Experiments on Nonlinear Vibrations of Graphite/Epoxy Composite Curved Panels

M. Amabili, S. Carra, R. Garziera and M. Pellegrini

Dipartimento di Ingegneria Industriale,
Università di Parma, Area delle Scienze 181/a, Parma, 43100 Italy
marco@me.unipr.it

1. SUMMARY

Large-amplitude (geometrically nonlinear) vibrations of circular cylindrical panels, subjected to radial harmonic excitation in the spectral neighborhood of the lowest resonances, are investigated. Vibration response to harmonic excitation in the neighborhood of the natural frequencies of two different thin circular cylindrical panels of graphite-epoxy composite material (a plain-weave fabrics panel and a three-layer panel with 90°-0°-90° orientation) has been measured for different force levels. The experimental boundary conditions approximate (i) on the curved edges: zero radial and circumferential displacements; all rotations were allowed; (ii) on the straight edges: zero radial displacements; all rotations and circumferential displacements were allowed. The different levels of excitation made it possible to reconstruct the relatively strong, softening type nonlinearity of the panels.

2. INTRODUCTION

Amabili and Païdoussis /1/ recently compiled an extensive review of work on geometrically nonlinear (large-amplitude) vibrations of shells and curved panels. It was found that not many experimental investigations on large-amplitude vibrations of circular cylindrical shells are available. In particular, it seems that experiments on nonlinear vibrations of circular cylindrical panels were reported only in the work of Nagai *et al.* /2/, but the trend of nonlinearity, which is obtained by performing tests with different levels of the excitation force, was not given. For this reason, Amabili *et al.* /3/ performed experiments on a stainless steel circular cylindrical panel and found a relatively strong nonlinearity of softening type. Also other experiments on nonlinear dynamics of curved panels are very scarce. Palazotto *et al.* /4/ tested composite circular cylindrical panels subjected to impact.

Many theoretical and numerical studies are available on geometrically nonlinear vibration of circular cylindrical panels with different boundary conditions; see *e.g.* the review by Amabili and Païdoussis /1/ and the papers of Leissa and Kadi /5/, Hui /6/, Fu and Chia /7/ and Raouf /8/.

The experimental response of panels as function of the frequency of harmonic excitation and vibration amplitude is fundamental for the validation of nonlinear shell theories and for solution algorithms, as shown by the results that have been already published by the authors in references /3-9/. In particular, in the present study experiments on large-amplitude vibrations of composite curved panels with rectangular boundaries were performed. The experimental boundary conditions approximate (i) on the curved edges: zero radial and circumferential displacements; all rotations were allowed; (ii) on the straight edges: zero radial displacements; all rotations and circumferential displacement were allowed.

3. EXPERIMENTAL SET-UP

Tests have been conducted on two circular cylindrical panels made of graphite/epoxy composite material: (i) a plain-weave fabrics panel and (ii) a three-layer panel with 90°-0°-90° orientation (assuming 0° as the longitudinal direction). The common dimensions of the two panels are: length between supports L = 522 mm, radius of curvature R = 150 mm and angular width between supports 72° (curvilinear width length b = 185 mm).

The first panel has global thickness h = 0.56 mm and is made of two symmetric layers of plain-weave fabrics of graphite/epoxy with global Young's modulus $E = 55 \times 10^9$ Pa in the two principal orthogonal directions, mass density $\rho = 1010$ kg/m^3 and Poisson ratio $\nu = 0.2$.

The second panel has total thickness h =0.66 mm, mass density ρ = 1335 kg/m^3 and each layer is characterized by $E_x = 160 \times 10^9$ Pa on the direction x of the fibers, $E_y = 7.9 \times 10^9$ Pa on the cross direction y and $\nu_{x,y}$= 0.16.

Each panel was inserted into a heavy rectangular frame, see Figure 1, having grooves designed to hold the panel. The frame was built with a mobile side, a screw and a load cell to give the desired axial load to the panel. Silicon was placed into the grooves to hold better the edges of the panel and then avoid radial displacements at the edges. The circumferential displacements at the curved edges were prevented by friction between the panel and the grooves (due to a small axial compression) and by silicon. The axial and circumferential displacements on straight edges were allowed because the constraint given by silicon on this displacement was small.

