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CHAPTER

51

UK RULES FOR UNFIRED PRESSURE VESSELS

David H. Nash

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51.1 INTRODUCTION

The present code PD 5500, formerly BS 5500 [1] evolved partly from the well-known BS 1500 [2] in the 1950's and BS 1515 [3] first published in 1965; the latter permitted higher level allowable stresses and more advanced rules. In 1969, following a report from the Committee of Enquiry into the Pressure Vessel Industry, the British Standards Institution brought all the pressure vessel interests together under one general committee in order to rationalise the activity. This became PVE/ and presides over a large committee structure. There are a series of functional sub-committees who deal with specific aspects and a large number of technical committees as well as many additional sub committees and working groups. Most of these meet regularly. The technical committee PVE/1, Pressure Vessels, has overall responsibility for BS 5500. The functional committee PVE/1/15 Design Methods has an overall responsibility relating to 'Design' with particular reference to the design section of BS 5500 (Section 3).

The first edition of BS 5500 was issued in 1976. The actual issue was delayed for some time because, in the early 1970's, there was an attempt in Europe to produce an international pressure vessel standard. A draft of the international standard appeared as ISO DIS 2694 [4] in 1973 but it was not generally accepted and the attempt was abandoned in the mid 70's. It was decided to use some of the material from 2694 within BS 5500 so that although the Standard was long delayed it benefited to some extent from the international efforts. Initially, committee PVE/l set out the concept of a "master" pressure vessel standard which could readily be applied to any vessel in either ferrous or non-ferrous materials and for highly specialised application with the minimum of supplementary requirements. The layout of BS 5500 is consistent with this concept and although the Standard has perhaps not fulfilled this high ideal, it has certainly been employed widely in many industries including non pressure vessel type applications. When issued it had a number of distinctive features compared with other pressure codes viz; weld joint factors were removed, the present three categories of construction were introduced, there was a new novel external pressure section, it has a loose leaf format and an annual updating was introduced. Further editions of BS 5500 have been issued every three years since 1982.

51.1.1 Withdrawal of BS 5500 and Issue of PD 5500

In May 2002 the first issue of the European Standard EN 13445 Unfired Pressure Vessels [5] was published. This standard has been developed to facilitate the provision of vessels subject to the European pressure equipment directive 97.23.EC [6] commonly known as the 'PED'. Under the CEN rules BSI was obliged to withdraw BS 5500 when the European Standard was published in 2002.

The first edition of EN 13445 is not as comprehensive as BS 5500, and due to demands from industry it was decided that the British pressure vessel standard should continue to be available and become a published document (PD) under the new reference *PD 5500*, with equal content, validity and application to the previous *BS 5500*. Its principle difference is that it does not have the status of a "national standard".

It should be noted that most other European pressure vessel codes are not national standards (i.e. they are not published by the national standards body of the country in which they apply).

51.1.2 The Pressure Equipment Directive and PD5500

The main provisions of the Pressure Equipment Directive (PED) are summarised, and are covered in detail in Chapter 47.

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The Pressure Equipment Directive, 97/23/EC, is what is termed a 'new approach' directive which prescribes Essential Safety Requirements (ESRs) which are intended to maintain existing safety levels within the European Community.

The European Parliament and the Council of Ministers approved the PED in May 1997. All member states were required to introduce national laws and provisions necessary to comply with the PED by 29 November 1999. A transition period applied through to 29 May 2002 when the Directive became fully enforced. With its advent, member states must not permit the placing on the market of pressure equipment or assemblies that do not comply with the regulations in force.

To implement the requirements of the PED in the UK the Department of Trade and Industry (DTI) published the Pressure Equipment Regulations 1999 (SI 1999/2001) [7], which became law in February 2000. The DTI have also produced a free guidance booklet URN 99/1147. The Health & Safety Executive is responsible for the enforcement of the legislation in the UK.

The Pressure Systems and Transportable Gas Container Regulations 1989, have been revoked and are replaced by the Pressure Systems Safety Regulations 2000 (SI 2000/128), which apply to the design and construction of pressure equipment not covered by the Pressure Equipment Regulations, and also to the use and ongoing integrity of pressure systems.

The PED applies to products which are first placed on the market / put into service / supplied in any EU member state, and applies to equipment produced for the home market as well as for products destined for use in other EU countries. It also applies to products imported from outside the EU.

The PED covers the design, manufacture and conformity assessment of pressure equipment and assemblies with a maximum allowable pressure greater than **0.5 barg**. The PED does **not** apply to modifications, servicing and repair of equipment unless there is a substantial change of use.

Pressure Equipment means:

Pressure vessels – housings designed and built to contain fluids under pressure.

Piping – piping components intended for the transport of fluids when connected together for integration into a pressure system.

Safety accessories – devices designed to protect pressure equipment against the allowable limits being exceeded.

Pressure accessories – devices with an operational function and having pressure bearing housings.

In general, the duties fall on the **manufacturer**, but care is needed in determining who the manufacturer is - it is not necessarily the fabricator. Duties may also fall to a designer, an importer, a user, a supplier, or the manufacturer's authorised representative.

Each item of equipment is classified into one of the four conformity assessment categories I to IV according to the tables in Annex II of the PED, or as ‘sound engineering practice’ (SEP) for very low risk equipment. The classification depends on the type of equipment (vessel, steam generator, or piping), the state of the fluid contents (gas or liquid), the fluid group of the intended contents (group 1 or 2) and the pressure/volume of the equipment.

Fluid Group 1 comprises those fluids classified, according to the EC Directive on the classification of dangerous substances (Directive 67/548/EEC of 27 June 1967), as explosive, extremely flammable, highly flammable, flammable, very toxic, toxic or oxidising. Group 2 comprises all other fluids including steam.

The pressure/volume of the equipment is the design pressure in bars multiplied by the volume in litres, or for piping, the pressure multiplied by the nominal size in mm.

Equipment classified as (SEP) must be designed and manufactured according to ‘sound engineering practice’. CE marking must **not** be affixed to SEP equipment.

