EN 13445 "Unfired pressure vessels"

Background to the rules in Part 3 Design

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Introduction

The European standard EN 13445 "Unfired pressure vessels" provides a precedent in that after 10 years of discussion between experts, a European consensus was achieved in the field of pressure equipment. Part 3 which was prepared by a group of leading European experts under the guidance of Dr. Fernando LIDONNICI, Sant'Ambrogio, (Milano, Italy), represents a major advance in European technical convergence. The adoption of the first issue of EN 13445 in May 2002 was the first step of a continuous process for development & improvement.

This new standard benefits from the contribution of all the European expertise; as such, it includes innovative capacities and supplies solutions for modern subjects.

The CEN Design rules promote Limit Analysis and Design By Analysis – Direct Route. Design has a strategic importance for the future and the competitiveness of the pressure equipment industry. Optimum design allows substantial advantages such as: thickness reduction and damage control in service with safety increase and drastic maintenance cost reduction.

The objective of this book is to explain the background of these rules, to help industry to apply them in the most effective way. It was initiated by EPERC, the European Pressure Equipment Research Council, and and was awarded a contract of CEN, the European Standardization Committee, with support of the European Commission.

The release of this booklet was made possible by the co-operative efforts of the experts involved in the discussion of Part 3, namely:

- Guy BAYLAC	Design criteria
- Matteo CANNEROZZI	Openings in shells
- Joris DECOCK	Additional non-pressure loads
- Richard FAWCETT	Shell under internal pressure, shell under external pressure, Design of flanges and domed ends
- Alain HANDTSCHOEWERCKER	Simplified assessment of fatigue life, Design by Analysis based on stress categories
- Fernando LIDONNICI	Flat ends
- Stephen MADDOX	Detailed assessment of fatigue life
- Olavi VALTONEN	Rectangular pressure vessels
- Francis OSWEILLER	Heat exchangers, Expansion bellows
- Joachim WOELFEL	Advanced design for flanges, Advanced design of tubesheets
- Josef ZEMAN	Design by Analysis – Direct Route

UNM, which leads the maintenance Help desk of EN 13445 (EN 13445/MHD), provides its logistical support to compile and format the contributions, and agrees to put the booklet on the EN 13445/MHD website for free uploading.

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Presentation

This book shows the background to Part 3 "Design" of the European standard EN 13445 "Unfired pressure vessels". To facilitate for the reader the cross-referencing between the explanations provided and the normative content of the standard itself, the booklet is organized according to the same clause numbering than the standard. In each clause, a different numbering, with letters, is provided.

Each explanatory clause address the following topics:

- Background and references to the rules (with, where relevant a bibliography included in each clause)
- Detailed description of the method and comparison to other methods
- Future developments

The following clauses are included:

NOTE Some clauses are missing in the 1st issue of 9 July 2004. They will be included in the 2nd issue, the target date of which is 1 September 2004.

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Annex P Classification of weld details to be assessed using principal stresses

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Annex ZA Clauses of this European Standard addressing essential requirements or other provisions of the EU Directives

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1 Scope

Part 3 of EN 13445 gives the rules to be used for design and calculation under internal and/or external pressure (as applicable) of pressure bearing components of Pressure Vessels, such as shells of various shapes, flat walls, flanges, heat exchanger tubesheets, including the calculation of reinforcement of openings. Rules are also given for components subject to local loads and to actions other than pressure.

For all these components the DBF (Design by Formulae) method is generally followed, i.e. appropriate formulae are given in order to find stresses which have to be limited to safe values. These formulae are generally intended for predominantly non-cyclic loads, which means for a number of full pressure cycles not exceeding 500.

However general prescriptions are also given for DBA (Design by Analysis) which can be used either to evaluate component designs or loading situations for which a DBF method is not provided, or, more generally, as an alternative to DBF.

Methods are also given where a fatigue evaluation is required, due to a number of load cycles being greater than 500. There are two alternative methods: a simplified method based on DBF (valid mainly in case of pressure variations) and a more sophisticated method based on a detailed determination of total stresses using, for example, FEM or experimental methods. This can be used also in the case of variable loads other than pressure.

For certain components (such as flanges and tubesheets) also an alternative DBF method (based on limit analysis) has been provided; the choice of which method has to be used in each particular case is left to the Designer.

For the time being, the scope of Part 3 is limited to steel components working at temperatures lower than the creep range of the specific material concerned.

2 Normative references

Clause 2 includes the list of the referenced documents cited in EN 13445-3 in such a way as to make them indispensable for the application of the standard. These references are dated, that means that subsequent amendments to, or revisions of, dated references will need to be incorporated by amendment of the document referring to them.

3 Terms and definitions

Clause 3 gives the definition of terms applicable to the whole Part 3 such as calculation pressure/temperature, design pressure/temperature, governing weld joint. Specific definitions are also found in the clause of the standard where they are used.

4 Symbols and abbreviations

Clause 4 establishes symbols and units, needed to apply Part 3. Specific symbols are also found in the clause of the standard where they are used. The units are SI-units, consistent with the ISO 31 standard series.

5 Basic design criteria

5A General

Basic design criteria for Part 3 are given in Clause 5.

It is essential to remember that:

EN 13445-3:2002 does not contain rules to design in the creep range. Creep design rules are under development and will be introduced later, probably in 2006.

The rules are not applicable in case of localised corrosion. In this case the material shall be changed or adequate protection provided.

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Clause 5 deals successively with:

- Corrosion, erosion and protection
- Load cases
- Design methods
- Weld joint coefficient
- Design of welded joints

5B Corrosion, erosion and protection

In the standard "corrosion" is a very general term to be understood as all forms of wastage. Thus, it is impossible to give rules to protect against corrosion, due to the multitude of cases to consider; only general advice can be provided in informative notes.

The two cases where the standard can be prescriptive are:

1. When an additional thickness is sufficient to protect against corrosion during lifetime. Then the design shall take into consideration the corroded condition at the end of life.

2. Or when an adequate coating or lining is a reliable protection against corrosion.

Figure 3-1 of the Standard gives the relation between the various thicknesses. This figure is applicable to plates. It is relatively complex since it aims at providing guidance to order the plates.

5C Load cases

Load cases to consider are in conformity with the requirements of Annex I of the Directive on Pressure Equipment [1]. The classification of the load cases in three categories is classical, but may be modified in the future.

5D Design methods

This Part provides currently two design methods:

1. Design by formulae which is used in Clause 7 to 16

2. Design by analysis, which is covered by Annexes B and C. Annex C uses the classical approach of stress portioning while Annex B Design by Analysis – Direct Route is totally new.

A new amendment is in preparation on experimental techniques.

This Standard mainly addresses welded construction and the communication between Part 3 and the other parts [2] is ensured by the testing groups.

This architecture has been built from elements borrowed from the German Code AD-Merkblatt [3], the British Specification for unfired pressure vessels PD 5500:2000 [4] and the French Code CODAP 2000 [5].

In Part 1 of the Standard, the testing group of a weld is defined as "one of the four groups designed to specify the extent of non destructive testing and destructive testing necessary in association with weld joint coefficient, material grouping, welding process, maximum thickness, service temperature range".

The table of testing groups is given in annex (Table 1). Testing groups are classified from 1 to 4 in decreasing extent of NDT. The manufacturer may select a high extent of NDT (testing groups 1 or 2), a reduced extent of NDT (testing group 3), or just a visual inspection (testing group 4).

However testing groups are designed to offer the same safety by a combination of several factors, as represented in table 1. Material grouping of the table is per CR ISO/TR 15608:1999 [6]. More detailed information on testing groups can be found in EPERC Bulletin Nr 2 [7].

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When the weld is a governing one (longitudinal weld on a cylinder or a cone, or main weld on a sphere or a dished end), the testing group controls the thickness of the weldment and generally the thickness of adjacent plates by the weld joint efficiency factor.

It is intended that a single testing group shall be applied to the entire vessel. Nevertheless, where there is more than a single governing joint on a vessel and provided the requirements of table 1 are met, combinations of testing groups 1 and 2 or 1, 2 and 3 are permissible. Thus the concept of testing group is more flexible than the concept of vessel category used in certain codes.

However testing group 4 cannot coexist with any other testing group on the same vessel.

5D-1 Pressure loading of non-cyclic nature

Many pressure vessels are designed for pressure loading of non-cyclic nature. The requirements specified in the Design By Formulae (DBF) section of EN 13445-3 provide satisfactory designs when the number of full pressure cycles or equivalent full pressure cycles is less than 500. This value is similar to the value 1000 of the ASME Code [8], but reduced to take account of a higher nominal design stress (safety coefficient of 2,4 instead of 4 on the ultimate strength)

$$n_{eq} \leq 500 \qquad \text{(5D-1)}$$

Then no fatigue analysis is necessary and the standard requirements of non destructive testing given in EN 13445-5 shall be applied.

For n_i pressure cycles of pressure range ΔP_i less than the full calculation pressure P, the number of equivalent full pressure cycles is given by:

$$n_{eq} = \Sigma\, n_i \cdot \left(\frac{\Delta P_i}{P_{max}} \right)^3 \qquad \text{(5D-2)}$$

In the above formula, the exponent 3 is the exponent related to the design fatigue curve, P_{max} is the maximum permissible pressure based on the analysis thickness. P_{max} is greater than the calculation pressure P. This increases the number of allowed full pressure cycles.

For usual components, P_{max} expression is generally given in the different clauses of the DBF section. If necessary, P_{max} may be replaced by the calculation pressure P.

Pressure vessels to testing group 4, are intended for non-cyclic operation and are limited to 500 full pressure cycles or equivalent full pressure cycles.