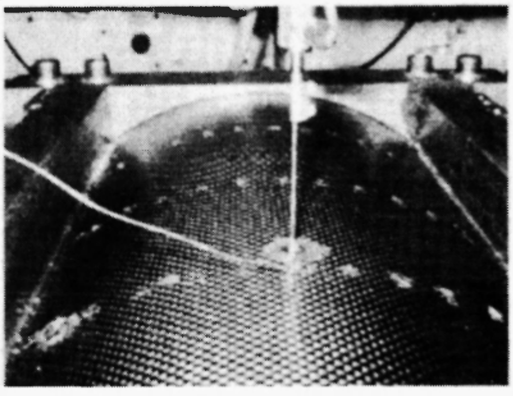

Fig. 1: Photography of the experimental set-up. Plain-weave fabrics panel.

Each panel has been subjected to (i) burst-random excitation to identify the natural frequencies and perform a modal analysis by measuring the panel response on a grid of points, (ii) harmonic excitation, increasing or decreasing by very small steps the excitation frequency in the spectral neighbourhood of the lowest natural frequencies to characterize nonlinear responses in presence of large-amplitude vibrations.

The excitation has been provided by an electrodynamical exciter (shaker), model LDS V406 with power amplifier LDS PA100E, connected to the shell by a thin stinger glued in a position close to the centre of the panel, in particular 11 mm away axially. A piezoelectric force transducer, model PCB M209C11, placed on the shaker and connected to the panel with a stinger, measured the force transmitted. The shell response has been measured by using a sub-miniature accelerometer, model Endevco 22, of mass 0.14 g. For all nonlinear tests, the accelerometer has been glued at the middle of the panel length at different angular positions corresponding to the antinode of the mode excited. The time responses have been measured by using the Difa Scadas II front-end connected to a HP c3000 workstation and the software CADA-X of LMS for signal processing and data analysis; the same front-end has been used to generate the excitation signal. The CADA-X closed-loop control has been used to keep constant the value of the excitation force for any excitation frequency, during the measurement of the nonlinear response.

4. NATURAL FREQUENCIES AND MODES

The Frequency Response Functions (FRFs) have been measured between 91 response points and one single excitation point. Both excitation force and measured responses have been in the radial direction. The response points have been located on a grid of seven equidistant circumferential arcs and 13 positions on each arc.

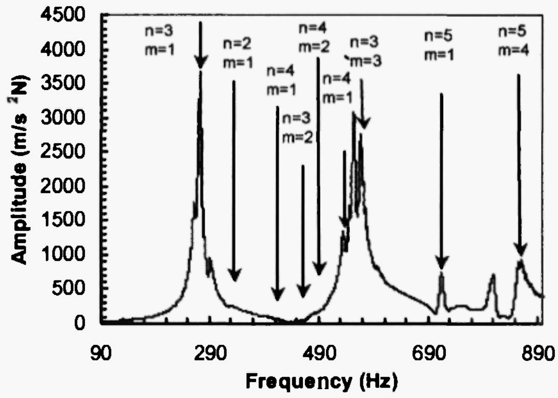

Fig. 2: Measured FRF of the driven point with identification of natural modes for the composite plain-weave fabrics panel.

The experimental modal analysis has been performed by using the software CADA-X 3.5b of LMS and burst-random excitation with the following parameters: burst length 88%, frequency resolution 0.29 Hz, 17 averages, Hamming windows, for the plain-weave fabrics panel; burst length 81%, frequency resolution 0.44 Hz, 20 averages, Hamming windows, for the three-layer panel. The level of excitation was kept low in order to give small amplitude vibrations (approximating a linear system). The FRFs have been estimated using the H_V technique. The modal parameters have been estimated by using the Frequency Domain, Direct Parameter Identification technique. The analysis of the experimental data has been validated by using the Modal Assurance Criterion and the Modal Phase Collinearity.

4.1 Composite plain-weave fabrics panel

The measured FRF in correspondence of the excitation (driven point) is shown in Figure 2 for the composite fabrics panel with identification of natural modes. The measured natural frequencies are presented in Table 1 and they are compared to the results of the numerical calculations (Table 2) that have been obtained by the FEM code ADINA 7.5 (using shell elements with 8 nodes). The indexes *n* and *m* represent the number of circumferential and axial half-waves for each mode shape, respectively.

