of the same specimen with good results (Figure 4) in terms of predicted frequencies and amplitudes.

If the three couples of eigenmodes are considered in Table 3, it can be seen that one is in good agreement with the experimental result (i.e. small percentage error) whereas the second is not. This is related with the lack of homogeneity of the material. In fact, with a density change of the 1% between the two sides of a specimen (with constant total mass) the amplitude of the second peak is double of the first.

These results can be used in FEM models of foam filled structures if the damping behavior of the contact between foam and skin is known. This will be the object of a future work.

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