INELASTIC BUCKLING ANALYSIS, EXPERIMENTAL TESTS ON VESSEL HEADS

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1 - INTRODUCTION

The pool type architecture adopted in France for fast breeder nuclear power plants, both for Phénix /1/ /2/ and for Creys-Malville /3/, allows operation at a pressure approaching atmospheric pressure /4/.

In this type of reactor, the main vessel is very large. Owing to the low operating pressure, however, its thickness remains within moderate proportions. Hence it may be noted that the cylindrical portion of the main vessel of Super-Phénix is about 20 meters in diameter and about 25 millemeters thick. The relatively thin structures implies that considerable attention must be paid to any buckling risks. Since the relevant analyses showed that buckling only occurs after plastic deformation, the need arose to develop methods for determining the critical elastic/plastic buckling loads (in French design practice most containments are at temperatures at which creep can be ignored). These methods are part of the CEASEMT general structure analysis system /5/.

In the absence of a complete, consistent general theory of elastic/plastic buckling, it proved necessary to develop and expand mathematical models designed to obtain the critical loads with technically sufficient accuracy /6/ /7/ /8/ /9/ /10/. This undertaking cannot be based merely on intellectual speculations. It requires solid experimental foundations. In determining adequate mathematical models, developing and improving them, as well as for their indispensable validation, it is necessary to refer to a suitable body of experimental results. Use was made of the available results in the literature /8/, and an experimental program was launched at the Saclay Nuclear Research Center. The most significant aspect of this program was a series of tests on dished elliptical heads subjected to internal pressure. This discussion describes these tests and the results obtained. The tests covered a series of eighteen dished elliptical heads mounted on cylindrical shells approximately 500 millimeters in diameter. These heads were fabricated by flaturning, and a summary of the definition of each of these heads is given in Table 1. A complete description of these tests is available in a CEA Note /11/.

2. PREVIOUS EXPERIMENTAL WORK

A good review of the state of the matter has been provided by Esztergar /12/. However, it appears that the risks of buckling of dished heads under internal pressure were only taken into account relatively recently, as shown by the unforeseen experience of the collapse of a pressure vessel head at Avon, California, in 1956 /13/ /14/, which occurred during the hydraulic test.

Although most work on head buckling is intended for the postulation of theories, they will not be mentioned here. The first tests were reported by Mescal /15/ followed by Adachi and Benicek /16/. Transitions between elastic and plastic behavior do not appear to have been pointed out. Another series of experimental investigations was carried out at Manchester University in Great Britain, by Kirk and Gill /17/ and by Patel and Gill /18/. With respect to cases of buckling in which plasticity appears to have played a major role, the authors stress that the critical loads are considerably lower than those calculated by theoriticians, and that they are in fact relatively comparable to pressures corresponding to limit loads (in the sense of limit analysis). A third experimental study was carried out at Liverpool University by Galletly /14/ /19/ /20/. In his conclusions, the author stresses that with respect to steel heads, a large discrepancy exists between the critical pressure determined experimentally and the calculated pressure. In the foregoing investigations, the authors insist on the problems encountered in obtaining a constant, clearly defined thickness. The work of Save /21/ although not specifically an investigation of buckling, must be mentioned as dealing with the experimental determination of limit pressures.

3. OPERATIONAL DEFINITION OF THE CRITICAL BUCKLING LOAD

The purpose of this investigation is to determine the critical pressure at which buckling appears. Hence it is necessary to have an operational definition of this pressure, so that its value can be derived from test results without any ambiguity. Unfortunately, the buckling of industrial structures has never been clearly defined /19/.

The question is posed fairly simply for elastic structures with a geometry corresponding to theoretical buckling conditions, i.e. which possess several simultaneous states of equilibrium. In this case, buckling corresponds to the "stability exchange" postulated by Poincaré in the stability theory /22/. Note that this author showed that the loss of stability is normally associated either with a limit point, with a point of bifurcation at which the fundamental equilibrium path intersects another equilibrium curve. Buckling corresponds to the stability exchange at the point of bifurcation. Buckling does not always correspond to a loss of stability. The new state may also be stable, and several bucklings may occur successively, each time accompanied by a change in form.

Structural forms in which a true stability exchange can occur are relatively rare. Nevertheless there is a strong tendency to select them, owing to their tendency to be simple, symmetrical and effective. This applies in particular to thin shells and hence to dished heads, where the best use of the material imposes the selection of forms, in which the load only generates membrane stresses. These particular shapes generally exhibit a branch point, so that the problem is frequently faced by engineers. It arises in an imperfect manner, and owing to fabrication tolerances, while the real form approaches one which is susceptible to an equilibrium exchange, the latter does not necessarily occur. While in certain cases a true instability exists with a "limit point", occurring at a load lower than that under which a perfect structure would buckle, it is difficult in other cases to observe "buckling" in the practical sense of the word. If one also considers the geometric changes which occur progressively, it is easy to understand the problems faced in using experimental results to determine the critical buckling load.

