EXPERIMENTAL INVESTIGATION OF THE DYNAMIC BEHAVIOR OF STOCKBRIDGE DAMPERS

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Abstract. *The purpose of this work is to present an experimental investigation on the dynamic behavior of Stockbridge dampers. Stockbridge dampers are dynamic absorbers widely used in overhead transmission line conductors to suppress the vibration amplitudes of the conductor due to wind excitation. Stockbridge dampers are composed of a flexible steel wire cable (also referred to as messenger cable), two inertial masses and a suspension clamp to attach it to the conductor. The messenger cable is composed of one core wire and one or two layers of wires wound around. The friction among the constituent wires of the messenger cable during its flexural vibrations is the mechanism responsible for the dissipation of the mechanical energy on these devices. Most of the works in the literature treats these absorbers as linear systems, computing the conductor response in the presence of Stockbridge dampers through the use of their complex mechanical impedance obtained for only one clamp velocity amplitude. We have verified a strong nonlinear behavior for such mechanical systems, since both their natural frequencies and impedance curves vary significantly with the clamp velocity amplitude. The experimental measurements presented here encourage the development and/or application of existing nonlinear models in order to better predict the dynamic response of Stockbridge dampers .*

Keywords*: wind-induced vibration, Stockbridge dampers, nonlinear behavior, experimental data*

1. Introduction

Wind-excited mechanical vibrations in single conductors of overhead transmission lines are understood as critical problems for the safety and reliability of the transmission line. Different types of mechanical vibrations may occur; however, the most common type corresponds to wind-excited vibrations in the frequency range of 3 Hz to 150 Hz, caused by vortex-shedding (Rawlins, 1979; CIGRE, 1989). The aerodynamic lift forces arising from the periodic shedding of vortices are responsible for the subsequent conductor vibrations. Since the span length of a typical transmission line is of the order of hundred to thousand meters, the frequency spectrum of the conductors is almost continuous; two consecutive natural frequencies of the conductors are quite close, separated by approximately 0.1 to 0.2 Hz; therefore, the conductors are frequently excited into forced resonant vibrations. Depending on the pattern of the wind flow and on the mechanical damping of the transmission line, the dynamic stresses and strains induced on the constituent wires of the conductors may become dangerously high, mainly near the conductor clamps and near the attachment points of Stockbridge dampers. These stresses and strains may lead to fatigue damage on the wires with catastrophic consequences such as the complete rupture of the conductor and interruption on the supply of electric energy.

Due to the low self-damping inherent to transmission line conductors, mainly at low frequencies, Stockbridge dampers are frequently attached to the conductors for the suppression of the wind-induced vibrations, as shown on Fig. 1. The Stockbridge damper was originally designed by George Stockbridge in 1925 and since then different modifications on the original design were proposed; nevertheless, most of the Stockbridge dampers used nowadays are composed of a flexible steel wire cable (also referred to as messenger cable), two inertial masses and a suspension clamp to attach it to the conductor. The messenger cable is composed of a core wire and one or two layers of wires wound around. The friction among the constituent wires of the messenger cable during its flexural vibrations is the mechanism responsible for the dissipation of the mechanical energy on these devices. The Stockbridge dampers exert a vertical force and a bending moment on the conductor at the attachment point. Indeed, they act as dynamic absorbers, absorbing part of the mechanical energy input by the wind only in narrow frequency bands in the neighborhood of its natural frequencies. These forces and moments may distort the wave form traveling through the conductor in such a way that the strains may become dangerously high in the neighborhood of the attachment points. There are several reported cases of failures of transmission line conductors due to excessive dynamical strains in the neighborhood of the suspension clamps of Stockbridge dampers.

