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REGARDING THE DYNAMIC CHARACTERISTICS OF TECHNOLOGICAL PRODUCTS OBTAINED FROM RECYCLING COMPOSITE INSULATORS WITH RUBBER SILICONE COATING

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Abstract—The paper presents an alternative method for determining the dynamic characteristics of an epoxy resin rod obtained from recovered components of composite silicone insulators, in the case of an encased rod [1], [4].

Keywords—Composite silicone insulator, epoxy rod, recovery, dynamic analysis

I. Introduction

TECHNOLOGIES for recovering used technological products for recycling into other similar types of products have advanced recently given the accelerated decrease of natural prime material availability [2], [3].

Using rubber in general and silicone rubber in particular provides possibility of manufacturing products with various shapes and usage in important industries such as automotive, transportation, aerospace, chemical electro technical but also in medical and adjacent domains [5], [8], [9].

Recovery of components elements from used electrical composite insulators represents a large interest technology [6]. A special importance is given to the central rod made of fiberglass reinforced epoxy resin.

Fig. 1 illustrates the main components of a composite silicone insulator with partial or total recovery options.

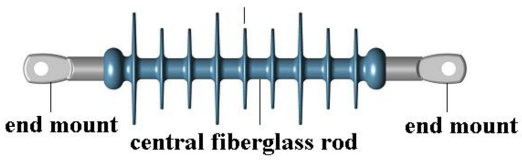

Fig. 1. Structure of a composite silicone insulator.

II. Theoretical Aspects Regarding The Use Of Frequency Response Characteristics

A. Fourier Transform

The Fourier transform (named after the French mathematician and physicist Joseph Fourier) applied on a complex function produces another complex function that contains the same information as the original function, but re-arranged into component frequencies. For example, if the initial function is a time-dependent signal, its Fourier transform decomposes the signal by frequency and produces its frequency spectrum.

The Fourier transform of *y(t)* is represented by $\Im(y(t)) = \Im(f))$.

The formal definition of the Fourier transform of an integrable function is:

$$Y(f) = \int_{-\infty}^{\infty} y(t) \cdot exp(-i \cdot 2 \cdot \pi \cdot t \cdot f)\, dt \qquad (1)$$

Conceptually the *f* argument represents a frequency, while *t* represents a dimension (temporal or spatial).

This property of the Fourier transform of re-arranging information by frequency is extremely useful in processing different types of signals, particularly vibratory signals.

The function *y(t)* and its Fourier transform *Y(f)* form a biunivocal correspondence:

$$Y(f) = \int_{-\infty}^{\infty} y(t) \cdot exp(-i \cdot 2 \cdot \pi \cdot t \cdot f)\, dt \qquad (2)$$

Direct Fourier Transform

$$y(t) = \int_{-\infty}^{\infty} Y(f) \cdot exp(i \cdot 2 \cdot \pi \cdot t \cdot f)df \qquad (3)$$

Inverse Fourier Transform

The functions $Y(f)$ and $y(t)$ are refered to as a Fourier transform pair.

B. Unit Impulse Response Function

The unit impulse function, or Dirac delta function, has the following properties:

1. Its value is 0 over the entire time interval, except for the initial moment $t=t_0$, where its value is infinity;
2. Its integral over the entire time interval, including the initial moment $t=t_0$ is 1;
3. The integral over the entire time interval, including the initial moment $t=t_0$, of the product between the delta function and an arbitrary function $f(t)$ is the function value at t_0.

The delta function can be regarded as a limit case of an unit area impulse having infinitesimal small duration and infinite height. The unit delta function can be weighted by a weight factor (with or without physical dimension) such that the result of the integral over entire time axis of the weighted function is the value of the weighting factor at the initial moment $t=t_0$.

