OPTIMAL DESIGN AND PLACEMENT OF VISCOELASTIC DYNAMIC NEUTRALIZER FOR OVERHEAD TRANSMISSION LINES

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Abstract. *Wind excited vibrations in overhead transmission lines normally occur in the frequency band of 10 to 100 Hz. These may be dangerous vibrations because they may eventually induce cable fatigue close to the suspension points. A classical means to reduce wind induced cable vibrations is by attaching to it the so called dynamic vibration neutralizar (also known as dynamic vibration absorber) at some distance from the suspension points. The well known Stockbridge damper, or Stockbridge vibration neutralizer has been largely used since its invention in 1928. Approximate numerical considerations and past experience recommend the neutralizer attachment point to be at 80 to 100 cm from the suspension points. Stockbridge damper has a very low loss factor provided by its "spring", which is, in fact, a stranded cable. Theoretical analysis and experimental identification, carried out by the authors have shown that this loss factor is about 0.006. That explains why the Stockbridge neutralizers are so efficient in reducing vibrations at the point of attachment and at their natural frequencies. However, because of that low loss factor, the efficacy at other frequencies, even only slightly different from the natural ones, is highly reduced, hindering the protection it should provide to the cable. A viscoelastic vibration neutralizer for overhead transmission lines, in which the "spring" is now provided by viscoelastic elements, was designed and patented by the author. It provides a loss factor of about 0.25 to 0.30 and is able to reduce wind induced vibrations over a wide frequency range. This paper aims at describing how to design this viscoelastic vibration neutralizer and to compute its optimum location. In the process of designing and positioning, an objective function based on the Frobenius norm is used. Computations for a 30 meter line span are provided and numerical results are produced and discussed.*

Keywords*: Vibration Control, Optimization, Viscoelastic materials, Optimal placement, Optimal physical parameters.*

1. INTRODUCTION

Dynamic neutralizers are simple systems which, when attached to a mechanical structure (primary system), reduce its vibration levels. In overhead transmission lines, the main kind of vibration is that induced by the wind, due to the formation of von Karman vortices. Given certain conditions, these vortices give rise to pressure variations, which oscillate between the lower and the upper parts of a cable, leading to a periodical excitation, perpendicular to the wind direction. The characteristic frequency of that excitation is a function of the wind speed and the cable diameter.

As overhead transmission lines have a high modal density in the frequency range of concern (10 a 100 Hz), the frequency of the aeolian excitation, at some moment, will coincide with one of the natural frequencies of the line, forcing it to work in resonance. Due to the low capacity of energy dissipation of this kind of structure, the vibration amplitude will be very high. That may lead to fatigue by vibration in points close to the line supports.

The breakage of such a cable may cause interruption in the distribution of electrical energy (electric breakdown), bringing severe losses both to companies and people in general. In Teixeira (1997), a failure caused by aeolian vibrations in one of the transmission lines of the Itaipu power-station is presented.

To minimize this problem, dynamic neutralizers are added to overhead lines. The current devices, known as Stockbridge dampers (after its inventor, in 1928), are extremely efficacious at some discrete frequencies (natural frequencies of the devices), but very disappointing at the remaining frequencies in the range of concern. It also has to be pointed out that the complete reduction of vibration in a particular frequency may transfer the problem of fatigue from the suspension point, at the tower, to the point where the neutralizer is attached, which is unwanted.

Given that, it was proposed, in Bavastri et al (1998) and Teixeira (1997), the use of viscoelastic dynamic neutralizers which, due to their high damping, reduce the vibration levels over the whole frequency range in a suitable way, as shown in Espíndola and Bavastri (1995) and Espíndola and Bavastri (1997). These devices, apart from applying reaction forces on the system to be controlled, also dissipate vibratory energy.

To design these neutralizers, it is of foremost importance the precise knowledge of the dynamic characteristics of the viscoelastic material employed in the devices. These characteristics vary with frequency and temperature. The dynamic characteristics of some elastomeric materials, based on butyl rubber and neoprene and manufactured in Brazil, were determined by the authors. The model employed to the characterization of those materials was the four parameter fractional derivative model, as described in Espíndola *et al* (2005) and Lopes *et al* (2004).

