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IMPERFECTION SENSITIVITY ANALYSIS OF ALUMINIUM CYLINDERS

ANALISI DELLA SENSIBILITÀ ALLE IMPERFEZIONI DI CILINDRI IN ALLUMINIO

[1]F.M. Mazzolani, [2]A. Mandara, [2]G. Di Lauro, [2]A. Maddaloni

[1] Department of Structural Analysis and Design, University of Naples Federico II, Naples, ITALY
[2] Department of Civil Engineering, Second University of Naples, Aversa (CE), ITALY

ABSTRACT

This paper represents a first basis for the evaluation of imperfection sensitivity of aluminium cylinders subjected to axial load, uniform external pressure and torsion, aimed at defining a specific design criterion similar to the procedure set out in the European Prestandards ENV1993-1-6 "Shell structures". To this purpose, a comprehensive non linear F.E.M. analysis has been carried out, considering a wide imperfection pattern and accounting for actual inelastic properties of material. Results show the necessity to define specific buckling curves for aluminium alloy shells, in order to account for peculiar features of such materials. The work, developed within the activity of CEN/TC250 SC9 Committee, devoted to the preparation of Eurocode 9 "Design of Aluminium Structures", is intended to be a background for the new Part 1-5 "Supplementary rules for shell structures" of Eurocode 9, which is presently under development.

SOMMARIO

Questa memoria costituisce una prima base per la valutazione della sensibilità alle imperfezioni di cilindri in alluminio soggetti a carico assiale, pressione esterna e torsione, finalizzata alla proposta ed alla messa a punto di un criterio di verifica simile a quello implementato nell'Eurocodice 3 (ENV1993-1-6 "Shell structures"). A tal scopo è stata condotta una vasta analisi F.E.M. non lineare, in cui è stato considerato sia l'effetto delle imperfezioni che quello del comportamento inelastico del materiale. I risultati evidenziano la necessità di curve di instabilità *ad hoc* per i gusci in leghe di alluminio, onde considerare gli aspetti peculiari del comportamento di tali materiali. Il lavoro, sviluppato nell'ambito dell'attività della Commissione CEN/TC250 SC9 dedicata alla preparazione dell'Eurocodice 9 "Design of Aluminium Structures", è di supporto alla stesura della nuova parte dell'EC9 "Supplementary rules for shell structures", attualmente in fase di sviluppo.

1. INTRODUCTION

In spite of the great effort carried out in the field of codification in the recent years, the problem of buckling of shell structures is not yet definitively assessed, with particular regard to the effects of geometrical imperfection and material plasticity. In most cases, in fact, the effect of imperfection is evaluated through the traditional, empirical "Lower Bound Design Philosophy", according to which a knock-down factor of buckling loads, usually denoted by α, is roughly fitted as lower limit of the scattered experimental data. In spite of its excess of conservativeness, this approach is kept in most of codes dealing with shell structures. As far as the inelastic behaviour of material is concerned, its effect is only approximately considered in the codes, usually by means of an empirical regression curve, which is not able to account for the actual inelastic material features. Moreover, all codes are concerned with mild steel shells, whereas no allowance is made for round-house-type materials, such as stainless or high strength steels and aluminium alloys, whose behaviour is peculiarly hardening.
Only in the last years, an attempt to relate the α factor to the actual imperfection magnitude has been made, leading to the latest issue of Eurocode 3 (ENV1993-1-6 "Supplementary rules for the shell structures") (Rotter, 1998, Schimdt, 2000), dealing with steel shells, in which the actual shell imperfection level is accounted for by means of quality classes. This part of EC3, which is now undergoing the conversion into the EN stage, indeed represents a significant step ahead, even though its interpretation of buckling in plastic range can not be extended to materials other than mild steel.
Based on these considerations, the result of almost 6000 F.E.M. simulation runs on aluminium shells is reported in this paper, in order to delineate a preliminary numerical data-set for the definition of buckling curves for aluminium alloys shells, liable to be introduced into the new part of Eurocode 9, entitled EN1999-1-5 Supplementary rules for shell structures. For the sake of homogeneity, the basic layout of ENV1993-1-6, and in particular the concept of quality classes, is referred to as a basis for comparison. At the same time, some typical issues of EC9, and mainly those concerning the classification of alloys, have been addressed. Cases under consideration are the same covered in ENV1993-1-6, namely cylinders under axial load, external pressure and torsion/shear. For such shell-load combinations numerical simulations have been carried out by accounting for a wide geometrical imperfection spectrum, in order to consider the most dangerous distributions. The non linear F.E.M. code ABAQUS has been used, whose reliability has been checked in turn on the basis of experimental tests carried out by the authors (Mandara & Mazzolani, 1993, Mandara, 1993a,b). The analysis is mainly dealing with the case of inelastic buckling, as this aspect is of major concern in a comparison between steel and aluminium structures. The case of purely elastic buckling, namely that involving very thin cylinders, has been assumed to be not material-dependent and, for this reason, no specific imperfection sensitivity analysis has been performed.

2. THE PARAMETRIC ANALYSIS

2.1 General

As it is well known, the most important aspect of the buckling behaviour of cylinders is their high imperfection sensitivity, due to the asymmetric stable-unstable response at bifurcation point. This usually involves a remarkable reduction of the actual buckling loads as compared to theoretical predictions, depending on both load case and imperfection magnitude. Most of the existing data confirm that the greatest reduction of buckling loads pertains to axially loaded thin cylinders (R/t > 300 ÷

R/t	R [mm]	t [mm]	L [mm]	L/R
Cylinders under axial compression				
200	1000	5	2000	2
100	1000	10	2000	2
50	1000	20	2000	2
25	1000	40	2000	2
12.5	1000	80	2000	2
Cylinders under external pressure				
200	1000	5	4000	4
100	1000	10	4000	4
50	1000	20	4000	4
200	1000	5	2000	2
100	1000	10	2000	2
50	1000	20	2000	2
200	1000	5	1000	1
100	1000	10	1000	1
50	1000	20	1000	1
Cylinders under torsion				
200	1000	5	4000	4
100	1000	10	4000	4
50	1000	20	4000	4
200	1000	5	2000	2
100	1000	10	2000	2
50	1000	20	2000	2

Table 1 – Geometric data of analysed cylinders (R mean radius, t wall thickness, L overall length).