The Dirac unit impulse, or delta function, $\delta(t)$, is given by:

$$\begin{cases} \delta(t - t_0) = \begin{cases} \infty \ for \ t = t_0 \\ 0 \ for \ t \neq t_0 \end{cases} \\ \text{with :} \\ \int_{-\infty}^{\infty} \delta(t - t_0)dt = 1; \\ \int_{-\infty}^{\infty} \delta(t - t_0) \cdot f(t)dt = f(t_0) \end{cases} \qquad (4)$$

Considering an invariable linear system, at whose input is applied the function $f(t)$,determined by the infinite series $f(t_n)$ formed by the samples acquired at time moments t_n, equally spaced on the time axis with increment $\Delta t \to 0$. Thus it can be considered that the function $f(t)$ at time t_n is the delta function weighted by the function value at time t_n:

$$f(t_n) = \int_{-\infty}^{\infty} f(\tau)\delta(t_n - \tau)d\tau \qquad (5)$$

The response function of the linear system to the Dirac unit impulse is represented by $h(t)$. $h(t)$ is called the unit impulse response function.

Each sample $f(t_n)$, comprised of the weighted Dirac impulse, produces a response for the unit impulse whose level is proportional to $f(t_n)$, while the time origin corresponds to the origin of impulse application. The signal at the output of the linear system in the moment τ, $y(\tau)$ consists of a sum of pulses, each being delayed by an interval proportional to the time duration passed from excitation to measurement. Since each point of the response curve is a sum of components excited at different time moments, it is necessary to integrate the response over the entire time duration τ .

The response at t_0 corresponding to the $f(t)$ signal application is given by the sum of all unit impulse responses corresponding to t_0 :

$$\begin{aligned} y(t_0) &= \sum_{j=-\infty}^{\infty} \left(\int_{-\infty}^{\infty} f(\tau)\delta(\tau - t_j)d\tau \right) h(t_0 - t_j) \\ &= \sum_{j=-\infty}^{\infty} f(t_j)h(t_0 - t_j) \end{aligned} \qquad (6)$$

At limit when $\Delta t \to 0$ the sum from the previous relation becomes the convolution integral:

$$y(t) = \int_{-\infty}^{\infty} f(\tau) \cdot h(t - \tau)d\tau = f(t) * h(t) \qquad (7)$$

Thus the response of a linear system is represented as the convolution of the input signal with the unit impulse response function.

The restrictions on the convolution limits are determined by the causality condition. Causal systems are those systems in which the output signal at moment t depends only on the past and present of the input signal, but not on its future. In other words, the system output is independent of the input signal future behaviour.

The causality condition imposed on the linear system impacts the unit impulse response function. Reconsidering the convolution definition relation:

$$\begin{aligned} y(t_0) &= \int_{-\infty}^{\infty} f(\tau)h(t_0 - \tau)d\tau = \\ &= \int_{-\infty}^{t_0} f(\tau)h(t_0 - \tau)d\tau + \int_{t_0}^{\infty} f(\tau)h(t_0 - \tau)d\tau \end{aligned} \qquad (8)$$

It can be observed that the terms from the first sum require input signal values $f(\tau)$ for $\tau \in (-\infty, t_0]$, namely the past and present of the input signal. The terms from the second sum require input signal values $f(\tau)$ for $\tau \in (t_0, +\infty)$, namely the future input signal.

If the output signal must depend only on the past and present input signal, it is clear that the unit impulse response function must satisfy the following condition:

$$\begin{aligned} &h(t_0 - \tau) = 0 \ for \ \tau \rangle t_0 \\ &h(t) = 0 \ for \ t \langle 0 \end{aligned} \qquad (9)$$

Consequently, a linear system is causal if and only if their unit impulse response is 0 for negative time values. The causality condition imposes that the response of a linear and causal system should take the form of the following convolution product:

$$y(t) = \int_{-\infty}^{t} f(\tau)h(t-\tau)d\tau = \int_{0}^{\infty} h(\tau)f(t-\tau)d\tau \tag{10}$$

C. Frequency Response Function

For introducing the frequency response function, we consider a linear, causal and invariable physical system characterized by its unit impulse response function *h(t)*. The system input is the signal *f(t)*. Depending on the characteristic properties and on the input signal, the system will output the *y(t)* signal.