By the methodology presented in Bavastri *et al* (1998), the optimal parameters of the viscoelastic dynamic neutralizers were calculated considering a point excitation on a 30 meter long experimental line. In Bavastri *et al* (2004), it was considered an excitation distributed along the line. Thus, the excitation modelling is more realistic, resembling the action of the wind. A methodology to simultaneously compute the optimal location and the optimal physical parameters of a set viscoelastic dynamic neutralizers was implemented in Bavastri *et al* (2005). The system to be controlled there was a steel plate without constraints along its boundaries.

In Espíndola *et al* (2006), it was introduced a global objective function which does not depend on the excitation acting on the system to be controlled, known as the primary system. This objective function was the norm of Frobenius of the frequency response function matrix, defined on the modal subspace of the primary system. It is particularly suitable and efficacious for problems in which the excitation location is not known.

Herein it will be presented an alternative simultaneous optimization technique, based on the norm of Frobenius. It will be applied to 30 meter long experimental overhead line. By this technique, both the optimal physical parameters and the optimal placement of a set of viscoelastic neutralizers will be determined. The overhead line consists of a Partridge cable. The modal parameters which define the mathematical model of the primary system will be obtained by the continuum mechanics theory, allowing a fine discretization of the line and a very accurate optimal location. The numerical results obtained as previously indicated will be compared to experimental values measured on the line.

2. FRACTIONAL DERIVATIVE MODEL

For an accurate modelling of the viscoelastic material and, thus, of the control device, it was employed the fractional derivative model. This model was firstly introduced by Nutting (1921), modelling the relaxation of tension in viscoelastic materials by means of fractional powers of time. After that, Gemant (1936) observed that the elasticity and damping of viscoelastic materials were proportional to fractional powers of frequency. In Bagley and Torvik (1986), the description of the viscoelastic behaviour by fractional calculus was tackled. In that work, it was shown that the fractional model is closely related to the molecular theory which describes the microscopic behaviour of most viscoelastic materials.

The constitutive relationship in shear regarding the four parameter fractional derivative model is given by :

$$\tau(t)+\varphi_0\frac{d^{\beta}\tau(t)}{dt^{\beta}}=G_L\gamma(t)+G_H\varphi_0\frac{d^{\beta}\gamma(t)}{dt^{\beta}} \tag{1}$$

where $\tau(t)$ and $\gamma(t)$ are the stress and strain time histories, respectively, and φ_0, G_L, G_H *e* β are the four parameters to be experimentally determined. The fractional derivative model given by (1) describes the linear behaviour of thermorheologically simple viscoelastic materials (Bagley and Torvik, 1986 and Pritz, 1996). These materials present a complex modulus of elasticity, where the real part accounts for the storage of energy (spring effect) and the imaginary part for the dissipation of energy (damping effect).

In the frequency domain, the complex shear modulus may be given by (Lopes, 1998):

$$\bar{G}(\Omega,T)=\frac{G_L+G_H\varphi_0\left[i\alpha_T(T)\Omega\right]^{\beta}}{1+\varphi_0\left[i\alpha_T(T)\Omega\right]^{\beta}} \tag{2}$$

where the shift factor α_T is determined by

$$\log_{10}\alpha_T(T)=\frac{-\theta_1(T-T_0)}{\theta_2+(T-T_0)}, \tag{3}$$

Ω is the circular frequency [rad/s], T is the absolute temperature [K] and T_0 is the reference temperature [K].

Once the shear modulus of a viscoelastic material is known, it is possible to determine the corresponding stiffness of any simple system made of this material. From now on, the temperature will be regarded as constant and, therefore, omitted in the shear modulus.

3. GENERALIZED EQUIVALENT PARAMETERS

A simple neutralizer attached to an overhead transmission line may be regarded as a single degree of freedom system, as in Bavastri *et al* (1998). As such, the neutralizer consists of a lump of mass connected to the primary system by a resilient element of viscoelastic nature (see Fig. 1), the complex stiffness of which is:

$$\bar{K}(\Omega) = L_G \bar{G}(\Omega) = L_G G(\Omega)\left[1 + i\eta(\Omega)\right] \tag{4}$$

where L_G, in general, is a factor which depends on the geometry of the viscoelastic element, as defined in Espindola and Bavastri (1995) and $\eta(\Omega)$ is the loss factor, resulting from the ratio between the imaginary and the real parts of the complex shear modulus, as also defined in Espindola and Silva (1992) and Espindola and Bavastri (1995).

