Structural Mechanics

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Assessment of the Systems Dynamic Characteristics Using Identification Technique

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Summary

This study, presents the assessment of the systems dynamic characteristics using different identification techniques.

KEYWORDS: the Fourier transformation of the answer, transfer function, amplitude, phase, frequency, mass matrix, damping matrix, rigidity matrix,

1. INTRODUCTION

The dynamic identification of structures can be considered a set of techniques allowing the determination of the physical parameters occurring in equations, which describe the behaviour of structure, subject of dynamic actions.

These techniques were developed due to the difficulties in the exact assessement of rigidity, dampings and masses of real structures.

In most of the identification methods it is acted from the outside upon the system with an iposed, known excitation, making easier the interpretation of the measurements. It is necesary that during the experimentation, the influence of other disturbing sources should be reduced to the minimum and the equipment used in the excitation of the structure, as well as the one used in the measurement of the answer should not considerably change its parameters.

2. TYPES OF EXCITATIONS USED IN THE STRUCTURES IDENTIFICATION

The dynamic characteristics of a system where are considered a single input and a single output are described in the time domain by the weight function h(t), and in the frequencies domain by the answer function in frequency (FRF, transfer function) and H(jω) with which a Fourier transformation couple is formed:

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$$H(\omega) = \frac{Y(\omega)}{U(\omega)} \tag{1}$$

Ussing the following relations, the FRF will have a value:

$$U(\omega) = \int_{-\infty}^{\infty} u(t)\, e^{-j\omega t}\, dt \qquad Y(\omega) = \int_{-\infty}^{\infty} y(t)\, e^{-j\omega t}\, dt \tag{2}$$

where, Y(ω) is Fourier transformation of the answer, and U(ω) is Fourier transformation of the excitation.

2.1 The Harmonic Excitation

In this case the structure is excited by means of an electrodynamic generator, execising upon the structure a sinusoidal punctual force whose amplitude, phase and frequency are ruled.

The identification of the dynamic characteristics is performed in the frequency domain.

In the linear vibration they correspond to the peaks oscillation from the curve of the transfer function.

The modal damping ratios are determined through the classic method of the semipower. The modal forms are determinated by the ordinate function of these peaks and their sign is obtained by the phase function of the transfer function.

The imput and the output are:

$$u(t) = u_0\, e^{j\omega t} \qquad y(t) = y_0\, e^{j(\omega t + \varphi)} \tag{3}$$

and the transfer function is:

$$H(\omega) = \frac{y_0}{u_0} e^{j\varphi} \tag{4}$$

The modulus of the transfer function is obtined from the amplitude-pulsation characteristic (y_0/u_0-ω) and the argument of this (φ) function from the phase-pulsation characteristic (φ–ω). These two characteristics can be traced either point by point, performing measurements on discrete frequencies in fixed regime, or continuously in quasistationary regime, using a frequent scavenging slow enough to allow the establishment of the answer on every frequency especially in the area of weak damping resonance.

Now, the identification methods with harmonic excitation are the most used, but in case of model coupling or of structure nonlinear behavior, the result can be often wrong.

2.2 The Excitaton With Transitory Signals Or Impulses

The simplest shape is represented by rectangular, triangular, trapezoidal, sinusoidal form, a.s.o.

The amplitudes spectrum of these impulsee are annulled on certain frequencies and theoretically have an infinite domain of frequencies (fig. 1).

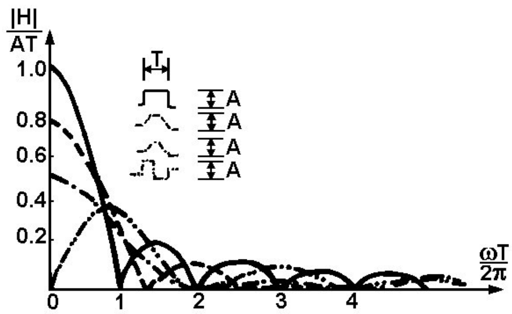

Fig.1 The amplitudes spectrum of the impulsee

In case of the system with nGLD there exists the posibility of certain resonaces failure to be excited, as well as the possibility of excitation of resonances outside the interest domain.

Using a cosntant amplitude sinusoidal excitation and a time variable frequency, these shortcomings can be eliminated.

In case of studies on reduced scale models the excitation through striking with a special hammer is enough in order to obtain u(t) and y(t) signals which, undertaken by Fourier analyzer in real time give directly the answer function in frequency.

2.3 The Excitaton With Accidental Signs

The excitation force can be the wind or the earthquake. The behavior of the structure is considered linear. In this case all the spectral components are simultaneously excited and analyses in real time can be made.

Ussing the following relation, results the FRF:

$$H(\omega)=\frac{S_{uy}(\omega)}{S_{uu}(\omega)} \tag{5}$$

where, $S_{uu}(\omega)$ is the power spectral density of the excitation

$S_{uy}(\omega)$ is the power interspectral density of the excitation and the answer.

