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Dynamic buckling of thin cylindrical shells under shear loading

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ABSTRACT

This paper deals with the buckling of cylindrical shells under a dynamic shear load. The aim of our study is to compare static buckling load and buckling load during a sweep frequency excitation. First we describe the special experimental device and the two finite element codes used in this study. In a second part static tests and corresponding finite element calculations are presented in order to have a reference buckling load and to understand the effect of initial imperfections. In the last part, we discuss about dynamic results. Finite element and experimental results show a good agreement for load-displacement curve, buckling load and buckling mode. When we reach the first eigen frequency, the buckling load drops and the buckling deformations increase.

I. Introduction

LMFR thin shell components have to withstand buckling in shear mode during an earthquake. A relevant example is the connecting course of the Main Vessel (MV). The dynamic buckling of thin shells was investigated in the eighties at the Commissariat à l'Energie Atomique (CEA) : experimental and numerical tests were performed on cylindrical and spherical shells under pressure [1]. A diagram was established giving the ratio between the dynamic and the static critical loads as a function of the frequency excitation.

This paper presents the methodology used to study the dynamic shear buckling of a cylinder in order to deduce a similar rule for this situation.

Some investigations on dynamic shear buckling have been presented in the beginning of the nineties [2], [3] but some questions on effect of the frequency excitation are still remaining.

In the present study, a special experimental device is used to apply dynamic shear loading and buckling results are compared to those given by finite element simulations.

II. Description of the experimental device and finite element codes

II. 1. Test machine

For our study, the shell geometry is representative to SUPERPHENIX Main Vessel. The radius-thickness ratio is equal to 450 and the length-radius ratio is 1. We use an electrodeposited nickel shell in order to have a very little initial imperfection (thickness is equal to 270 μm). At each extremity, the shell is connected to rigid insert in order to have a boundary condition close to an embedment. The material properties are measured during the experiment. We take the mean values (ν = .3, E =180000 Mpa) for finite element simulations. The stress-strain curve is given in table 1. The density of the material is 8700 kg/m^3.

σ_e (Mpa)	ε
180	0.001
300	0.0018
400	0.0027
500	0.0038
600	0.0052
700	0.0075

Tableau 1 Stress-strain curve

In our tests, the bottom of the shell is clamped to a rigid base plate and the top is connected (with a rigid plate) to an horizontal hydraulic ram which applies a displacement controlled load (figure 1). An automatic system enables us to measure the initial imperfection and the buckling deformations during static tests. A constant tensile load, simulating the effect of the self-weight of the MV and the internals, can be applied.

II. 2. Finite element codes

We use two different finite element codes for test simulations :

- the first one, INCA developed at the CEA [4], uses a quasi axisymmetric shell element (COMU). This element is a two node shell element. At each node the geometry is defined by the r, z coordinates and an initial non axisymmetric displacement. The initial imperfection is given on a set of FOURIER circumferencial harmonics defined by the user. The displacement response is decomposed in FOURIER series and the plasticity is evaluated on discrete points around the circumference. In order to take into account the added mass of the thick top plate in dynamic tests, the plate is meshed. In dynamic analysis, we realize an incremental calculation, taking also into account a small initial imperfection (< 10% of the thickness) in order to simulate the post-buckling behavior.

- in the second one, ABAQUS, we use a 3D modelisation. We choose the isoparametric S8R5 shell element (8 nodes). In order to simulate the post-buckling behavior, during static nonlinear calculations, we use the modified RIKS method. For dynamic simulations, we don't use modal analysis but we realize a direct integration with an implicit operator.

III. Static results

In static tests, the shell has a bilinear behavior with a stable post-buckling (figure 2). The buckling load is 610 daN with a corresponding critical displacement of 150 μm. We have a multimodal (10 to 21) buckling mode with 3 waves on each side of the shell (figure 3). After a static test the buckling deformations are still remaining with an amplitude of 20 μm and if we realize a cyclic loading test (up to buckling load) the buckling deformations don't increase. FE simulations and test results are in good agreement for buckling mode, buckling load and post-buckling behavior. FE simulations show the weak effect of an initial imperfection (even for an initial imperfection affine to the elastic buckling mode) on buckling load.

The constant tensile load increases the stiffness of the shell.