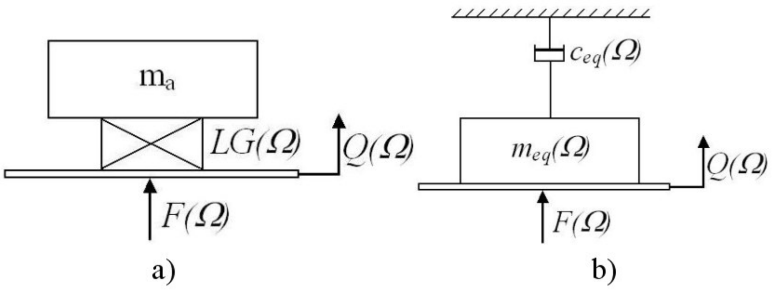

Fig. 1. a) A single degree of freedom system model b) Equivalent dynamic system model.

It is easily shown that the impedance and the dynamic mass for the model of Fig. 1a) are represented, as in Espindola and Silva (1992), Espindola and Bavastri (1995) and Espindola *et al* (2006), by:

$$Z_a(\Omega) = \frac{-im_a \Omega L_G \bar{G}(\Omega)}{m_a \Omega^2 - L_G \bar{G}(\Omega)} \tag{5}$$

$$M_a(\Omega) = -m_a \frac{L_G \bar{G}(\Omega)}{m_a \Omega^2 - L_G \bar{G}(\Omega)} \tag{6}$$

The neutralizer antiresonance frequency is defined in such a way that, in the absence of damping, the denominator of (5), or (6), equals to zero, that is:

$$\Omega_a^2 = \frac{L_G G(\Omega_a)}{m_a} \tag{7}$$

Defining the relationship $L_G\, G(\Omega) = L_G\, G(\Omega_a)\, r(\Omega)$, (5) and (6) may be written as:

$$Z_a(\Omega) = -im_a \Omega_a \frac{\varepsilon_a r(\Omega)[1 + i\eta(\Omega)]}{\varepsilon_a^2 - r(\Omega)[1 + i\eta(\Omega)]} \tag{8}$$

$$M_a(\Omega) = -m_a \frac{r(\Omega)[1 + i\eta(\Omega)]}{\varepsilon_a^2 - r(\Omega)[1 + i\eta(\Omega)]} \tag{9}$$

where $\varepsilon_a = \Omega / \Omega_a$.

From (8) and (9), the generalized equivalent parameters of mass and viscous damping are defined by:

$$c_{eq}(\Omega) = \Re\left(Z_a(\Omega)\right) = m\,\Omega_a \frac{r(\Omega)\eta(\Omega)\varepsilon^3}{(\varepsilon^2 - r(\Omega))^2 + (r(\Omega)\eta(\Omega))^2} \tag{10}$$

$$m_{eq}(\Omega) = \Re(M_a(\Omega)) = -m\frac{r(\Omega)\{\varepsilon^2 - r(\Omega)[1+\eta(\Omega)^2]\}}{(\varepsilon^2 - r(\Omega))^2 + (r(\Omega)\eta(\Omega))^2} \tag{11}$$

With these equivalent parameters, the models of Fig. 1a) and b) are dynamically equivalent, as shown in Espindola and Silva (1992). Thus, the dynamics of the compound system (primary system plus neutralizer) can be formulated in terms of the generalized coordinates of the primary system only, although the neutralizer has added a degree of freedom to the compound system. This is the main advantage introduced by the generalized equivalent quantities.

