A Theoretical Experimental Comparison of the Buckling Caused by Fluid Structure Interaction During a Seismic Load

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The buckling of shells subjected to seismic type of loads is not very well known. To study this type of phenomenon we have performed theoretical and experimental investigations on structures consisting of two shells separated by a thin fluid layer, and submitted to a seismic type of load.

The objectives of these investigations are the following:
- Study the coupling between buckling modes and vibrations modes and buckling of the effects of this coupling on the level of the pressure.
- Study of the appearance on such structures of dynamic instabilities processes.
- Qualification of computer codes of the CEASEMT system.
- Qualification or criticism of the methodology used in the design based on a "static equivalent" idea.

The experiments are made on two types of structures: spherical and cylindrical shells. For the spherical shells ($e/R = 2 \times 10^{-3}$, $e$ thickness, $R$ radius), the material has been chosen such that the buckling occurs in the plastic regime. The Euler elastic bifurcation load is three times the plastic bifurcation load. For the cylindrical shells ($e/R = 2 \times 10^{-3}$), the elastic bifurcation load is three times less than the plastic bifurcation load. These structures buckle in the elastic range.

The load applied on the shells consists of a permanent pressure and of a dynamic pressure due to fluid structure interaction.

The system is put on the vibrating table and excitation is vertical for the hemispherical case, and horizontal for the cylindrical cases. Six models of each type are tested, with sinusoidal excitation at resonance.

The tests on the spherical shells are presented and compared with calculations. The correlation is good and the main results is, as predicted by numerical calculation, that if the sum of the permanent and oscillatory pressure is greater than the static buckling load, the shells buckle. This result validates the static methodology. The tests on the cylindrical tanks will be exploited by the end of the year and presented in this paper.
1. INTRODUCTION

A precedent theoretical study \( \sqrt{17}/\sqrt{2} \) has been done showing qualitatively and quantitatively what is the behavior of a thin sphere under external dynamic pressure. The main result being that this sphere buckles under dynamic pressure at the same instantaneous pressure as if it was loaded with a static pressure. This study relies on dynamic non linear fluid structure interaction computations. It is clear that this type of computations cannot be performed in 3D for the design of a reactor. It is necessary to validate a more simple way to proceed in such cases.

This paper intends to compare the theoretical computed buckling loads under seismic loadings and experimental values. It is attempted to validate on this case the method of the statics equivalent (i.e. to compute the Dynamic Buckling as if the dynamic load were applied in static.)

The general aim of the experimental investigation is to study if there is a difference in the Buckling under static and dynamic Buckling of two types of structures. The first type of structures consists of rather stiff structures which buckle in the plastic regime.

The second type of structure are rather soft structures that buckle in the elastic range.

2. SEMI-SPERICAL SHELLS

2.1 - Structure, geometry, material and loadings

The first type of structure is a thin semispherical shell which is put inside a thicker spherical shell; the 2 spheres are separated by a thin layer of water. The inner shell has a nominal thickness of 0.001 m for a diameter of 1.03 m. The thickness of the water layer is 0.01 m. The outer shell has a thickness of 0.006 m (see figure 1). All the spheres are made of stainless steel. It is loaded by two types of pressure a permanent pressure \( P_o \) applied in the air gap and a dynamic pressure associated with the vibrations.

2.2 - Static buckling

The static buckling has been obtained by increasing the level of the permanent pressure \( P_o \). The critical value of the pressure if of 0.35 MPa for one of the spheres and 0.22 for another one. The calculations and experiments show that this buckling is a buckling on the plastic range. It shows also clearly that taking into account the measure shape imperfections into the analysis gives a good prediction of the observed critical static pressure (see table 1) for more details see \( \sqrt{3} \).

2.3 - Vibration analysis

The vibration experiments method at permanent pressure show a first eigen frequency of about 36 Hz with a damping of about 1 %. The influence of \( P_o \) on the first eigen frequency has been studied experimentally. Table 2 shows this evolution. This experimental result is in contradiction with the computed results that show that the presstress induced by the permanent pressure \( P_o \) should have no effect on the first eigen frequency (see figure 2). The reason of this contradiction is not well understood yet except that the boundary conditions may depend on the permanent pressure \( P_o \). On figure 2 are pointed the variation of the eigen frequencies with the presstress \( P_o \). And we observe that the eigen frequency comes to zero very suddenly when \( P_o \) approaches the value of the Euler load (\( P_e = 0.7 \) MPa). The figures 3.1, 3.2 and 3.3 give an idea of what happens. Figure 3.1 is the axisymmetric vibration mode for \( P_o = 0 \). Figure 3.2 is the axisymmetric vibration mode for \( P_o = 0.7 \) MPa. Figure 3.3 is the buckling mode.
When the vibration and buckling modes are very different (orthogonal) there is no effect of the prestress on the eigen frequency when they are close, there is an important influence as we shall see on the case of the cylinders.