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If $S_{uu}(\omega)$ = S_0= const. ("the white noise"), than, $S_{uy}(\omega)$ = $S_0H(\omega)$, and the intercorrelation function between excitation and answer $R_{uy}(t)$ can be determinate with the following relation:

$$R_{uy}(\tau) = \int_{-\infty}^{\infty} S_{uy}(\omega) e^{j\omega\tau} d\omega \qquad (6)$$

The trials in accidental regime have the advantage of simultaneously excite all the spectral components of the interest domain, allowing an identification of the structures with time variable parameters. Using an exponential mediation in the calculus of the specters S_{uu} and S_{uy} an analyses in real time can be made. The procedure is useful in the optimization of a structure answer through the modification of the masses, rigidity, and damping distribution.

3. IDENTIFICATION PROCEDURES OF THE PROPER VIBRATION FREQUENCIES AND FORMS

The various identification procederes propose to minimize a comparision criteria starting from the different between the answering of the model and the one measured in the real system. The experimenter is confronted with the problem of determine the characteristics of the system represented by the masses (**M**), damping (**C**) and rigidity (**K**) matrixes, which are not directly measurable out of such quantities as frequency, modal forms.

The tests of the complex great structure is performed in two stages:

A – it is roughly determited the number of the proper vibration mode and the resonance frequency using a single vibrator;

B – the modes are isolated through a distribution corresponding to a vibrators number along the structure, matching the excitation forces from the vibrators, so that only the mode subject of interest should be dominatly excited.

The most simple technique used for the determination of the resonance frequencies is the method of the amplitude peak. The structure is acted by sinusoidal force from a sole vibrator and the answer curves are registered under the form of the answer function in frequency (FRF).

As the answer of the structure is due to the answer of all proper modes simultaneuosly, the measured proper forms are seldom distorsion. This problem is amplified in case of structures with close natural frequencies when the separation of the modules becomes difficult.

As the precision of the modal parameters dipend by the precision which the FRF can be measured, we must give a special importance to the excitation module of the structure, to mentain the vibration amplitudes at the same level in every cases.

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This thing becomes easy to achieved in the frame of simple structures. In case of great structures with numerous links and un-proportioned damping the nonlinear effects and the proximity of the modes are frequent and not only that hinders the localization of frequencies where the modules has to be identified but also the forms established for a certain positions of the vibrator might not coincide with those established on other positions of the vibrator.

In order to eliminate these difficulties the structure can be excited in several point simultaniously.

3.1. The Identification Of The Frequencies And The Modal Forms Using The Sole Excitation

A linear and elastic structure excited by a sinusoidal force will answer directly proportional with the excitation force, having also the same frequency.

The measurement of the excitation force and the answer in a number of points from the whole domain of frequencies is enough to describe the behavior of the structure. The information can be represented through drawing the relation between answer and excitation as a frequency function, the answer being in displacement, velocity or accelerations.

It is considered a system where the action is after the GLDk direction:

$$F_k = \overline{F}_K e^{j\theta t} \tag{7}$$

the answer being measured in the joint ”i”, having the expression:

$$y_l = \overline{y}_l e^{j\theta t} \tag{8}$$

The complex admitance, which represents the ratio between the answer in displacement and the action (force) can be also written:

$$\overline{H}_{lk} = \frac{y_l}{F_k} = \overline{H}_{lk} e^{j\varphi} = \overline{H}' + j\,\overline{H}'' \tag{9}$$

where, H’ is the real part of the admittance
H" is the imaginary part of the admittance.

In the case of the system with proportional damping, the complex admittance is graphically represented usually by one of the forms:

- as amplitude – pulsation diagram and phase – pulastion diagram;
- as vibration diagram with pulsation of the answer component in phase with the force and the component in quadrature with the force;
- as polar diagram representing the geometrical place of the extremity in complex plane, having the pulsation ω as parameters.

The various analyses methods of the answer in frequency are different first of all, through the hypothesis made upon the contributions of the ”nonresonant” vibrabration modes at the total answer in the proximity of resonance frequency and separation procedure of the vibration modes when they close the pulsations.

No matter the adoptive method, the aim is the same, that is to determine the following measures:

- the resonance pulsation;
- the proper forms $\{Y_{ki}\}$ and modal matrix [Y];
- the modal parameters ν_i, m_i, k_i.

so, the matrixes [M], [C], [K] can be know.

3.2 Identification Of The Linear Systems, Harmonically In Several Joints

A complex of the structure vibrates simultaneously in several modes. For a correct analyses of the results the undesired modes must be eliminated. This can be achievd through the excitation of the structure using several vibrators forcing the structure to vibrate in its main modes. When the structure is proportionally damped it can be excited in any frequencies through a paricular set of forces which are in phase or antiphase, one with the other, so that the answer measured in every point should be all in phase or antiphase and the characteristic phase delay between the force and the answer is unique. In the frequency there exists a number of characteristic phase delays associeted to the distribution of independent linear forces equal to the GLD number. The structure excited in this way for a particular ratio of the forces shall vibrate in the main mode as a system with 1 GLD.