4. OPTIMUM DESIGN OF DYNAMIC NEUTRALIZERS

If various (say, p) neutralizers are attached to a multi-degree of freedom system, the equation of motion can be written as:

$$\tilde{M}\ddot{q}(t) + \tilde{C}\dot{q}(t) + Kq(t) = f(t) \tag{12}$$

Applying the Fourier transform and considering the generalized equivalent parameters defined previously, it gives:

$$\{-\Omega^2\tilde{M} + i\Omega\tilde{C} + K\}Q(\Omega) = F(\Omega) \tag{13}$$

with

$$\tilde{M} = M + M_{eq} = \begin{bmatrix} m_{11} & \cdots & m_{1n} \\ \vdots & \ddots & \vdots \\ m_{n1} & \cdots & m_{nn} \end{bmatrix}_{nxn} + \begin{bmatrix} 0 & 0 & 0 & 0 \\ 0 & m_{e_1} & 0 & 0 \\ 0 & 0 & \ddots m_{e_p} & 0 \\ 0 & 0 & 0 & 0 \end{bmatrix}_{nxn} \tag{14}$$

and

$$\tilde{C} = C + C_{eq} = \begin{bmatrix} c_{11} & \cdots & c_{1n} \\ \vdots & \ddots & \vdots \\ c_{n1} & \cdots & c_{nn} \end{bmatrix}_{nxn} + \begin{bmatrix} 0 & 0 & 0 & 0 \\ 0 & c_{e_1} & 0 & 0 \\ 0 & 0 & \ddots c_{e_p} & 0 \\ 0 & 0 & 0 & 0 \end{bmatrix} \tag{15}$$

In the above, M is the mass matrix of the primary system, M_{eq} is the equivalent mass matrix, C is the damping matrix of the primary system and C_{eq} is the equivalent viscous damping matrix. All those matrices are of order n x n, where n is the number of degrees of freedom considered in the primary system.

Calling Φ the modal matrix of the system, it is possible to carry out the following transformation of coordinates:

$$Q(\Omega) = \Phi P(\Omega) \tag{16}$$

Substituting (16) in (12) and pre-multiplying by Φ^T, it results, from the properties of the modal vectors, that:

$$\left\{-\Omega^2\left[diag(m_j) + m_A(\Omega)\right] + i\Omega\left[diag(c_j) + c_A(\Omega)\right] + diag(k_j)\right\}P(\Omega) = N(\Omega)\ ,\quad \text{j = 1, n} \tag{17}$$

where

$$N(\Omega) = \Phi^T F(\Omega) \tag{18}$$

The matrix elements m_j, c_j and k_j are the coefficients of the mass, damping and modal stiffness matrices and are, respectively, equal to 1, $2\xi_j\Omega_j$ and Ω_j^2, if the modal vectors are orthonormalized by modal mass matrix. The matrices m_A and c_A are the equivalent mass and damping matrices in the modal space.

In general, a few modes only are considered in the modal matrix, denoted accordingly by $\hat{\Phi}_{nx\hat{n}}$, with $\hat{n} << n$.

These modes are those within the frequency range of concern along with one below and one above that range (known, in experimental modal analysis, as residuals). Thus, the system of equations, in the modal subspace of the primary system, becomes of order $\hat{n} \times \hat{n}$, which makes it extremely attractive from the computational point of view.

Equation (17) can be written as:

$$[D_1 + D_2]P(\Omega) = N(\Omega) \tag{19}$$

where

$$D_1 = \{-\Omega^2 diag(m_j) + i\Omega diag(c_j) + diag(k_j)\} \tag{20}$$

$$D_2 = \{-\Omega^2 m_A(\Omega) + i\Omega c_A(\Omega)\} \tag{21}$$

The system response in the modal space, P(Ω), for a given excitation vector, $F(\Omega)$ (which can be defined as having a unit value in a certain generalized coordinate and null elsewhere or as a full one vector), is:

$$P(\Omega) = [D_1 + D_2]^{-1} N(\Omega) \tag{22}$$

from which the receptance in the modal space can be defined as:

$$H(\Omega) = [D_1 + D_2]^{-1} \text{ or } H(\Omega) = [D_1 + D_2]^{+} \tag{23}$$

where $[\;]^+$ denote the pseudo-inverse matrix. With (16), (18) and (22), it is possible to obtain the system response, $Q(\Omega)$, as shown in (24).

$$\{Q(\Omega)\} = [\hat{\Phi}][D_1 + D_2]^{-1}[\hat{\Phi}]^T \{F(\Omega)\} \tag{24}$$