The frequency response function (FRF) is defined as the Fourier transform of the unit impulse response function *h(t)*. By applying the Fourier transform to both members of the previous relation, followed by the convolution theorem, which states that the Fourier transform of a two-function convolution equals the product of the functions' Fourier transforms, it results:

$$\begin{aligned} &\Im(y(t)) = \Im(f(t) * h(t)) \\ &Y(\omega) = F(\omega) \cdot H(\omega) \end{aligned} \tag{11}$$

where $H(\omega)$ represents the frequency response function of the system under test.

The previous relation gives the frequency response function (FRF):

$$H(\omega) = \frac{Y(\omega)}{F(\omega)} \tag{12}$$

So the frequency response function of a system is the Fourier transform of the unit impulse response function *h(t)* and is determined as the ratio of the Fourier transforms for the response and input functions.

The frequency response function is a complex quantity that can be expressed by its real and imaginary components. Multiplying (9) with the complex conjugate $\bar{F}(\omega)$ gives the equivalent frequency response function:

$$H(\omega) = \frac{S_{yf}(\omega)}{S_{ff}(\omega)} = \frac{S_{yy}(\omega)}{S_{fy}(\omega)} = |H(\omega)|e^{i\varphi(\omega)} \tag{13}$$

Where:

$$\begin{aligned} &S_{yf}(\omega) = Y(f) \cdot \bar{F}(f), \\ &S_{fy}(\omega) = F(f) \cdot Y(f) \end{aligned} \tag{13.1}$$

Power interspectrum of input/response

$$\begin{aligned} &S_{ff}(\omega) = F(f) \cdot \bar{F}(f), \\ &S_{yy}(\omega) = Y(f) \cdot \bar{Y}(f) \end{aligned} \tag{13.2}$$

Power autospectrum of input/response

The following definitions are used in conventional system analysis:

1. Frequency response function – the ratio between the Fourier transforms of the response (acceleration, speed or displacement) and the excitation force;

2. Transfer function – the ratio between the Fourier transforms of the response (acceleration, speed or displacement) and the excitation force;

3. Transmissibility – the ratio between the Fourier transform of the response in two different points (similar responses in acceleration, speed or displacement).

III. DETERMINING THE DYNAMIC CHARACTERISTICS OF THE EPOXY RESIN AND FIBERGLASS ROD

A. Experimental Modal Analysis

The experimental modal analysis (EMA) is the procedure of establishing the mathematical model of a structure based on the experimental data obtained by measurements performed on the structure in a controlled vibratory state. The structure is subject to controlled excitation, and based on the evolution laws of the vibratory excitation and response there is identified a minimal number of parameters that describe the proper vibration modes: self oscillations, damping factors, modal forms. The modal parameter set defines the modal model uniquely associated to the real system and allows theoretical evaluation of the structure response to different excitations applied, concentrated in real point (weight lift, carriage displacement) or distributed among the structure (wind). The modal model allows changes to the real structure and theoretical evaluation of the modified system's response to external or internal actions, determining the optimal modifications for obtaining the desired vibratory response system.

The response is expressed in accelerations, displacements, bending moments or mechanical tensions in the structure, currently defined for points in which it was experimentally determined the vibratory excitation or response.

Identifying self oscillations is important, since localization of structural resonances in the excitation frequency domain leads to undesired amplifications of the vibratory response. For a given structure, resonation displacement emphasizes material wear, resistance structure cracks and weakening of the assembly components.

Modal forms are geometric representations of the dominant structure movements at own frequencies. Analysis of the modal forms shows the weak points and the loosening or breaking areas. The dangerous frequencies for the structure or the equipment can be identified. Vibration sensitive equipment is not to be placed in the antinodes of the predominant excitement areas. If there is no possibility of changing equipment position, the structure must be optimized for an acceptable vibration level in the sensitive equipment area.