IV. Dynamic results

IV. 1. Dynamic tests

During dynamic tests we realize a sweep frequency (1 to 100 Hz) with a fixed displacement amplitude u_0 :

$$u = u_0 \sin(\omega t) \qquad (1)$$

In the first tests, we impose a small displacement amplitude (10% of critical static one). Then, we increase the displacement amplitude up to the static critical one. The elastic behavior of the shell enables us to perform several tests on the same shell.

When we reach the experimental eigen frequency (75 Hz) the load at the top of the shell decreases and the radial deformation increases whatever the displacement controlled amplitude is (figure 4). For frequencies higher than 75 Hz, the load level remains weak and radial displacement is still very important.

The load drop is quite the same for the different excitation amplitudes and can be explained by the proximity of the first eigen frequency.

High speed camera is necessary to see the buckling deformations at high frequencies and some films will be realized during the next tests.

IV. 2. FE eigen mode analysis

The first step in the dynamic analysis of the structure is the eigen frequency calculation. We calculate them with and without the added mass at the top of the shell (this added mass acts only for the first eigen frequency). Without the added mass, the first eigen frequency is 243 Hz. With the added mass the first FE eigen frequency is equal to 80 Hz and the corresponding eigen mode is a shear mode 1 (figure 5). Upper eigen modes are bending mode, torsional mode, and tensile mode. The first pure circumferencial mode is at 940 Hz (figure 6).

The effect of an initial imperfection was also studied [5]. In the case of a small imperfection amplitude (less than 50% of the shell thickness), circumferencial eigen modes and eigen frequencies are the same.

IV. 3. FE dynamic simulations

For dynamic simulations, we realize load control and displacement control calculations both with INCA and ABAQUS.

In displacement control incremental simulations, we always have the same results for frequency up to 100 Hz : buckling appears when critical displacement is reached. In our FE simulations, we can only calculate the reaction load at the bottom of the shell. So the buckling load is similar to static one. For frequencies higher than 100 Hz, this load increases linearly with the frequency.

In load control simulation, we are able to calculate the minimum load inducing buckling. It is necessary to calculate the response for several cycles in order to know the cyclic behavior of the shell. Three different behaviors are observed :

- the top displacement of the shell remains constant and never reaches the critical one.
- the top displacement of the shell increases at each cycle and after 2 or 3 cycles the critical displacement is reached and buckling occurs (figure 8).
- the top displacement increases very slowly and it is necessary to stand more than 10 cycles in order to reach the buckling of the shell.

The reaction load/displacement curve always has the same slope as the static one (figure 7).

The buckling mode (figure 9) does not change up to 40 Hz but for higher frequencies the added mass produces a compression load near the azimuth 0° and deformations appear in this zone (figure 10).

We can draw the minimum load necessary to obtain buckling as a function of excitation frequency (figure 11). It is close to the experimental one.

V. Conclusions

The behavior of cylindrical shells under a sinusoidal shear load was investigated. The effect of initial imperfection and added mass on eigen frequency analysis is explained. When we reach the first eigen frequency, a drop of the buckling load occurs. FE simulations are in good agreements with experimental results and we can use such calculations in order to simulate the behavior of the shell for higher frequencies. Propositions for a future design rule about dynamic buckling should be drawn from these results.