It follows that the receptance matrix of the compound system will be:

$$[\alpha(\Omega)] = [\hat{\Phi}][D_1 + D_2]^{-1}[\hat{\Phi}]^T \tag{25}$$

For single degree of freedom systems, the relationship between the neutralizer mass and the primary system mass, according to Den Hartog (1956), should be defined as $\mu = m_a / m = 0.1 \; to \; 0.25$. In Espindola and Silva (1992), it was proposed a modal mass relationship for multi-degree of freedom systems, aiming to a mode by mode control. This relationship was:

$$\mu_j = \frac{m_a \sum_{i=1}^{p} \Phi_{k_i j}^{\;2}}{m_j} \tag{26}$$

After, in Espindola and Bavastri (1995), this relationship was used for a wide band control, defining the mass of the neutralizers by means of an arithmetic mean. That is, a single mass value for the neutralizers is calculated for each mode within the frequency range of concern and, then, it is taken the mean value of all values obtained over the whole range. This procedure will also be employed herein.

5. OPTIMIZATION PROCESS

In order to determine the optimal location and the optimal physical parameters of the neutralizers (for which the mass should be as small as possible), the optimization process should contain two optimization procedures, one internal to the other. The optimization process starts by

$$min \; f_{obj}(\overline{x}) : R^{np} \to R \tag{27}$$

where $\bar{x}^T = \{p_1, p_2, \ldots, p_{np}\}$ is the design vector, specifying the location of the neutralizers, and *np* is the vector size.

This optimization problem is subjected to the following inequality constraints $x_i^L < x_i < x_i^U$, where x_i^L is the lower limit and x_i^U the upper limit of each component on the design vector, *i=1, np*. The vector components are all integer values and determine the attachment point of each neutralizer on the primary system.

The corresponding objective function is defined by:

$$f_{obj}(\bar{x}) = w_1 f_1(\bar{x}) + w_2 f_2(\bar{x}) \tag{28}$$

where w_1 e w_2 are weight functions, f_1 is the total mass of the neutralizers and f_2 is

$$f_2(\bar{x}) = min\ f_{obj2}(\bar{x}, \bar{x}_1) \tag{29}$$

with $\bar{x}_1^T = \{\Omega_{a1}, \Omega_{a2}, \ldots, \Omega_{ap}\}$, subjected to $x_{1i}^L < x_{1i} < x_{1i}^U$. The vector $\bar{x}_1^T$ contains the characteristic frequencies of the neutralizers, since the neutralizer mass values are previously defined by Eq. (26) and the damping is defined by the viscoelastic material employed in the vibration control. To compute f_2, an internal optimization procedure should be carried out, with the following objective function:

$$f_{obj2}(\bar{x}, \bar{x}_1) = \left\| \max_{\Omega_1 < \Omega < \Omega_2} \left| H(\Omega, \bar{x}, \bar{x}_1) \right| \right\|_2 \tag{30}$$

where $H(\Omega, \bar{x}, \bar{x}_1)$ is obtained by Eq. (23).

The objective function given by Eq. (30), based on the norm of Frobenius, represents the compound system in the modal subspace of the primary system. This function does not depend on the excitation point and considers the model of the compound system in its global form.

It is also possible to define the objective function by the response of the compound system in the modal subspace of the primary system, $P(\Omega, \bar{x}, \bar{x}_1)$, computed by Eq. (22). However, in this case, the excitation point must be known. As shown in Espíndola *et al* (2006), when the excitation point is known, the objective function given by $P(\Omega, \bar{x}, \bar{x}_1)$ is more suitable. On the other hand, when the excitation point is not known, as regarded herein, the objective function of Eq. (30) yields a better convergence.

Schematically, the complete optimization process can be seen in the block diagram of Fig. 2. The presented methodology uses, for the external optimization procedure, where the optimal location is sought, a genetic algorithm. For the internal procedure, where the optimal physical parameters are determined, it is employed the method of non-linear optimization of Davidon-Fletcher-Powell, which is a quasi Newton method. The optimization parameters used in both cases can be found in Bavastri *et al* (1998).