Currently, experimental modal analysis is used in conjunction with the finite element analysis which uses a similar mathematical model obtained by a discrete representation of the real structure into predefined finite element types between which are defined interaction laws and external factor influences. Depending on the software used (Ansys, Nastran, Abaques, etc.) and the complexity of the structure under test, the results of the analysis can accurately describe the behavior of the real structure.

As general rule, any finite element analysis contains errors than can prove significant due to incorrect specification of the material or geometric structure specification, even if the geometric model seems accurate. For this reason, every finite element model must be validated through experimental data.

B. Measuring Equipment Used

1. Spider 8 acquisition system, 12-bit resolution;
2. NEXUS 2692-A-0I4 signal conditioner, linearity 0.01%;
3. Bruel & Kjaer 4391 accelerometer, linearity 2%;
4. Bruel & Kjaer 8309 accelerometer, linearity 2%;
5. Bruel & Kjaer tip 8206 impact hammer, linearity 1%;
6. Notebook IBM ThinkPad R51 – 1 pc.

C. Recorded Parameters

1. Acc1, ..., Acc4(m/s^2) – response accelerations;
2. F(N) – excitation force;

D. Impact Hammer

The function principle is similar to that of an accelerometer, the difference consisting in the force being applied directly to the piezoelectric crystal, due to missing of the spring between the crystal and the seismic mass.

Fig. 2. 8206 Bruel & Kjare impact hammer.

It has the following frequency characteristics:

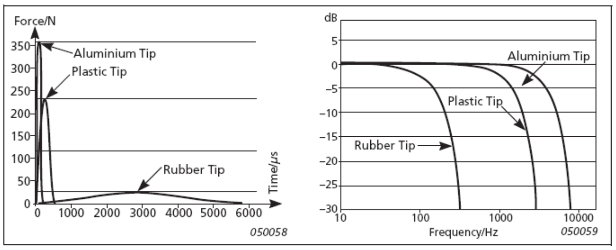

Fig. 3. Frequency characteristics of 8206 Bruel & Kjare impact hammer.

Some of the functional technical characteristics are:

1. Load sensitivity: 22.7 mV/N
2. Force domain: 220 N
3. Frequency domain depending on the attenuator used
4. Measurement error: ≤ 1%

IV. TRIALS

The experiments were performed within the TCM Mechanical Faculty laboratory of the University of Craiova on a cylindrical rod with ϕ10 mm and length 290 mm. There were two types of experiments, one with the console mounted rod and the other with the encased rod.

A. Determining the Dynamic Characteristics of the Encased Rod

The ϕ10 mm, 290 mm long cylindrical rod was mounted in the chucks of the SNA 500 universal lathe. The rod was divided into 4 equal segments of 72.5 mm each, having the measurement points P1, P2, P3 and P4 at L/4, L/2, 3L/4 and L relative to the left hand side head.

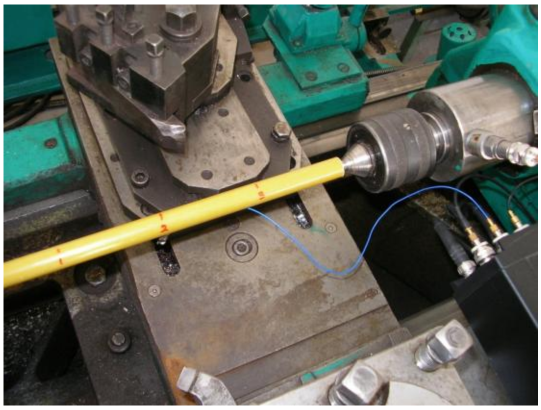

Fig. 4. Experimental setup, showing the position of the measurement points.

The B&K 8309 accelerometer was mounted successively into P1, P2, P3 and P4, measuring the response accelerations Acc1, Acc2, Acc3 and Acc4. For each accelerometer position a force impulse was emitted with the impact hammer B&K 8206 (P4 corresponds to hitting the loose end of the rod).