For equipment classified in Categories I to IV, the technical requirements of the PED are presented as a set of basic principles, the Essential Safety Requirements (ESRs), which have to be met. Article 3 and Annex I of the PED state the Technical Requirements and the ESRs respectively.

The ESRs cover general requirements, design, manufacturing and materials. The use of the European harmonised standards being developed, such as EN 13445 on unfired pressure vessels, confers a presumption of conformity to the ESRs. PD 5500 *Annex Z* gives guidance on the application of PD 5500 to pressure vessels falling within the scope of the PED.

To demonstrate that the essential safety requirements are satisfied, the PED requires equipment to be subjected to conformity assessment procedures. The conformity assessment modules have been designed to reflect current industrial practice, and manufacturers are given a choice of modules depending on the category.

Equipment in Category I is subject to the manufacturer's own internal production control. The modules for products in Categories II, III and IV require the involvement of ‘notified bodies’ appointed by Member States either in the approval and monitoring of the manufacturers' quality assurance system or in direct product inspection.

Materials used in the construction of equipment which falls within the scope of the PED must comply with the directive. For equipment which falls within the scope of the PED the materials must comply with the directive.

Three routes are available to demonstrate conformity with the Directive:

- by using materials which comply with harmonised European standards
- by using materials covered by a European Approval of Materials (EAM)

- by a Particular Material Appraisal (PMA)

The EAM approval will be performed by those notified bodies specifically appointed for this task. The result of an EAM is a European Data Sheet, which will contain all the necessary information for the design engineer as well as the inspector. Reference to the EAM will be published in the Official Journal of the European Communities.

A Particular Material Appraisal follows a similar assessment route without subsequent publication of the information. Materials approved by this route can only be used by the manufacturer who obtained the approval on the job concerned. If the same material is used on another job, or by a different manufacturer then a new approval must be obtained.

Some European materials standards have been published, such as EN 10028 Flat products made of steels for pressure purposes, and EN 10222 Steel forgings for pressure purposes, but many of the materials are not yet readily available from stock. European material standards for pipe, tube and fittings have not yet been published. In many cases it will be necessary to follow the EAM or PMA routes where harmonised product or material standards are not yet available, or when the manufacturer wishes to use materials to BS, ASTM, DIN or other standards.

At present only fifteen EAM approvals have been issued, all relating to Nickel 201 and nickel alloy materials. The PMA route must be used for all other non-European materials.

The PED, Annex I sub-section 7.1.2 gives specific requirements for the evaluation of allowable stresses. In some cases these requirements are more conservative than PD 5500 resulting in lower design stresses. In particular, the allowable stress for ferritic materials is limited to the smaller of $R_e/1.5$ or $R_m/2.4$ (see where R_e is the yield strength and R_m is the ultimate tensile strength). The safety factor of 2.4 for the tensile strength is slightly higher than the value of 2.35 used to derive the design strengths for ferritic materials in PD 5500 *Tables K.1-2* to *K.1-12*. For equipment which must comply with the PED the design strength at ambient temperature may need to be reduced accordingly. For example, the design strength to the PED for BS 1501-224-490A or 490B material at 50°C would be 204.17 N/mm^2 compared with 208.0 N/mm^2 from PD 5500 *Table K.1-2.*

Similarly, the allowable stress to the PED for austenitic stainless steels where the elongation after rupture exceeds 35%, is limited to the smaller of $R_e/1.2$ or $R_{m/t}/3.0$. In general this only affects the S61 and S63 grades. $R_{m/t}$ is the tensile strength at the design temperature. This data is not generally available for *BS* materials, but for ASME/ASTM materials values are given in ASME II, *Part D, Table U*. For example, the design strength to the PED for BS 1501-304-S61 material at 50°C would be 183.33 N/mm^2 compared with 203.0 N/mm^2 from PD 5500 *Table K.1-4.*

For equipment manufactured from carbon steel which must comply with the PED, the factor for R_m should be increased from 2.35 to 2.4. For equipment manufacturer from austenitic stainless steel which must comply with the PED, the factor for R_m for austenitic stainless steels where the elongation after rupture exceeds 35%, should be increased from 2.5 to 3.0. In the PED R_m should strictly be $R_{m/t}$ (the tensile strength at the design temperature), but this data is often not available. Most austenitic plate materials to *BS 1501 Part 3* or *ASTM A-240* have a specified elongation which exceeds 35%.

For austenitic stainless steels where the elongation after rupture exceeds 30% but does not exceed 35%, the design strength to the PED is limited to 2/3 of $R_{e(T)}$ at all temperatures, but is not affected by R_m. Many austenitic stainless steel forging materials have a specified elongation which does not exceed 35%.

For equipment which must comply with the PED, the design strength for aluminium alloys, excluding precipitation hardening alloys, is the smaller of $R_{p0.2}/1.5$ or $R_m/2.4$.

51.2 PD5500

PD 5500 specifies requirements for the design, construction, inspection, testing and verification of compliance of unfired fusion welded pressure vessels. The responsibilities of the purchaser, the manufacturer and the Inspecting Authority are defined in *sub-section 1.4*. On completion of the vessel the manufacturer must issue “Form X” to certify that the vessel has been designed, constructed and tested in accordance with PD 5500 and with any additional requirements specified by the purchaser.

Vessels which are required to comply with the PED must also be accompanied by a Declaration of Conformity and, where relevant, operating instructions.

PD 5500 covers five basic material types:

- ferritic steels - such as carbon, carbon manganese and low alloy steels
- austenitic steels - such as type 304, 316, 321 and 347 stainless steels

- aluminium and aluminium alloy
- nickel and nickel alloys
- copper and copper alloys

Although other non-ferrous materials are not specifically covered by *PD 5500*, the code is quite commonly used for designing vessels in other materials. Requirements for titanium will be published as an Enquiry Case in 2005.