5D-2 Pressure loading of cyclic nature

If the number of full pressure cycles or equivalent full pressure cycles is likely to exceed 500, the calculations of vessels of testing groups 1, 2 and 3 shall be completed by a simplified fatigue analysis, as given in clause 17 of EN 13445-3 or, if necessary, by a detailed fatigue analysis, as given in clause 18.

In addition clauses 17 and 18 specify limiting values D_{max} of the cumulative damage for the determination of critical zones where additional requirements on weld imperfections and NDT shall be applied, as defined in Annex G of EN 13445-5.

Figure 5D-1 shows the correlation between fatigue analysis and NDT.

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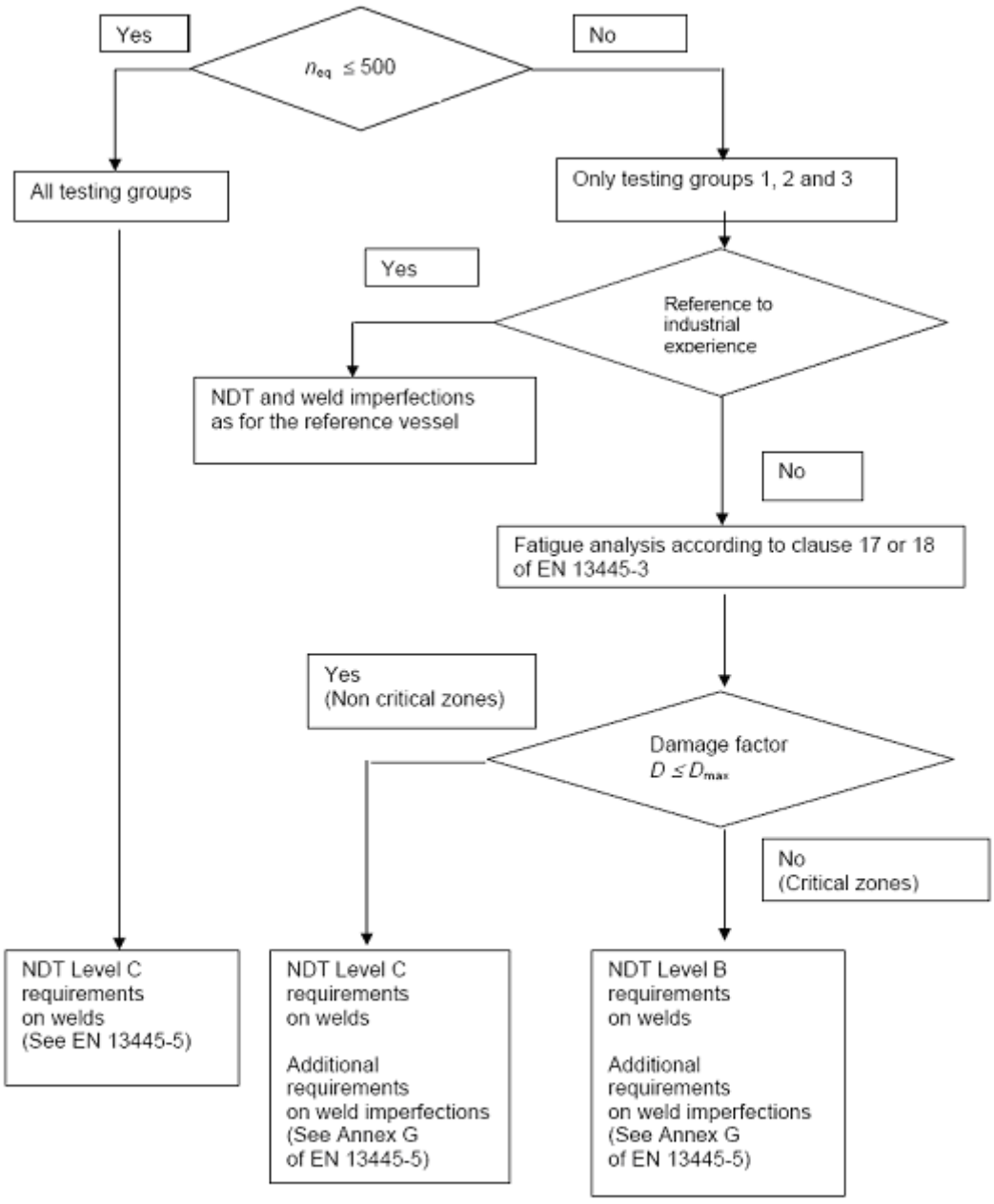


Figure 5D-1 Fatigue analysis and NDT

6 Maximum allowed values of the nominal design stress for pressure parts

Table 6.6.1-1 of Part 3 gives the maximum value of the nominal design stress for ductile steels, according to the definition of ductility given in EN 13445-2. This value is to be used in the DBF section of EN 13445-3 for pressure components other than bolts.

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Table 6.6.1-1Testing groups for steel pressure vessels

Requirements	Testing group a						
	1		2		3		4
	1a	1b	2a	2b	3a	3b	[b,j]
Permitted materials [g]	1 to 10	1.1, 1.2, 8.1	8.2, 9.1, 9.2, 9.3, 10	1.1, 1.2, 8.1	8.2, 9.1, 9.2, 10	1.1, 1.2, 8.1	1.1, 8.1
Extent of NDT for governing welded joints [e,h]	100 %	100 %	100 % - 10% [d]	100 % - 10 % [d]	25 %	10 %	0 %
NDT of other welds	Defined for each type of weld in Table 6.6.2-1						
Joint coefficient	1	1	1	1	0,85	0,85	0,7
Maximum thickness for which specific materials are permitted	Unlimited [f]	Unlimited [f]	30 mm for groups 9.1, 9.2 16 mm for groups 9.3, 8.2 [i], 10	50 mm for groups 1.1, 8.1 30 mm for group 1.2	30 mm for groups 9.2, 9.1 16 mm for groups 8.2, 10	50 mm for groups 1.1, 8.1 30 mm for group 1.2	12 mm for groups 1.1, 8.1
Welding process	Unlimited [f]	Unlimited [f]	Fully mechanical welding only [c]		Unlimited [f]	Unlimited [f]	Unlimited [f]
Service temperature range	Unlimited [f]	Unlimited [f]	Unlimited [f]	Unlimited [f]	Unlimited [f]		Limited to (-10 to +200) °C for group 1.1 (-50 to+300) °C for group 8.1

[a] All testing groups shall require 100 % visual inspection to the maximum extent possible
[b] Testing group 4 shall be applicable only for:
- Group 2 fluids; and
- $P_S \leq 20$ bar; and
- $P_S V \leq 20\,000$ bar·L above 100 °C; or
- $P_S V \leq 50\,000$ bar·L if temperature is equal or less than 100 °C;
- higher pressure test (See clause 10);
- maximum number of full pressure cycle less than 500;
- lower level of nominal design stress (See EN 13445-3).

[c] Fully mechanised and/or automatic welding process (See EN 1418:1997).
[d] First figure: initially, second figure: after satisfactory experience. For definition of “satisfactory experience", see 6.6.1.1.4
[e] Testing details are given in Table 6.6.2-1
[f] Unlimited means no additional restriction due to testing. The limitations mentioned in the table are limitations imposed by testing. Other limitations given in the various clauses of the standard (such as design, or material limitations , etc.) shall also be taken into account.
[g] See EN 13445-2 for permitted materials.
[h] The percentage relates to the percentage of welds of each individual vessel
[i] 30 mm for group 8.2 material is allowed if delta ferrite containing welding consumables are used for depositing filling passes up to but not including the capping run.
[j] Limited to single compartment vessels and single material group.

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For ferritic steels, the safety factor of 2,4 put on the ultimate strength at 20° C impedes efficient use of the new modern high yield strength steels (Thermo-Mechanically rolled and Quenched and Tempered steels). Therefore Annex B of EN 13445-3, DBA Direct Route, allows the use of a reduced safety equal to 1,875, but still giving a margin of safety of 2 towards burst for vessels with moderate notch effect (e.g. weld details of testing group 1 in accordance with Annex A of EN 13445-3).

For austenitic steels with a rupture elongation greater or equal than 35 %, the nominal design stress based on the ultimate strength at calculation temperature T is safe, but may induce large strains. Therefore the nominal design stress cannot exceed the value:

$$\left(\frac{R_{\mathrm{p1,0}/T}}{1{,}2} \right) \qquad \text{(6-1)}$$

Many pressure vessels are built to testing group 4 without NDT, except for the cone to cylinder junction when the cone angle is greater than 30°. Although these vessels are built from easy-to-weld steels belonging to material groups 1.1 and 8.1, the nominal stress is limited 90 % of the current nominal design stress.

A safety equivalent to the vessels of other testing groups has been obtained by:

1. Reducing the manufacturing tolerances (peaking and excess weld of the longitudinal weld)

2. Increasing the test pressure to reduce residual stresses, obtain crack blunting and correction of shape imperfections.

The beneficial effect of a higher test pressure was shown by the Research and Development programme HYDFAT, sponsored by DG Research of the European Commission. For more information, see EPERC Bulletin Nr 4 [9].

6A Bibliography

[1] Directive 97/23/EC of the European Parliament and of the Council of 29 May 1997 on the approximation of the laws of the Member States concerning pressure equipment, Official Journal of the European Communities, No L 181/1, 9 July 1997.

[2] EN 13445, Unfired pressure vessels, Issue 1(2002-05), Part 1: General, Part 2: Materials, Part 3: Design, Part 4: Manufacture, Part 5 Inspection and Testing, Part 6: Requirements for design and fabrication of pressure vessels and vessel parts constructed of spheroidal cast iron.

[3] AD-Merkblatt, 2000 edition, English translation, Carl Heymans Verlag KG, D-50939 Köln.