If the structure is nonproportionally damped, it can be excited in its main mode only at the natural frequency corresponding through a set of monophasic forces. We are actually interested in the conditions in which the exciting of proper vibration can be achieved.

It considered the system with nGLD with a viscous damping, whose movement equation is:

$$[M]\{\ddot{y}_k(t)\}+[C]\{\dot{y}_k(t)\}+[K]\{y_k(t)\}=\{F_k(t)\} \qquad (10)$$

or in generalized coordonate:

$$[M_{gi}]_i\{\ddot{q}_i(t)\}+[C_{gi}]\{\dot{q}_i(t)\}+[K_{gi}]\{q_i(t)\}=\{F_{gi}(t)\} \qquad (11)$$

We have to determine the pulsation harmonic excitation θ, that makes the system to vibrate in the main mode at the pulsation ω_i, that is generating an answer:

$$\{y(t)\}=A_0\{Y_i\}e^{j(\theta t-\varphi)} \qquad (12)$$

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where, φ is trail behind of the answer confronted by the excitation;
A_o is the amplitude factor.

If A_o=1 we can know the excitation force:

$$\{F\} = \{\overline{F}\} e^{j(\theta t + \varphi)} \tag{13}$$

which will generate an answer:

$$\{y(t)\} = \{Y_i\} e^{j\theta t} \tag{14}$$

Knowing the folowing relation:

$$\{y(t)\} = \sum_{i=1}^{N_m} \{Y_i\} q_i \tag{15}$$

the relations (14) şi (15) must be equal:

$$\{y(t)\} = \sum_{i=1}^{N_m} \{Y_i\} q_i = \{Y_i\} e^{j\theta t} \tag{16}$$

So, for isolate of the mode ”i”, it is necessary that in the equation (11) to replace:

$$\{q_i(t)\} = \{I\}_i e^{j\theta t} \tag{17}$$

where, $\{I\}_i$ is the "i" column in the matrix [I]:

$$\{F_{gi}(t)\} = \{\overline{F}\} e^{j(\theta t + \delta)} \tag{18}$$

Replacing (17) and (18) in (11), simplifying with e^{j0t} and spareting the real part from the imaginary one, we get:

$$([K_{gi}] - \theta^2 [M_{gi}])\{I\}_i = \{\overline{F}\} \cos\varphi \tag{19}$$

wher, $\{C\}_i$ is the "i" column in the matrix $[C_{gi}]$.

It comes out that in order to excite a structure in one of the main vibration modes the force distribution must have the following form:

$$\theta\{C\}_i = \{\overline{F}\} \sin\varphi \tag{20}$$

The relations (19) and (20) can be written in another way:

$$(\omega_i^2 - \theta^2)[Y]^T [M]\{Y_i\} = [Y]^T \{F'\} \tag{21}$$

$$\theta [Y]^T [C]\{Y_i\} = [Y\}]^T \{F''\} \tag{22}$$

where, are evidencing the two vectorial components of the excitation:

$$\{F\} = \{F' + j\,F''\}\,e^{j\theta t} \tag{23}$$

Simplifying with $[Y]^T$, we get:

$$\{F'\} = (\omega_i^2 - \theta^2)[M]\{Y_i\} \qquad \{F''\} = \theta[C]\{Y_i\} \tag{24}$$

It results that for excitate a structure in one of the main vibration mode, the distribution of the forces must be:

$$\{F\} = (\omega_i^2 - \theta^2)[M]\{Y_i\} + \theta[C]\{Y_i\} \tag{25}$$

The component {F'}in phase with the displacement is necessary for the balancing of the harmonization elastic and inertial forces and the component {F"} is in quadrature before the displacement and it is necessary for the balancing the damping forces. Between the two components, there is a relation:

$$\{F''\} = \frac{\theta}{\omega_i^2 - \theta^2}[C][M]^{-1}\{F'\} \tag{26}$$

For the excitation of the pure modes of the system distribution of forces in phase, is used:

$$\{F\} = \{\overline{F}\}\,e^{j\theta t} \tag{27}$$

where, {F} is a vector with real elements.

4. CONCLUSIONS

In case of great structures with numerous links and non-proportional damping, the effects of nonlinearity and the proximity of the modes are the frequencies, a fact implying the positioning of the frequencies where the modes have to be identified. Out of this reason there were determined the identification techniques of the frequencies and modal forms specific to the structure using the unique excitation or excitations from several points.

The excitations for the identification of the system can be hamonic, transitory, accidental type or excitations of the normal functioning.

According to the goal of the identification the results can be used for the verification and validation of the analytical models, the determination of some measures, which are not directly measurable the foreseeing of the structural modification effects, determination through calculus of the answer at other excitations or several simultaneous excitations based on the answer measured for a certain type of action.

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