For equipment which falls within the scope of the PED the materials must comply with the directive

Certain special types of vessel are covered by specific standards:

BS 1113 Water-tube steam generating plant (partially replaced by BS EN 12952)

BS 2790 Shell boilers of welded construction (partially replaced by BS EN 12953)

BS 4975 Prestressed concrete pressure vessels for nuclear engineering

BS 4994 Vessels and tanks in reinforced plastics

BS 5169 Fusion-welded steel air receivers (excluding those vessels covered by *BS EN 286*)

BS 7005 Carbon steel vessels for use in vapour compression refrigeration systems

BS EN 286 Simple unfired pressure vessels designed to contain air or nitrogen (Parts 1 to 4)

Other vessels not covered by the above standards would normally be designed to PD 5500 Specification for Unfired Fusion Welded Pressure Vessels. This standard is divided into five sections together with various appendices and Code enquiry cases.

Section 1 - General

Section 2 - Materials

Section 3 - Design

Section 4 - Manufacture and Workmanship

Section 5 - Inspection and Testing

There are also a number of other documents published by BSI which give background information relating to the requirements of PD 5500.

PD 6439 - A review of the methods of calculating stresses due to local loads and local attachments of pressure vessels

PD 6497 - Stresses in horizontal cylindrical pressure vessels supported on twin saddles – a derivation of the basic equations and constants

PD 6550 - Explanatory supplement to *BS 5500:1988*
Part 1 Domed ends (heads)
Part 2 Openings and branch connections
Part 3 Vessels under external pressure
Part 4 Heat exchanger tubesheets

51.2.1 Materials

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The basic material type to be used will normally be specified by the process engineer, often in conjunction with a metallurgist. Typical factors which might affect this selection are the corrosion resistance, the presence of aggressive contents such as hydrogen sulphide, hydrogen, chlorides, etc., exposure to high temperature and low temperature, cost and weight. While selecting the material type the process engineer will also consider what corrosion allowance should be applied. The purchaser and the manufacturer shall give joint consideration to the likely effect which corrosion (internal & external) will have upon the useful life of the vessel (*Section 3.3.1*).

From the basic material type the specific material grades for the various components of the vessel are chosen. These material grades or specifications are selected from British Standards such as *BS 1501*, *BS 1503*, or European Standards such as *BS EN 10028*, or other national standards such as *ASME II* or *ASTM*. It is noted that many British Standards, such as *BS 1501* and *BS 1503*, have been superseded by European Standards, but materials to these superseded standards are still available.

The material design strength is usually established from the basic material properties of yield strength and ultimate tensile strength (UTS). For carbon steels for example, the nominal design strength, f_E, is established as the lower of

$$f_E = \frac{R_e}{1.5} \quad \text{or} \quad \frac{R_m}{2.35}$$

These factors on yield and UTS are essentially those proposed in ISO/DIS 2694 [4]. If the material is operated at temperature, then suitable reductions in strength are enforced. Where a material testing standard specifies 0.2% or 1.0% proof stress, these values are taken as R_e. Using this basis for establishing the design strength, values are tabulated in *Annex K for* various product forms and vary according to thickness and temperature. If the design temperature is in the creep range, the calculation of stress levels can be complex and so for membrane regions, the stress limit is set to be less than the creep rupture stress/1.3. Tabulated design strength values at high temperature are also provided and vary according to length of time in hours at elevated temperature.

For carbon steel vessels operating at low temperature, resistance to brittle fracture is addressed by ensuring that the material has sufficient toughness. Consideration is given to the level of membrane stress, the level of inspection and also the use of post weld heat treatment which are seen as adjustments to the basic design temperature. The procedure makes sure that the material at the appropriate thickness has sufficient ductility at room temperature. If however it does not, then the method provides the low temperature value to which material must be tested. Using a Charpy V Notch test, for most carbon steels an impact energy value of 27J must be achieved using a 10×10mm specimen. For high strength steels, 40J must be achieved. Austenitic stainless steels and aluminium alloys are not susceptible to low stress brittle fracture so no special requirements are necessary for their use at temperatures down to -196°C.

PD 5500 has little to say on the strength of welds. It is considered that the weld process controls the quality and that welding standards EN 287 [8] and 288 [9] ensure that the weld procedures and welder approval affirm that the strength and ductility is compatible with the parent material. This implies the joint efficiency factor is equal to unity.

51.2.2 Design

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Section 3 of PD 5500 contains specific rules for performing calculations for shells, heads, cones, nozzles, flat covers, flanges and tubesheets. Most methods in this section are 'design-by-rule' and minimum thicknesses can be calculated if the leading dimensions, allowable strength and design pressure are known. These are well established rules and have much in common with the major international pressure vessel codes. As such, only a brief presentation of the methods is made and differences with other codes are highlighted where appropriate.

51.2.3 Shells under Internal Pressure

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Cylinders and spheres

PD 5500 allows the design of thin shells under internal pressure loading using membrane stress analysis. Thick walled pressure vessels are normally analysed using the Lame equations. Design equations based on this analysis are given in ASME VIII *Division 1, Appendix 1* [10]. Thin cylindrical shells are treated as a closed-end cylindrical shell under internal pressure and the stresses can be found from the conditions of static equilibrium and by evaluating the governing hoop stress. Rearranging the equation as per *sub-section 3.5.1.2(a)* allows the thickness to be evaluated. In the case of spherical shells, a similar set of equations is given in *sub-section 3.5.1.2(b)*. These are approximately based on Lame's equations and incorporate a safety factor which means that the pressure term has a multiplier.

$$e = \frac{pD_i}{2f - p} \quad \text{for cylindrical shells, and} \qquad e = \frac{pD_i}{4f - 1.2p} \quad \text{for spherical shells}$$

Dished Ends

Dished ends follow a similar pattern for spherical ends. However, fabrication of hemispherical ends (and indeed, spherical vessels) is expensive, normally using a labour intensive cap and petal method. The most commonly used closures for pressure vessels are torispherical and ellipsoidal dished ends. Ellipsoidal ends are usually specified as 2:1 (the ratio of major to minor axes) but other ratios may be used. A torispherical end consists of a spherical portion (the crown) and a toroidal portion (the knuckle). This type of end is normally made from a disc, which is held at the centre and spun and cold-formed into the desired shape. Torispherical ends generally have crown radius of between 80% and 100% of the shell diameter, and a knuckle radius of between 6% and 15% of the diameter.