Fig. 5 presents a typical recording obtained by successive excitation in P1 and P2 and measuring the vibratory response in P1. Similar representations are obtained for all excitation/measurement cases. From the graphical representation, it can be observed that in this case the vibratory response is the composition of two vibratory movements.

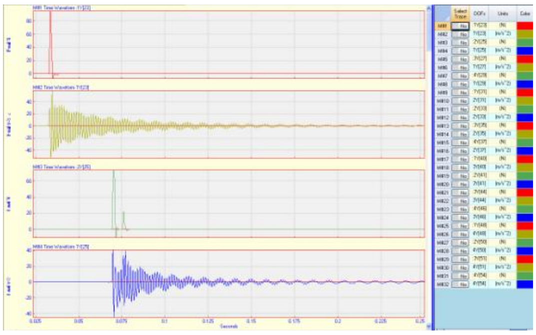

Fig. 5. Mounted rod. Original recording. Excitation P1, P2. Measurement P1.

Using the previously presented methodology, there are determined the frequency response functions for all excitation/measurement combinations. Fig. 6 – 10 present the response functions corresponding to successive excitations applied to P1 – P4 and the measured vibratory response.

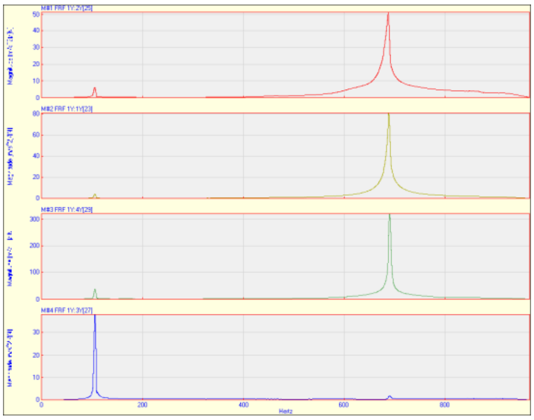

Fig. 6. Frequency response functions. Excitation P1 … P4. Measurement P1.

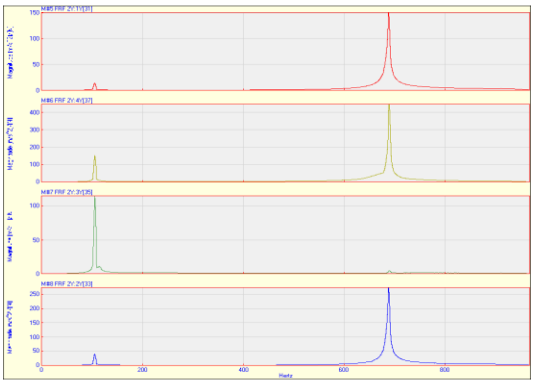

Fig. 7. Frequency response functions. Excitation P1 … P4. Measurement P2.

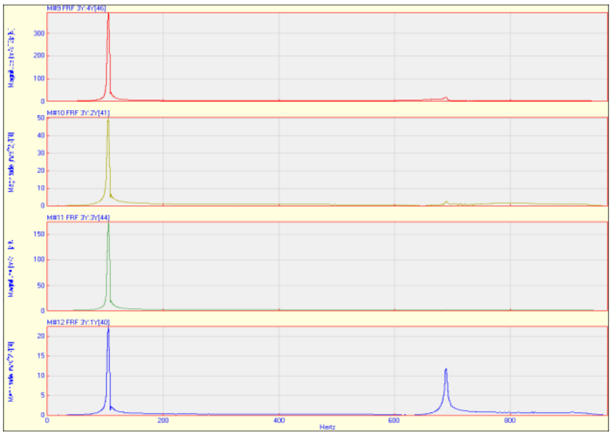

Fig. 8. Frequency response functions. Excitation P1 … P4. Measurement P3.