2.4 - Seismic experiments

The same system is now excited with a periodic sinusoidal acceleration at the resonant frequency of the system. For a pressure $P_0$ of 0.13 MPa the maximum pressure observed in the fluid has been 0.3 MPa and no buckling occurred. When the permanent pressure $P_0$ has been increased to 0.2 MPa the buckling of the inner sphere has occurred very suddenly (0.003 second for the decrease of pressures) at a pressure of 0.33 MPa. The buckling mode is identical to the static buckling mode. For more details see \( \text{[4,7]} \).

3. CYLINDRICAL TESTS

3.1 - Geometry, loading, material

The soft structures are cylinders, the dimensions of which are given on figure 4. The material is again a stainless steel and there is a water layer that separates the two cylinders.

This system is again loaded by two types of pressure. A permanent pressure $P_0$ and a dynamic pressure $P$ resulting from the pressures due to the vibrations induced by shaking the cylinder in mode 1.

The shape of the cylinders are measured with a rotating system at ten different levels. The shape is then analysed through a Fourier transform and the Fourier decomposition is given. You can observe on figure 5 a typical curve. This shape measurement is done after each tests so that we can see with a great accuracy if the test has changed the shape of the structure. With this methodology it was observed that the shape near the clamping of the inner shell changed of about 20% after the research of eigen frequency which indicated some stress redistribution in this part of the shell even at a very low level of loading. The general tendency is to have a high contribution of the low order modes 2, 3, 4 and a weak of the others. The amplitude of the imperfection is of the order of magnitude of the thickness. For more details see \( \text{[5,7]} \).

3.2 - Vibration characterization

During the research of eigen frequency we have found a great number of resonances between 10 and 45 Hz. (see Figure 6). The characterization of the eigen modes has then been made and we obtain the following frequencies given in table 3.

The most interesting fact is that even at a very low level of excitation this structure excited in mode 1 has a response that is not in mode 1. The hypothesis taken for the calculations of such shells under seismic load are not valid. The structure has a response which is not colinear to the excitation modes. This proves that the classical seismic methodology is not valid for this type of structure. The physical reason of this experimental evidence is the coupling of harmonics via the defects of the structure and the fluid coupling. This has already been remarqued by Clough and also by Veletsos and Turner \( \text{[6,7]} \).

We have also found a very large influence of the permanent pressure ($P_0$) on the eigen frequencies. See Figure 7. You remark on these figures that as predicted by the theory the square of the measured frequency depends linearly of the permanent pressure $P_0$. 

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V 8/1
3.3 - Static buckling
The static buckling of these shells is obtained for a value of the pressure of about 0.03 MPa. The circumferential mode is 11. The amplitude of the bumps created by the buckling is of 0.04 m. The buckling starts in the elastic range.

3.4. - Dynamic buckling
The shaking table has been excited in mode 1 for different increasing permanent pressure $P_0$ in two different ways:
- 1st at the eigen frequency of the vibration circumferential mode 12.
- 2nd in a range of plus and minus 10% of this frequency.
The buckling has been observed for a permanent pressure of 0.02 MPa. The total pressure being 0.04 MPa and for the 2nd type of excitation. For the 1st type no buckling happened for the same maximum value.
The buckling mode was identical to the one in static.
For lower value of the pressure $P_0$, no buckling was observed but it is not obvious that the total pressure were higher than in the case that led to buckling.

4. CONCLUSIONS
This series of experimental and theoretical investigations show three main points.
For the type of thin structures studied here, that may buckle, who are subjected to the sum of a significant permanent pressure and a dynamic pressure, the buckling is obtained when the total pressure is greater than the static pressure. This proves that the methodology of the static equivalent is valid for this case.
The prestress can have a very important effect on the eigen frequencies if the buckling modes are colinear to the vibration modes.

Thin structure excited with a seismic type of load (mode1) can have a response in the circumferential modes and not only mode 1 as predicted by the theory of perfect shells. The defects explain the coupling of modes.

ACKNOWLEDGEMENT
This experimental work was made through the financial support of Electricité de France (SEPTEN), La Défense 92 - PUTEAUX France

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Rapport DMT/SMTS/LAMS (à paraître)
 TABLE 1

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<tr>
<th>SHELL NO</th>
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<td>P_E</td>
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<td>P_B</td>
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<td>δ_b/ε</td>
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P_E  elastic bifurcation pressure
P_B  plastic bifurcation pressure
P_B/ε_max  plastic imperfect shell pressure
P_E/ε_max experimental buckling pressure
δ_b/ε ratio of imperfection over thickness

TABLE 2

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<tr>
<th>P_o</th>
<th>0</th>
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TABLE 3

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PHOTO 1: Semi spherical inner shell after buckling

PHOTO 2: Cylindrical shell after buckling

FIGURE 1: Semi spherical shells geometry

FIGURE 2: Eigen frequency evolution
FIGURE 3-1: Axisymmetric vibration mode for $P_0 = 0$

FIGURE 3-2: Axisymmetric vibration mode for $P_0 = 0.7$ MPa

FIGURE 3-3: Buckling mode

FIGURE 4: Cylindrical shells geometry
FIGURE 5: Typical curve Fourier decomposition shape

FIGURE 6: Cylindrical inner shell frequency response

FIGURE 7: Permanent pressure influence