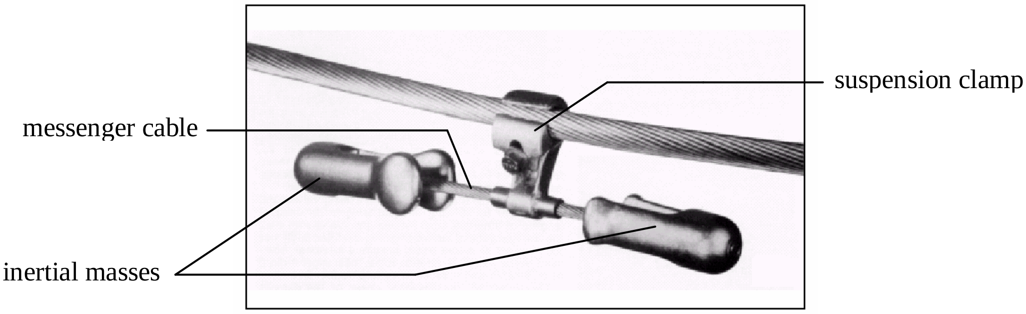

Figure 1. Stockbridge damper attached to a transmission line conductor.

The response of the conductor is sharply influenced by the number, the point of attachment and the dynamic behavior of the Stockbridge damper, and also by the conductor self-damping (Wagner et al., 1973; Dhotarad et al., 1978; Hagedorn, 1982; Vecchiarelli et al., 2000; Sauter, 2003). In order to better predict the dynamic response of a transmission line with Stockbridge dampers under wind excitation, we need first to understand the mechanical behavior of the Stockbridge damper, as an isolated mechanical system, when subjected to harmonic excitations. The purpose of this work is to present a better understand on the dynamics of Stockbridge dampers from experimental measurements performed on a laboratory. To this end, we have developed an experimental approach to obtain the forces and moments exerted by the damper, the mechanical impedance and the power absorbed by it as a function of the excitation frequency and clamp velocity amplitude. At the same time that little experimental data about the dynamic behavior of Stockbridge dampers are available in the literature, most of the works encountered treats these devices as linear mechanical systems, with their frequency-dependent mechanical impedance measured for only one clamp velocity amplitude (Wagner et al., 1973; Hagedorn, 1982; Hagedorn, 1987; Vecchiarelli et al., 2000). However, we have observed a strong nonlinear behavior for the Stockbridge damper tested, with its natural frequencies and frequency-dependent impedance being strongly dependent on the clamp velocity amplitude. Our experimental results suggest that such available linear models are unable to completely understand the response of transmission line conductors in the presence of Stockbridge dampers. The experimental measurements presented here encourage the development and/or application of existing nonlinear models in order to better predict the dynamic response of Stockbridge dampers.

This paper is organized as follows: in section 2, we describe in detail the experimental tests performed on laboratory to measure the dynamic behavior of Stockbridge dampers; in section 3, we present and analyze the experimental results obtained for the natural frequencies, mechanical impedance and power absorbed; and, finally, in section 4, we finish by offering the final remarks and suggestions for future works.

2. Experimental Tests

On the laboratory tests performed, the suspension clamp of the Stockbridge damper was subjected to harmonic forced vibrations by means of a frequency- and amplitude-controlled hydraulic shaker. The suspension clamp was also instrumented with two force transducers and one piezoelectric accelerometer for the measurement of the force and acceleration signals, respectively. We have developed a computational tool with appropriate virtual LabView instruments and acquisition hardware to acquire and analyze in real time the force and acceleration signals, for sinusoidal sweep tests performed at a rate of 0.2 Hz/s. This sinusoidal sweep test is standardized by the international standard IEC 61897 (IEC, 1998). With the measurement data acquired on the tests, we computed the mechanical impedance, the power absorbed, the force and the moment exerted by the damper as a function of both the excitation frequency and clamp velocity amplitude.

2.1. Stockbridge damper tested

The damper used in the laboratory tests was an asymmetrical Stockbridge damper, as shown in Fig. 2. Stockbridge dampers with different inertial masses and different lengths of messenger cable on each side of the clamp are referred to as asymmetrical dampers. The clamp is made of an aluminum alloy weighting around 0.500 ± 0.005 kg. The two inertial masses are made of galvanized casting iron; the left one weights 1.450 ± 0.005 kg while the right one weights 0.760 ± 0.005 kg. The messenger cable is composed of one core wire and one layer with six wires wound around in a helix geometry. All the wires are made of galvanized steel and the overall diameter of the messenger cable was measured to be 9.00 ± 0.05 mm. The main dimensions of the Stockbridge damper tested are shown in millimeters in Fig. 2.