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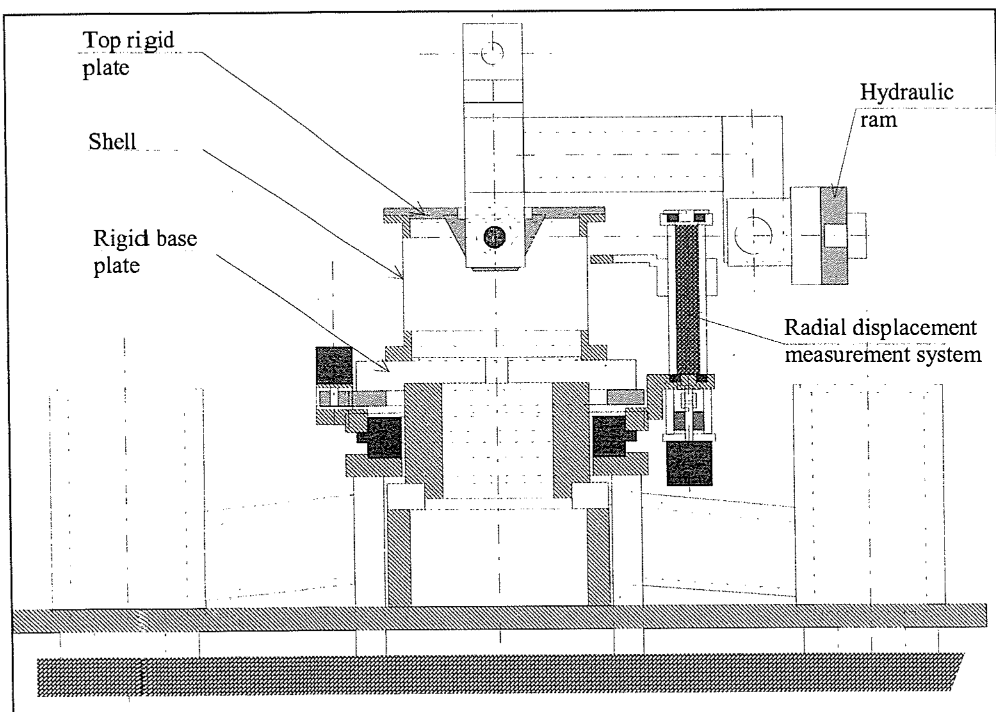
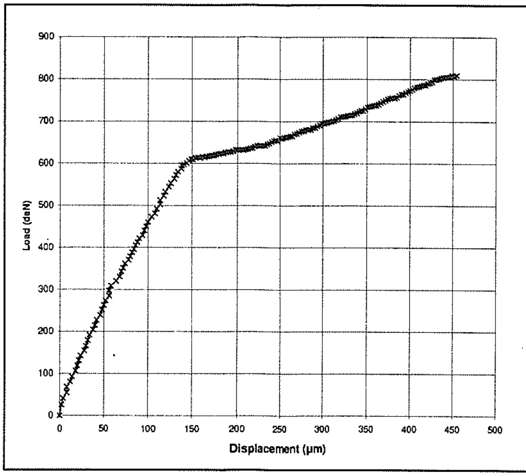