	$f_{0.2}$ [MPa]	$n_{R.O.}$
Strong hardening alloys	100	10
Weak hardening alloys (Heat-treated alloys)	200	20
	300	30

Table 2 – Mechanical features of alloys under consideration.

500), whose instability predominantly occurs in elastic range. Nevertheless, imperfections play an important role also in load cases other than axial load, as well as in the case of thicker shells (R/t < 200), in which the interaction between imperfections and plasticity effect can occur, with a strong influence on the buckling modes and postcritical behaviour. When significant plastic deformations arise before buckling, in fact, both behaviour and deflected shape at buckling may be completely different, being affected by the plastic flow in material rather than by the geometrical non linear effect of surface imperfections. This effect is particularly evident in case of axially loaded cylinders.

2.2 Description of the analysis

In order to cover cylinders characterised by elasto-plastic buckling, the range of geometric properties shown in Table 1 has been considered. R/t and L/R values have been chosen in order to fall within the range of civil engineering applications. They also involve significant plastic deformation before buckling. Three types of alloys have been considered in the analysis (Table 2). All of them have been taken into account for cylinders under axial load, whereas for cylinders under external pressure and torsion the first two have been allowed for, only. Alloys in Table 2 have been chosen according to the distinction between Hardening alloys and Heat-treated alloys, already considered in the European Recommendations on Aluminium Alloys of ECCS (1978), and also introduced in the EC9. The exponent $n_{R.O.}$ of the *Ramberg-Osgood* law has been evaluated according to the *Steinhardt* proposal, i.e. by assuming $n_{R.O.} = f_{0.2}/10$, with $f_{0.2}$ expressed in N/mm^2 (Mazzolani, 1995). A further group of alloys with higher mechanical features ($f_{0.2}$ = 300N/mm^2 and $n_{R.O.}$ = 30) has been considered in order to cover high strength alloys (such as the 6000 and 7000 alloy series).

A static analysis of the unstable structural behaviour has been performed by means of F.E.M. simulation. In this case the limit load of the imperfect structure is evaluated together with the postcritical response as a function of the initial imperfection. Such a kind of approach is suitable when no snap-through or mode jumping is expected, as usually happens in the case of relatively thick cylinders which buckle in plastic range. In addition, a quasi-static procedure emphasises the buckling response regardless of testing procedures, by highlighting the postbuckling equilibrium paths.

The ABAQUS F.E.M. analysis has been carried out using the *RIKS option for the solution algorithm, the *DEFORMATION PLASTICITY option for material law and four-node, reduced integration S4R5 shell elements (Mandara, 1993a,b and Mandara & Mazzolani, 1993). The ABAQUS option *DEFORMATION PLASTICITY has been adopted, as it is slightly more conservative than the option *PLASTIC. Due to the fact that it does not take into account the elastic unloading arising after buckling, the *DEFORMATION PLASTICITY option provides a better interpretation of the structural response at buckling, which is essentially governed by the tangent modulus of material. In addition, such option is based on a pluriaxial formulation of the classical *Ramberg-Osgood* law, and this is particularly suitable when dealing with aluminium alloys.

2.3 Modelling of structural imperfection

The definition of structural imperfection is one of the basic concerns of the research on shell stability. The imperfection model assumed herein is:

$$w = w_0 \sum e^{-k_{1x}(x-x_o)^2} \cos\left[k_{2x}\pi \frac{(x-x_o)}{L}\right] e^{-k_{1y}(y-y_o)^2} \cos\left[k_{2y} \frac{(y-y_o)}{R}\right] \tag{1}$$

Even though this model does not contain higher order terms, it can be profitably used for interpreting the imperfection affecting relatively thick shells, being these imperfections generally smoother as compared to those potentially present in thin shells. By assigning suitable values to k_{1x}, k_{1y}, k_{2x}, k_{2y}, x_0 e y_0, Equation (1) is able to describe an imperfection distribution similar to both axisymmetric and asymmetric critical and postcritical modes. As a rule, an initial imperfection distribution similar to single or multiple critical modes has been assumed in the analysis. This corresponds to give the parameters k_{2x} and k_{2y} a value corresponding to the number of longitudinal (m) and circumferential (n) waves at buckling, respectively. In this way the most severe condition for the buckling response has been investigated, so as to determine a lower bound of the ultimate load carrying capacity as a function of the initial imperfection magnitude. Other types of imperfection (say a concentrated dimple placed at cylinder midlength or a continuous longitudinal groove) have been also considered. Such imperfections, obtained by giving parameters k_{1x}, k_{1y}, k_{2x}, k_{2y}, x_0 e y_0 specific values, have been allowed for in case of axially loaded cylinders only. Examples of imperfection distributions according to Equation (1) are illustrated in Figure 1.

In order to widen the field of investigated imperfection patterns, the following alternative expression has been also used (Chryssanthopulos, Baker & Dowling 1991, Arbocz & Hol, 1991):

$$w = w_0 \sum_{k=1}^{n_1} \sum_{l=1}^{n_2} \sin\frac{k\pi x}{L}\left(C_{kl}\cos\frac{ly}{R} + D_{kl}\sin\frac{ly}{R}\right) \tag{2}$$

where the coefficients A_{kl} and B_{kl} depend on the constructional features of the shell. By using such an imperfection model it is possible to interpret the sharp gradients which are common in the imperfection pattern of very thin shells. In this analysis such imperfection model has been considered for axially loaded cylinders only, with coefficients A_{kl} and B_{kl} assumed to have a random distribution. A detailed description of the imperfection distributions assumed in the analysis is given in Mazzolani & Mandara, 2003.

