A Numerical Method to Fit the Need of a Straightforward Characterization of Viscoelastic Materials for Marine Applications

Giovanni ROGNONI[a,1], Giada KYAW OO D'AMORE[a], Emanuele BROCCO[b] and Marco BIOT[a]

[a] *Department of Engineering and Architecture, University of Trieste, Via Alfonso Valerio 7/4, 34127 Trieste, Italy*

[b] *C.S.N.I. Scarl, Italy, Corso Perrone 27/R, 16152 Genova, Italy*

Abstract. In the field of green shipping the reduction of acoustic noise partially transmitted into water and the need of guarantee high comfort levels are important aspects in the view to agree with the UN 2030 Agenda in respect to life below water and good health and well-being. Both these aspects imply actions to increase absorption and dissipation of vibrational energy radiated towards the hull. To accomplish this effect, viscoelastic materials (VEM) characterized by high levels of damping are commonly used on board ships. In the last times, new strict requirements led to the development of Isocyanate free VEM, so the necessity of a provisional method to investigate in an efficient way new VEM is required. Experimental tests are essential in order to obtain performance indicators (non-standard procedure) or material physical characteristics (Oberst's beam test, ASTM E756 – 05). The implementation of the usual experimental setup could result rather complicated and it needs a high degree of accuracy, so in the last times finite element methods (FEM) has been increasingly used. Knowing VEM physics parameters allows numerical simulation in both the provisional and the optimization phase to be accurate and reliable. In this paper, an experimental-numerical method is proposed, with the aim of overtake the issues linked to the small-scale traditional cantilever beam test and paving the way to the selection of the most appropriate shape of the specimen. The innovation proposed through this method lies in the evaluation of the VEM complex modulus based on a reverse engineering approach, in which the loss factor estimation, contrarily from the traditional methods, is free from peak sharpness dependence. The proposed procedure is validated by comparison with the traditional method.

Keywords. Hysteretic Damping, Loss factor, Viscoelastic material, FEA, Oberst's beam test

[1] Corresponding Author; E-mail: giovanni.rognoni@phd.units.it.

1. Introduction

Viscoelastic materials (VEM) are commonly used on ships as a countermeasure for shock and vibration on board. Therefore, the most important characteristic for a VEM is the so-called damping capacity, namely the energy dissipated by a damper, or a material, in a complete cycle of motion [1]. Many methods have been developed through years in order to measure the loss factors, however does not exist an easy and straightforward way universally applicable to any vibrational system. One of the most widely used method to achieve the loss factor is the well-known half-power bandwidth method, however, in the last times alternative methods have been proposed: circle fit [2], modal strain energy [3], wave coefficients [4][5], hysteresis loop evaluation [6].

For VEMs, a relatively simple procedure to obtain the principal quantities governing the dissipating effect has been designed by H.V. Oberst [7]. Oberst's technique results relatively easy to set up and has been used through years by many researchers [8][9][10]. Although valid, this procedure presents some limitations, especially when the frequency response results highly damped, as in the case of marine VEMs.

The aim of this research is the development of a procedure, more practical and accessible than those currently applicable, to predict damping, integrating to the Oberst's experimental technique a finite element (FE) approach. The method could be a useful practical instrument for the characterization of new green and eco-friendly Isocyanate free VEM, mandatory in future ships. In this paper, a reverse engineering methodology based on the complex representation of the frequency response function of a vibrating beam is proposed. Damping is calculated basing on the phase angle between displacement ad exciting force, combining the data obtained by experiments and numerical simulation.

2. Background method

A convenient model for VEMs is the based on the complex modulus approach [11] which describes the relationship between stress σ and strain ε:

$$\sigma = E^{'}\varepsilon + \frac{E^{''}}{\omega}\frac{d\varepsilon}{dt} \tag{1}$$

The last term of Eq. (1) represents the energy dissipation characteristics of the material under the harmonic excitation:

$$\sigma(\mathrm{t}) = E\varepsilon(\mathrm{t}) = E\varepsilon \sin(\omega \mathrm{t}) \tag{2}$$

and it physically arises from relaxation and recovery of the polymer network after deformation [12]. This concept is at the base of the hysteretic damping model, which is described by the following 1 degree of freedom (DOF) system:

$$m\ddot{x} + \frac{h}{\omega}\dot{x} + kx = F \tag{3}$$

where m is the mass, k the stiffness, h is called hysteretic damping constant and F is the force. The constant h can be considered as the dissipating part of a complex stiffness, proportional to k with a factor η called loss factor. Eq. (3) can be rewritten as:

$$m\ddot{x} + k(1+i\eta)\ x = F \tag{4}$$

The focus of this paper is the evaluation of the loss factor through a method based on the phase angle measured between the excitation acting on the system and its response. In the 1 DOF hysteretic damping model as expressed by Eq. (4), the phase angle φ between force and displacement is given by the loss factor divided by the term 1- $(\omega/\omega_n)^2$. The relationship is valid whatever is the nature defining the damping of a specific system, provided that the model describing it is hysteretic damping. Knowing the phase angle between the force F and displacement x should theoretically provide the necessary data in order to calculate the loss factor η, whatever value the frequency assumes:

$$\eta = \tan\varphi \cdot \left(1 - \left(\frac{\omega}{\omega_n}\right)^2\right) \tag{5}$$