Such heads are prone to buckling under internal pressure and so *sub-section 3.5.2.2* makes recommendations limiting the shape of the end in order to prevent this occurring. A composite graph is available which is based on a stress concentration factor along with limit pressure data, and allows the minimum thickness to be evaluated as a function of head height, design pressure and design stress. Recent FE elastic-plastic work [11] has influenced the design procedure and this has been incorporated along with the PD 5500 approach into EN 13445-3.

Conical Shells

Where a vessel has sections with different diameters, these are usually joined by means of a conical section. Conical ends are sometimes used in place of dished ends, particularly when the end has a large central nozzle. The rules note that the design of nozzle reinforcement for dished ends is limited in *clause 3.5.4.2(d)* to nozzle diameters not exceeding one half of the diameter of the equivalent sphere for the crown portion of the end. For larger nozzles a conical end would be used. Knuckles may be provided at the large and small ends of cones. In addition to calculating the maximum required thickness of the cone for internal pressure, the reinforcement of the cone to cylinder junctions at the large and small ends must be checked to ensure that the discontinuity stresses at the junction are acceptable. Although it is possible to analyse these discontinuity stresses, the method in PD 5500 *section 3.5.3* contains a simplified calculation for the reinforcement.

In the January 1996 amendments the design method for the reinforcement of cones was completely revised. The new method has its origins in the TGL standards [12] from the former East Germany and is very similar to method in BS EN 13445-3 *sub-section 7.6*. The method is based on a limit analysis, and some supporting information is given in the background to the rules in EN 13445-3 [5]. The rule is cast in the form of establishing the thickness based on pressure loading and taking the effect of the discontinuity like a stress concentration factor β.

$$e_j = \frac{pD_c\beta}{2f} \quad \text{where} \quad \beta = \frac{1}{3}\sqrt{\frac{D_c}{e_j}} \times \frac{\tan\alpha}{1+1/\sqrt{\cos\alpha}} - 0.15$$

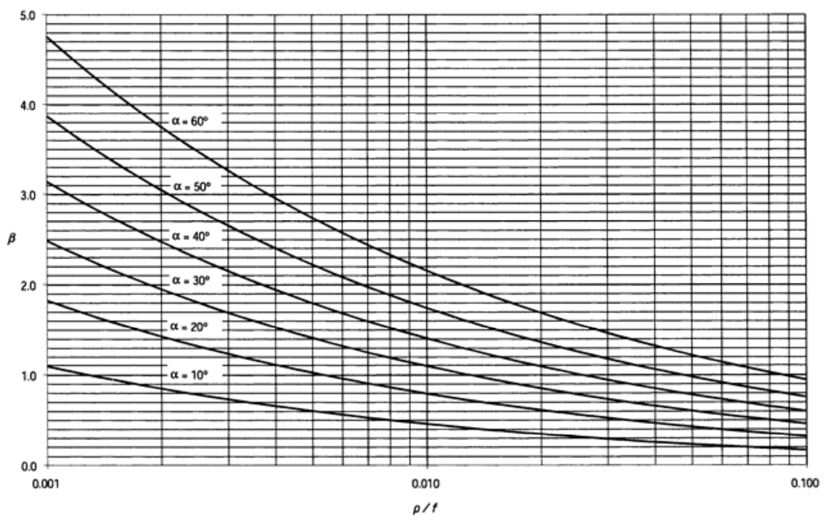

Figure 3.5-4 Values of coefficient β for cone/cylinder intersection without knuckle

Figure 51.1 Values of coefficient β for cone/cylinder intersection without knuckle (Source: Figure 3.5-4 of PD 5500:2003 edition)

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The reinforcement must not be reduced near the discontinuity for a distance of the form $l = \sqrt{De}$. This is typical of a 'die-out' distance from shell theory.

For vessels subject to combined loading, PD 5500 does not provide explicit equations for the minimum thickness for cylindrical, spherical and conical shells subjected to loads in addition to that of internal pressure, so a trial and error solution is necessary (see *PD 5500 Annex B*).

A first approximation for the required thickness for cylindrical shells subject to an axial load *W* and a bending moment *M* is outlined in *sub-section 3.5.1.3.2*. When this approximate analysis indicates that an increase in thickness is required then *Annex B* should be use to determine the minimum thickness.

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51.2.4 Shells under External Pressure

In addition to internal pressure loading, covered in the previous section, many vessels are also subject to External Pressure**.** This may be due to a vacuum condition inside the vessel or an applied external pressure as occurs for the inner shell of a jacketed vessel. The design of externally pressurised vessels involves a completely different approach from that used for internally pressurised vessels. In addition to analysing the compressive membrane stresses, the problems of **elastic and plastic buckling** must be considered. A detailed discussion of the theory involved is given in PD 6550*: Part 3*[13]

When analysing a cylindrical shell subject to external pressure it is not possible to calculate the required shell thickness directly. The procedure in PD 5500 enables an allowable external pressure to be calculated for a given shell diameter, length and thickness. Because **the allowable external pressure reduces as the shell length is increased,** it is often necessary to use ring stiffeners to produce an efficient design. An efficient design is one which minimises the shell thickness and, as such, must have a short maximum length between any two stiffening planes.

Shape imperfections are also of importance since these will normally increase under the action of external pressure. The calculations in PD 5500 *Section 3.6.1* are valid for cylindrical shells that are circular to within **±0.5%** on the radius. A procedure to measure and calculate the departure from a true circle is given in *clause 3.6.8*. A method is given in *Annex M* for the determination of the safe external working pressure for cylindrical shells outside this circularity limit.

In the design of stiffened cylindrical shells, three conditions are considered:-

(a) **Interstiffener buckling** - ie. local collapse of the shell but the stiffeners remaining circular. Generally occurs when the stiffeners are placed too far apart.

(b) **Overall buckling of the shell** - ie. general gross collapse of the shell and any stiffeners which are attached. Generally occurs when the stiffeners are too weak too resist the external pressure.

(c) **Stiffener instability** - ie. local buckling of the stiffener which may result in an overall shell buckling situation. Generally occurs when the stiffener proportions are incorrect, namely tall thin stiffeners, in which the individual webs may initially collapse.