In order to assume imperfection patterns distributed according to shell critical modes, for each of cylinder geometric types referred to in Table 1, the critical bifurcation modes and corresponding critical loads have been evaluated for both elastic and plastic instability. Effects of plasticity on the buckling load have been taken into account by means of a suitable plasticity reduction factor η, obtained from literature (Gerard, 1962). A synopsis view of bifurcation loads, critical modes and plasticity factors is given in Table 3 for all relevant load cases considered in the analysis.

In case of axially loaded cylinders failing in purely elastic range, the equation (see Table 3):

$$\frac{(m^2 + (nL/\pi R)^2)^2}{m^2} = \frac{2}{\pi}\frac{L^2}{Rt}\sqrt{3(1-\nu^2)} \tag{3}$$

shows that many critical modes are possible for the same value of the buckling load. These modes can be individuated by calculating the minimum value of the bifurcation load for couples of integer values of *m* and *n*. Equation (3) can be plotted in the *m,n* plane where, by means of an appropriate choice of axis scale, it represents the equation of a circle ("*Koiter* circle").

In case of plastic buckling, critical axisymmetric modes derived by a bifurcation analysis has been considered. The minimum critical load in plastic range is expressed in the general form:

$$\sigma_{cr,p} = \eta \sigma_{cr,el} \qquad (4)$$

where η is a reduction factor introduced to take into account the plasticity effect. Values obtained by *Gerard* (1962) have been indicated in Table 3, evaluated under the assumption of incompressible material in the plastic range and following the plastic deformation approach.

The values of both tangent $d\sigma/d\varepsilon$ and secant moduli σ/ε at the elastoplastic bifurcation stress can be calculated by means of the *Ramberg-Osgood* law $\varepsilon = \sigma/E + 0.002(\sigma/E)^n$ (Mazzolani, 1995).

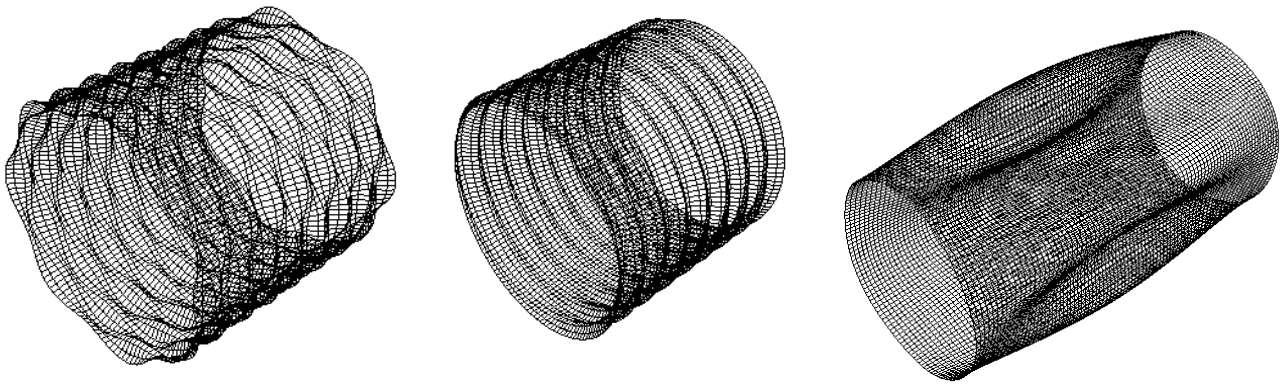

Figure 1 - Imperfection distributions according to Equation (1) (n = 7, m = 15), (m = 16), (n = 4).

<table>
<tr><th colspan="4">Cylinders under axial load</th></tr>
<tr><td>Elastic bifurcation load</td><td>Number of circumferential (n) and axial (m) waves at elastic buckling</td><td>Plasticity reduction factor</td><td>Number of axial (m) waves at plastic buckling</td></tr>
<tr><td>$\sigma_{cr,el} = \frac{E}{\sqrt{3(1-\nu^2)}} \frac{t}{R}$</td><td>$\frac{(m^2 + (nL/\pi R)^2)^2}{m^2} = \frac{2}{\pi} \frac{L^2}{Rt} \sqrt{3(1-\nu^2)}$</td><td>$\eta = \frac{\sqrt{E_t E_s}}{E}$</td><td>$m = \frac{L}{\pi} \sqrt{\frac{12}{Rt\sqrt{6 + 9\frac{E_t}{E_s} + \frac{E_e}{E_t}}}}$</td></tr>
<tr><th colspan="4">Cylinders under external pressure</th></tr>
<tr><td colspan="2">Elastic bifurcation load</td><td>Number of circumferential (n) waves at elastic buckling</td><td>Plasticity reduction factor</td></tr>
<tr><td colspan="2">$p_{cr,el} = \frac{Et/R}{n^2 - 1 + \frac{1}{2}\left(\frac{\pi R}{L}\right)^2} \left[\frac{1}{\left(\left(\frac{nL}{\pi R}\right)^2 + 1\right)^2} + \frac{t^2}{12R^2(1-\nu^2)} \left(n^2 - 1 + \left(\frac{\pi R}{L}\right)^2\right)^2 \right]$</td><td>$n = 2.7 \left(\frac{R}{L}\right)^{0.5} \left(\frac{R}{t}\right)^{0.25}$</td><td>$\eta = \frac{1}{4}\frac{E_s}{E} + \frac{3}{4}\frac{E_t}{E}$</td></tr>
<tr><th colspan="4">Cylinders under torsion</th></tr>
<tr><td colspan="2">Elastic bifurcation load</td><td>Number of circumferential (n) waves at elastic buckling</td><td>Plasticity reduction factor</td></tr>
<tr><td colspan="2">$\tau_{cr,el} = 0.75\, E \left(\frac{R}{L}\right)^{1/2} \left(\frac{t}{R}\right)^{5/4}$</td><td>$n = 4.2\, (0.75)^{1/8} \sqrt{\frac{R}{L}} \sqrt[4]{\frac{R}{t}}$</td><td>$\eta = \frac{E_s}{E}$</td></tr>
</table>

Table 3 – Synopsis of elastic and elasto-plastic buckling loads and corresponding critical modes for load cases under consideration.

