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Imperfection sensitivity of cylindrically curved steel panels

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ABSTRACT

Unstiffened cylindrically curved panels constitute a common subset of shell structures and therefore share the advantages and disadvantages of such structural solution. One of the bigger disadvantages of shells is their sensitivity to imperfections that lower significantly their ultimate strength.

At the European level, in the scope of plate design by FEM, imperfections are treated in EN 1993-1-5 (Annex C) and in shell structures designed by global numerical analysis using GMNIA, recommendations are given in EN 1993-1-6 (clause 8.7.2). Since curved panels are neither flat nor full revolution cylinders, rules for estimating equivalent geometric imperfections remain unclear. In order to tackle this problem, this paper introduces a numerical parametric study on the imperfection sensitivity of cylindrically curved panels. The effect of the amplitude and shape of initial geometric imperfections (together with different values of curvature and aspect ratio) on the ultimate strength of unstiffened cylindrically curved steel panels is studied, results are presented and conclusions are drawn.

1. Introduction

In the field of plate and shell stability, imperfections (geometric, residual stresses and eccentricities) are known to be the cause of poor correlation between theoretical and experimental results. Generally, the presence of imperfections decreases the ultimate load that a structure can support (Fig. 1).

Classically, design formulae were calibrated exclusively with experimental results (*e.g.* Winter's formula) where the measurement of imperfections was not important since the goal was, after identifying relevant parameters, to establish a large scatter of points representing the ultimate resistance of a certain structural component and later perform a regression to obtain an expression capable of predicting the structural element ultimate load on the safe side. Nowadays, experimental campaigns are complemented with numerical simulations of the structural component being studied, allowing a much larger scatter of points in a much shorter time period. The downside of the numerical approaches is that it becomes crucial to know and to model imperfections accurately. Additionally, high-performance software tools (almost exclusively based on the finite element and on the finite strip methods) are widespread and are used in most design offices around the world. Therefore, definition and guidelines on how to model imperfections is an urgent task.

At the European level, when numerically modelling plated structures, imperfections may be treated according Annex C of EN1993-1-5 [2]. It is stated that both geometric and material related imperfections should be taken into account or, alternatively, in a more straightforward way, only equivalent geometric imperfections may be considered. If the first approach is chosen, the shape of the geometric imperfections may be defined with relevant eigenmode shapes with a recommended amplitude of 80% of the fabrication tolerance limits (this recommendation is based on engineering judgement [3]) which are defined in EN 1090-2 [4], and the material related imperfections (residual stresses) should be represented by a stress field on the element related to the fabrication process (welding and forming). On the other hand, if the second approach is chosen, it is recommended to use an eigenmode shape or shapes defined in Figure C.1 of EN 1993-1-5 with amplitudes defined in Table C.2. In the case of unstiffened isolated plates or sub panels under in-plane loading (axial and shear stresses) the amplitude proposed by EN 1993-1-5 is given by expression (1)

$$\Delta w_{0,eq,EN1993-1-5} = \min(a/200; b/200) \quad (1)$$

where a is the panel's length and b is the panel's width. When comparing Winter's curve to numerical results following the second approach given by EN 1993-1-5, it can be concluded that the latter returns results somewhat conservative. Rusch and Lindner [5]

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Nomenclature

a	panel's length
b	panel's width
E	Young's modulus
f_u	material ultimate stress
f_y	material yield stress
k_σ	elastic buckling coefficient
R	panel's radius of curvature
t	panel's thickness
Z	curvature parameter $(=b^2/Rt)$
α	aspect ratio $(=a/b)$
$\overline{\lambda}$	non-dimensional slenderness parameter
ν	Poisson's coefficient
χ_{GMNIA}	maximum load factor obtained from numerical analysis
ψ	stress ratio $(=\sigma_1/\sigma_2)$
w	out-of-plane displacement
$\Delta w_{0,eq,EN1993-1-5}$	initial amplitude for geometric imperfections given by EN 1993-1-5
$\Delta w_{0,eq,EN1993-1-6}$	initial amplitude for geometric imperfections given by EN 1993-1-6
$\Delta w^{mod}_{0,eq,EN1993-1-5}$	modified initial amplitude for geometric imperfections
$\Delta w^{mod}_{0,eq,EN1993-1-6}$	modified initial amplitude for geometric imperfections
l_{Tw}	length of the biggest transverse wave
l_{Lw}	length of the biggest longitudinal wave
l_g	relevant gauge length given by EN 1993-1-6
U_n	dimple imperfection amplitude parameter given by EN 1993-1-6

