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Experimental collapse of thin cylindrical shells submitted to internal pressure and pure bending

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Abstract

A thin-walled pressurised cylindrical shell is sensitive to buckling phenomena when it experiences locally a compressive stress. It is often considered that its behaviour under bending is rather similar to pure compression, but very few are the experimental investigations that precise the real behaviour of a thin pressurised cylinder submitted to a bending load. A large amount of experimental results is presented here, obtained on thin shells ($550 < R/t < 1450$) of moderate length ($L/R \approx 2$). The evolution of the cylinders' behaviour that has been recorded when internal pressure increases is outlined. It is shown that one must distinguish between local buckling and global collapse of the structure. A comparison of our experimental data to design recommendations given by two standards (NASA SP8007 and Eurocode 3) is finally achieved, putting in advance safety margins provided by these codes.

Keywords: Buckling; Collapse; Thin shells; Bending; SP8007; Eurocode 3

1. Introduction

Thin-walled cylindrical shells are widely used in numerous applications such as tanks, silos, space launchers. A correct design has to pay attention to buckling phenomena, which may occur under specific loading conditions, and might cause global collapse of the structure.

Real loading conditions often result of a combination of fundamental loads (axial compression, internal or external pressure, torsion, and bending). Whereas the case of a pressurised cylindrical shell submitted to axial compression has been extensively studied, fewer investigations consider the interaction of bending moment with internal pressure, especially from an experimental point of view.

We present here a large amount of experimental results, obtained on thin shells ($550 < R/t < 1450$) of moderate length ($L/R \approx 2$). A comparison of our results to design codes (NASA SP8007 and Eurocode 3) is also provided, and safety margins are evaluated.

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2. Statement of problem

2.1. Buckling of unpressurised shells submitted to pure bending

The first experimental campaigns were conducted on thin shells in the 1930s [2–7]. Due to the moderate length of the tested specimens, the well-known progressive flattening, which characterizes the collapse mechanism of long cylinders under bending as predicted by Brazier [1], is not noticeable: the boundary conditions prevent the flattening, which is in general not visible prior to failure. The behaviour is rather similar to the one observed on a compressed cylinder: the increasing bending moment causes the buckling of the compression half of the specimen, the typical failure pattern consisting of diamond-shaped buckles. Relying on these experimental results, Flügge [8] is the first to treat the problem as a bifurcation one; he obtains a bifurcation stress equal to $1.30\sigma_{CL}$, where σ_{CL} is the buckling stress of a cylindrical shell under axial compression, independently identified by Timoshenko [9], Lorenz [10] and Southwell [11].

Suer et al. [12] performed in the 1950s new tests under both compression and bending loads, and compared their results to available experimental data. A statistical analysis puts forward that the experimental buckling stress is from 20 to 60% higher in bending, depending on the R/t ratio, which confirms previous

Nomenclature

R, L, t	radius, length, wall thickness of cylinder
E, ν	Young's modulus, Poisson's ratio
σ_L	material elastic limit
σ_U	material ultimate stress
Loads	
P	internal pressure, $P>0$
N	axial load
M	bending moment
p^*	dimensionless pressure parameter $p*=PR/[t\sigma_{CL}]=\sigma_\theta/\sigma_{CL}$
Stresses	
σ_{CL}	theoretical compressive buckling stress $\sigma_{CL}=Et/\left[R\sqrt{3(1-\nu^2)}\right]$
σ_P	axial stress due to internal pressure $\sigma_P=PR/[2t]$
σ_θ	circumferential stress due to internal pressure ($\sigma_\theta=PR/t$
σ_N	axial stress due to N $\sigma_N=N/(2\pi Rt)$
σ_M	axial compressive stress due to M $\sigma_M=M/[\pi R^2 t]$
σ_c	maximum axial compressive stress on tested cylinder $\sigma_c=\sigma_N+\sigma_M-\sigma_P$

observations [5,6]. Nevertheless, it must be noted that among the collected experimental data, the classical stress is never reached, neither in compression nor under flexural moment. This discrepancy is for a large part attributed to the presence of geometrical imperfections on tested specimens.

In 1961, Seide and Weingarten [13] show that when a linear prebuckling state is assumed, the predicted buckling axial stress due to pure bending for finite length simply supported cylindrical shells is equal to that for uniform axial compression. These authors also re-examine Flügge's results and conclude that his incorrect estimation was related to a specific R/t ratio and a particular assumed longitudinal half-wavelength of the buckling shape. Consequently the main reason generally suggested to explain the noticeable higher stress experimentally obtained under a bending load, is that this case is less sensitive to geometrical imperfections (behaviour in bending will be affected by defects only in the region of the greatest compressive stress, whilst under compression, any imperfection on the surface can initiate buckling).

The interaction between bifurcation buckling and Brazier's flattening effect has been studied by Stephens and Starnes [14], Fabian [15], Libai and Bert [16] or Tatting et al. [17]. We retain here Stephens and Starnes' results [14], which show that for short cylinders ($L/R<3$) Brazier effect can be excluded.

2.2. Buckling of pressurised shells

Consequences of internal pressure on the buckling of cylindrical shells under uniform axial compression have been extensively studied. Flügge, Hutchinson or Yamaki (among others) have shown [18–20] that for a perfect shell, pressurisation does not affect the theoretical buckling stress, once the axial pretensional stress has been balanced:

$$\sigma_{bif}=\sigma_{CL}+\frac{PR}{2t} \tag{1}$$

Lo et al. [21], Weingarten and Seide [22], and Limam [23] observed that the experimental critical stress increases with internal pressure to reach a maximum equal to the reference stress given by Eq. (1). The cylindrical shell is progressively less sensitive to geometrical defects. Schnell and Thielemann [24,25] examined the modification of the post-critical behaviour with internal pressure; for sufficiently high levels of pressure, this behaviour is found to be stable, provided that the shortening of the shell remains small. Limam [23] confirmed these results through numerical FE simulations: the pressurisation inhibits asymmetrical bifurcation modes but has no effect on the axisymmetrical 0-mode. Buckling occurs for this reason at the same stress with or without pressure; the only change is the shape of the critical mode.

Concerning a pressurised cylindrical shell under flexure, Weingarten [26] uses the same methodology as for unpressurised cylinders [13], and points out that the actual bifurcation stress is slightly increased by the pressure, depending on the R/t ratio (Fig. 1) (for the lack of simplicity we will consider in the following that the stress given by Eq. (1) is a good approximation of the buckling stress for a pressurised cylinder under bending). From an experimental point of view, only a few campaigns dealing with pressurised shells under a bending moment have been achieved. Suer et al. [12] expose a set of 58 tests where the pretension stress ($PR/2t$) due to internal pressure is sometimes balanced by an axial compressive load before the bending moment is applied. It can be seen from these results that when the pretension is balanced, the critical stress increases to the classical value with internal pressure. When the pretension is not balanced, the critical stress also increases, and for some cases exceeds from a considerable amount the stress given by Eq. (1). In the early 1960s, new experiments [27] carried out on Mylar specimens confirmed that the collapse stress of a pressurised cylinder could be higher than the theoretical bifurcation stress. Nevertheless, Weingarten noted [26] that for high values of pressure the critical buckling stress given by Eq. (1) 'closely represents the stress at which the load-deflection curves become non-linear'.

2.3. Collapse moment of a pressurised cylinder

Stein et al. proposed in [28] an evaluation of the maximal moment supported by a pressurised cylinder. Adopting a purely membrane behaviour, they obtain the following closed-form expression:

$$M_{coll}=\pi R^2 t\left(\frac{pR}{t}\right) \tag{2}$$