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The Damping and the Dynamic Stabillity of Thin Plates Parametrically Excited

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Abstract: In the paper are presented the experimental data for the structural damping of thin rectangular plates parametrically exited. The tests were carried into a hydro-pulse system MTS onto different specimens of plates. For the same plates were experimental obtained the frontiers of the instability areas in the region of principal parametric resonance. The data were compared with theoretical results. In addition, the paper deals with the correction for the theoretical model; correction established starting from the values of the structural damping.

Key words: damping, dynamic stability, plates

1 Introduction

The simply supported thin rectangular plates parametrically excited represent a field of research for which an extensive effort and considerable amount of investigations have been concentrated.

Excellent reviews on the subject can be found in articles written by Hui [2-8] , as well as Ilanko and Dickinson [9] and Bugaru [1].

The survey of the literature reveals that the work on the subject has been devoted to the investigation of various types of shapes, loadings and boundary conditions [11-13].

The present paper covers the area of experimental data concerning the structural damping of such plates and the data for the frontiers of the instability areas in the region of parametric resonance.

2 Experimental Apparatus and Test Procedures

The entire experimental system can be divided into 3 sub-systems:

A. Sub-system for the excitation of the plates ;

B. Sub-system for digital and analogical measurements ;

C. Sub-system for the recording of experimental data.

The sub-system for the parametrical excitation of this rectangular plates consists of: a hydro-pulse MTS Model 891, the special frame design to fix the plates so that the boundary conditions are simply supported. In figure 1 is shown the hydro-pulse scheme

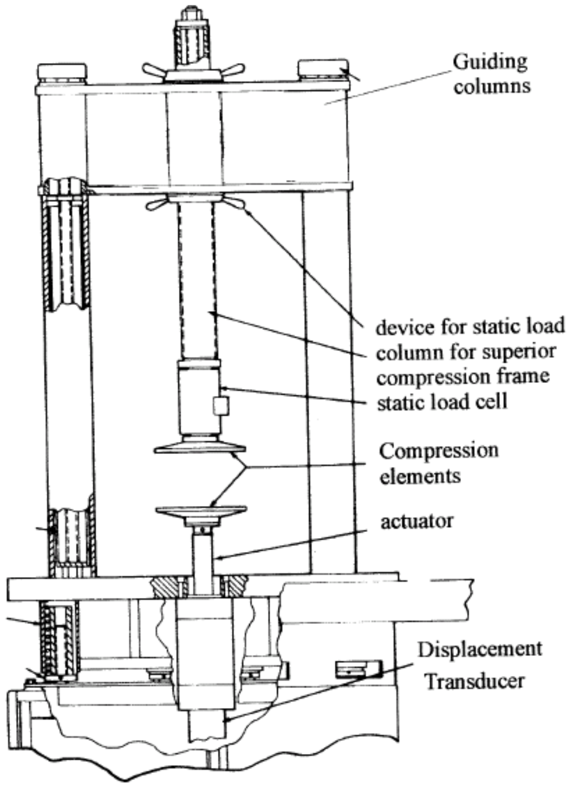

Figure 1

and in figures 2 and 3 are illustrated the special frame and a detail of it, frame in which is "fixed" the test specimen (the plate) having the boundaries simply supported.