[4] PD 5500:2000, Specification for unfired fusion welded pressure vessels, British Standards Institution, London, UK.

[5] CODAP 2000, French code for the construction of pressure vessels, SNCT, F-92400 Courbevoie

[6] CR ISO/TR 15608:1999(E), Welding – Guidelines for a metallic grouping system for fabrication purposes.

[7] EPERC Bulletin Nr 2, October 1999, European Approach to Pressure Equipment Inspection, Ed. Jean-Bernard Veyret, Guy Baylac, European Commission JRC, NL-1755 ZG Petten, S.P.I. 192.

[8] ASME Code, Section VIII, Division 2, Alternative Rules, 2000.

[9] EPERC Bulletin Nr 4, June 2001, European R&D on fatigue strength and hydrotest for pressure equipment, European Commission JRC, NL-1755 ZG Petten, S.P.P.01.42.

7 Shells under internal pressure

7A Introduction

Rules for cylinders, spheres and cones as given in EN 13445-3 clauses 7.4.2, 7.4.3, and 7.6.4 require no comment, the equations being standard and familiar. It is only when one comes to the intersection of a sphere and a cone (on the same axis) in 7.6.5 to 7.6.9 that comment is needed, as described below under 7B.

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Dished ends are used in most pressure vessels so their economic design is of importance. They are dealt with in sections 7.5 and 7.7 of part 3 of EN 13445-3, and explained under 7C.

7B Cones and conical ends

There are three situations to consider: the large end of the cone, with and without a knuckle, and the small end. In each case it is necessary to consider gross deformation and shakedown criteria, for which limit analysis and elastic analysis with a stress limit of 2f are used respectively. Not only must the rules give a minimum thickness but they must also tell the user how far from the junction the thickness has to extend.

Rules for determining the minimum thickness are, it will be seen, all based on limit analysis based procedures supplied by the German delegation. They are taken from the East German pressure vessel code [1]

7B-1 The large end of a cone without a knuckle

This has traditionally been dealt with by limiting the stress at the junction to 3f. However it is found that the limit pressure is close to but slightly less than the 3f pressure for all D/e ratios and angles, the difference becoming greater at greater angles. The method based on stress analysis has therefore been replaced by a formula provided by Germany and based on limit analysis.

The formula originally supplied was

$$e = \frac{pD\beta}{2f} \text{ where } \beta = 0{,}4\sqrt{\frac{D}{e}}\frac{\tan(\alpha)}{1+1/\sqrt{\cos(\alpha)}} - 0{,}25$$

and α is the cone semi-angle. The formula is an approximation to a fuller and far more complicated solution.

It was accepted that deformation by the internal pressure loading results in a more favourable shape and that there has been successful experience with rules based on the 3f criterion. It was therefore decided to replace the above equation for β by:

$$\beta = 1/3\sqrt{\frac{D}{e}}\frac{\tan(\alpha)}{1+1/\cos(\alpha)} - 0{,}15$$

which brings us very close to the 3f method. The upper limit on angle is set at 60°, though it has been established that the formula is safe up to 90°.

While a formula based on limit analysis for intersections in which the thicknesses of cone and cylinder are different exists, it has not been used. Stresses hardly come down at all as the thickness of one member is increased, so the stress limit soon becomes controlling and the stress would have to be compared with 3f.

Once it is realised that the limit load controls design, fresh thought has to be given to the distance over which the increased thickness must be maintained.

Looking first at the minimum distance along the cylinder, conventional elastic stress analysis shows no increase in stress even when the junction thickness is maintained only for a distance well below $\sqrt{D \cdot e}$ However work carried out by the author using a thin shell limit analysis computer code (named LASH - details in 7B-5) showed unacceptable loss of limit load - about 15% - with junction thickness extending for a distance $\sqrt{D \cdot e}$ but only a minor loss of < 5% for a distance of $1{,}4\sqrt{D \cdot e}$, which has therefore been adopted as the minimum for the new standard.

It is also necessary to consider the acceptable distance between two major features, such as two cone/cylinder intersections. Calculations of typical examples with LASH showed that this distance could be $2\sqrt{D \cdot e}$ and not the $2{,}8\sqrt{D \cdot e}$ that might be expected. One reason for this observation is the importance in these problems of shell bending, so that any thinning of the shell greatly reduces the available moment in the limit analysis at the point where thickness changes.

7B-2 The large end with a knuckle

The method is again based on limit analysis and has been modified to tie in with the formula without a knuckle. It

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$$e = \frac{pD\beta}{2f\gamma}$$

where β is given above

$$\gamma = 1 + \frac{\rho}{1{,}2\left(1 + 0{,}2/\rho\right)},$$

$$\rho = \frac{0{,}028r}{\sqrt{D \cdot e}} \times \frac{\alpha}{1 + 1/\cos(\alpha)}$$ and r is the radius of the knuckle.

Use of LASH showed that the distance over which the thickness calculated above had to be maintained could be measured, not from the tangent line with the knuckle but from the junction, as defined in the standard.

7B-3 The small end of the cone

Turning now to the small end of the cone, we find a situation where the limit load controls at all times and stress concentrations are moderate. It is therefore possible to provide a method allowing for different thicknesses of shell and cone. It is also found that the limit load improves only little with the introduction of a knuckle, so no separate method need be given for it.

The method now is the CEN draft again comes from East Germany. As before, it is an acceptable approximation to the full analysis, and better in that respect than other sets of formulae such as area replacement methods.

7B-4 Future work

Recommendations were submitted to CEN TC 54 WG"C" for removing the angle limitation, allowing the rules for angles up to 90°, but there is little incentive to make the change to the text of EN13445-3. Of course as the cone becomes very flat the cone rules become very conservative due to the cos(α) factor, but it is then possible to turn to the flat plate rules as given in clause 10.

Confirming the various minimum distance rules would make a nice masters degree project.

7B-5 Thin Shell Limit Analysis Program LASH

The computer program LASH was written to carry out a limit analysis on an axisymmetric thin shell. The thin shell itself is modelled by a sandwich of four shells, each representing 1/4 of the total thickness. The four shells individually can only take membrane loads but can combine to give the full plate the ability to accept a bending moment. (Shear is also of course allowed). The normal Tresca yield criterion is applied to each of the layers. The shell is split up into 'finite elements' about $0{,}25\sqrt{(re)}$ long.

The equations relating the loads in the elements to bending moments and direct loads in the whole plate and the equations balancing the various loads are all linear. In lower bound limit analysis one is trying to find a stress distribution that maximises the applied load without infringing the yield condition. We therefore have a linear programming problem, which can be solved by computer, though with much more effort than required for an elastic analysis.

How good a model does the 4 layer sandwich provide? Considering direct load and bending in one direction only, the sandwich approximation to the shell replaces the well known yield parabola by two straight lines. The parabola referred to is:

$$\left(N/N_0\right)^2 + M/M_0 = 1$$

There is a maximum departure from the true curve of 5%, always on the safe side. The 4 layer sandwich also gives a good approximation to the interaction between direct load in one direction and bending in the other, which is important in this class of problem.

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7C Dished ends

7C-1 Introduction

Dished ends are used in most pressure vessels so their economic design is of importance. It is found that two modes of failure have to be considered in drawing up design rules – excessive axisymmetric plastic deformation and, for thin ends, buckling of the knuckle under the compressive circumferential stresses found there. Over the years the problem has attracted the attention of many workers and a number of different design rules are to be found in pressure vessel codes. There is a great fund of experience of the use of dished ends.

However it is only very recently that the power of the computer has enabled us to produce consistent design rules. The rules provided in the UPVS to protect against excessive deformation are based on one report, Welding Research Council Bulletin 364 [2] by Kalnins and Updike (known here as K&U) modified by the CEN committee to take account of European experience.

A number of papers have been produced on the phenomenon of buckling in dished ends but the latest one to suggest design rules is by Galletly (1986) [3] and these rules have been adopted as they stand.

The question of a nozzle in or intruding into the knuckle region is dealt with in this part of the standard since the solution is to increase thickness over the whole of the knuckle region.

7C-2 Review of EN 13445-3 rules for dished ends

Dished ends are dealt with in sections 7.5 and 7.7 of part 3 of EN 13445-3.

The limitations in 7.5.3.1 follow from the data used, except that the committee considered a limitation by Galletly of his method to ambient temperatures only could be ignored. Galletly argued from the fact that the experimental work was all at ambient, but the committee considered the extensive computer work justified a different view.

Equation (7.5-1) and (7.5-6) are concerned with the membrane stress in the spherical cap.

Equations (7.5-2) and (7.5-7), together with (7.5-9) to (7.5-17) are the main part of the method and are intended to prevent excessive deformation.

The K&U results appear in the method as polynomials that closely match the data in their report. These polynomials are the expressions in brackets in equations (7.5-13), (7.5-15) and (7.5-17).

The modifications made by the CEN committee appear in the standard as factor N in equations (7.5-13) and (7.5-15), the 0,95 and 0,5 in equation (7.5-19) and the factor $\{(0,75\,R + 0,2\,D)$ (replacing R) in equations (7.5-2) and (7.5-7).

Equations (7.5-3) and (7.5-8) are based on Galletly's method.

The rules in 7.5.4 for ellipsoidal ends provide for a nominally equivalent torisphere. The simple formulae therein are approximations to the two well-known equivalence formulae, namely

$$(0,5/K - 0,08) \cong \left\{K^2 + 1 - \left(K-1\right)\sqrt{K^2+1}\right\}/\left(4K^2\right)$$

and $$(0,44\,K + 0,02) \cong \left\{K^2 + 1 + \left(K-1\right)\sqrt{K^2+1}\right\}/\left(4K\right)$$

Moving now to section 7.7 on nozzles in the knuckle region, the rules are based on AD2000-Merkblatt B-3, the main difference being that formulae have been found (equations (7.7-3) to (7.7-10) that are close to the crude graphical method in the ADM. These rules are limited to the two standard ends since there is no way of interpolating when two shape variables are involved.