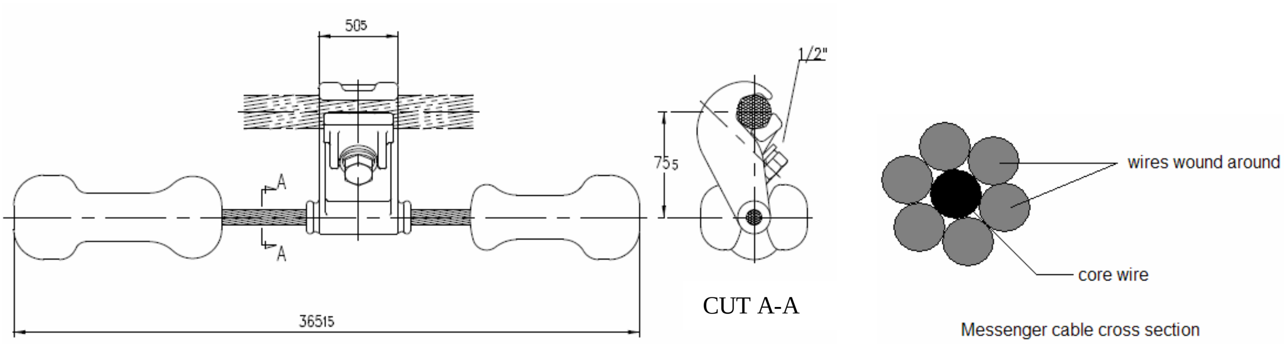

Figure 2. The Stockbridge damper used in the laboratory test.

2.2. Experimental setup

The basic setup used in the experiments consists of a remote frequency- and amplitude-controlled hydraulic shaker, two force transducers, one ICP piezoelectric accelerometer, two signal conditioners, a data acquisition board with sixteen channels (eight input and eight output) and the LabView software, as can be seen schematically on Fig. 3. The ICP piezoelectric accelerometer contains an integrated circuit that pre-amplifies the measured acceleration signal. The two force transducers do not have this same feature; hence, two signal conditioners were used in order to amplify the signal to a measurable value. The accelerometer was located on the top of a steel base plate connected to the shaker cylinder and the force transducers were positioned on the same base plate, symmetrically around the center of the Stockbridge damper tested. The acceleration signal was sent to the shaker control unity and to one channel of the data acquisition board. The two force signals were pre-amplified in the signal conditioners and, then, sent to the other input channels of the data acquisition board.

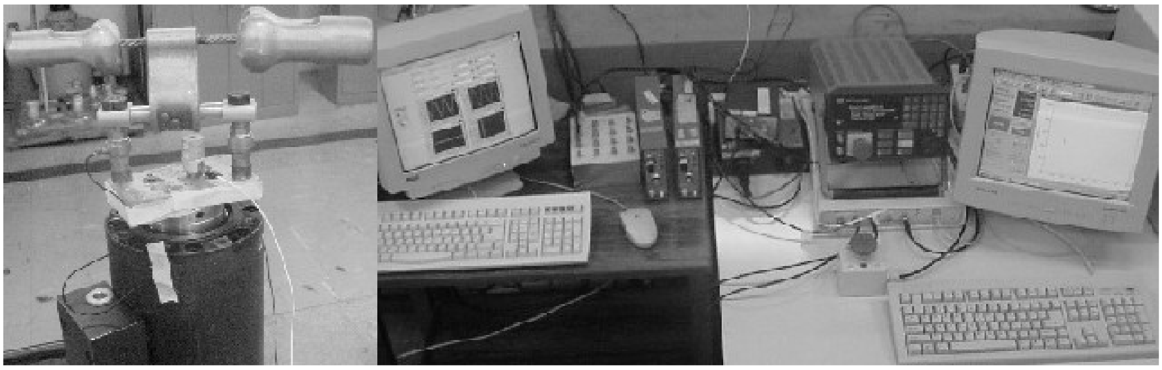

Figure 3. Experimental set-up for the frequency-response test performed with Stockbridge dampers.