If the system modes of vibration result sufficiently frequency spaced so they don't interfere one to each other, the loss factor could be estimated by Eq. (5) for each $\omega = \omega_n$. Although this conclusion is basically correct, a practical limitation in the use of Eq. (5) arises. In the very correspondence of the resonance, in fact, φ goes to $\pi/2$ and the ω/ω_n ratio is equal to 1, therefore, loss factor results indefinite. To overcome this impediment, the approximation that Eq. (5) is valid for a limited frequency range where $\omega \approx \omega_n$ assumption has to be made. If loss factor in this interval is calculated and averaged, excluding the values obtained in the very proximity of the resonance, the result well represents the actual loss factor for $\omega = \omega_n$.

3. Methods and Materials

3.1. Reverse engineering based numerical approach

ASTM E756-05 standard [13] regulates the experimental procedures in order to calculate los factor of VEMs. However, when the damping is too high or too law, some limitations in the usage the method arise. To overcome these problematics, a methodology combining experimental tests, numerical simulation and the loss factor estimation by phase angle analysis has been developed. The method is based on a reverse engineering approach, the aim is simple: to replicate the beam behaviour measured in the experiment using a FE model developed in MSC Nastran in order to define the physical parameters of the VEM.

A traditional cantilever beam test has to be operated measuring the displacement or the acceleration in a significant point of the specimen. Simultaneously, the excitation force has to be measured with a load cell. It is suggested, for free layer (FL) treatment, the usage of a non-contact transducer, while for the constrained layer (CL) treatment specimens, contact accelerometers does not seem to interfere in a way that they disturb the measurement.

The data collected has to be used to deliver the transfer function. This function needs to be complex, so that it can be expressed either by real and imaginary or magnitude and phase representation. The transfer function will be the benchmark for the reverse engineering cycle implemented in the FE software. The research of the correct solution is a recursive process performed numerically. Basing on the experimental benchmark, the values of the elastic modulus can be estimated by matching the natural frequencies. When the frequency response function (FRF) displacement or acceleration peaks calculated by numerical analysis coincide with the peaks measured experimentally, the values of E for each natural frequency are defined. If peaks are not well detectable, a comparison can be made between the real part of the experimental and numerical frequency response curves. In correspondence of a natural frequency, in fact, the real part of the displacement transfer function is equal to zero, as the whole motion is dissipative. Once the elastic modulus is characterized, the loss factor values need to be set. The result will be achieved comparing the imaginary part of experimental and numerical response. When the two characteristics result close, then the VEM loss factor used as an input for the calculation has to be considered correct and eventually the system loss factors calculated for both the cases should coincide, except for a tolerance property selected case by case. The estimation of the system loss factor could be executed by any method, in this case the tangent method (TM) previously descripted has been used and the results compared with those obtained by the half-power bandwidth method (HPBM), when the latter has been demonstrated usable.

3.2. Limitations and uncertainties

Although theoretically there is no limitation in the use of the methodology, the are some situations in which it results complicated to apply the method. Experimentally, some difficulties in the usage of the loss factor estimation through analysis of the phase arise when the specimen tested is characterized by low damping, i.e., bare beam or FL damping treatment. In order to be applied, being the method based on the mean of the values calculated in proximity of the natural frequency, the range of frequency around it should be well set. Practically, that means that the frequency resolution has to be small enough to describe the phase change by an adequate number of points. A further source of inaccuracy can derive from the laser instrumentation which could present a rather significant noise when measuring displacement. The estimation of damping through phase analysis results more suitable in the case of high damping, as for the CL treatment, when peaks are large and there is not a need of increase the resolution in a significant way. At the contrary, when damping is very low, actually, it results more convenient the usage of the traditional half-power bandwidth method to estimate loss factor.

Some other issues outcome when the numerical analysis is carried out. When the FL treatment beam is studied, values of damping lower than 80 MPa could bring uncertainties in the setting of the elastic modulus. In this case, in fact, the natural frequencies depend more on the values of the base beam, generally much stiffer than the VEM layer. Lastly, there is an uncertainty concerning the FL treatment. The Oberst's equations assume that the whole damping layer attached to the metal plate works in order to dissipate the energy. This could be not true as the damping material full works only on part of the thickness. If this condition occurs, then all the results may be miscalculated. For all these reasons, in this paper the CL treatment has been preferred, presenting less uncertainties compared to the FL one.