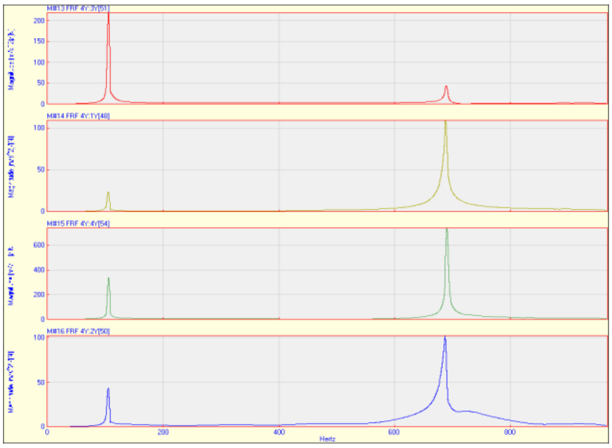

Fig. 9. Frequency response functions. Excitation P1 … P4. Measurement P4.

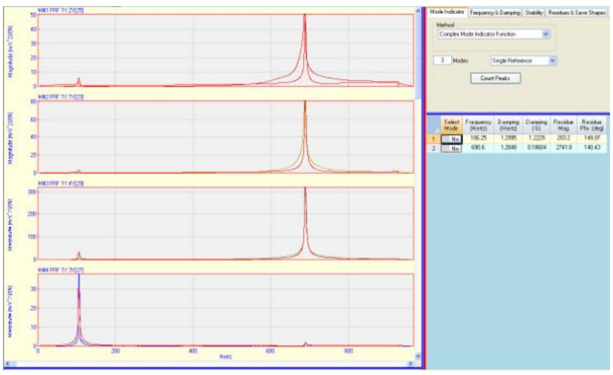

Fig. 10. Modal parameters indentification. Excitation P1 … P4. Measurement P4.

FRF analysis shows that the encased rod presents two resonance frequencies at 106.25 Hz and 690.6 Hz. In the second vibration mode the rod has a vibration node in P3.

The modal parameters of the rod are determined by running the modal identification module.

The modal form of the mounted rod in its own vibration modes is presented in Fig. 11 and Fig. 12, confirming that P3 is a vibration node for the second vibration mode.

TABLE I
MODAL PARAMETERS

Mode	Resonance frequency [Hz]	Damping ratio [Hz]	Critical damping ratio [Hz]
1	106.25	1.2995	1.2229
2	690.6	1.2848	0.18604

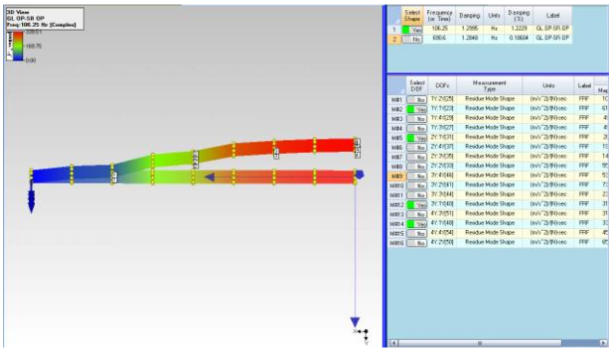
Fig. 11. Modal form at 106.25 Hz

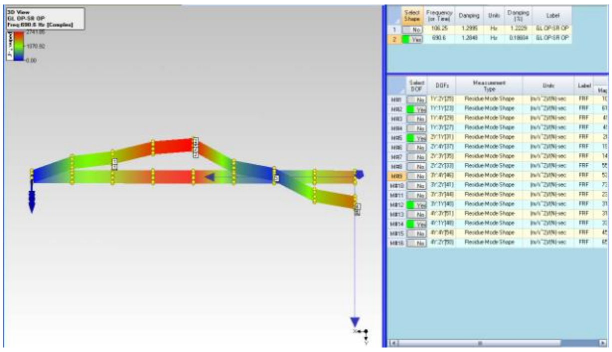
Fig. 12. Modal form at 690.6 Hz

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