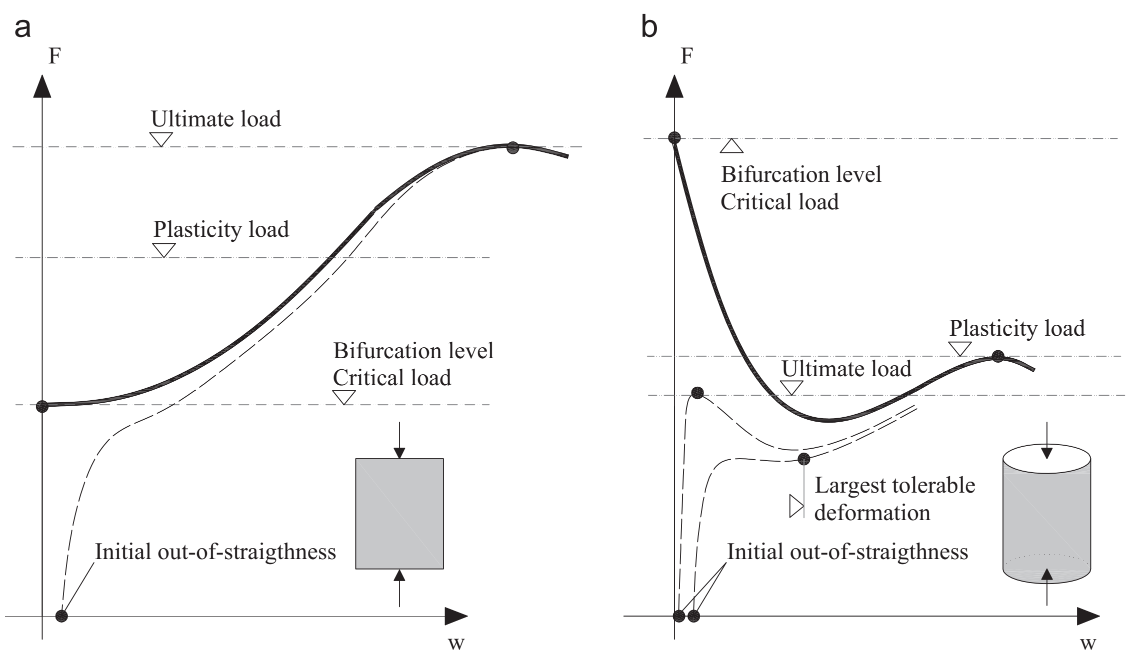

Fig. 1. Post-buckling behaviour examples: (a) plated structure and (b) shell structure [1].

performed a study where one of the main conclusions was that the best fit to Winter's curve is found with equivalent geometric imperfection modelled with a half-sine function in both longitudinal (with m half-waves) and transverse (with 1 half-wave) directions (see expression (2)) and with amplitudes of $b/420$. This conclusion is evident from Fig. 2, although it is seen that amplitudes equal to $b/200$ are also well adjusted to the EN1993-1-5:2006 curved, *i.e.* Winter curve (results shown in Fig. 2 are for simply supported square plates where the longitudinal edges are free to wave in the plate's plane and the loaded edges are constrained, see Fig. 3).

$$w = \Delta w_0 \sin\left(\frac{m\pi x}{a}\right)\sin\left(\frac{\pi y}{b}\right) \tag{2}$$

For shell structures (shells of revolution), recommendations on how to consider geometric imperfections are given in clause 8.7 of EN 1993-1-6 [6], where it is stated that "imperfections should generally be introduced by means of equivalent geometric imperfections". The amplitude of the equivalent geometric imperfections is given by the following expression:

$$\Delta w_{0,eq,EN1993-1-6} = \max(l_g U_n; 25tU_n) \tag{3}$$

where l_g is the relevant gauge length according to clause 8.4.4(2) in EN 1993-1-6 (for curved panels l_g relates only to the meridional direction as stresses in the circumferential direction are not relevant), U_n is the dimple imperfection amplitude parameter depending on the fabrication tolerance quality class and may be taken from Table 8.5 of EN 1993-1-6, and t is the shell's thickness. In contrast to what is considered for plates, EN 1993-1-6 only recommends the eigenmode affine shape for geometric imperfections, unless no other more unfavourable pattern can be justified. Additionally, it is recommended that the imperfection's maximum amplitude should always be applied inwards.

In conclusion, since unstiffened cylindrically curved panels are neither flat panels nor shells of revolution, the following question is raised: what standard should be followed to define imperfections in cylindrically curved panels? If neither of the options described above