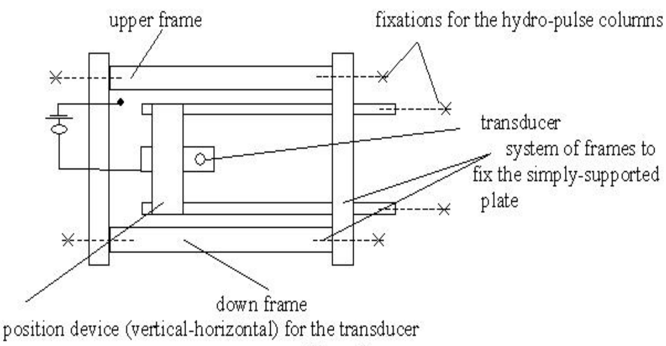

Figure 2

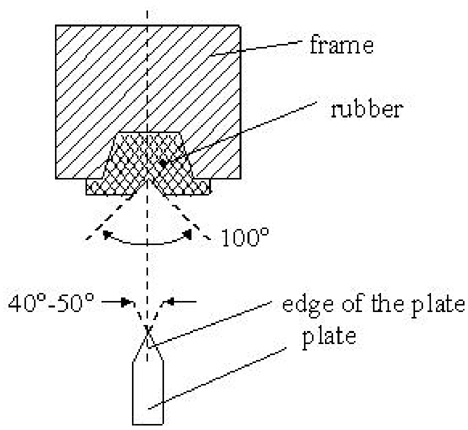

Figure 3

To obtain the above-mentioned boundary conditions for the plate the special frame (see fig.2 and fig.3) consists of two parts a metallic one and a rubber part. The rubber part has an opening with an angle of 100^o while the edge angle of the plate is of 40^o ÷ 50^o. This geometrical aspect of the rubber part permits the free rotation of the mid-plane of the plate with ± 25° ÷ 30° and represents the simply supported practical boundary condition.

The elements of compression of the hydro-pulse permit the static and dynamic load, through the actuator, of the plate in its mid-plane realizing the parametric excitation of the plate.

By this way it is transmitted to the plate the parametric excitation, uniform distributed over two opposite edges, having the mathematical expression:

$$N_y(t) = N_{yo} + N_{yt} \cos(\theta(t)) \tag{1}$$

where N_{yo} is the static load, N_{yt} is the amplitude of the dynamic load and $\eta = \frac{d\theta}{dt}$ is the frequency excitation.

The sub-system for digital and analogical measurements consists of:
- a central unit Model 436 that controls the function generator for the excitation ;
- a transducer Model 440.21&22 which measures the static and dynamic loads ;
- a controller Model 442 which measures the elastic equivalent constant and the loss factor β of the structural damping ;
- a controller-computer Model 429 which controls the input data (amplitude of dynamic load and frequency excitation) ;
- amplifier, filter and displacement transducer contactless to detect the smallest amplitudes of the lateral vibrations of plate.

The sub-system for the recording of experimental data consists of:
- multi digital input switch Model 430;
- oscilloscope to visualize the hysteretic loop of the plate under dynamic load.

For experiments it were used 4 test specimens of rectangular plates having the dimensions : 150x150, 150x225, 150x300, 150x450 [mm x mm] and the thickness h=1mm. All the plates are of polycarbonate, therefore they have the following physical properties:
- the Young's modulus E = 2.3856 GPa
- the Poisson's ratio ν = 0.45
- the density ρ = 1205.5 kg/m^3

To obtain the electrical conductivity of the plates for measuring contactless the lateral displacements, it was used a solution of ECCOCOAT. CC-2 produced by Emerson & Cuming Ltd. (USA).

To have a high precision for the measurements it was necessary to calibrate the entire system for compensating the mechanical effects of the special frame used to fix the plate.

This was done, in virtue of ISO/DIS 4664/1996, by using an etalon rubber bar in the range of excitation 5 ÷ 100 Hz.

3 Experiments and Results

The experiments were carried out in the region of principal parametric resonance that is for

$$\eta \cong 2\,\Omega \tag{2}$$

where Ω is the free vibration circular frequency of the rectangular plate loaded by a constant component of in plane force.

Using the contactless transducer for displacements it were determined the frontiers of instability region in the space (μ,η) and (μ,η/2Ω), where μ represents the load parameter defined by the relation

$$\mu = \frac{N_{yt}}{2\left(N_{cr} - N_{yo}\right)} \tag{3}$$

N_{cr} being the critical buckling load of the plate. The results are presented in figures 4 and 5, where by continuos line were represented the frontiers of instability regions computed theoretical [1]. As can be seen there exist a close agreement between experimental data and theoretical values. The experiments were conducted for the test specimen having the dimensions 150 x 300 mm.