An attempt is made here to relate the definition of buckling to the appearance of a new geometric form. In most tests, this new geometry appears to change so rapidly with pressure that plastification occurs, causing pinching of the metal in the form of a fold, and this is easy to observe. Furthermore, the formation of this fold causes an increase in the volume of the head and, since the pump delivery is fairly low, a sharp drop in pressure occurs since this fold continues to form despite this pressure drop. The value of the critical buckling pressure is thus clearly indicated by recordings. In thicker heads, on the other hand, the formation of the wave is far more progressive and folds do not appear suddenly. Hence the critical buckling pressure adopted in these cases is the value at which the waves become sufficiently visible to the naked eye.

In conclusion, the critical buckling load adopted is that at which geometric modifications appear which are technically appreciable and durable after depresurization, in other words, observable damage.

4. EXPERIMENTATION See reference /11/ in particular.

4.1 Geometry of vessel heads.

The form of the meridian of each head was checked by means of elliptical templates, and the heads only accepted if the difference between the real geometry and the nominal ellipse did not exceed 1 mm.

Owing to the fabrication process for the heads, it was not possible to obtain the nominal thickness accurately ; the thickness obtained varied along the meridian but not along the parallel. The heads were thus perfectly axisymmetrical. The thickness was measured along the meridian after cutting each vessel head /11/.

4.2 Material characteristics.

The proper use of the results of these tests can only be made if the mechanical properties of the materials are well known /11/. Consequently, the material characteristics adopted were the results of tensile tests performed on samples cut out from the knuckle area of the heads after the tests (see Table 1).

4.3 Results.

The most interesting results are given in Table 2.

5. COMMENTS AND CONCLUSIONS.

5.1 - Buckling always occurred in these tests at pressures substantially higher than the appearance of non-linearities. When buckling occurred, the deformations were already significant enough for the initial geometric form to be substantially altered.

The behavior of the cylindrical shell on which the head is fixed is extremely important. If swelling due to pressure occurs very early, this results in an overall deformation which prevents any subsequent buckling (see head 11). Consequently, a vessel head buckles if it is fixed on a thick shell, but will not buckle if it is fixed on a thin shell which swells at a pressure lower than the buckling pressure.

In no case did buckling give rise to instability, and it was always possible to subject the head to pressures far higher than the buckling pressure, although the head was considerably deformed.

Buckling is a clear, sudden occurrence in thin heads (thickness/diameter ratio ranging from 0.001 to 0.002), in which successive folds are formed abruptly (see Figures 1 and 2).But this occurrence is generally diffuse and progressive in thicker heads (thickness/diameter ratio about 0.004), in which a general waviness occurs progressively.

5.2 - This paper would be incomplete without an empirical formula combining the results obtained here. This formula is certainly likely to render service to designers responsible for dimensioning their vessel heads, but it will also be useful for future experiments.

The proposed formula is the following :

$$P_{cr} = 700 \left(\frac{\sigma_y + \sigma_u}{2}\right) \left(\frac{t}{D}\right)^{5/3} \left[\left(\frac{D}{H} \right)^2 - 8 \right]^{-2/3}$$

where the following notations are employed :

- P_c : buckling pressure
- σ_y : yield strength
- σ_u : ultimate stress

} same units

- t : thickness in the knuckle area
- H : height of vessel head
- D : diameter of vessel head

} same units

Table 3 gives the results obtained and compared with experimental results.

5.3 - Tests were performed on 17 dished elliptical heads of three different materials and different geometries. The results include :

- a suitable definition of the geometry and, in particular, a reading of a direct thickness measurement along the meridian,
- a precise definition of the characteristics of the materials, measured on test pieces taken from the heads themselves,
- a plot of deformation/pressure curves with an indication of the pressure at which buckling occurs (see example in Figure 1).

Apart from their practical use in designing vessel heads, the results are employed to validate and qualify the plastic buckling modules of the CEASEMT finite element computation system /9/ /10/ /23/.

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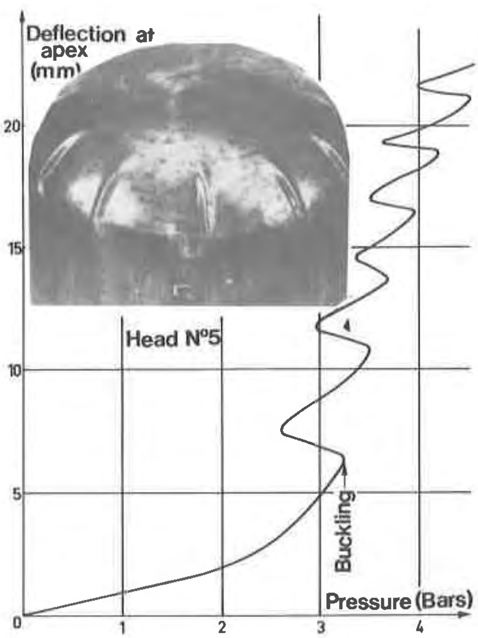