2.3. Equipments, sensors and software utilized

A LabView virtual instrument (VI) was developed in order to acquire, simultaneously and in real time, the acceleration and the two force signals. In the LabView virtual instrument developed, we also made some elementary computations with the acquired signals to obtain the mechanical impedance, the power absorbed and the moment exerted by the damper. All the measured data were automatically recorded in an Excel worksheet for further graphical presentation of the data. In the LabView virtual instrument developed, the input data, which must be provided by the user before the beginning of the tests, are the sampling rate, the number of samples to be acquired and the sensitivities of all sensors used. In the following paragraph, we briefly describe the sensors used in the tests performed. The sampling rate was always chosen to be at least two times the excitation frequency, according to the Nyquist criterion. Before the beginning of the tests, all sensors were carefully calibrated and the conditioners gains were set accordingly.

One ICP piezoelectric accelerometer with a sensitivity of (85.0 ± 0.2) mVpeak/g (g = 9.81 m/s^2) was used to measure the acceleration of the suspension clamp. The acceleration signal was first sent to the shaker control unity and then to the data acquisition board for further processing. Two force transducers connected in two signal conditioners were used. The sensitivities of the force transducers used were calibrated to (19.0 ± 0.2) mVpeak/N. The two force signals were sent to two different input channels of the data acquisition board. The total force exerted by the damper was then computed by summing the two force signals, while the moment exerted was computed by subtracting the two force signals and then multiplying the result by the distance between the two force transducers. The distance between the two force transducers was measured to be (130.00 ± 0.05) mm.

2.4. Methodology

Previously, the frequency-response tests with Stockbridge dampers were done without automation, with a sequence of steps that demanded time, effort and careful attention. To conclude a frequency response test in a frequency range of 5 Hz to 80 Hz with sweep rate of 0.2 Hz/s we took approximately two hours. The main disadvantage of our previous methodology was the manually data acquisition. For each excitation frequency, the operator of the test read the values of the frequency, acceleration and force in an oscilloscope or spectral analyzer and then manually wrote those data on an Excel worksheet. The LabView virtual instrument developed increases significantly the accuracy of the data measured and also significantly reduces the duration of the test to approximately ten to fifteen minutes, depending on the sweep test rate and frequency range.

The experimental methodology behind the frequency-response tests performed with Stockbridge dampers comprises the measurement of the force exerted by the damper, its acceleration and the phase angle between the force and the velocity signals. These measurements have to be done for each excitation frequency of the test. The first task is to define the frequency range of the test. The international standard IEC 61897 recommends that the frequency range be the one expected for the wind-induced vibrations. Therefore, we computed the frequency range through the Strouhal relationship:

$$f_{\min} = 0.22 \frac{v_{\min}}{D} \tag{1}$$

$$f_{\max} = 0.22 \frac{v_{\max}}{D} \tag{2}$$

where $f_{\min}$ and $f_{\max}$ denote, respectively, the minimum and maximum excitation frequency expected; $v_{\min}$ and $v_{\max}$ are, respectively, the minimum and maximum wind speeds; and D is the diameter of the conductor which will be protected by the Stockbridge damper tested. For wind-induced vibrations on transmission line conductors, the Strouhal number frequently used is 0.22 (Schäfer, 1984). The Stockbridge dampers tested were designed to damp wind-induced vibrations on transmission line conductors with diameters of about 16 mm for wind speeds in the range of 0.5 to 6 m/s. Therefore, the frequency range for the tests, computed from Eqs. (1) and (2), is approximately 7 Hz to 82 Hz.

For linear mechanical systems, we can define the complex mechanical impedance, Z, as (Ewins, 2000; Sauter, 2003)

$$Z = \frac{F}{v} = \hat{Z}\,\mathrm{e}^{i\varphi} \tag{3}$$

where

$$v(t) = \hat{v}\,\mathrm{e}^{i\Omega t} \tag{4}$$

and

$$F(t) = \hat{F}\,\mathrm{e}^{i(\Omega t+\varphi)} \tag{5}$$

are, respectively, the velocity of the clamp (excitation) and the force exerted by the damper (response); $\hat{v}$ and $\hat{F}$ are, respectively, the velocity and force amplitudes, both assumed to be real quantities; φ is the phase angle between the force and velocity signals; and i is the complex unity ($i^2 = -1$). In a somewhat more general way, the impedance can be defined by the quotient of the Fourier transforms of the two signals $F(t)$ and $v(t)$. In Eq. (3), $\hat{Z}$ is the magnitude of the impedance and it is simple computed as $\hat{F}/\hat{v}$. For linear systems, $\hat{Z}$ and φ depend only on the excitation frequency Ω, while for nonlinear systems, as Stockbridge dampers, they depend on both the frequency and amplitude of the excitation (displacement, velocity or acceleration). Strictly speaking, the mechanical impedance is defined only for linear systems. For nonlinear systems, a harmonic velocity input will lead to a periodic output force, which can be decomposed into the following Fourier series:

$$F(t) = \hat{F}_0 + \hat{F}_1\,e^{i(\Omega t+\varphi_1)} + \hat{F}_2\,e^{i(2\Omega t+\varphi_2)} + \ldots \tag{6}$$

where

$$\hat{F}_k = \frac{1}{T}\int_0^T F(t)\ \mathrm{e}^{-i(k\,\Omega t+\varphi_k)}\ dt \tag{7}$$

k = 0, 1, 2, …, and $T = 2\pi/\Omega$ is the fundamental period of the steady-state oscillation. If we terminate the infinite series after the first harmonic and further assume $\hat{F}_0 = 0$ we can write

$$F(t) = \hat{F}_1\ \mathrm{e}^{i(\Omega t+\varphi_1)} \tag{8}$$

and, then, we can use Eq. (3) to compute the mechanical impedance for a nonlinear system.

As the steady-state has been reached, the average power absorbed by the damper can be computed from the average power provided by the excitation source (here, the hydraulic shaker). The average power provided by the excitation source, P_a, can be computed as

$$P_a = \frac{1}{T}\int_0^T F(t)\,v(t)\,dt = \frac{1}{2\pi/\Omega}\int_0^{2\pi/\Omega} \hat{F}\sin(\Omega t+\varphi)\,\hat{v}\sin(\Omega t)\,dt = \frac{1}{2}\hat{F}\,\hat{v}\cos(\varphi) = \frac{1}{2}\hat{F}\,\frac{\hat{a}}{\Omega}\cos(\varphi) = \frac{1}{4\pi f}\hat{F}\,\hat{a}\cos(\varphi) \tag{9}$$

where $\Omega = 2\pi f$ and $\hat{a}$ stands for the acceleration amplitude, measured by the piezoelectric accelerometer. Finally, the moment exerted by the damper, M_d, was computed as the difference, in absolute value, between the amplitudes of the two force signals recorded, multiplied by the distance between the transducers.

After acquiring all the signals waveform in time, the main frequency of each signal was found and their corresponding amplitudes were also measured. The LabView virtual instrument automatically measures the phase angle between the force and velocity signals. The following data were recorded until the end of the test: the excitation frequency, the amplitude of force signal recorded by each transducer, the amplitude of the total force exerted by the damper, the amplitude of moment exerted by the damper, the amplitude of the acceleration signal and the phase angle between the total force and velocity. The above data were recorded in a tabular form inside an Excel worksheet. Using our LabView tool, we can further build the graphics of the force and moment exerted, mechanical impedance, power absorbed as functions of the excitation frequency, for a given clamp velocity amplitude.

3. Results and Conclusions

In this section, we present and analyze the experimental results obtained during the frequency response tests. The curves of force and moment exerted, mechanical impedance and power absorbed by the Stockbridge damper tested as functions of the excitation frequency for four different velocity amplitudes are shown, respectively, in Figs. 4, 5, 6 and 7. The four velocity amplitudes tested were: 0.05 m/s, 0.10 m/s, 0.15 m/s and 0.20 m/s. These velocity clamp amplitudes were chosen based on typical values of maximum amplitudes of transmission line conductors in the range of wind speeds frequently expected for wind-induced vibrations.