Figure 1 Test machine



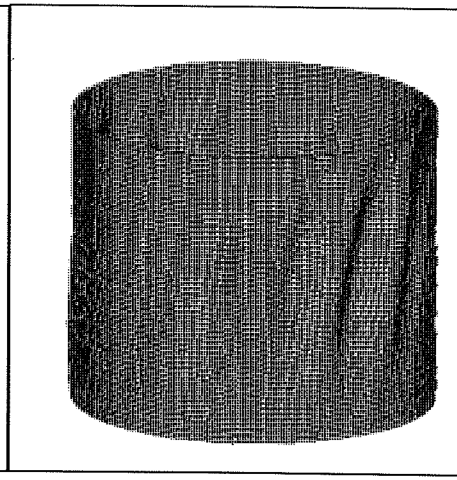


Figure 2 Static load/displacement curve **Figure 3 Buckling mode**

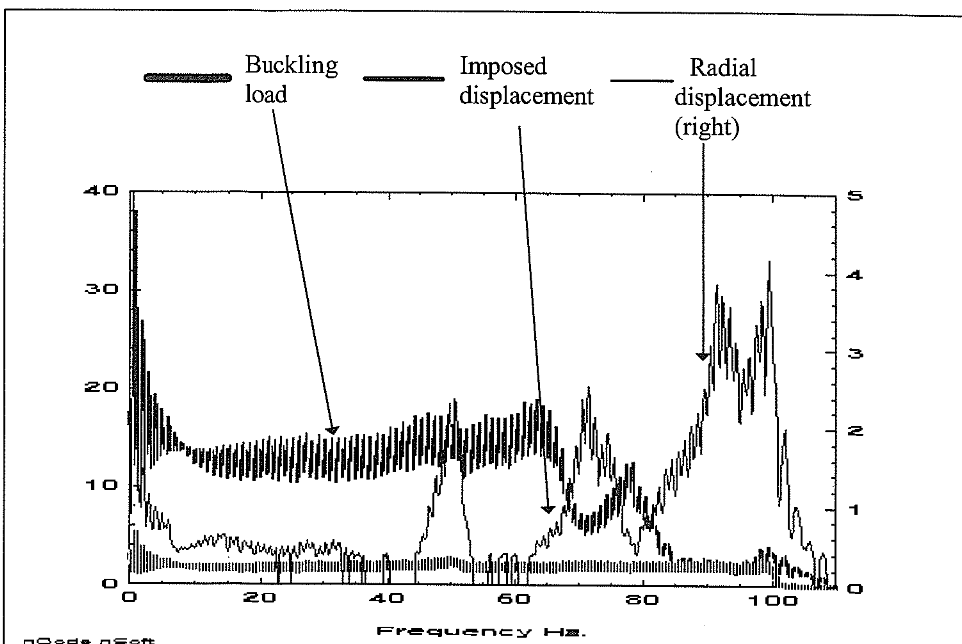


Figure 4 Buckling load, displacement at the top of the shell and radial displacement versus excitation frequency

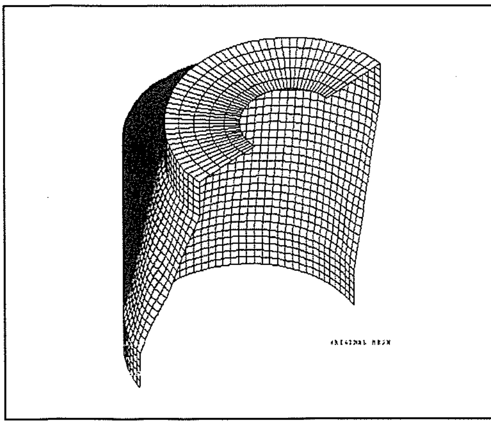


Figure 5 First eigen mode (80 Hz)

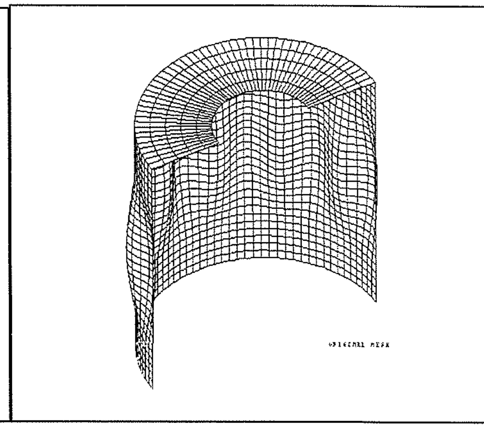


Figure 6 Second eigen mode (940 Hz)