The **'Interstiffener Buckling'** check must be performed for all cylindrical shells subject to external pressure. The **'Overall Buckling'** and **'Stiffener Instability'** check are only performed when additional stiffeners are introduced.

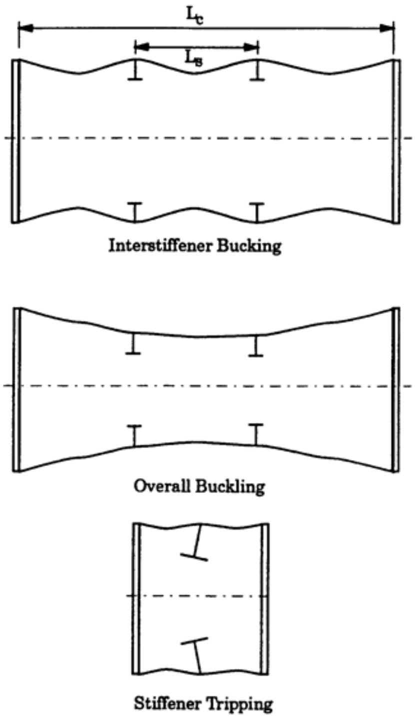

Figure 51.2 Buckling forms for stiffener cylindrical shells (Source: Figure 3.6-2 of PD 5500:2003 edition)

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The design procedure allows the determination of two important pressure quantities. The first is the pressure, p_y, at which the mean circumferential stress in the cylindrical shell midway between stiffeners reaches the yield point of the material, from *equation (3.6.2-7)*, as shown:

$$p_y = \frac{sfe}{R(1-\gamma G)}$$

where s is a material factor (1.1 for austenitic steels and 1.4 for carbon steels), f is the design strength, e the vessel thickness, R the mean radius and γG, a second order term when stiffeners are present.

The second quantity, the elastic instability pressure, p_m, for collapse of the cylindrical shell is found from *equation (3.6.2-8)* as shown:

$$p_m = \frac{Ee\varepsilon}{R}$$

Where E is the young's modulus of the material and ε is the theoretical buckling strain for a perfectly circular cylinder. This strain is a function of the mode shape of collapse of the vessel and it in turn is a function of the unsupported length, diameter and thickness of the shell. The theoretical bucking strain is given in graphical form and also by a supporting equation.

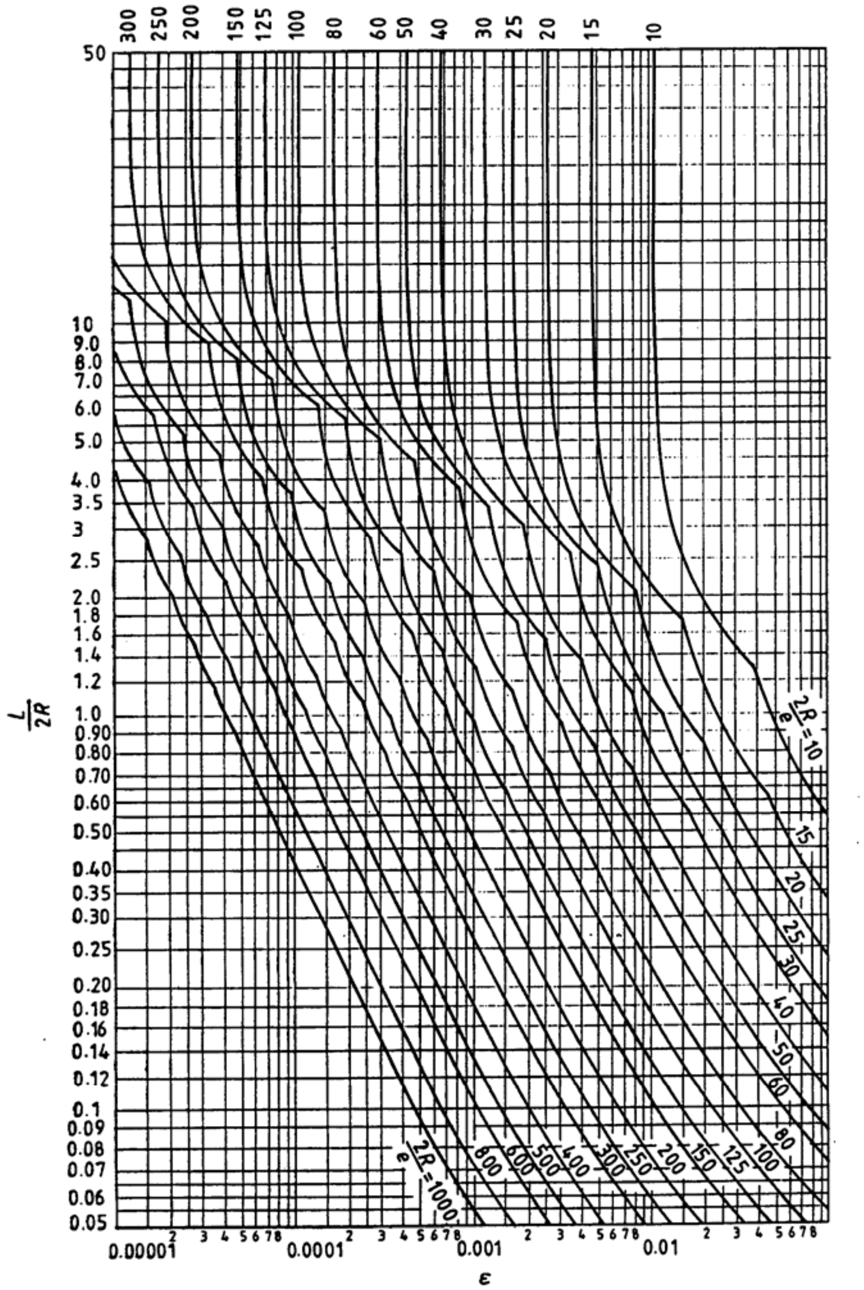

Figure 3.6-2 Values of ε

Figure 51.3 Theoretical bucking strain ε as a function of shell length, radius and thickness **(SOURCE: Figure 3.6-2 of PD 5500:2003 edition)**In order to correct for real vessels which may have slight shape imperfections and some residual compressive stress, a graph was provided by Kendrick [15] which relates the theoretical collapse pressure to the actual experimental collapse pressure with a 1.5 safety factor. This graph is non-dimensionalised by dividing by the yield pressure found previously i.e. $K=P_m/P_y$ and $\Delta=P_{allow}/P_y$.