3. DISCUSSION OF RESULTS

3.1 General

The F.E.M. imperfection sensitivity analysis has emphasised the strong influence of the initial imperfection pattern, in particular when directed according to critical modes. Typical imperfection sensitivity curves are plotted in Figure 2, showing the great reduction of the buckling load P_u with the imperfection magnitude w_0, compared with the theoretical elastoplastic critical load $P_{cr,th}$. Limits corresponding to quality classes A, B and C, as defined in ENV1993-1-6, are also drawn (see chapter 4). In general, elasto-plastic bifurcation loads according to *Gerard*'s η coefficients are in good agreement with F.E.M. prediction. As a rule, values provided by simulation for $w_0 = 0$ fall on the conservative side with a slight discrepancy, decreasing as long as both the *R/t* ratio and the elastic limit strength $f_{0.2}$ decrease. In general, it can be confirmed that the effect of initial imperfection increases as *R/t* and *L/R* ratios, as well as the elastic limit strength $f_{0.2}$ increase, while in the same conditions the effect of plasticity is relatively smaller. Some buckling deflected shapes are shown in Figure 3.

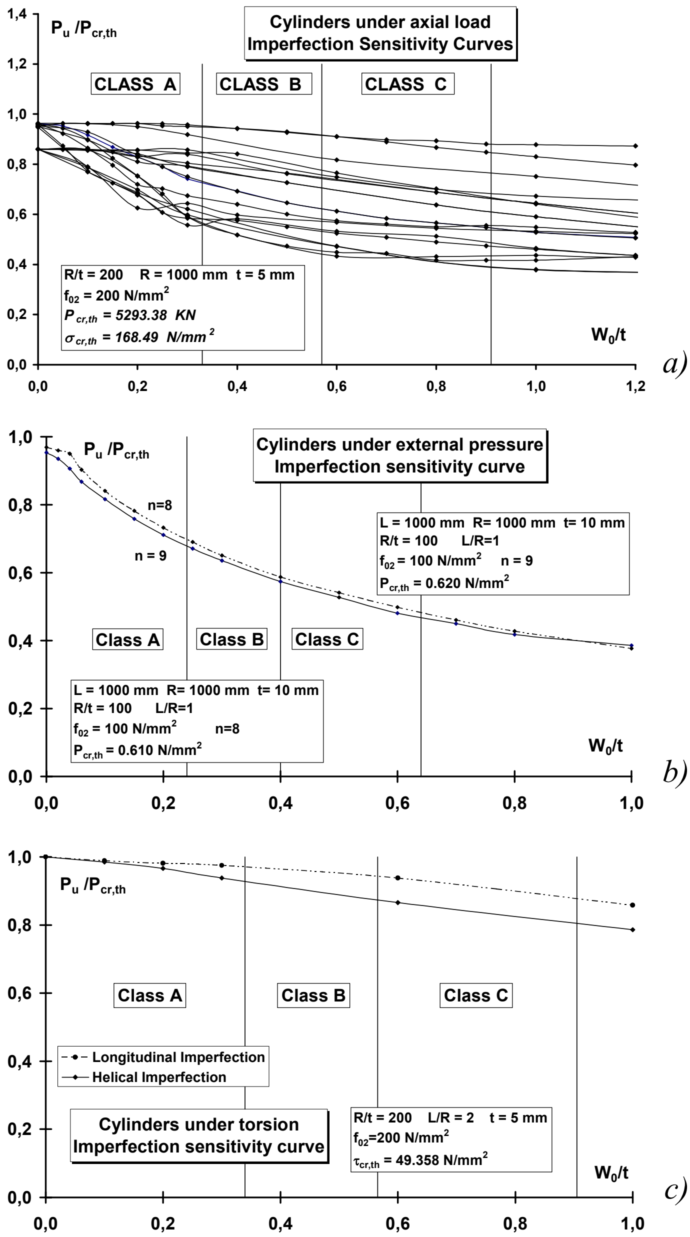

Figure 2 - Imperfection sensitivity curves for cylinders under axial load, external pressure and torsion.

3.2 Cylinders under axial load

F.E.M. analysis confirms that buckling may be either axisymmetric or asymmetric depending on the initial imperfection distribution, as well as on the value of buckling stress. For an asymmetric imperfection the typical snap-through behaviour of perfectly elastic shell tends to disappear when plasticity effects occur, at least for small imperfection magnitudes. Transition from asymmetric (Figure 3a) to axisymmetric (Figure 3b) buckling corresponds to a critical value of imperfection w^* (Mandara, 1993a,b, Mandara & Mazzolani, 1993). When considering an axisymmetric imperfection, buckling is always symmetric with the ultimate load decreasing with the imperfection magnitude. In case of imperfections different from those corresponding to critical modes, the buckling behaviour is rather different. In general, imperfection sensitivity for concentrated or linear defects is remarkably lower than for imperfection distributions similar to critical modes.