A note on the standard German ends – the method in section 7.5 is a complicated calculation and to supplement the graphs the following tabulation for factor $C = \frac{ef}{pR}$ is provided.

e/R	0,001	0,002	0,003	0,004	0,006	0,008	0,01	0,012
Kloepper	1,082	1,044	0,990	0,941	0,862	0,802	0,758	0,726
Korbogen	0,809	0,788	0,760	0,734	0,692	0,659	0,634	0,615

e/R	0,014	0,016	0,02	0,024	0,028	0,032	0,036	0,04

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Kloepper	0,703	0,685	0,654	0,624	0,595	0,568	0,545	0,524
Korbogen	0,603	0,593	0,578	0,563	0,550	0,537	0,527	0,518

7C-3 Basis for design rules

7C-3a Excessive plastic deformation

Although there are a number of available methods for dished end design in current use and many papers on the subject, the method presented in this standard is based on the work of Kalnins and Updike. The first task here is to state what they did, then how it was adapted for CEN use. It is not the purpose here to provide a general review of the history of dished end design or to show that other failure mechanisms are catered for; that is done already by K&U.

Failure of dished ends is difficult to define. Those made of typical ductile pressure vessel materials will deform into stronger and stronger shapes, eventually taking on a nearly spherical form, before bursting. The shape improvement on hydro-test also goes to reduce stresses in subsequent operation, of importance when considering shakedown and fatigue.

The failure criterion chosen by K&U was the twice elastic slope criterion (as prescribed by ASME) applied to the deformation of the centre of the head.

The theoretical model included shape change. There was no work hardening. The ratio of Young's modulus to yield was set at 1000. The von Mises yield criterion was used.

Geometries studied were for just three values of *r/D* (inside knuckle radius/inside diameter) of 0,06, 0,1 and 0,2. The value of *e/R* (thickness/inside crown radius) covered the range from 0,002 to 0,04. There was no straight flange of thickness equal to that of the end – the thickness of the whole cylinder was set in the calculations at the minimum required by code rules.

Results were presented in the form $\frac{p}{f} = func\left(\frac{e}{R}, \frac{r}{D}\right)$. Note that the number of dimensionless variables is one less than expected. It was found that results varied little with *R/D* and so in the interests of ease of use this variable was omitted.

The arguments used to justify the modifications made by committee are in the next section.

Interpolation has to be used in a design procedure between the values of *r/D* considered by K&U. A simple but conservative interpolation rule is needed. The one chosen is a linear interpolation for β since it is more conservative than interpolation for 1/β.

In the absence of other data, design for *e/R*>0,04 is dealt with by using the value at 0,04.

7C-3b CEN modifications to K&U

Since the Kloepper form is used so much in Europe and is typical of shapes used elsewhere, comparison is concentrated on it. It also happens to be close to one of the shapes considered by K&U, who looked at *r/D* values of 0,06, 0,1 and 0,2. There is a slight difference because their *D* was the internal diameter whereas the Kloepper *r/D* is based on the external diameter.

The attached graph shows how the ratio *C = (e/R)/(p/f)* varies according to the above data. It is seen that ASME/CODAP is closest to the K&U data.

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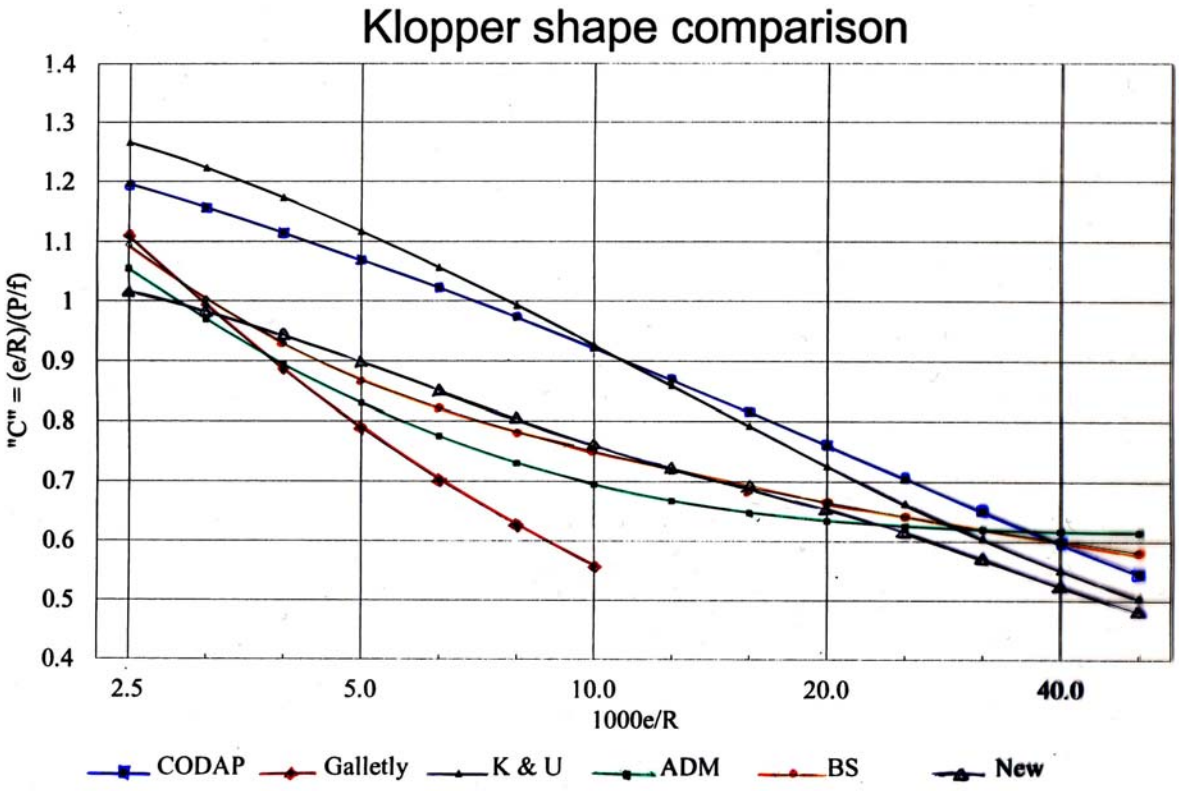


Figure 7C-1 Klopper shape comparison

PD 5500 and ADM are relatively close to each other and, in the middle of the range, well below the other lines.

Faced with the difference between Kalnins and Updike's results and the ADM rules in particular, of which there has been such wide experience, it is first necessary to consider how German ends can be satisfactory when theory suggests they are not. The difference is greatest at e/R = 0,007, the values of C being 1,028 and 0.754 respectively, a ratio of 1,36

It was considered that the two may be reconciled as follows:

The ADM rules are only tested to a pressure 1,3 times design, whereas K&U were aiming for the conventional 1,5 safety factor.

K&U's failure criterion was based on deflection at the centre of the end using the twice elastic slope criterion. However dished ends continue to behave in an acceptable manner well past that point, to say 3 or 4 times elastic slope. This is worth another 17% approximately, based on the observed shape of load deflection curves.

The comparison is about the ability of a certain thickness to carry pressure, the pressure allowed according to K&U being lower. Unlike previous workers in this field K&U did not work with a cylinder thickness equal to the thickness of the end, rather they used the cylinder thickness corresponding to the p/f for the end. Thus the actual geometry on which the ADM experience is based would have had a thicker cylinder. This is worth another 4%.

K&U's figure is for all R/D, an approximation. Although they found that C is nearly constant as R/D varies, it is greater at R/D = 0,8 than at 1,0. It is estimated that this accounts for another 5% of the difference. (R/D is 1,0 for the Kloepper shape.)

Design is based on minimum thickness after forming. Actual heads are of non-uniform thickness, worth another 5% say.

Most heads will also have the benefit of a corrosion allowance, reduced design stresses for elevated temperatures and greater yield stress than that specified by up to 50%.

Multiplying the factors in 1) to 5) above together gives an overall factor of 1,55, much greater than the ratio of 1,36 that had to be explained. The academic calculation and experience can be reconciled. The problem is to go from there to a design method.

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Note that we do not wish to use the German rules directly. They cover only certain geometries and we wish to be able to permit a higher pressure than 1,3*p* at pressure test, namely 1,43*p*.

It was decided that it is reasonable to adjust the K&U data by the 17% in 2) above together with an additional term to take credit for the variation of '*C*' with *R/D* in 4) above. It was also agreed that the nominal factor of safety could be reduced by 2,5% giving an overall adjustment factor of $\frac{1,17}{0,975(0,75+0,2\times D/R)} = \frac{1,2}{0,75+0,2\times D/R}$.

In the Kloepper case this is worth 1,2/0,95 = 1,26. The Kloepper experience would justify a slightly smaller factor, namely 1,36*1,3/1,43=1,24.

This credit is extended to *r/D* = 0,06 and to *e/R*<0,007. Above *e/R* of 0,007 the 1,2 adjustment tapers away to unity because shape change is less for such geometries and '*C*' is getting closer to the membrane limit for the crown.. For *r/D* = 0,2, the same comment about the crown applies but it was still felt possible to make a small adjustment to the K&U data as well as retaining the *D/R* factor.