Table 1

Natural frequencies and damping ratios of the plain-weave fabrics panel from experimental measurements.

n	m	Frequency (Hz)	Damping (%)
3	1	271.87	0.94%
2	1	353.48	0.79%
4	1	448.51	1.11%
3	2	472.39	0.80%
4	2	512.87	0.50%
4	1	536.06	0.94%
3	3	568.80	0.65%
5	1	717.53	0.54%
5	4	850.56	0.86%

Table 2

Natural frequencies of the plain-weave fabrics panel calculated by using Adina and error with respect to experimental results.

n	m	Frequency (Hz)	Error (%)
3	1	282.9	3.9
2	1	342.6	- 3.1
3	2	430.7	- 8.8
4	2	508	0.96
3	3	521.4	- 9

Theoretical and experimental results are in good agreement for both natural frequencies and mode shapes. This assures that the experimental boundary conditions approximate the constraints used in the FEM model, which are those summarized in Table 3.

Table 3

Boundary conditions for the two panels.

	Axial displacement	Radial displacement	Circumferential displacement	Rotations
Straight Edges	*Constrained*	*Constrained*	*Free*	*Free*
Curved Edges	*Constrained*	*Constrained*	*Constrained*	*Free*

4.2 Three-layer (90°-0°-90°) composite panel

The measured FRF in correspondence of the excitation (driven point) is shown in Figure 3 for the three-layer (90°-0°-90°) composite panel with identification of natural modes. Higher natural frequencies than in the fabrics panel are observed.

The measured natural frequencies are presented and compared in Tables 4 and 5 to numerical calculations with the FEM code ADINA 7.5.

Table 4

Natural frequencies and damping ratios of the three-layer (90°-0°-90°) composite panel from experimental measurements.

n	m	Frequency (Hz)	Damping (%)
2	1	470.25	2.5
3	1	627.98	0.51
3	2	809.35	0.01
3	3	852.65	0.85
3	4	1158.15	1.46

Table 5

Natural frequencies of the three-layer (90°-0°-90°) composite panel calculated by using Adina and error with respect to experimental results.

n	m	Frequency (Hz)	Error (%)
2	1	482.1	2.46
3	1	685	8.32
3	2	791.6	-2.15
3	3	885.9	3.71
3	4	1108	-4.51

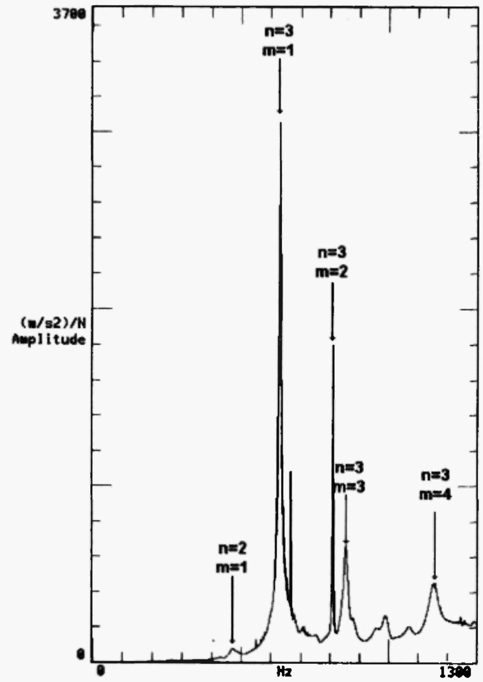

Fig. 3: Measured FRF of the driven point with identification of natural modes for the three-layer composite panel.

5. NONLINEAR RESULTS

5.1 Composite plain-weave fabrics panel

Figure 4 gives the measured displacements in the spectral neighborhood of the fundamental frequency ($n = 3$, $m = 1$; natural frequency 271.9 Hz) *versus* the excitation frequency for four different force levels.

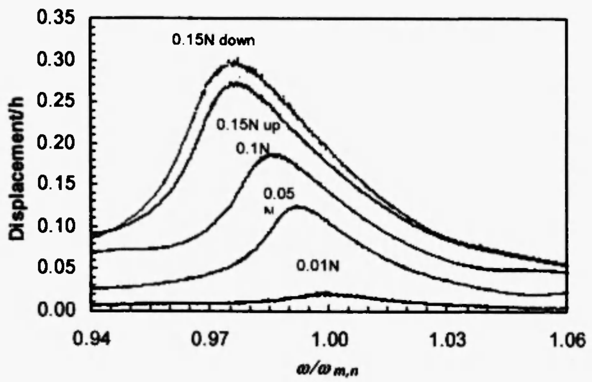

Fig. 4: Experimentally measured displacement (nondimensional) *versus* excitation frequency (nondimensional) for different harmonic force levels; first mode (n = 3, m = 1).