The influence of boundary conditions deeply depends on the imperfection type and, as a consequence, on the buckling character. In particular, it is rather significant when the buckling is axisymmetric, due to the fact that buckles take place near the loaded ends. On the contrary, the influence of boundary conditions drops down when the buckling deflection is predominantly developed in shell intermediate region. This happens in case of asymmetric instability and, in general, in all cases when buckling occurs without significant precritical plastic deformations. Eventually, a great scatter of results can be observed (Figure 2a), which confirms the commonly acknowledged experimental performance of axially loaded cylinders. Such scatter increases as long as *R/t* ratio and elastic limit $f_{0.2}$ increase. The lower bound of numerical data corresponds, with good approximation, to imperfection distributions directed according to a single or to a combination of critical modes obtained from Equation (3).

3.3 Cylinders under external pressure

Both theoretical and experimental predictions on the buckling pattern are fully confirmed by F.E.M. analysis. The buckled configuration always consists of one longitudinal halfwave and of a number of circumferential halfwaves increasing with the *R/t* ratio of the cylinder (Figure 3c). This buckling pattern is kept in inelastic range, too. The imperfection sensitivity analysis has shown a lower influence of geometrical imperfections on the load bearing capacity compared with axially loaded cylinders (Figure 2b). This is confirmed also by a reduced scatter of results. In addition, a relatively small sensitivity to localised defects has been detected. The effect of restraint conditions is significant for very short cylinders, only, and the theoretical prediction of the critical load is well caught in case of hinged ends. In conclusion, imperfection distributions directed according to the elastic critical mode give place, as a rule, to the lowest buckling response of the shell.

3.4 Cylinders under torsion

For this load case the same considerations made for cylinders under external pressure hold. The buckling shape is well interpreted by F.E.M. analysis, showing that most of deformation is concentrated in the central region of the cylinder (Figure 3d). As in the case of cylinders subjected to external pressure, the influence of restraint condition is negligible in most cases, apart from the case of very short cylinders. F.E.M. results, however, best agree with theoretical prediction in case of hinged ends. Finally, the observed scattering of results is very reduced and, also, a slightly lower global imperfection sensitivity is observed (Figure 2c).

4. CONSIDERATIONS ON THE USE OF EC3 PROVISIONS FOR ALUMINIUM SHELLS

In this chapter a first assessment of the applicability of EC3 provisions to aluminium shells is given. According ENV1993-1-6, the characteristic buckling strengths are obtained from the characteristic yield strength by means of the relationships $\sigma_{xRk} = \chi_x f_{y,k}$, $\sigma_{\theta Rk} = \chi_\theta f_{y,k}$, $\tau_{x\theta Rk} = \chi_t f_{y,k}/\sqrt{3}$, for axial load, external pressure and torsion, respectively, in which the reduction factors χ_x, χ_θ and χ_t are given as a function of the relative slenderness of the shell λ from:

$$\chi = 1, \text{ for } \lambda < \lambda_0; \qquad \chi = 1 - \beta\left(\frac{\lambda-\lambda_0}{\lambda_p-\lambda_0}\right)^{\eta}, \text{ for } \lambda_0 < \lambda < \lambda_p; \qquad \chi = \alpha/\lambda^2, \text{ for } \lambda_p < \lambda_0 \tag{5}$$

in which α is the elastic imperfection reduction factor, β is the plastic range factor, η is the interaction exponent and λ_0 is the squash limit relative slenderness, all of them depending on the load case (Tables 4 and 5). The value of the plastic limit slenderness λ_p is given by $\lambda_p = \sqrt{\alpha/(1-\beta)}$.