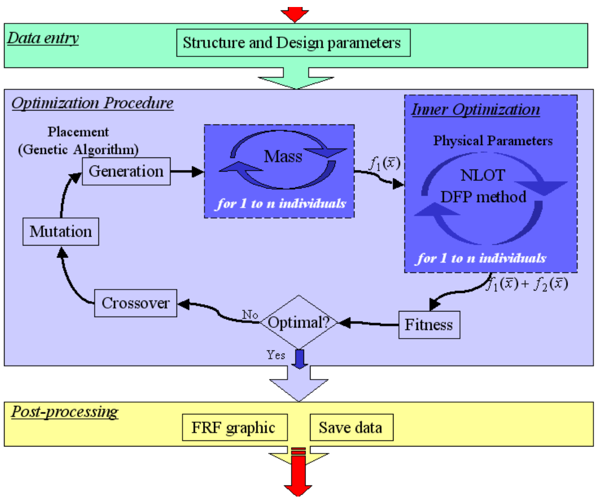

Figure 2. Schematic diagram of the complete optimization process.

6. EXAMPLE OF APPLICATION

The ensuing example shows the steps to be followed in order to compute the optimal location and the optimal physical parameters of a set of viscoelastic dynamic neutralizers, applied to a 30 meter long experimental overhead line.

Firstly, the modal parameters of the experimental line must be known. In this work, these parameters were obtained in analytical form by the continuum mechanics theory. Thus, it is possible to achieve a fine discretization of the cable, which allows an increased accuracy in the optimization process, particularly to the optimal location of the neutralizers.

The cable under focus is a Partridge cable, composed of aluminum wires in the outside and steel wires in the core. It is usually used as a lightning conductor cable. As noted in Teixeira (1997), its outer diameter is 16.3 mm, its mass density per unit length 0.5468 kg/m and its E_cI_c product 124.83 Nm^2. The tension load applied to the line was 900 N.

Due to the dynamic behaviour of the cable at high frequencies, the modal parameters are obtained from the equation of motion of a simply supported beam with an axial load.

Based on the schematic of Fig. 3, the governing equation of motion is given by:

$$-EI\frac{\partial^4 \omega(x,t)}{\partial x^4}+P\frac{\partial^2 \omega(x,t)}{\partial x^2}-\rho A\frac{\partial^2 \omega(x,t)}{\partial t^2}=f(x,t) \tag{31}$$

where $\omega(x,t)$ represents the transversal displacement of the cable, P is the axial load, A is the cross section area and ρ is the mass density.

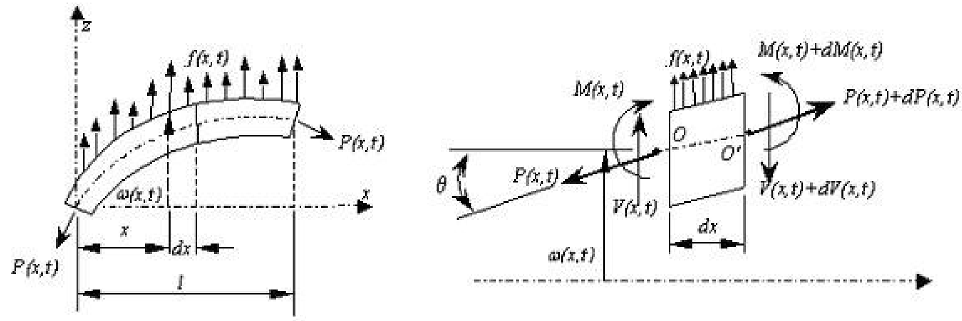

Figure 3. Free body diagram of the cable under axial load.

In Teixeira (1997), the solution of Eq. (31) is presented. The resulting natural frequencies and mode shapes are:

$$\omega_r^2=\left(\frac{n\cdot\pi}{L}\right)^2\frac{P}{m}\left[1+\left(\frac{n\cdot\pi}{L}\right)^2\frac{E_c\cdot I_c}{P}\right] \tag{32}$$

and

$$\Phi_n(x)=sin\left(\frac{n\cdot\pi\cdot x}{L}\right) \tag{33}$$

where $n=1,2,3,...$, L is the cable length and m is the cable mass per unit length (which equals to ρA).

The orthonormalization of the mode shapes is essential to the application of the methodology presented in this paper. Therefore, by the theory of orthogonality in continuum mechanics (Clark, 1972), it results that:

$$\Phi_n=\frac{sin\left(\frac{n\cdot\pi\cdot x}{L}\right)}{\sqrt{m\cdot\alpha_i}} \tag{34}$$

with

$$\int_0^L \Phi_n^2(x)\cdot dx=\alpha_n \tag{35}$$

Once the natural frequencies and the mode shapes are known, the modal damping factors of the primary system should be obtained. These parameters were determined by experimental modal analysis on the experimental line (see Teixeira, 1997), which showed that all modal loss factors (assumed the model of structural damping) could be regarded as constants and equal to 0.001.