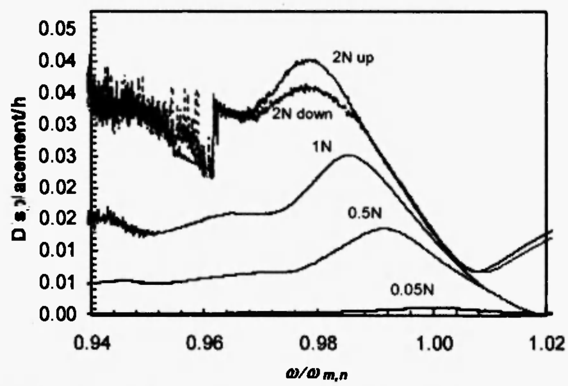

Fig. 5: Experimentally measured displacement (nondimensional) *versus* excitation frequency (nondimensional) for different harmonic force levels; second mode (n = 2, m = 1).

The panel response measured for excitation level of 0.15 N (harmonic excitation) increasing (up) and decreasing (down) the excitation frequency presents significant differences due to the high distortion of the excitation, whose first harmonic is kept constant by the closed loop control. However, the two curves have the same shape.

Figures 5-7 show the measured displacements around the second, third and eighth natural frequencies *versus* the excitation frequency for different force levels. Remarkable softening type nonlinearity has been identified for all these modes.

The first harmonic of the measured acceleration (in periodic regime) has been converted to displacements, dividing by the excitation circular frequency squared. The graphs have been made non-dimensional dividing the displacements by the panel thickness h and the excitation frequencies dividing by the natural circular frequency.

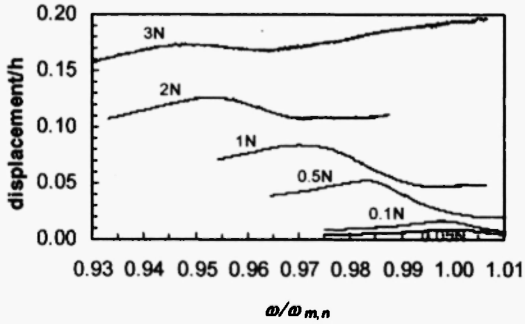

Fig. 6: Experimentally measured displacement (nondimensional) *versus* excitation frequency (nondimensional) for different harmonic force levels; third mode (n = 4, m = 1).

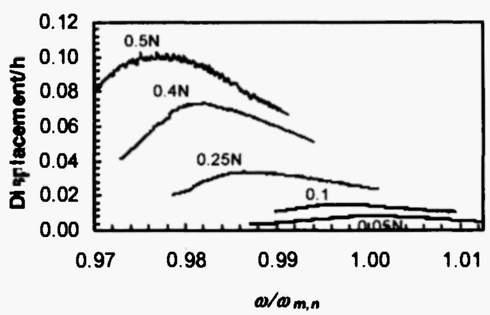

Fig. 7: Experimentally measured displacement (nondimensional) *versus* excitation frequency (nondimensional) for different harmonic force levels; eighth mode (n = 5, m = 1).

5.2 Three-layer (90°-0°-90°) composite panel

Figure 8 gives the measured displacements in the spectral neighborhood of the second natural frequency at 627.98 Hz (n = 3, m = 1) *versus* the excitation frequency for different force levels. As a consequence that this panel is thicker than the previous one and that the natural frequency is higher, smaller nondimensional amplitudes have been reached in this case. Remarkable softening type nonlinearity has been identified.

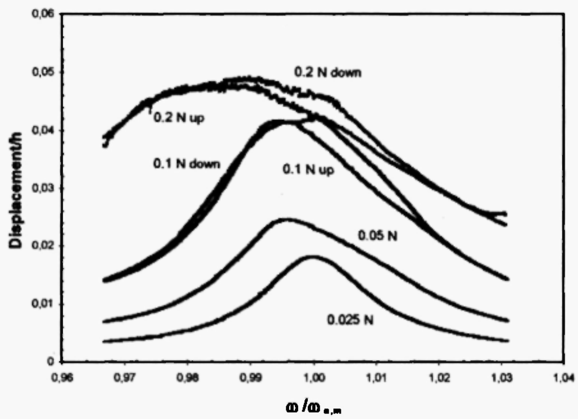

Fig. 8: Experimentally measured displacement (nondimensional) *versus* excitation frequency (nondimensional) for different harmonic force levels; second mode n = 3, m = 1.

6. CONCLUSIONS

Experiments show that the two tested composite curved panels present a relatively strong geometric nonlinearity of softening type. A theoretical model for the boundary conditions used in the experiments is under development in order to provide comparison of theoretical and experimental results.

ACKNOWLEDGEMENTS:

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7. REFERENCES

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