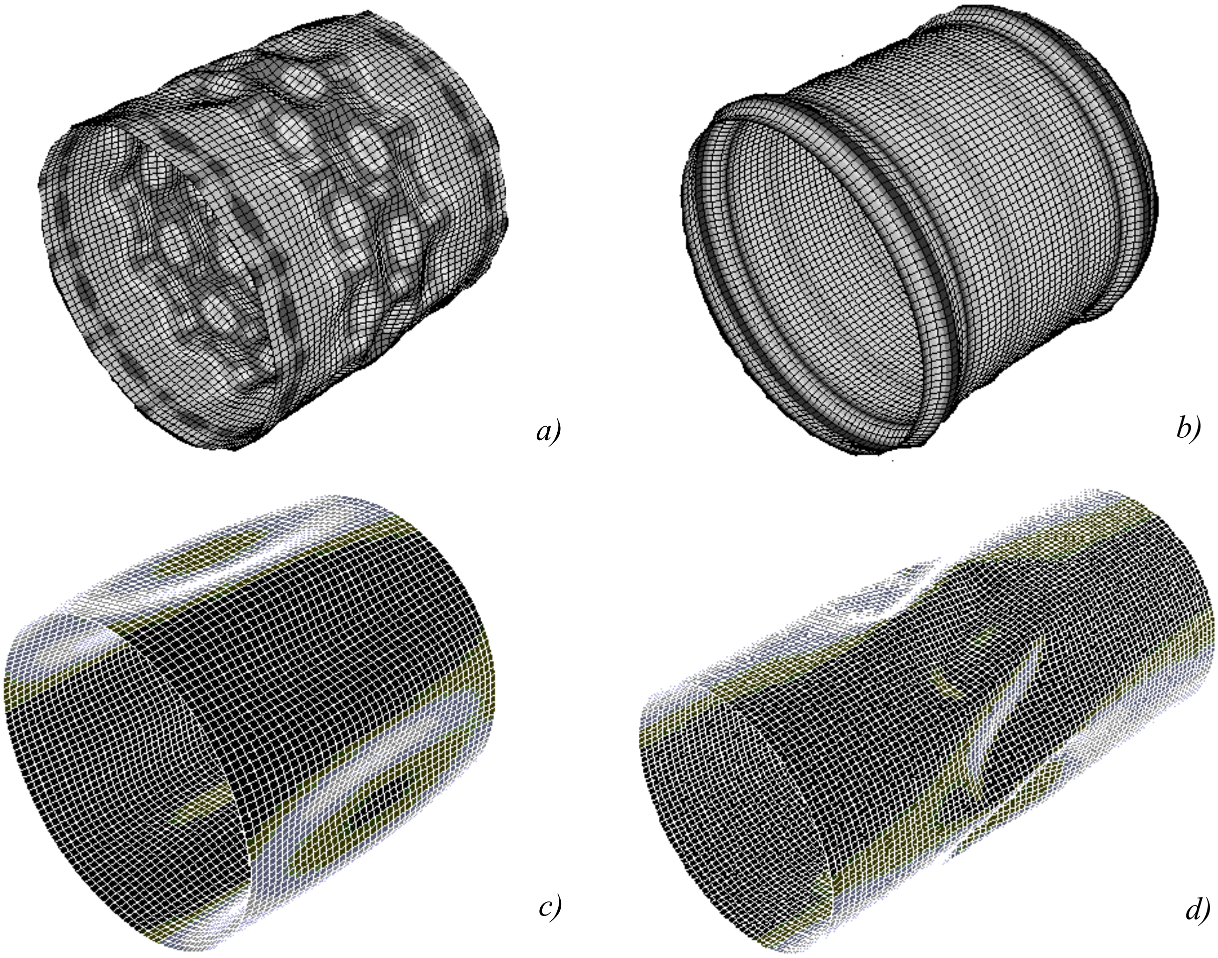

Figure 3 – Buckling deflected shapes: a) axial load, elastic buckling; b) axial load, plastic buckling; c) uniform external pressure; d) torsion/shear.

	Axial (meridional) load	External pressure and torsion (shear)
λ_0	0.20	0.40
β	0.60	0.60
η	1.00	1.00

Table 4 - Expressions of factors λ_0, β and η according to ENV1993-1-6.

Fabrication tolerance quality class	Description	Axial (meridional) load		External pressure (α_θ) and torsion (shear) (α_τ)
		Q	α_x	α_θ or α_τ
Class A	Excellent	40	$\alpha_x = \dfrac{0.62}{1+1.91(\Delta w_k/t)^{1.44}}$	0,75
Class B	High	25		0,65
Class C	Normal	16		0,50

Table 5 - Fabrication tolerance quality classes and expressions of factor α according to ENV1993-1-6.

The shell slenderness parameters for stress components corresponding to considered load cases are determined from $\lambda_x = (f_{y,k}/\sigma_{xRc})^{1/2}$, $\lambda_\theta = (f_{y,k}/\sigma_{\theta Rc})^{1/2}$, $\lambda_\tau = ((f_{y,k}/\sqrt{3})/\tau_{Rc})^{1/2}$, respectively, where the critical buckling stresses σ_{xRc}, $\sigma_{\theta Rc}$ and τ_{Rc} are obtained from the code according to the loading case.

By observing Figures 4 to 6, it is clear that rules given in ENV1993-1-6 for steel shells cannot be applied to aluminium cylinders as they are, but need a deep transformation in order to take into account the peculiar aspects of aluminium alloys. This is evident when comparing χ - λ curves of ENV1993-1-6 with the results of simulation analysis. Each quality class A, B and C, and both strong and weak hardening alloys are represented in the comparison. With the only exception of shells loaded in torsion, matching code curves with numerical results clearly demonstrates that EC3 curves are unacceptably unconservative, in particular in case of strongly hardening alloys in the slenderness range $0.5 < \lambda < 2$. This stands in particular for axially loaded cylinders. In order to try an adaptation of EC3 curves, a modification of λ_0, β and η parameters would be necessary, so as to fit numerical results with a better degree of approximation. Nevertheless different values of λ_0, β and η, as shown in Mazzolani & Mandara (2003), would result in too much conservative values of the buckling factor χ for $\lambda < 1$. This would also cause the distinction between strongly and weakly hardening alloys to miss its meaning, and this is evidently contrary to what commonly accepted in both codification (including Eurocode 9) and literature. This is the reason why a completely new formulation of buckling curves for aluminium shells is needed.