7C-3c Instability

The paper by Galletly concludes with a suggested design equation, namely

$$\frac{P}{\sigma_y} = \frac{80\left(r/D\right)^{0,825}\right)}{\left(D/e\right)^{1.5}\left(R/D\right)^{1,15}} .$$

This was modified slightly for use in EN 13445-3 to give equations (7.5-3) and (7.5-8).

The basis of the equation is a curve fit to the results of many large deflection plastic buckling calculations, with a knock-down factor based on the lower bound of a limited number of test results. Bearing in mind the difficulty of modelling the actual shape, thickness profile and work hardening found on actual fabrications, theory and practice were in reasonably close accord. The paper confirms that the thickness to be used in the formula is the minimum at the knuckle after forming and the proof stress is that in the plate as received, both in conformance with definitions of these quantities in the standard.

The graphs referred to above also show the Galletly design equation on the same basis as the other criteria. The comparison is a fair one for carbon steel since the same value of yield is used as in the main calculation but the comparison becomes more complicated for stainless steel. Ends that are not cold spun require the 0,2% proof stress, somewhat lower than the 1% proof stress, so that stability becomes important at slightly larger e/R. On the other hand with cold spun ends the design stress is multiplied by 1,6, in which case stability no longer controls. There may be doubt about the extrapolation of Kalnins and Updike's results to e/R< 0,002, but on the other hand the increase in yield observed for cold spun ends will more than compensate for this uncertainty.

7C-4 Nozzle in Knuckle rules

As mentioned above, the ADM contain rules for increased thickness if there is a nozzle intruding into the knuckle region. It is a crude rule - a thickness adjustment that takes no account of the location of the branch or its wall thickness. It was reviewed by the CEN working group and compared with available stress analysis data and experience. It was concluded that the ADM adjustment should continue to be used in the new code. The curves in the ADM are incomplete and leave doubts about interpolation but equations have been fitted to them.

It is frequently said that some other codes prohibit the use of this arrangement. That is not strictly true; they merely fail to provide design rules for them. Those who carry out a Design by Analysis or use strain gauges are surprised at how low the stresses turn out to be.

Unfortunately no method has been found for applying the curves to other than the standard German geometries.

Recent work described in The Design by Analysis Manual [4] found that there is an area of large deformation if the nozzle gets close to the tangent line with the cylinder, so the arbitrary penalty in the last paragraph of 7.7.2 was imposed.

7C-5 Future work

The following items are suggested, in no particular order

K & U themselves clearly recognised the conservative and arbitrary nature of their failure criterion. Since with a dished end “gross deformation” is not an intimation of failure by any other mode and merely makes the end less like the shape on the drawing, there is a need to reconsider in this case what end deflection can really be accepted.

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There is a large gap between two of the values of *r/D* selected by K & U. Their work should be repeated at *r/D*=0,14

The rules limit *R/D* to be less than 1,0. This is traditional and therefore K & U did not consider working with values greater than unity. Industry tends to press to be able to continue doing what it has been doing rather than seek improvements. It seems probable that the rules can safely be applied as written to R/D>1, without limit, but this has yet to be properly demonstrated. This could be of interest to manufacturers of ends for low pressures.

There is a need to look at thicker ends (*e/R*>0,04)

The nozzle in knuckle rules consider only pressure. The absence of information on stresses due to pipe loads is a severe restriction on their use in much of industry

Galletly noted the lack of a sufficient number of tests and felt that with more there could be scope for less conservative rules. In particular it might be possible to consider the effect of work hardening on the resistance of spun stainless steel dished ends to axisymmetric failure.

7D Bibliography

[1]. Fachbereitstandard, Behalter und Apparate, Festigskeitsberechnung, Kegelschalen, TGL32903/06. Standardsversand, Leipzig, April 1989

[2]. KALNINS A and UPDIKE D P; New design curves for torispherical heads, Welding Research Council Bulletin 364, June 1991

[3]. GALLETLY G D; Design equations for preventing buckling in fabricated torispherical shells subjected to internal pressure. Proc I Mech E, London 1986.

[4]. Design by Analysis Manual. Pressure Equipment Directive Joint Research Centre. To be found at http://ped.eurodyn.com/jrc/jrc_design.html.

8 Shells under external pressure

8A Introduction

Rules for external pressure design have to cover a variety of components and modes of failure. They are therefore complicated. The rules in clause 8 of EN 13445-3 are based almost entirely on the rules in British Standards Institution (BSI) document PD 5500 [1], itself a continuation of the previous BS 5500. The rules have been in BS 5500, the British Standard for unfired pressure vessels, since about 1973.

The British rules were rewritten for EN 13445-3 and some modifications made, mainly in presentation. The safety factor is now shown explicitly in the formulae and the distinction between light and heavy stiffeners is clearer. The values of elastic limit (yield stress) have been changed slightly.

A number of documents have been produced to justify or as an aid to understanding the rules in PD5500. The author of this paper has drawn heavily on PD 6550: Part 3: 1989, an explanatory supplement to BS 5500 on external pressure, published by BSI [2]. Other papers are in the list of references.

Although the rules are said to be rules for design, the user will in all cases have to choose a thickness or in the case of stiffeners a complete set of dimensions. Application of the rules will then tell the user whether his proposal is adequate for the pressure.

The rules are discussed in the order in which they appear in EN 13445-3 clause 8.

8B General comments

8.2.2 and 8.2.3. A typical ‘heavy stiffener’ would be a girth flange. If it is designated as a heavy stiffener then the cylinder is split into two and the two shorter cylinders are calculated independently. However the girth flange may not be stiff enough to meet the requirements of 8.5.3.7. It would then be permissible to redesignate it a ‘light stiffener’ and recalculate, to see if a more favourable result is obtained. A worked example is appended to assist in the understanding of this point.

8.4.2 and 8.4.3 The important strength parameter in stability studies is the elastic limit. In the case of low alloy steels with a distinct yield point, elastic limit and yield are the same. In most places where design against external pressure is discussed, as in the rest of this paper, the word yield is used instead of elastic limit

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However in the case of austenitic stainless steels and others materials with stress/strain curves showing work hardening and great plasticity, there is no such thing as a yield point. The elastic limit for these materials is not generally available from a standard tensile test - indeed it is difficult to define and measure precisely - and so design is based on the nearest readily available parameter, the 0,2% proof stress. Study of some typical stress/strain curves for stainless steel gave the factor 1,25 in equations (8.4.3-1) and (8.4.3-2).

The possibility of increasing the nominal elastic limit while at the same time taking account of the consequential reduction in Young's modulus was considered, but calculations showed there was no benefit from so doing.

The tolerance on radius (deviation from theoretical shape) is an important feature of the method. The experimental results that went into deriving figure 8.5-5 were limited to those for cylinders that met the 0,5% tolerance, so it is not possible to apply the rules to cylinders outside the specified tolerance without further consideration. In the case of stiffener design the tolerance appears explicitly in the equations, for example equation (8.5.3-46).

Measurements of diameter are not sufficient to demonstrate circularity - one only has to consider a three lobed shape to see this. They may however be considered as evidence of good workmanship.

8.5.1.2 and 8.5.1.3 The simple safe rule in 8.5.1.2 can be applied at the design stage whereas 8.5.1.3 and Annex F are for use after fabrication is complete and the actual out of tolerance shape is known.

8.5.2.1 The simple rules for L when the vessel end is a cone, a sphere or a dished end, namely the 0.4h term and the effect of the cone angle, are based on computer instability studies of many geometries.

Equation (8.5.2-6) is the well known Mises equation. Other versions of it appear but this is the most accurate. It is possible to replace the calculation of ε by a simple formula not containing n_{cyl}, but this lacks accuracy at low values of n_{cyl}.

Curve 1) of Figure 8.5-5 represents the lower bound of 700 well documented tests available at the time of the study. It gives the pressure above which failure might occur as a function of the Mises instability pressure and the pressure to circumferential yield.

The knock down factor at the 'slender' end (low p_m/p_y) is 2,0. There was much scatter in that part of the plot of test results. Those on the line were checked and proved to be good experimental results.

8.5.3.1 Attention is drawn to the note in 8.5.3.1 and to the comments above on paragraphs 8.2.2 and 8.2.3.

8.5.3.4 It was found that the particular pressure at yield calculated in equation (8.5.3-15) (the maximum circumferential stress in the cylinder) gave the best correlation of experimental results.

8.5.3.5 Equation (8.5.3-23) is the result of a limit analysis of the shell, considering it as a flat plate fully supported at the ends, and taking account of the weakening effect of the circumferential membrane stress.. For guidance on the use of heating/cooling channels as shell stiffeners, reference may be made to PD 5500 clause 3.11.4

8.5.3.6.2 The first term in equation (8.5.3-24) represents the contribution of the shell and the second term the contribution of the stiffener. Ignoring the first term and setting n=2 leads to equation (8.5.3-51) for a heavy stiffener.

Equation (8.5.3-31) is only a preliminary filter before going on to 8.5.3.6.4.

Factor S_f adds an additional safety factor for stiffeners to cover any interaction between inter-stiffener and overall collapse. It is made larger still when residual stresses may be high.

8.5.3.6.3 The full precise method for calculating Le, the length of shell acting with the stiffener is extremely long and complicated but the method provided here is a close approximation..

8.5.3.6.4 Here the stress in the stiffening ring is calculated, the first term in equation (8.5.3-46) being the direct stress and the second term the bending stress due to out of round, magnified by the effect of external pressure. This stress is limited to the nominal elastic limit. Note that the resulting allowable external pressure is a lower bound, not a prediction of failure, so no experimental knock down factor is called for.

8.5.3.7 The method for heavy stiffeners is based on that for light stiffeners with the omission of the effect of the shell (first term in equation (8.5.3-24) and setting n=2).