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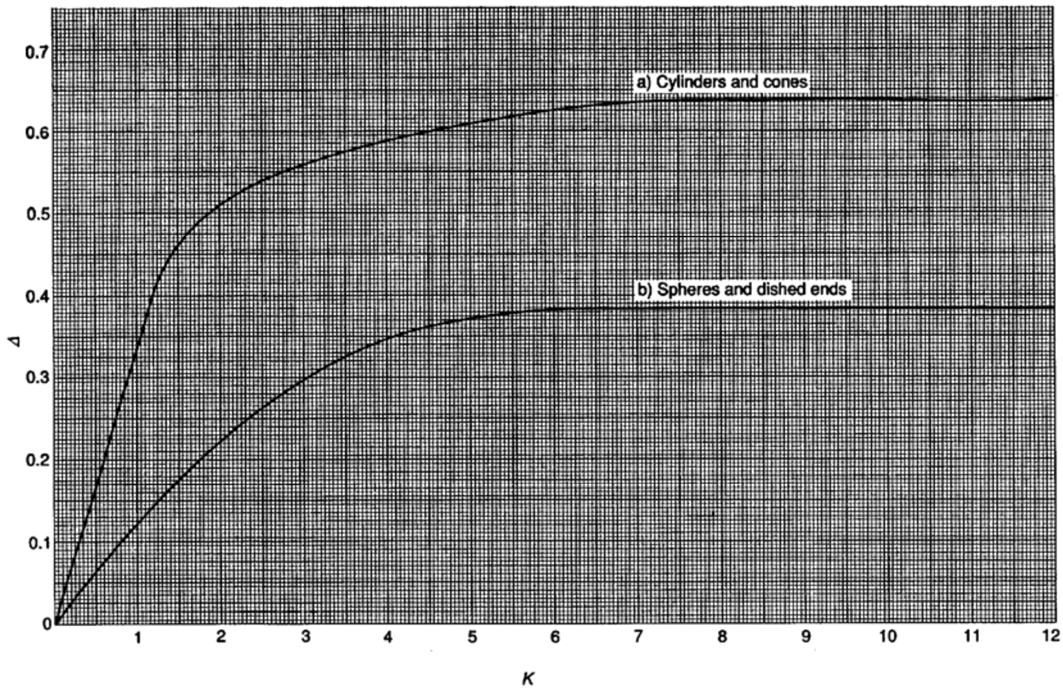

Figure 3.6-4 Values of Δ

Figure 51.4 Graph of non-dimensionalised allowable external pressure versus theoretical collapse load (SOURCE: Figure 3.6-3 of PD 5500:2003 edition)

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Spherical shells and dished ends are designed in a similar manner using appropriate equations for a spherical shell. Again, two pressure quantities are evaluated. The theoretical buckling pressure is related to the actual collapse load using the lower of the two curves shown above.

The calculations are valid for cylindrical shells that are circular to within ±0.5% on the radius. This tolerance may be increased at the design stage if the design external pressure is increased in the ratio of (increased % radial tolerance / 0.5). For example, if the radial tolerance is increased to 0.8% then the design external pressure must be increased by a factor of 1.6. A more refined method for evaluating the allowable external pressure when the radial tolerance exceeds 0.5% is given in *Annex M*, but this procedure requires measurements of the as built shape which are not available until after the vessel has been fabricated.

This procedure satisfies the 'interstiffener buckling' condition. When stiffeners are present, interstiffener buckling must be rechecked however the stiffener shell and ring must satisfy the 'overall buckling' requirements. In addition, the proportions of the stiffener must not exceed prescribed limits. This ensures that the 'tripping' condition will not prevail.

51.2.5 Nozzle Reinforcing

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The traditional way of reinforcing an opening or nozzle is to provide material near the hole in excess of the minimum thickness required for the individual components considered as unpierced shells. Planes through the centre of the opening and normal to the vessel surface are considered, and the area (in the plane) of additional material is required to be at least equal to the area removed by making the hole in a shell of minimum thickness. This type of design approach has been applied very widely indeed. It was used, for instance, in BS 1500 and BS 1515 pressure vessel codes and the water-tube boiler code BS 1113, and is to be found in European codes as well as various sections of the ASME Boiler and Pressure Vessel Code.

A different form of presentation, generally known as the "pressure area" method, is used in the French pressure vessel code (CODAP) [15]. Here in essence the product of the total area of material within the defined region and the design stress, *f*, is compared with the product of the design pressure, *p*, and the area over which the pressure acts. The pressure area method has been adopted for the European unfired pressure vessel standard EN 13445 and the simple unfired pressure vessel standard EN 286, and is now included as an alternative method in PD 5500 *Section 3.5.4.9*.

One of the main disadvantages of the area replacement approach is that it gives no information on the stresses and these can vary considerably from one design to another, resulting in differing performance especially under fatigue conditions. Also the method is not readily capable of being extended to situations where there are significant loads in addition to pressure.

In the UK, the main approach for the design of nozzles and their reinforcement has been stress analysis based and this thereafter being cast into a design-by-rule method.

For nozzles in spherical shells, the methods are based on the work of Leckie and Penny with their solution of the elastic, small displacement, thin shell equations for an intersecting cylinder and sphere. In this the maximum stress due to some loading at the nozzle intersection, when expressed as a stress concentration factor K by dividing by the nominal stress produced in the vessel. They did this by observing that for useful ranges of r/R and R/T, these variables can be combined, and the SCF expressed as a function of the ρ parameter independently of the precise values of r/R and R/T so long as ρ is fairly small. Curves were given for both flush and protruding nozzle designs, for internal pressure, for a direct thrust load on the nozzle, for a bending moment applied to the nozzle, and for a shear load at the nozzle to vessel intersection. These are included in PD 5500 *Annex G 2.5*. An important point to note is that, because of the thin shell assumptions concerning the joining of the nozzle and the sphere, the solutions can only be expected to describe the gross structural behaviour. Welds or curved profiles are not modelled and hence the localised stress concentrations at the toes of welds are not included in this analysis.