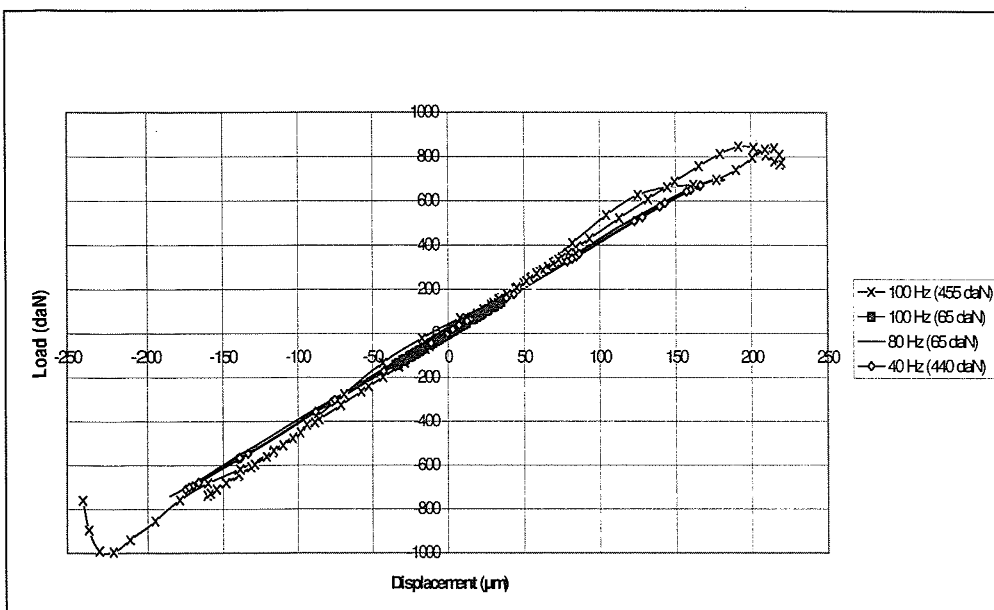


Figure 7 INCA FE dynamic simulations under controlled load

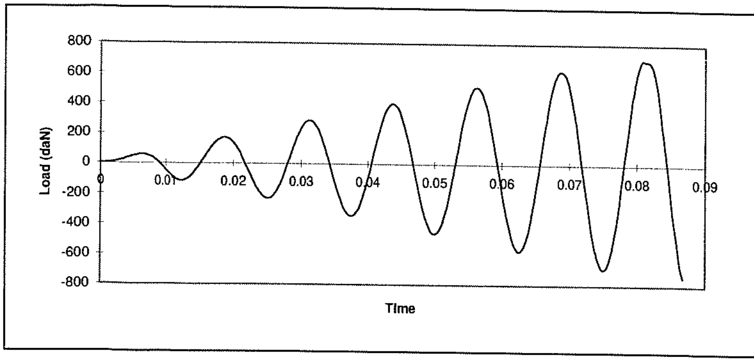
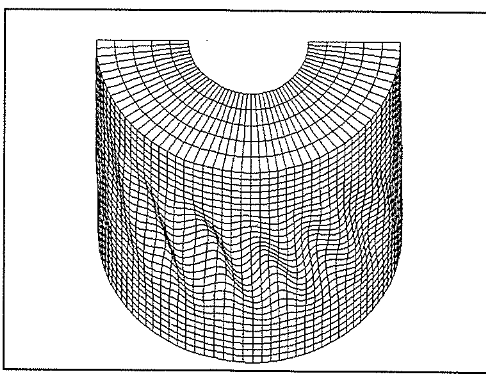


Figure 8 Load/time curve for a controlled load calculation (80 Hz, 65 daN)



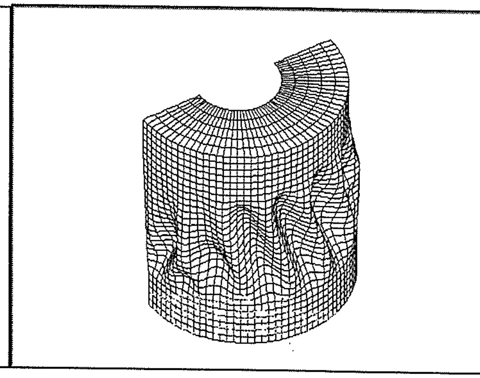


Figure 9 FE dynamic simulation (40 Hz)

Figure 10 FE dynamic simulation (80 Hz)

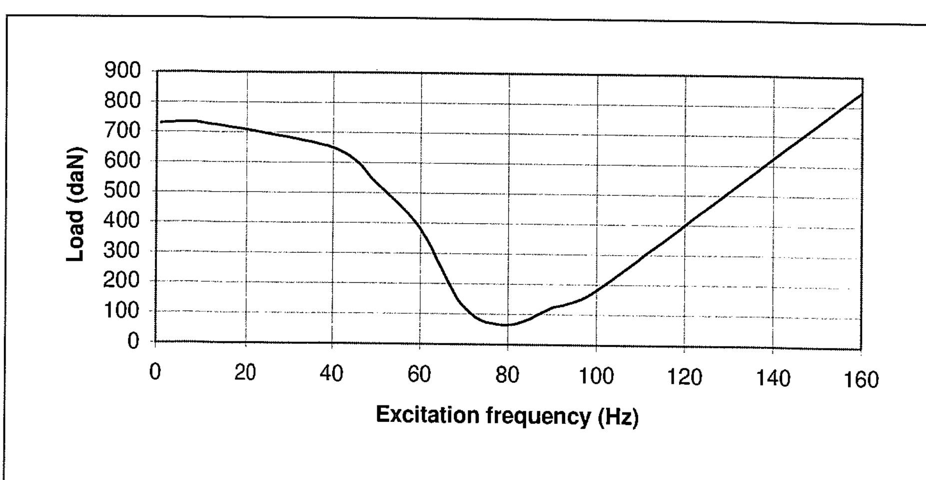


Figure 11 FE minimum load at buckling