3.3. Experimental setup

The experimental system set up for the tests has been characterized by different elements: the specimens have been clamped at one end on the head of an electrodynamic shaker controlled by a digital controller imposing a constant spectrum in the frequency between 10 and 2500 Hz; the data have been acquired using a PCB Piezotronics 352A24 accelerometer placed on the fixed extremity, since it has been verified that the accelerometer weight does not interfere in a significant way with the measurement; a further accelerometer has been located on the clamping support of the beam in order to measure the imposed accelerations; a load cell, interposed between the shaker head and the clamping support, has been also used to measure the force acting on the system; the data have been acquired through a Dewesoft Sirius acquisition system. The CL configuration specimens have a length of 250 mm and a width of 20 mm, the thickness of base and constraining plates is 1 mm and the thickness of the VEM core is 1.5 mm, for a total thickness of 2.5 mm. All tests have been performed at a room temperature ranging from 18 and 25°C.

3.4. Specimen materials

The base material for the bare beam and for the plates of the FL and CL treatments is steel with an estimated density of 7840 kg/m^3 and an elastic modulus calculated to be equal to about 206 GPa. Loss factor for steel has been evaluated following the ASTM E756-05 procedure.

Two VEM materials have been tested in order to define their damping characteristics: Type A material, a traditional viscoelastic widely used on board ships; Type B is a polymer modified cement designed for interior decks to reduce low frequency structure borne noise. The first material ($\rho \approx 1300$ kg/m^3) is soft and so appropriate for constraining plate applications, while the cement material ($\rho \approx 1125$ kg/m^3), characterized by a higher stiffness, is properly designed for FL applications and to be used in combination with electro-galvanized steel plates. The Poisson ratio has been assumed 0.4 for the Type A material and 0.3 for the Type B material, according to suggestions in technical literature [11]. These two different materials have been selected for the experimental validation of the procedure because of their different nature.

3.5. Numerical model

A numerical FE model has been created, simulating the sandwich beam of the CL configuration. The steel layers have been modelled with square shell elements having a 0.5 mm side. Their offset has been adjusted in a way to have a total thickness of 2.5 mm. The viscoelastic core has been modelled through cubic brick elements with a 0.5 mm side so the central layer is discretized by three elements along its height. The overall DOF of the model is of about 320˙000.

Physical parameters for steel plates and VEM core are those given in the previous paragraph. The model is excited by a 1 m/s^2 acceleration applied on the clamped extremity. The results of the FEA are the frequency response function on nearest grid point to the actual locations of the accelerometer in the experimental setup.

3.6. Operational Procedure

For each specimen, a 360 seconds session has been performed. The time records have been then analyzed in order to obtain their frequency spectrum. It has been shown that a 0.5 Hz resolution interval is appropriate for CL beam treatment permitting to accurately analyze the peaks.

After the experimental data have been collected, the numerical procedure has been implemented using the model previously descripted. For each resonance peak identified, the iterative research of the associated elastic modulus has been conducted until the correct values of E and η have been obtained. A MSC Nastran tool allows to perform a frequency analysis, direct or modal, on materials with frequency dependent characteristics.

For both materials, the recursive procedure has been repeated to obtain a sufficient agreement between experimental and numerical responses.

4. Results and Discussion

4.1. Results

The experimental-numerical procedure descripted provides the values for VEM elastic modulus and loss factor. In Figure 1(A), the response for both material specimens is shown in terms of transfer function of acceleration H measured at the reference point referred to that measured on the clamping system. Experimental characteristics (dotted line) are compared with those obtained numerically (straight line).

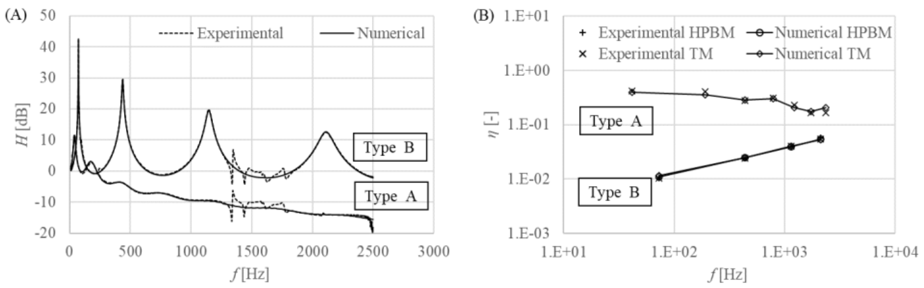

Figure 1. Experimental and numerical transfer functions (A) and system loss factors (B) for both materials.