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8.5.3.8 Equation (8.5.3-60) is a simple but conservative result derived from energy considerations. The stiffener is assumed to be hinged at the junction with the shell. Note that $\sigma_{es}\left(P/P_{ys}\right)$ gives the direct stress in the stiffener. It is necessary to be more precise with flat bar stiffeners and most of the calculation has been done for us and presented in table 8.5-5.

8.6 The necessary modifications to the methods for cylinders are presented here. Rules for heavy stiffeners have been omitted in error, though the capable user should be able to work them out for himself from what he has been told already.

8.6.5.1 and 8.6.5.2 The rules provided here for the large and small ends of the cone are based on extensive computer stability analysis, which shows them to be safe. .

8.7.1 The method for spherical shells is similar to that for cylinders - the two theoretical formulas are compared with experimental results and a lower bound curve to the experimental results is drawn, as shown in figure 8.5-5. The elastic instability pressure p_m is much greater than the equivalent pressure for a cylinder of the same thickness and diameter, due to the benefit of the double curvature, but the knockdown factor is much greater.

8.7.2 Conforming to the design shape is again important, though in the case of spheres it is only the local shape that matters. The requirement that the sphere should be spherical to 1% on radius is only a quality control measure. The requirement for a maximum on the local radius of curvature is vital.

8.8 With vessel ends we have to determine the maximum radius of curvature and then apply the rules for a sphere. For an ellipsoid the maximum radius of curvature occurs at the centre.

The general rule that internal pressure rules shall be applied without modification using the external deign pressure, does not apply to torispherical ends since the method in clause 7 claims some benefit from the improvement in shape that occurs under pressure, and under external pressure the shape should get worse. That is why we have the rule N=1 introduced.

8C Future work

Three items that require no further technical development

1. Add heavy stiffeners to the rules for cones.

2. Add something about use of heating/cooling coils as stiffeners along the lines of PD 5500 clause 3.11.4

3. Add something about the fact that a short cylindrical flange forming part of a dished end and thinner than the rest of the cylinder, is permissible - see PD 5500 3.6.4 final paragraph.

Three items that may be possible with further work.

3. Rules for cylinders of variable thickness - for example a tall column.

4. Less onerous rules for measurement of tolerances in which measurements of departure from design shape are more related to predicted deformed shape.

5. More precise design for inter-stiffener collapse of cylinders, based on separation of the two failure mechanisms associated with circumferential and longitudinal loads respectively. Longitudinal compressive loads on cylinders suffer in the same way as spheres under external pressure from a large knock-down factor. Maybe a study of test results will show that the circumferential load acting alone has a low knockdown factor, resulting in more economic design for longer cylinders.

8D Bibliography

[1] PD 5500:2003 Specification for unfired fusion welded pressure vessels, British Standards Institution

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9 Openings in shells

9A Introduction

9A-1 The first proposal of CEN/TC54/WG'C' was to use 'Area Replacement Method' that was present in ASME VIII Div. 1 and in a few European national Codes (based on method of replacement of cross sectional missing area of openings with cross sectional areas of material reinforcement of shell and nozzles), but during standardization works it was decided to adopt the 'Pressure-Area Method' because this one was well known and widely used in continental Europe.

The Pressure-area method is based on ensuring that the reactive force provided by the material is greater than, or equal to, the load from the pressure. The former is the sum of the product of the average membrane stress in each component and its stress loaded cross-sectional area (see Figure 9A-1). The latter is the sum of the product of the pressure and the pressure loaded cross-sectional areas. If the reinforcement is insufficient, it shall be increased and the calculation repeated.

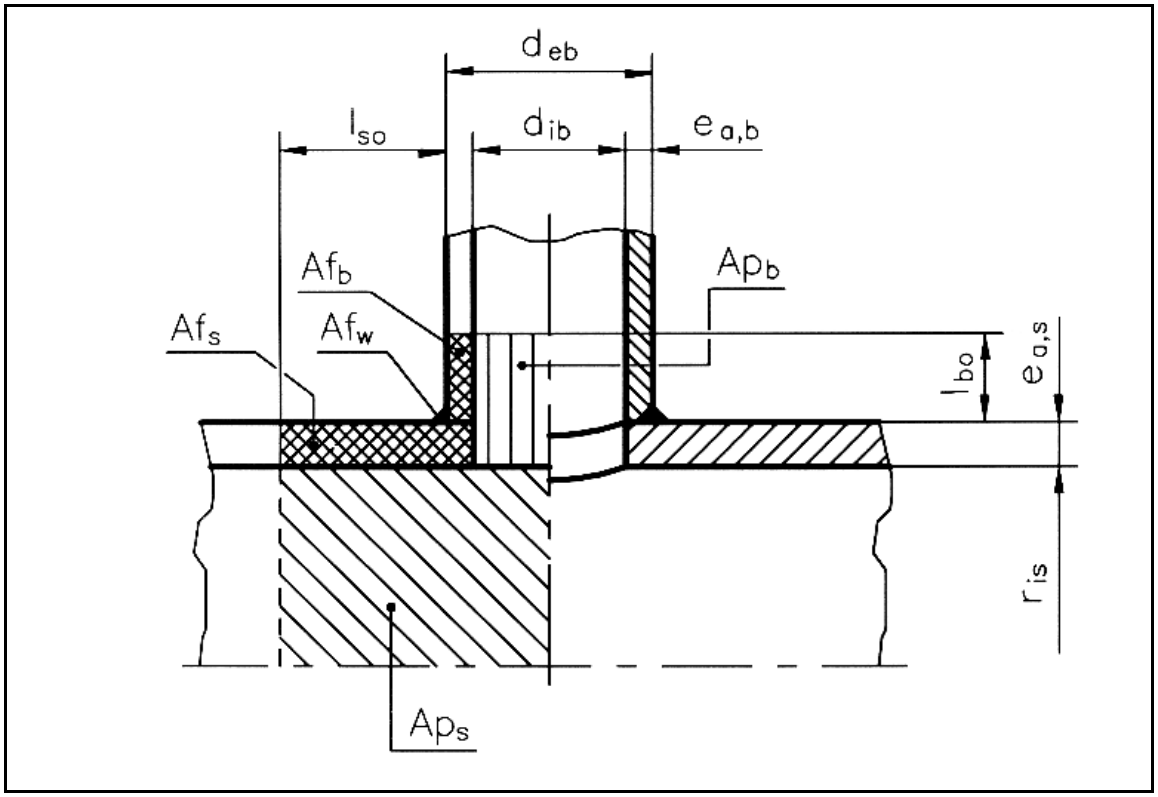


Figure 9A-1 Cylindrical shell with isolated opening and set-on nozzle (identical to Figure 9.4-7 in EN 13445)

9A-2 It was decided to start with Code ISO 5730 (International Code for shell boilers) based on balance between cross sectional pressure areas and stress compensation of cross sectional areas of shells and nozzles, but it seemed to be good only for the working conditions and limits of shell boilers. Therefore it was decided to increase and expand the rules also for all possible stress situations of unfired pressure vessels, and the rules of

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CODAP have been taken to complete and expand the code of ISO 5730 (with some other integrations by Belgian rules for limitations of calculation thickness of nozzles for too large openings).

After this decision, the Working Group spent a lot of time just to put into the code more possible stress situations and stress limitations (more general and comprehensive formulas for each geometrical possible situation).

9A-3 Actually in Clause 9 there are no restrictions on size of circular openings and some restrictions on shape of elliptical openings (coming from inclined circular nozzles), unless reinforcing pads are used.

Present rules include allowing openings close to a discontinuity and also small adjacent openings.

This makes EN 13445 Part 3 Clause 9 more comprehensive than ASME VIII DIV. 1 "Area Replacement " approach. Calculation of stress is not included.

9B Contents of Clause 9

9B-1 General

The design method specified in this clause is applicable to circular, elliptical or obround openings in dished ends or cylindrical, conical or spherical shells under internal or external pressure.

This clause is applicable to openings, nozzles and reinforcing plates in dished ends which are completely located inside the central area limited by a radius equal to $0{,}4D_e$. For different locations (i.e. nozzles in knuckle regions) the relevant design rules are given in Clause 7.

Design for non-pressure loads is covered by Clause 16.

A shell containing an opening shall be adequately reinforced in the area adjacent to the opening. This is to compensate for the reduction of the pressure bearing section. The reinforcement shall be obtained by one of the following methods:

- increasing the wall thickness of the shell above that required for an unpierced shell,
- using a reinforcing plate,
- using a reinforcing ring,
- increasing the wall thickness of the nozzle above that required for the membrane pressure stress,
- using a combination of the above.