Figure 1

Figure 2

TABLE I

Mechanical properties of vessel heads and materials

(E, σ_y and σ_u in MPa)

mock-up item N°	nominal thickness t (mm)	height h (mm)	material	E	σ_y 0.2 %	σ_u Rm	Ar %
1*	0,5	100	Carbon steel XE	-	-	-	-
2	1	100		18 7000	310	340	15
3	2	100		17 5000	300	330	10
4	0,5	50		16 5000	280	340	18
5	1	50		18 5000	260	330	22
6	2	50		18 5000	240	300	21
11*	0,5	100	Austenitic steel	-	-	-	-
12	1	100		17 2000	480	690	40
13	2	100		18 0000	520	700	30
14	0,5	50		17 5000	380	680	56
15	1	50		16 6000	470	770	49
16	2	50		19 7000	570	750	40
21	0,5	100	Aluminium alloy	6 9000	130	230	15
22	1	100		6 9000	180	230	11
23	2	100		6 9000	170	240	10
24**	0,5	50		-	-	-	-
25	1	50		6 7000	170	230	14
26	2	50		7 4000	150	200	21

* Heads N° 1 and 11 were not cut after test.

** Head N° 24 was not subjected to measurement.

TABLE II

Experimental results

head N°	P_{Li} (MPa)	$P_{buckl.}$ (PMa)	Deflection (mm)	Numb. of folds	P_{max} (MPa)
1	0,40 (7)	0,48 (1)	3,4 (4)	6	0,675 (3)
2	0,47 (8)	0,80 (1)	3,2	12	1,80 (3)
3	1,20 (8)	2,50 (2)	13	10 (5)	3,40 (3)
4	0,17 (7)	0,27 (1)	4,7	14	0,80 (3)
5	0,19 (7)	0,325 (1)	6,4	13	1,30 (3)
6	0,43 (8)	0,80 (2)	6,3	10 (5)	1,60 (3)
11	0,50 (8)	(9)	(9)	0	2,00 (3)
12	0,54 (8)	1,50 (1)	7,6	9 (6)	2,00 (3)
13	1,10 (8)	4,90 (2)	23	? (2)	4,90 (3)
14	0,25 (7)	0,36 (1)	5	12	0,80 (3)
15	0,32 (7)	0,575 (1)	7,9	12 (5)	1,80 (3)
16	0,75 (8)	2,10 (1)	17,8	7 (5)	3,20 (3)
21	? (8)	0,23 (1)	6,8	7	0,36 (10)
22	0,25 (7)	0,42 (1)	3	20 (5)	1,20 (10)
23	0,7 (7)	1,80 (2)	18	? (2)	1,80 (10)
24	-	(11)	-	-	-
25	0,125 (8)	0,165 (1)	5,2	16 (5)	0,40 (10)
26	0,25 (8)	0,56 (2)	32	? (2)	1,70 (3)

KEY :

- P_{Li} : pressure at end of linearity of the deflection/pressure curve.
- $P_{buckl.}$: critical buckling pressure.
- Deflection : polar deflection corresponding to $P_{buckl.}$
- folds : total number of folds formed at P_{max}
- P_{max} : maximum pressure applied to the head.

(1) Sudden drop in pressure on "formation" of the first fold.

(2) Progressive formation of waves. Pressure noted when they were visible.

(3) Test stopped. The head can withstand higher pressures.

(4) For 0.6 MPa, the polar deflection decreases. As the shell is deformed, the overall structure tends to assume the form of a sphere.

(5) For this head, certain folds are barely formed, so that the count is approximate.

(6) Highly deformed head, great difficulty in counting the folds.

(7) Fairly clear.

(8) Barely visible limit.

(9) The assembly assumed the form of a sphere. The head did not buckle.

(10) Fracture along a weld.

(11) Head destroyed during filling.

TABLE III

head	H (mm)	σ_y	(MPa)	σ_u	t (mm)	$P_{buckl.}$ (MPa)	P_{cal} (MPa)	$P_{buckl.}/P_{cal}$
1	100	(280)	(1)	(330)	0,64	0,48	0,49	0,98
2	100	310		340	0,85	0,80	0,83	0,96
3	100	300		330	1,74	2,50	2,66	0,94
4	50	280		340	0,85	0,27	0,26	1,04
5	50	260		330	0,95	0,325	0,30	1,08
6	50	240		300	1,95	0,80	0,90	0,89
11	100							
12	100	480		690	0,82	1,50	1,41	1,06
13	100	520		700	1,70	4,90	4,97	0,99
14	50	380		680	0,78	0,36	0,38	0,95
15	50	470		770	0,95	0,575	0,62	0,93
16	50	570		750	1,81	2,10	1,93	1,09
21	100	130		230	0,55	0,23	0,22	1,05
22	100	180		230	0,89	0,42	0,57	0,74
23	100	170		240	1,78	1,80	1,80	1,00
24	50							
25	50	170		230	0,89	0,165	0,18	0,92
26	50	150		200	1,88	0,56	0,55	1,02

- P_{cal} $$700\,\frac{\sigma_y+\sigma_u}{2}\left(\frac{t}{D}\right)^{5/3}\left[\left(\frac{D}{H}\right)^{2}-8\right]^{-2/3}$$

- $P_{buckl.}$ experimental values of critical buckling pressure.

- (1) means of σ_y and σ_u values for heads 2 to 6.