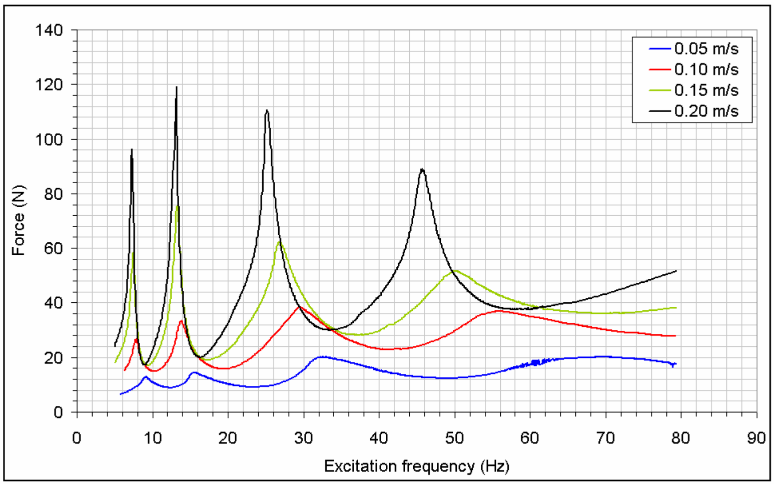

Figure 4. Force exerted by the damper as a function of the excitation frequency for four clamp velocity amplitudes.

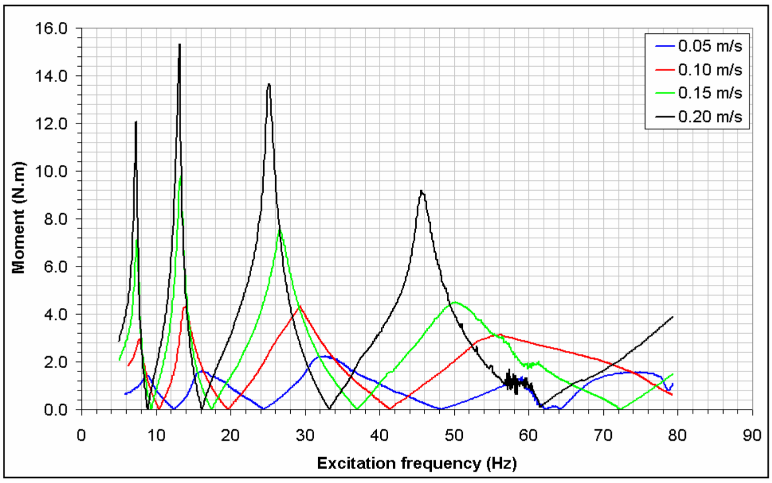

Figure 5. Moment exerted by the damper as a function of the excitation frequency for four clamp velocity amplitudes.

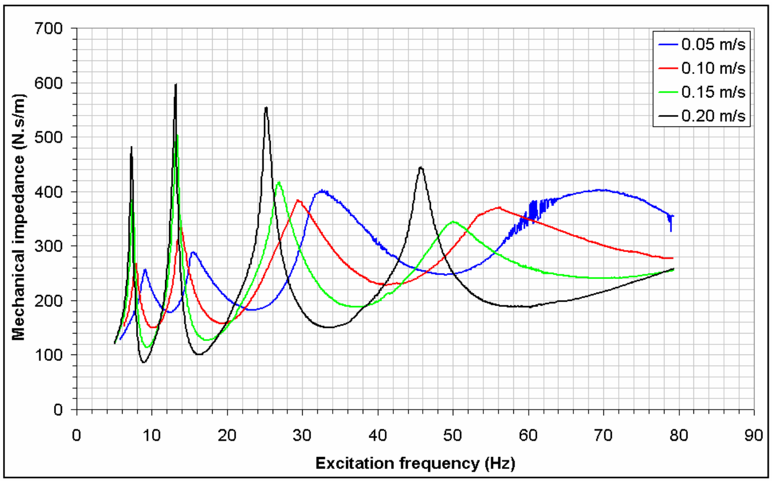

Figure 6. Impedance of the damper as a function of the excitation frequency for four clamp velocity amplitudes.