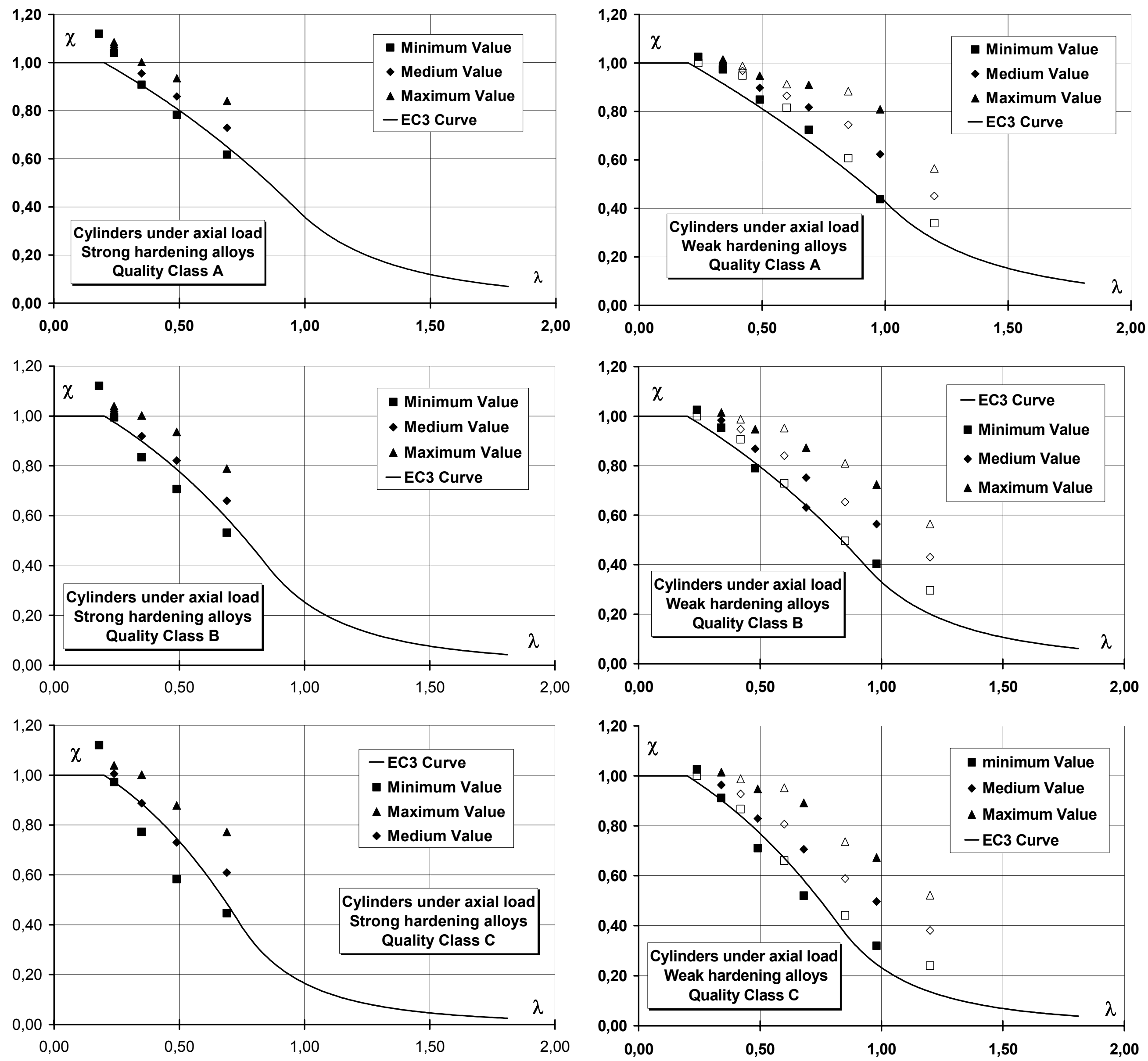

Figure 4 – Comparison of EC3 buckling curves with simulation data: axial load.

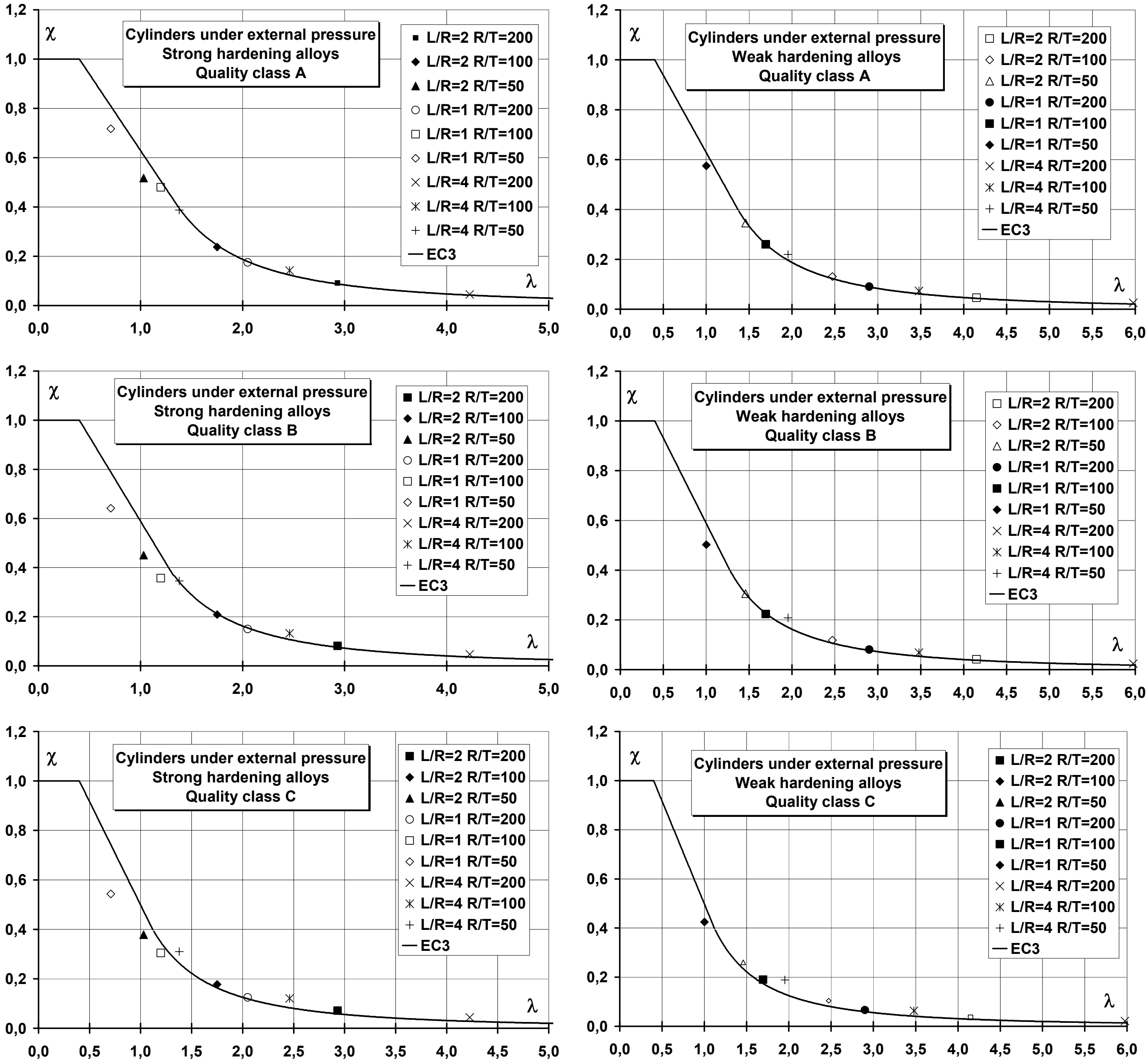

Figure 5 – Comparison of EC3 buckling curves with simulation data: external pressure.