For nozzles in cylindrical shells, the cylinder/cylinder geometry is much more difficult to analyse than the axisymmetric cylinder/sphere. To obtain a suitable stress concentration factor for a nozzle in a cylindrical vessel, some form of axisymmetric approximation is commonly employed. The method adopted in PD 5500 is to use a cylinder/sphere model generated from the actual transverse section of the vessel i.e. the sphere has the same diameter as the vessel. The stress concentration factor so obtained is then applied to the nominal hoop stress in the cylindrical vessel. This is equivalent to doubling the maximum stresses calculated at the intersection in the cylinder/sphere model and using this for the cylinder/cylinder connection. This approach is conservative and is regarded as justification for applying the approximation to rather larger *d/D* ratios than is perhaps appropriate for the other methods.

The design approach for nozzle reinforcement is based on elastic stresses used in conjunction with shakedown criteria. This is the same approach as used in older UK codes, namely, BS 3915 (steel vessels for primary circuits of nuclear reactors) in 1965 and in the 1968 edition of BS 1515. In BS 3915 and BS 1515 the permitted elastically calculated stress range was limited to $2.25f$ where f is the design stress. The same method is used in PD 5500 for nozzles in spherical vessels. For nozzles in cylindrical vessels, a slightly different approach has been employed. Shakedown factors are taken from a paper by Macfarlane and Findlay [16] Assuming a Tresca yield criterion and uniaxial condition at the crotch corner, it can easily be shown that $K_s = \frac{4K-3}{2K-1}$ where K is the elastic SCF obtained from a Leckie and Penny formulation. When the loading consists of cycling between zero and design pressure, limiting the stress range in effect means controlling the maximum elastic stress. The latter is the product of the SCF and the "membrane" stress near to the nozzle. If it is necessary to reduce the maximum stress, additional material can be placed near the intersection aimed primarily at reducing the SCF. Alternatively the vessel can be thickened by means of a reinforcing pad, or a general increase in plate thickness, giving a reduced local "membrane" stress. A combination of these methods can also be used.

The theoretical background to *Section 3.5.4.3* of PD 5500 involves many simplifications and is limited in scope. For example, only isolated, circular nozzles radial to the vessel are considered. The design curves apply only to nozzles sufficiently distant from each other and any further significant structural discontinuities that the maximum stresses at the junction are not affected. For the particular case of a nozzle or opening in a dished end, the minimum acceptable distance from the vessel/end junction as 1/10 of the vessel diameter for the provisions of *Section 3.5.4.3* to be applicable.

For multiple nozzles in some vessel applications it is necessary to have a large number of small nozzles (or tube connections) and close pitching often results. It is usually possible to arrange the tubes in rows in either a rectangular or triangular pattern. Reinforcement is provided by an increase in the general vessel thickness determined using a ligament efficiency factor. This approach was extensively developed for use in the water-tube boiler code BS 1113.

In addition to this design approach, PD 5500 gives guidance on the thickness of the nozzle for external force and moments loadings and also provides data for the calculation of shakedown loads for nozzles in spherical vessels.

In 1996, Appendix F in BS 5500 was completely revised. The previous calculation method was replaced with a pressure area method based on that used in the European unfired pressure vessel standard EN 13445 This method is now incorporated into PD 5500 S*ection 3.5.4.9*.The method is well established and is used with some variations in many European pressure vessel codes, such as CODAP and AD Merkblatter [17]. This is discussed later in this chapter.

51.2.6 Bolted Flanged Joints

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Most inspection openings or nozzles on vessels are provided with circular-type standard flanges for quick, easy disassembly of closing covers or connected piping. Under normal circumstances, when using standard "rated" flanges (such as *BS 1560*, *BS 4504*, *BS EN 1092*, *BS EN 1759* or *ASME B16.5*), calculations need not be performed since the design of the flange will have been previously covered and this noted on the supplied test certificate. Only non-standard flanges are designed or existing flanges checked for maximum working pressures.

There are three main types of circular bolted flange covered in PD 5500 *Section 3.8*:

(a) **Narrow-faced flanges**. These are flanges where all the gasket contact area lies inside the circle enclosed by the bolts and are designed in accordance with *sub-section 3.8.3*. PD 5500 also covers ungasketed seal welded flanges in *sub-section 3.8.5*.

(b) **Full-faced flanges**. These are flanges where the gasket contact area extends outside the bolt circle. Full-faced flanges with soft ring-type gaskets are designed in accordance with *sub-section 3.8.4*. A simple method for full-faced flanges with metal to metal contact outside the bolt circle is now included in PD 5500, *sub-section 3.8.8*.

(c) **Reverse flanges**. Theses are flanges where the shell is attached at the outer edge of the flange. They are used where there is a requirement to limit the maximum outside diameter of the vessel. **Narrow-faced** reverse flanges are designed in accordance with *sub-section 3.8.6*, and **full-faced** reverse flanges in accordance with *sub-section 3.8.7*.

PD 5500 *Enquiry Case 5500/133* covers **rectangular** narrow faced and full-faced flanges.

When standard flanges cannot be used or are not appropriate to the circumstances then it becomes necessary to design the joint in detail to match specific requirements. PD 5500 provides methods for analysis and design of a range of special joints, allowing the designer to create a design for the most appropriate joint in a given solution.

Flanged joint behaviour has been the subject of detailed research for many decades. In all this time, probably the most significant contribution was the paper published in 1937 by Waters, Wesstrom, Rossheim and Williams [18] in which the authors for the first time, the comprehensive flange design system which became the basis of the well known Taylor Forge method. The method of analysis used involves modelling the joint elements using simplified plate and shell theory with known boundary conditions, and then combining the elements to derive stresses in the various parts. This analysis was the first complete analysis which considered the flange, hub and shell as properly defined entities with minimal approximation.