The viscoelastic material chosen to the neutralizers was neoprene, the dynamic characteristics of which (modulus of elasticity and material loss factor) are shown in Fig. 4.

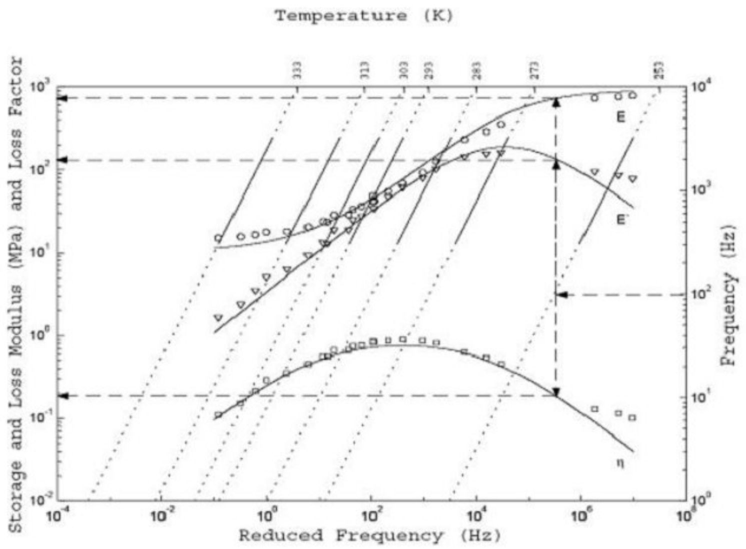

Figure 4. Dynamic characteristics of neoprene (reduced frequency nomogram).

Considering two viscoelastic neutralizers, the optimization process yielded approximately the same characteristic frequency of 24 Hz for both neutralizers, whereas their optimal locations resulted symmetrical, at 75 cm from the supports. In Fig. 5, inertances for which the response was taken at 84 cm from one of the supports and the excitation at 47 cm from the same support are shown. For sake of experimental validation, these inertances were computed without and with just one neutralizer at 75 cm from one of the supports.

To validate the above results, an experiment was performed on a 30 meter long experimental line, set up at Itaipu power station. In that experiment, by historical and practical reasons, just one viscoelastic neutralizer was attached at 75 cm from one of the supports. Inertances with and without the neutralizer, taken at the same location of the numerical case, were measured. The excitation was random (white noise), applied by a B&K 4809 shaker, whereas the response was captured by a B&K4370 accelerometer. Also used were a B&K8200 load cell, a B&K2365 pre-amplifier, a B&K2706 power amplifier and a HP35670A dynamic signal analyzer. The results are shown in Fig. 6.

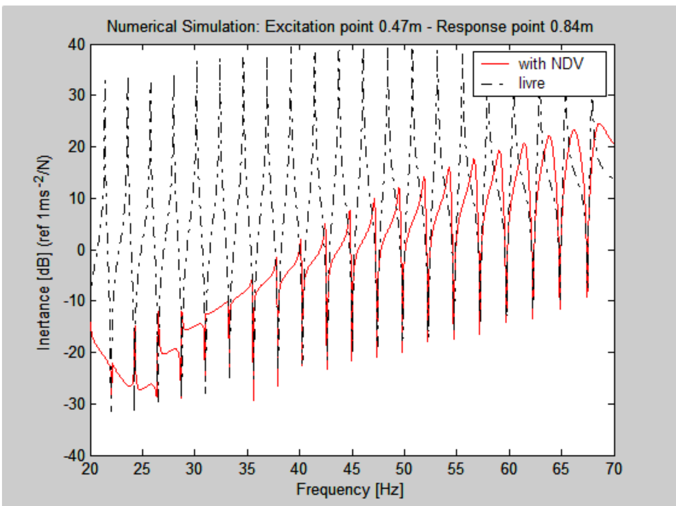

Figure 5. Numerical inertances with and without neutralizer.