5. CONCLUSIVE REMARKS

The study presented in this paper has intended to start a discussion on the definition of buckling curves liable to be introduced into European codification on aluminium alloy structures. To this purpose, a wide parametric F.E.M. analysis has been carried out by means of ABAQUS code. Load cases referring to axial load, external pressure and torsion (shear) have been addressed. Such cases are those considered in ENV1993-1-6, too, dealing with steel shells. In order to account for as many imperfection cases as possible, a global amount of almost 6000 runs have been carried out. This led to a thorough understanding of the problem, with particular regard to the case when relevant inelastic deformations occur before buckling. In particular, the interaction between the effect of geometrical imperfections and that of material inelastic behaviour has been emphasised.

Results obtained from simulation analysis have been matched with provisions given in ENV1993-1-6, showing that buckling curves used for steel shells are clearly unsuitable to aluminium shells, because of the strongly different behaviour in the transition region between elastic and plastic range. In short, buckling data for aluminium shells fall well below curves for steel, in particular in case of strong hardening alloys. This result was predominantly evident in case of axial load and external pressure for slenderness values around the plastic limit slenderness λ_p.

Results of F.E.M. analysis and comparison with EC3 buckling curves clearly indicate that a set of provisions purposely conceived for aluminium alloy shells is necessary, in order to get adequate accuracy in the prediction of buckling strength as well as to keep into account all peculiar features of these materials. In this view, a proposal regarding an alternative formulation for buckling curves is being developing within the activity of CEN/TC250 SC9 Committee and is at the moment under discussion (Mazzolani & Mandara, 2003). Hopefully, such proposal will be shortly accepted for the introduction into the new part 1-5 "Supplementary rules for shell structures" of Eurocode 9.

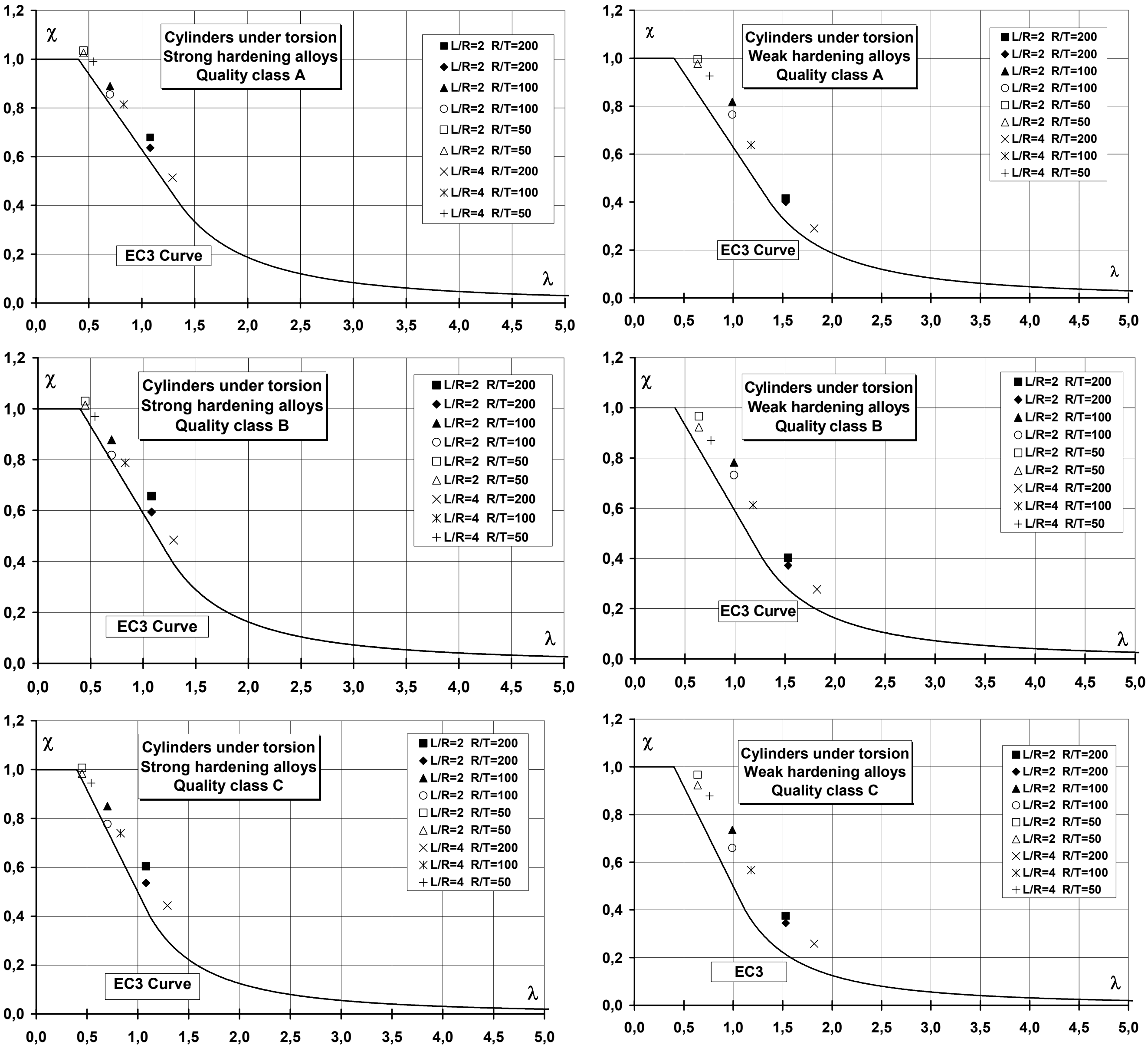

Figure 6 – Comparison of EC3 buckling curves with simulation data: torsion (shear).

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