9B-2 General equation and its derivates

General equation for the reinforcement of an isolated opening is given (see Figures 9A-1 and 9B-1) by:

$$(Af_s + Af_w)\,(f_s - 0{,}5P) + Af_p\,(f_{op} - 0{,}5P) + Af_b\,(f_{ob} - 0{,}5P) \geq P\,(Ap_s + Ap_b + 0{,}5\,Ap_\varphi)$$

where, in general:

Ap Af	Pressure loaded area. Stress loaded cross-sectional area effective as reinforcement.	mm^2 mm^2
fs	Nominal design stress of shell material.	MPa
P	Internal pressure of shell.	MPa

and P_{max} shall be obtained as follows:

$$P_{max} = \frac{(Af_s + Af_w)\cdot f_s + Af_b \cdot f_{ob} + Af_p \cdot f_{op}}{(Ap_s + Ap_b + 0{,}5Ap_\varphi) + 0{,}5\,(Af_s + Af_w + Af_b + Af_p)}$$

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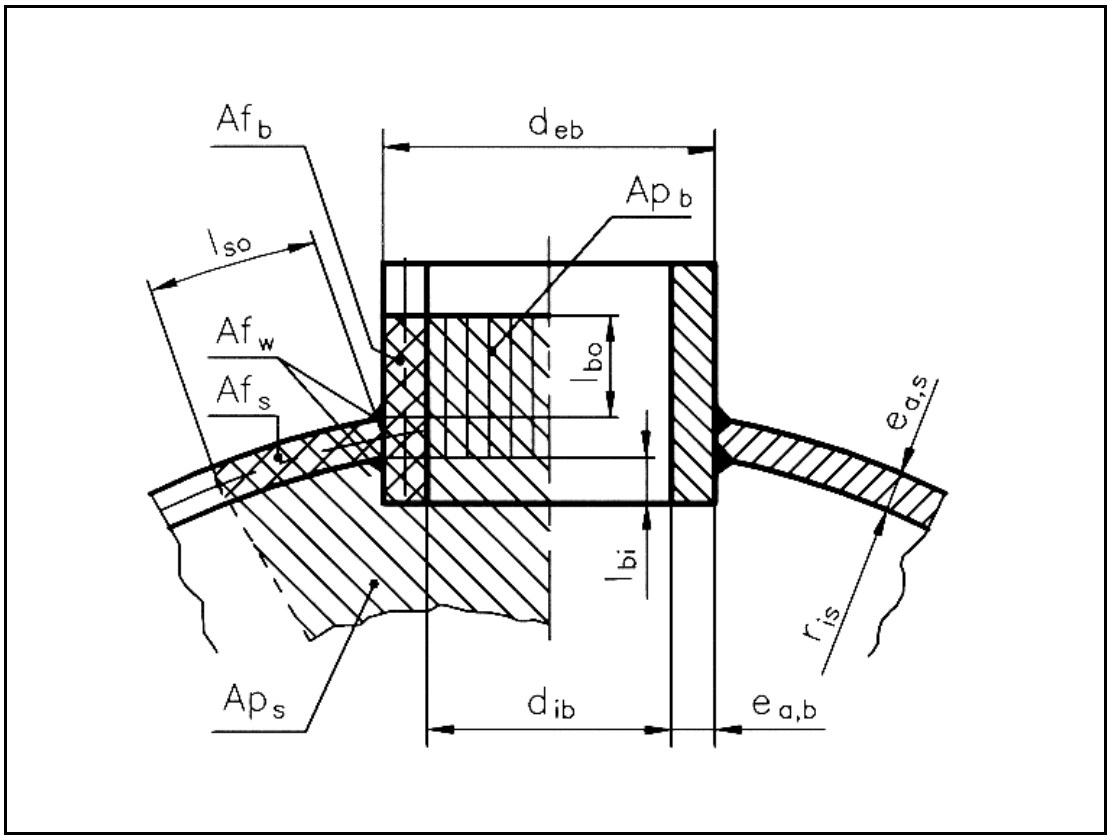


Figure 9B-1 Spherical shell or dished end with isolated opening and set-in nozzle (identical to Figure 9.4-8 in EN 13445)

9B-3 Single openings

In Clause 9 the general equation is adapted to each of the following configurations:

- Shells with openings without nozzle or reinforcing ring (with or without reinforcing pads) on cylindrical, conical and spherical shell, longitudinal and transverse cross-section.
- Shells with openings without nozzle, reinforced by reinforcing rings, on all same cases.
- Nozzles normal to the shell, with or without reinforcing pads, on all same cases.
- Nozzles oblique to the shell, with or without reinforcing pads, on all same cases.

9B- 4 Multiple openings

In Clause 9 the general equation is adapted also to each of the following configurations:

- Ligament check of adjacent openings.
- Openings in cylindrical and conical shells.
- Openings in spherical shells and dished ends.
- Overall check of adjacent openings.

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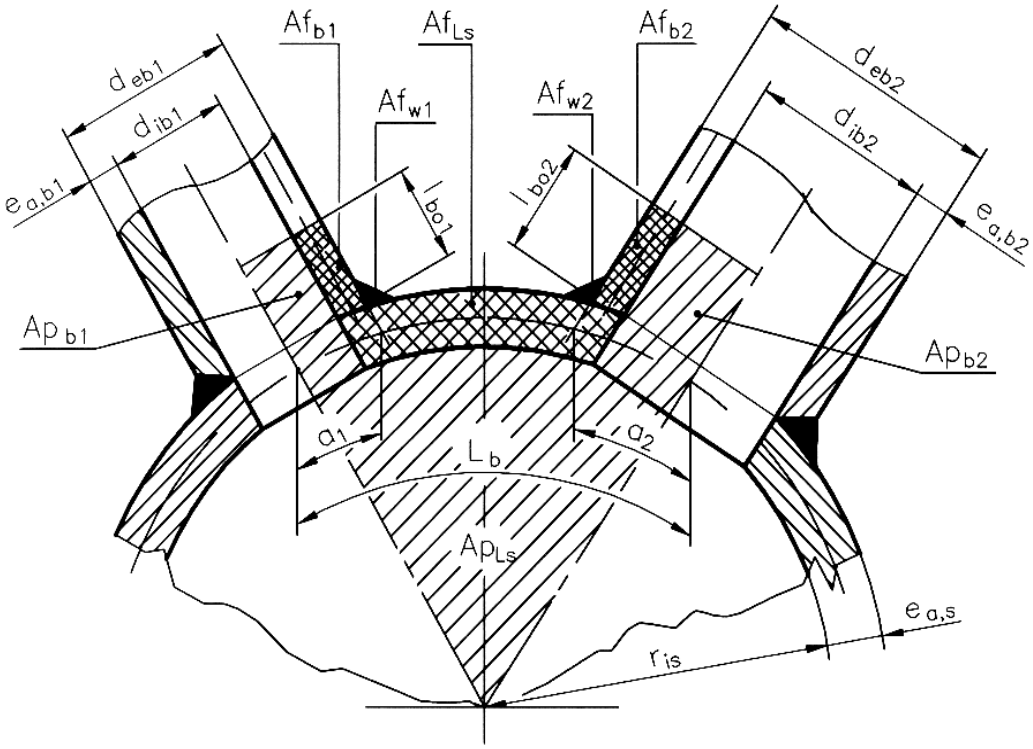


Figure 9B-2 Ligament check of adjacent nozzles normal to a spherical shell (identical to Figure 9.6-3 in EN 13445)

9B-5 Openings close to a shell discontinuity

In Clause 9 there are rules giving minimum distances of openings from discontinuities.

$$w \geq w_{\min} = \max\ (0{,}2\sqrt{(2r_{is} + e_{c,s}) \cdot e_{c,s}}\,; 3e_{a,s})$$

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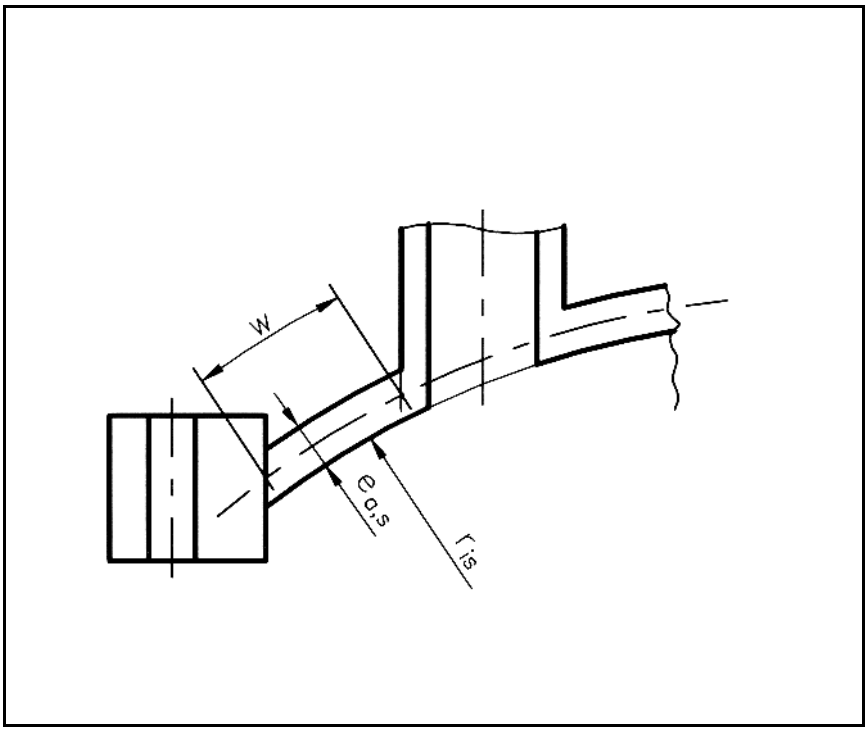


Figure 9B-3 Opening in a domed and bolted end close to the junction with the flange (identical to Figure 9.7-11 in EN 13445)

9B-6 Future work

There are other two items that need to be implemented:

- Openings with non circular nozzles,
- Openings in non circular walls.

10 Flat ends

The rules for flat ends are based partly on the rules of the French Pressure Vessel code (CODAP, in their 1995 ed. version) and partly on the rules of the Italian Pressure Vessel Code ISPESL/VSR. They deal with ends of both circular and non circular shape, either welded or bolted. The main purpose of the equations contained in the rules is to assure that a failure due to gross plastic deformation can never happen. For circular welded flat ends, this is achieved by setting the allowable pressure of the end to its limit pressure divided by 1,5.