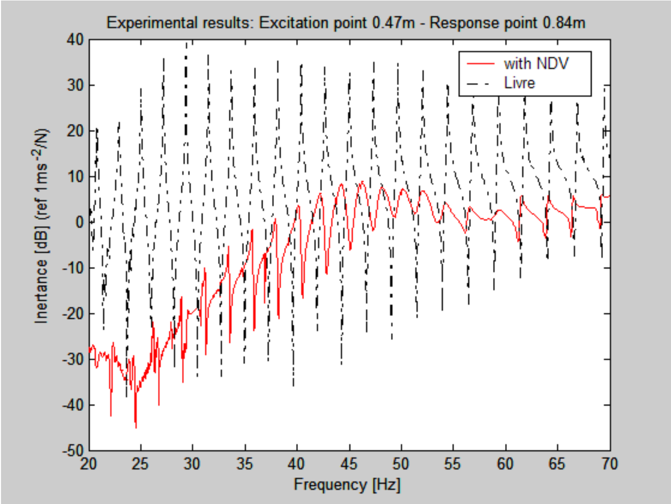

Figure 6. Experimental inertances with and without neutralizer.

There is a good correlation between the numerical and the experimental results, as observed by the inspection of Figs. 5 and 6. Both figures show that the maximum efficiency occurs for the frequency in which the neutralizer has its maximum impedance, as could be anticipated. That is indicated by a corresponding reduction in the inertances.

In order to check the optimal location, yet another experiment was performed, regarding two positions for the neutralizer: at 90 cm from one of the supports (position in the range of 80 to 100 cm, usually employed for Stockbridge dampers) and at 75 cm, as computed in the optimization process for viscoelastic neutralizers. The resulting inertances are portrayed in Fig. 7. As can be noted there, the efficiency obtained with the neutralizer at 75 cm is slightly better, if considered the whole frequency range of concern. The highest efficiency (approximately 13 dB) is obtained where the viscoelastic neutralizer has its best capacity of control (24 Hz).

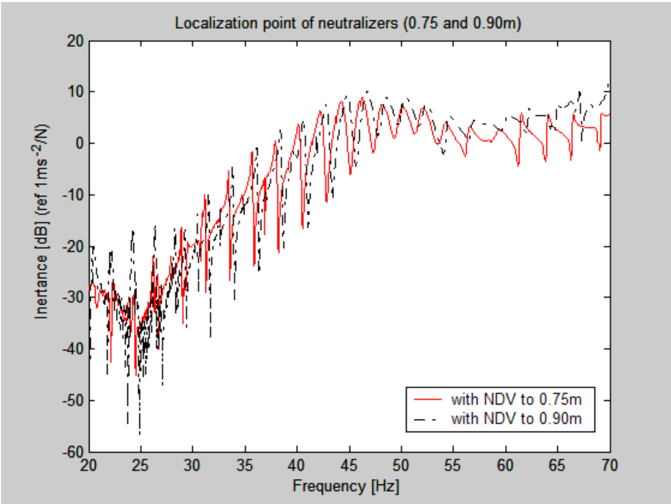

Fig. 7. Experimental inertances for the neutralizer at 75 cm and 90 cm from one of the supports.

7. CONCLUSÕES

A general methodology for vibration and irradiated noise control by viscoelastic dynamic neutralizers over a wide frequency range was reviewed.

A technique capable of simultaneously computing the optimal physical parameters and the optimal location of a system of viscoelastic dynamic neutralizers was presented and implemented. By this technique, the position of the neutralizers can be determined for geometrically complex structures with high modal density or with high modal coupling due to damping.

A method for computing the modal parameters of an overhead transmission line acted upon by an axial force was reviewed. The computed parameters were in close agreement with those obtained by experimental measurements.

The optimal physical parameters and the optimal placement of a system of viscoelastic dynamic neutralizers on an overhead transmission line were then computed. Numerical and experimental curves showed that the results were well correlated, reassuring the confidence on this kind of neutralizer for vibration control of overhead lines.

Frequency response curves (inertances), comparing the position usually adopted for Stockbridge dampers (between 80 and 100 cm) and the optimal position computed for viscoelastic dynamic neutralizers, showed that the latter leads to more efficient results.

8. ACKNOWLEDGEMENTS

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