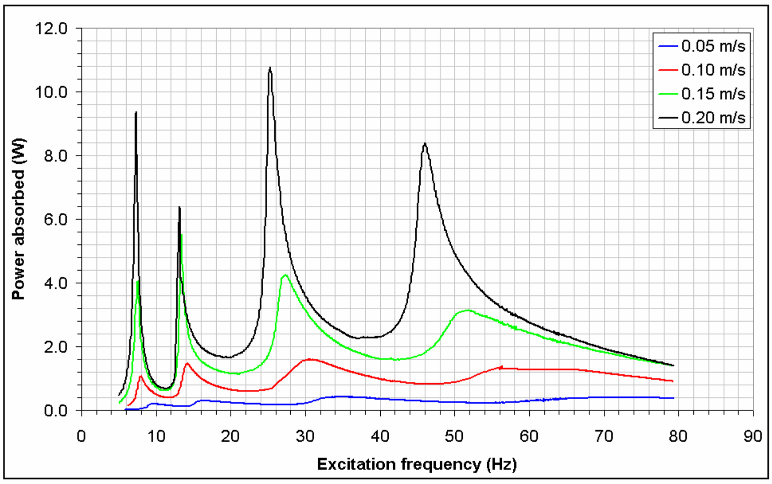

Figure 7. Power absorbed by the damper as a function of the excitation frequency for four clamp velocity amplitudes.

In Table 1, we present the four natural frequencies of the Stockbridge damper tested for each clamp velocity amplitude. These natural frequencies were extracted from the data recorded in the Excel worksheet. The first and second natural modes of vibration of the Stockbridge damper tested corresponded, respectively, to large flexural displacements of the messenger cable on the side of the higher and smaller masses. On the other hand, the third and fourth natural modes of vibration corresponded, respectively, to large rotations of the higher and smaller masses, with small flexural displacements of the messenger cable. In these latter two modes of vibration, the masses rotate around the points of attachment of the messenger cable to them.

Table 1. Natural frequencies of the Stockbridge damper tested.

Clamp velocity (m/s)	Natural frequencies (Hz)			
0.05	9.3	15.9	32.6	63.1
0.10	7.9	14.0	29.6	56.3
0.15	7.5	13.3	26.9	50.5
0.20	7.3	13.1	25.3	45.8

From the results presented on Table 1 and from the experimental curves of Figs. 4 to 7 we can draw the following conclusions:

- as the clamp velocity amplitude increases, the natural frequencies of the Stockbridge damper tested decrease, which can be easily seen from the force versus frequency graphic (see Fig. 4) whose peaks are moved to the left as the clamp velocity increases from 0.05 m/s to 0.20 m/s;
- as the clamp velocity amplitude increases, the force exerted and the power absorbed by the Stockbridge damper tested also increase, which were already expected; nevertheless, the rates of power increasing are not directly proportional to the increase in the velocity amplitude;
- as the velocity clamp amplitude increases, the peaks in the force and impedance versus frequency graphics became thinner; in other words, the frequency bands where the damper acts efficiently (the neighborhood of its natural frequencies) became narrower;
- as the velocity clamp amplitude increases, the third and fourth peaks became more clearly defined; and
- for the four velocity amplitudes tested, the maximum value of the force exerted by the Stockbridge damper was encountered at its second natural mode of vibration.

The first two conclusions stated above confirm a strong nonlinear behavior for the Stockbridge damper tested, a behavior already verified by other authors (Sauter and Hagedorn, 2002; Sauter, 2003). Thereby, the linear models available in the literature to compute the response of transmission line conductors with Stockbridge dampers seem to be inappropriate, since they are unable to reproduce the nonlinear dynamic behavior of Stockbridge dampers. To accurately predict the response of a transmission line conductor in the presence of Stockbridge dampers, their impedance must be known as a function of both the frequency and amplitude of the excitation. The decrease in the natural frequencies as the clamp velocity amplitude increases may be explained by a decrease in the stiffness of the Stockbridge damper tested since, roughly speaking, the natural frequencies of a mechanical system are directly proportional to the square root of its stiffness.

4. Final remarks and future works

In this work, we described an experimental approach to assess the dynamic behavior of Stockbridge dampers, based on frequency response tests performed on a laboratory. By means of a LabView virtual instrument and a data acquisition board we were able to acquire and analyze in real time the force and acceleration signals during standard sinusoidal sweep tests, and then computing the quantities of interest. With our approach, we were able to provide curves of force and moment exerted, mechanical impedance and power absorbed by the damper as functions of the excitation frequency for four different clamp velocity amplitudes. Based on the experimental data obtained, we verified a strong nonlinear behavior for the Stockbridge damper tested.