Figure 1(B) shows the system loss factor for both specimens. For Type A material two trends have been evaluated using the tangent method for both experimental and numerical responses, while, for Type B material, four trends have been evaluated, two experimental and two numerical, obtained with TM and compared with those obtained by HPBM: the values are nearly coincident and graphically indistinguishable.

The output of the entire procedure is represented by the complex modulus of the VEMs. For both Type A and Type B materials, in Figure 2(A) the Young’s modulus is provided, while in Figure 2(B), the loss factors are given. For Type B material, also the results obtained using the analytical formulations provided by ASTM E756-05 are reported. This evaluation could not have been applicated on Type A material because of the high damping value which causes a non-ideal behaviour of the specimen.

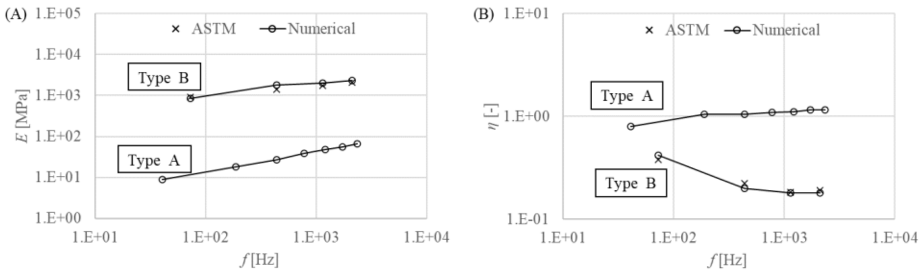

Figure 2. Elastic modulus (A) and loss factor (B) for both materials (for Type B material values according to ASTM E756-05 standard are reported).

4.2. Discussion

The main aim of this research is to demonstrate the capability of the exposed method to obtain, by small scale experiments coupled with a FEA, the characterization of a VEM for marine applications, in terms of complex modulus. As for both materials, Figure 1(A) shows an overall good correspondence between the experimental and numerical curves, so confirming the feasibility of the FEA procedure to replicate the laboratory test. The discrepancy around 1500 Hz is related to the specific fixing-exciting system which requires to be enhanced, however, the problematic does not compromise the implementation of the procedure.

Figure 1(B) shows the numerical-experimental comparison between the system loss factor for the two materials. The distance between the modes of Type B material makes possible to apply both the HPBM and the tangent method for loss factor extraction. The four curves are substantially coincident, suggesting the consistency of the tangent method proposed in the paper when compared to consolidated methods as the HPBM. Following this assumption, the method could be reasonably used to find the loss factor for each kind of vibrational system, even for systems with high damping, like that one characterized by Type A material. Strictly speaking, the 1 DOF theory could not be used in the case of modes overlapping, as for Type A material, however, if the result of Eq. (5) is taken as an estimate rather than the actual loss factor, the formula could still be used to provide a truthful result. The entire procedure can be developed to extract the approximated loss factor $\tilde{\eta}$ corresponding to each frequency investigated. In this way, a system loss factor parameter can be extrapolated from both experimental and numerical data and eventually compared. Regarding this approximate loss factor, the less the other modes participate in the dynamic of the model in correspondence of a natural frequency, the better $\tilde{\eta}$ approximate η. This approximated loss factor is the quantity compared in Figure 1(B), where the relative error is ranging between 2.3% and 11.2 % except for mode two (13.3%), influenced by the highly energetic mode one. On the other hand, the relative error (25.8%) related to mode seven is not significant because of the discrepancy due to the proximity to the boundary of the studied frequency interval.

The final results of the procedure show that the elastic modulus for both Type A and Type B (Figure 2(A)) materials has an upward trend in agreement with those reported as characteristic in the scientific literature on VEMs [11], where a transition range is typical between a lower stationary value at low frequencies and a higher one at high frequencies. For both materials the trend of loss factor is in agreement with the values given in literature, where the loss factor curve shows a peak value in correspondence of the transition zone of the elastic modulus. However, for Type A material, the increasing values with frequency seems to indicate that it is working in the rubbery-transition zone, while Type B material seems to operate in the transition-glassy zone.

As concluding remark about the quality of the two materials when used as damping materials, Type A appears to be appropriate when used in CL configurations, having a high loss factor and a relatively low elastic modulus, so allowing the reciprocal movement between the base and the constraining plates. Regarding Type B the high elastic modulus confirms that it is adequate for FL treatments, but the low loss factor classifies it as a not excellent damping material.

5. Conclusion

The validation tests here presented show that the proposed procedure is a valid method for deriving the damping characteristics of VEMs basing on small scale experiments. It can be inferred by the close correspondence between the numerically obtained characteristic of the transfer functions and those of the experimental benchmark. Thus, the proposed procedure can be used to broaden the operational field of the ASTM standard procedure to materials showing high damping even when the plate to VEM thickness ratio is over 1/10, so paving the way to experiments performed on samples with a VEM layer thickness typical of marine applications.

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