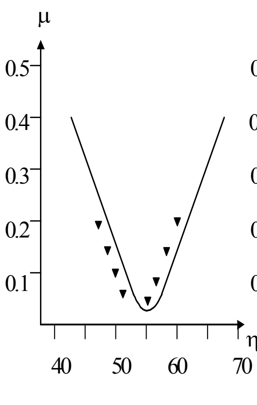

Figure 4

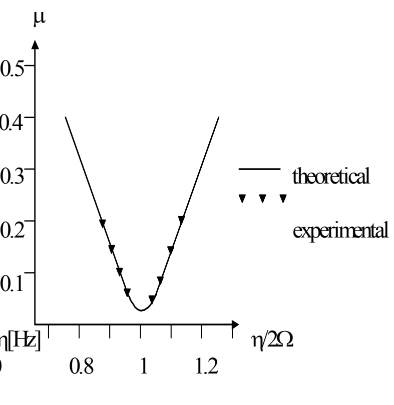

Figure 5

In figure 6 is illustrated the variation of the loss factor β = tg δ for two test specimens: 150 x 300 [mm x mm] (a) and 150 x 225 [mm x mm] (b).

It is obvious that the structural loss factor tg δ depends upon both frequency excitation η and the dynamic load amplitude.

In the region of principal parametric resonance (50 ÷ 60) Hz the loss factor is maximum, there fore we can conclude that the plate, through its internal mechanisms, increases the structural damping and there fore attemates the amplitude of lateral vibrations. Using the values of loss factor tg δ, by regarding ISO/DIS 4664/1996, it can be computed the equivalent logarithmic decrement of the plate on the basic equation:

$$tg\,\delta \cong \frac{\Delta}{\pi + 0.079577\,\Delta^2} \tag{4}$$

and also the equivalent viscous damping coefficient using the relation:

$$c = \frac{\Delta \cdot \Omega}{2\pi} \tag{5}$$

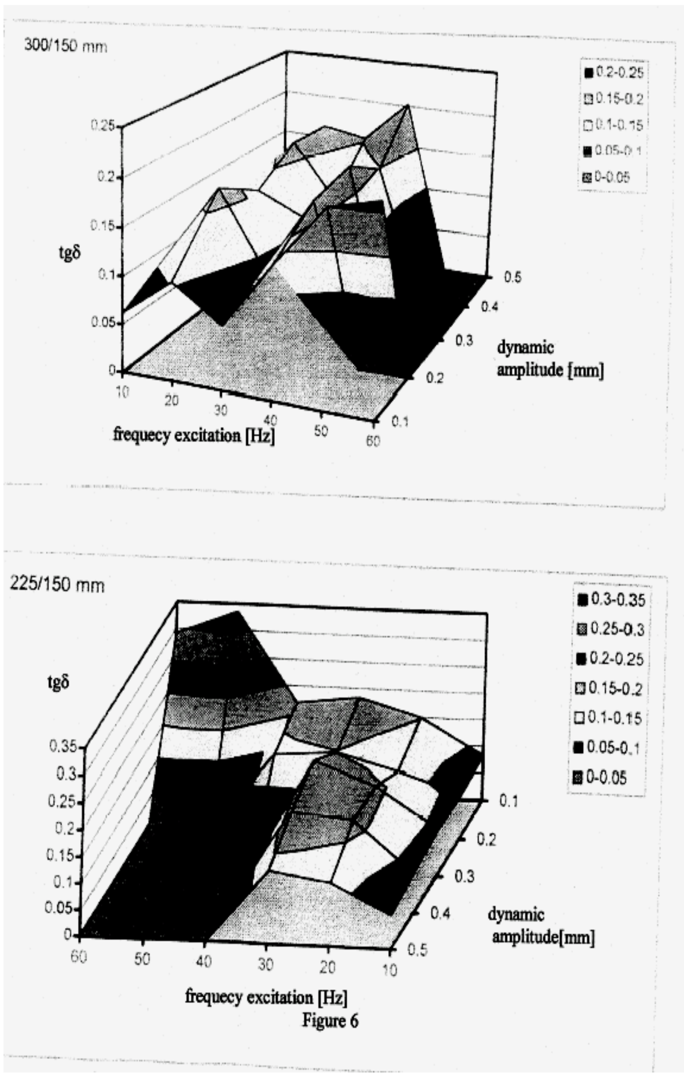

Figure 6

4 Conclusions

The present paper covers the area of experimental researches in the field of parametric vibrations of the rectangular plates simply supported.

By this way was validated the theoretical model, developed by the author [1], concerning the non-linear dynamic instability of thin rectangular plates parametrically excited.

Also, the paper reveals a new method to compute and predict viscous coefficient damping of the plates, based on the structural damping experimental determined.

This represents a correction to the theoretical model of parametric vibrations of thin rectangular plates developed by the author [1].

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