The results of this work encourage a questioning of the standard IEC 61897, concerning the recommended clamp velocity amplitude for the sinusoidal sweep tests performed with Stockbridge dampers. The international standard IEC 61897 recommends only 0.10 m/s for the clamp velocity amplitude. Nevertheless, as has been known for many years, wind-induced vibration amplitudes vary from 0.1 to 1 conductor diameter and, for the frequency range expected for wind-induced vibrations on transmission line conductors, such range of amplitudes may lead to clamp velocities from 0.05 m/s to 0.20 m/s approximately. Since the dynamic behavior of Stockbridge dampers may strongly depend on the velocity amplitude applied to its clamp, why not should we perform the frequency response tests for other clamp velocity amplitudes?

The experimental data presented here may be used for the validation of the available and new theoretical models proposed to predict the dynamic response of Stockbridge dampers. For future investigations, we are implementing in our LabView environment a procedure to calculate the experimental uncertainties of all measured data, in order to assess the validity and the limits of applicability of the phenomenological models of Bouc (Pivovarov and Vinogradov, 1987), Davidenkov (Vinogradov and Pivovarov, 1986) and Masing (Sauter and Hagedorn, 2002; Sauter, 2003). These models have been successfully used to predict the energy dissipated by stranded cables under flexural vibrations.

5. References

CIGRE SC 22 WG 01, 1989, “Report on aeolian vibration”, Electra, Vol. 124, pp. 41-77.

Claren, R. and Diana, G., 1969, “Mathematical analysis of transmission line vibration”, IEEE Transactions on Power Apparatus and Systems, Vol. 88(12), pp. 1741-1771.

Dhotarad, M. S., Ganesan, N. and Rao, B. V. A., 1978, “Transmission line vibrations”, Journal of Sound and Vibration, Vol. 60(2), pp. 217-237.

Ewins, D. J., 2000, “Modal testing: theory, practice and application”, Taylor & Francis Group, 2nd edition, 400p.

Hagedorn, P., 1982, “On the computation of damped wind-excited vibrations of overhead transmission lines”, Journal of Sound and Vibration, Vol. 83(2), pp. 253-271.

Hagedorn, P., Schmidt, J. and Nascimento, N., 1987, “Stochastic field processes in the mathematical modelling of damped transmission line vibrations”, Mathematical Modelling, Vol. 8, pp. 359-363.

IEC 61897, 1998, “Overhead lines – Requirements and tests for Stockbridge type aeolian vibration dampers”, International standard, first edition.

Pivovarov, I. and Vinogradov, O. G., 1987, “One application of Bouc’s model for nonlinear hysteresis”, Journal of Sound and Vibration, Vol. 118(2), pp. 209-216.

Rawlins, C. B., 1979, “Transmission Line Reference Book: Wind-Induced Conductor Motion”, Ed. EPRI, Palo Alto, California, 244p.

Sauter, D. and Hagedorn, P., 2002, “On the hysteresis of wire cables in Stockbridge dampers”, International Journal of Non-Linear Mechanics, Vol. 37, pp. 1453-1459.

Sauter, D., 2003, “Modelling the dynamic characteristics of slack wire cables in Stockbridge dampers”, Dr.-Ing. Thesis (in English), Von Fachbereich Mechanik, Technischen Universität Darmstadt, Germany, 89p.

Schäfer, B., 1984, “Dynamical modelling of wind-induced vibrations of overhead lines”, International Journal of Non-Linear Mechanics, Vol. 19(5), pp. 455-467.

Vecchiarelli, J., Currie, I. G. and Havard, D. G., 2000, “Computational Analysis of Aeolian Conductor Vibration with a Stockbridge-Type Damper”, Journal of Fluids and Structures, Vol. 14, pp. 489-509.

Vinogradov, O. G. and Pivovarov, I., 1986, “Vibrations of a system with non-linear hysteresis”, Journal of Sound and Vibration, Vol. 111(1), pp. 145-152.

Wagner, H., Ramamurti, V., Shastry, R. V. R. and Hartmann, K., 1973, “Dynamics of Stockbridge dampers”, Journal of Sound and Vibration, Vol. 30, pp. 207-220.

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