The limit pressure calculations on which the rules are based take into account the support brought by the connected shell. For bolted flat ends, gross plastic deformation is prevented by limiting the bending stress in the centre of the end to a value equal to 1,5 times the nominal design stress (which assures that the yield strength of the material will not be exceeded, when the nominal design stress is limited by this material characteristic). In the case of circular welded flat ends, prevention of cyclic plasticity has also been considered, and for this reason the corresponding equations and graph are intended to guarantee that the primary+secondary stress induced in the cylindrical shell does not exceed three times the lower value between the nominal design stress of the shell and the nominal design stress of the end (the so-called shakedown criterion, which is typically used in all the standards that provide design-by-analysis rules based on stress categorization).

The need of fulfilling the shakedown criterion has been contested in other clauses of EN 13445.3: however not in this one, where it was decided to stay in line with the selected source (Clause C3 of CODAP); but in the next issue of Clause 10, which is now ready for the shortened approval procedure provided by the CEN rules (the so-called UAP route), it will be permitted to waive the shakedown criterion under the condition that a simplified fatigue analysis is performed. In other words application of the shakedown criterion (which limits to 3*f* the primary+secondary stress range in the shell) will be required only in those cases where no fatigue analysis is made (note that EN 13445-3 states that no such analysis is needed when the number of full pressure cycles is less

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than 500). For vessels subjected to a small number of cycles, a higher primary+secondary stress in the shell will then be tolerated, resulting in much smaller thickness for the end.

It has to be noted that the shakedown criterion was up to now one of the main principles on which the traditional stress analysis of Pressure Vessels was based for years; and the fact that the primary+secondary stress range should be limited to 3 times the nominal design stress in all kind of Pressure Vessels, whatever their actual loading condition might be, was taken on board by all the main national Pressure Vessel Codes. With the introduction of the so-called "Direct Route" for Design By Analysis (that is, with the introduction of elastic-plastic calculations directed to finding the limit load of a given structure instead of the calculation of fictitious elastic stresses that in reality do not exist), it is possible to investigate the real behaviour of the same structure when it is subject to load variations; and therefore to predict which post-elastic behaviour will occur when the shakedown criterion is ignored: cyclic plasticity or incremental collapse. As far as the Design-By-Formulas rules are concerned, cyclic plasticity is assumed to be the only mechanism that is likely to occur, due to the uniqueness of the loading (pressure loading). So failure due to cyclic plasticity is in fact a failure due to fatigue, more specifically in the low cycle regime.

Looking at other clauses of EN 13445.3 (particularly flanges and tubesheets, where new methods based on limit analysis have been presented as an alternative to the traditional methods based on stress categorization) it is clear that this new standard is marking an important transition between these two different ways to consider Pressure Vessel design; probably the next issue of clause 10 will be the first clause in the standard where a synthesis of these two approaches will be made.

Coming back to the actual content of Clause 10, it has to be noted that the method used for opening reinforcement in circular (and non circular) flat ends is not based on area replacement (like in the case of openings in shells): the method, which comes from the Italian Pressure Vessel Code ISPESL/VSR, and has been successfully used for at least 30 years, is based on the replacement of the section modulus (which is also a rough approximation, but at least more logical, considering that we deal with walls subject to bending and not to tension). This rule aims to control the stress level in the end. In addition, there is also the need for a replacement of the moment of inertia, so that the overall bending behaviour of the pierced end is similar to that of the unpierced end, in order to assure similar bending stresses in the connected shell in both cases.

Clause 10 gives also the possibility of calculating ends of non-circular shape (that is, rectangular, elliptical or obround; even ends in form of a circular crown have been considered). The correlations used in this kind of calculations are also taken from the Italian Code.

11 Flanges

11A Introduction

Two sets of rules are provided for flange design in EN 13445-3, the traditional rules (in clause 11) and a new radical set of rules (in annex G).

The traditional rules are taken from the British PD 5500 and include a wide range of flange types apart from the common narrow face arrangement, namely full face flanges, reverse flanges, split flanges and welded flanges. The rules for narrow face flanges are based on the traditional Taylor Forge analysis.

Since Taylor Forge is so much used, it is not intended to explain or defend it here. Space will be devoted to listing some of its faults and the way it has been changed from the version still in the ASME code. Design stresses have been reduced at larger diameters in the light of wide experience of leakage problems when higher design stresses are used.

The new rules, known as the Alternative Method, cover narrow face joints only. While new to most countries, the method is based on a method used successfully in East Germany for some years. While most aspects of flange design were considered afresh by the authors of this method, the most notable features must be the use of limit analysis, the recognition that the bolt load has its own tolerance, the calculation of the way the bolt load changes as pressure (or other loads) are applied and the detailed analysis of gasket behaviour.

Any method that wishes to supplant Taylor Forge must be complete and attractive economically. It has to include taper hub flanges. It should assist understanding of what is happening in a flange assembly as an aid to good design and trouble-shooting. The Alternative Method does these things.

The Alternative Method as provided in EN 13445-3 includes a great deal of information on the mechanical model (see clause G.4.2), which there is no need to repeat here. Further explanation for many of the equations would be unduly complicated and not helpful to the user. It is nevertheless difficult for a new user to understand

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what is going on and have confidence in what he is doing, so some space is given in this paper to describing the overall structure of the method. The treatment of the gasket is especially novel and a simplified treatment is included below.

Although the Alternative Method is able to take account of thermal expansion effects due to differences in bolt temperature and/or material, no guidance is given on determining flange and bolt temperatures. This is also discussed below.

11B The Taylor Forge method

The Taylor Forge method goes back over 60 years. Many alternatives have appeared over the years and many papers have explained the faults in the method, but it has continued to be the choice in many national pressure codes. Its success has been put down to the fact that it is a complete method, including gasket parameters, and it gives answers that work most of the time. Its strong point is the elastic stress analysis of taper hub flanges. However it has many weaknesses.

11B-1 Weaknesses of Taylor Forge

a. The decision as to whether a flange is thick enough depends in the Taylor Forge method on a stress calculation which has nothing to do with leakage and is only indirectly associated with permanent deformation (yielding). A stress is useful for the study of cyclic failure, but that is not the normal problem with a flanged joint. Indeed, there are geometries where it will be found that the an increase in thickness of the cylinder or the hub can result in an increase in stress and consequent down-rating of the flange.)

b. The calculation fails to consider the behaviour of the bolt load as pressure is applied.

c. Neither differential thermal expansion between bolt and flange nor non-pressure loads are covered

d. The theoretical model ignores the effect of pressure directly on the shell.

e. The theoretical model for the flange applies the moment as equal and opposite loads at the two edges of the flange rather than at the bolt circle etc.

f. Not all flanges are best modelled as thin plates. Some can be closer to thin cylinders.

g. The weakening effect of the bolt holes is ignored. This is especially important in loose flanges in which the stresses are circumferential. On the other hand the weakening effect of any groove is greatly magnified.

h. Although the bolts are given low design stresses to compensate for uncertainty over actual bolt loads, the same logic has not been applied to the flange, which also has to carry those same uncertain bolt loads. The consequence is that if the bolts are over-tightened it is the flange that suffers, not the bolts.

i. The “optional” flange may be calculated in either of two ways but the integral flange of the same dimensions may be calculated in only one of those ways. This seems “unfair” on the integral flange.

j. Maximum gasket pressures are not provided.

k. The stub flange in a lap joint is not considered

l. The gasket factors m and y are not scientifically defined. They seem to be based on guess work.

m. All methods of bolt tightening are treated alike. No credit is given for close control of bolt load.

n. No change is made when a flange is mated to a flat cover, even though the elastic behaviour of a flat cover is very different.

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11B-2 Changes made to Taylor Forge Method

The major modifications made to the original Taylor Forge method as it appears in the ASME code follow. Note that only the first two modify the basic Taylor Forge method.

a. Following poor experience with simple substitution of the higher design stresses used in Europe, increased safety factors at larger diameters have been introduced.

b. Bolt stresses have been increased to reduce the disparity between flange strength and bolt strength mentioned in 8) above.

c. The complicated table for the calculation of effective gasket width has been removed. The biggest difference is the removal of the 1/64 in nubbin. The 1/64" nubbin is found in practice to have only a short life at higher temperature duties as it quickly gets worn away, so it is better removed from the standard.

d. The "loose" calculation may be applied to "integral" flanges.

e. For simplicity, the same moment is used in "loose" and "integral" calculations.

f. Formulae have been moved from the list of symbols into the text, making the method easier to apply.

g. Machining tolerances have been added.

h. Flange pairs of different design conditions (as found with a trapped tube sheet) are included.

i. Rules are provided for the stub flange in a lap joint.

11C Review of rest of Clauses 11 and 12

11C-1 Lap joints

The basis of the method for lap joints is that both components (stub flange and loose flange) are designed individually as flanges. The problem is to choose the diameter of the reaction force between the two. A rule is provided for calculating a diameter but, rather as in the use of limit analysis in the Alternative Method, it is permissible to choose an optimum. Note also that, in common with other traditional methods, no attempt is made to analyse the actual behaviour of the flange pair. The method is safe and may be very over-conservative when applied to existing designs.

11C-2 Full face flange with soft gasket

The method starts with some arbitrary but unimportant assumptions about gasket behaviour. Equation (11.6-16) is concerned with radial stress in the flange at the line of the bolt holes and inequality (11.6-17) is concerned with the flange being sufficiently stiff to be able to exert a pressure on the gasket over the whole span between bolt holes. The failure mechanism that equation (11.6-18) is aimed at preventing is rotation (twisting) of the complete flange assembly.

11C-3 Seal welded flanges

The method here follows Taylor Forge and therefore has the same weaknesses. In particular there is no check that the loss of bolt load on pressurisation will not be excessive.

11C-4 Reverse narrow face flanges

The only important piece of explanation needed to understand the logic of the rules is already in a note in the text. The method is simpler than that in ASME; results are the same for values of K not much greater than 1,0 but at higher values of K the CEN method is more conservative.

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