Wide acceptance and the relative simplicity in its application have made the Taylor Forge method the most widely used flange design technique in modern use and it forms the basis of the flange design sections of PD 5500, ASME VIII and many other codes around the world. However, this method involves a number of assumptions which potentially limit its applicability to certain classes of joints. The technique has suffered from a few problems, most notably the potential for joint leakage due to the increased flange flexibility of the inherently lighter (and more economic) joints. Although this problem was later cured by the introduction of an equation limiting flange rotation, the Taylor Forge method remains the prescribed method in the UK code.

The method requires that the geometry of the joint be specified along with details of the preferred sealing element or gasket. The gasket requires a certain amount of bolt preload to initialise and provide a seal. This is defined by the parameter '*y*' and this factor helps determine the required load for bolt-up. A second parameter, '*m*', is related to the ability to seal once compressed and is used to determine the amount of bolting required to maintain a seal in the operating condition. From these conditions, the actual bolting requirements can be established and the number and core area of bolts can be fixed. Thereafter, the various moments are calculated and the stresses in the flange ring and shell hub are evaluated. The stresses are categorised and limited according to their nature. Membrane stresses are limited to two third of yield and bending stress to yield. However, the method assumes that the *design* pressure is used to size the flange ring and if designed to the limit, then some limited yielding can occur during hydrotest. If leakage does occur during test, then the test should stop, the gasket be replaced and the test repeated. Some problems have been experienced in certain cases with bolted flanged joints in vessels with a diameter over 1m. A stress reduction factor, *k*, has recently been introduced to limit the stress levels in the shell hub connection. If $1000 < D < 2000$ then $k = \frac{2}{3}\left(1 + \frac{D}{2000}\right)$, and if $D > 2000$ then $k = \frac{4}{3}$.

Flange calculations are quite complex, and will usually involve several iterations before the design is finalised. Because of this most flange design is now performed using computer programs.

51.2.7 Flat Plates and Covers

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Flat ends or domed covers may be used to blank off flanges for pressure tests (blind flange), provide manway closures, or be used as fixed or removable end closures. This section covers the design rules given in PD 5500 *sub-section 3.5.5* for welded or bolted flat ends, and *sub-section 3.5.6* for domed and bolted ends. There are rules governing the design of three groups of flat ends and plates:

(i) Welded Flat Ends and Covers

(ii) Non-Welded Flat Ends and Covers

(iii) Flat Stayed Plates without openings (not covered in this course)

Any of these ends or covers may be non-circular, and the appropriate factors are included in the design methods. Welded flat ends are normally used only for small diameter vessels operating at low pressures. This is because of the large bending stresses induced in a flat plate subject to pressure loading. This results in flat ends being considerably thicker than the corresponding dished end.

The calculations for the minimum thickness of a circular unstayed flat end are given in *sub-section 3.5.5.3.1*, and for a non-circular flat end they are given in *Enquiry Case 5500/133*.

The basic thickness of a flat end can be evaluated by considering the analysis of a circular plate subject to pressure loading, see Roark and Young [19],. The most important consideration is the restraint imposed by the connection to the shell. This is established by the use of a factor, *C*, which ranges from 0.3 for a clamped edge to 0.41 for a simply supported edge.

$$e = CD\sqrt{\frac{p}{f}}$$

In addition, the stresses in the shell at the edge of the plate can be evaluated and these are limited to 2.7*f*. This should be compared with ***3f*** allowed in PD 5500 *Annex A*, *clause A.3.4.2.4*. The lower value provides some ability to accept additional loads.

For flat bolted covers, the methods are similar to those for flat plates with the addition of a term to account for the stresses induced by the bolting.

The method in *Section 3.5.6 of* PD 5500 allows the design of a dished closure or head connected to a flanged ring with a suitable **narrow faced gasket** to be evaluated. The procedure is similar to the ASME method and to other international codes. The analysis is somewhat simplified and empirical factors are introduced to take account of the discontinuity forces which interact between the two components.

5.1.2.8 Jacketed Vessels

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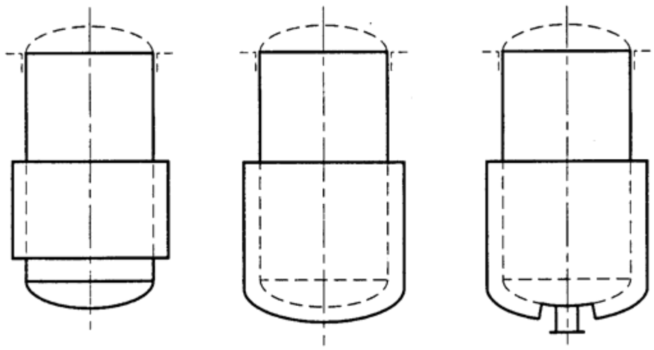
The rules for the design of jacketed vessels are given in *Section 3.11* of *PD 5500* as part of the main design section. The reason for the adoption of a jacket is to provide either **'heating or cooling'** to the main vessel contents. In addition, the jacket may provide a sealed insulation chamber for the vessel. The use of such jacketed vessels is primarily found in the process industry and these types of vessels are usually cylindrical in construction. Jackets are traditionally fabricated in the form of an additional shell belt, this encompassing part or all of the main vessel shell. In addition to this type, jackets can often encompass the lower dished head.



Type 1
Jacket of any length confined entirely to cylindrical shell

Type 2
Jacket covering a portion of cylindrical shell and one head

Type 2
Jacket covering a portion of cylindrical shell and one head, with bottom outlet

Figure 3.11-1 Some acceptable types of jacketed vessels

Figure 51.5 Various types of jacketed vessel

Since there are numerous types of design of jackets, the rules provided in *PD 5500* allow the main 'elements' of the jacketed vessel to be analysed and designed in accordance with the requirements for each individual element. However, due to the possibility of differing pressures in the jacket interspace, there are specific guide-lines provided which ensure that any additional pressure loading transferred to the main shell is adequately coped with. In September 2001 *Enquiry Case 5500/128* was issued, giving preliminary rules for a less conservative design of jacket blocking rings.

Clauses 3.11.4 and *4.3.5.4* cover the use of limpet coils, sometimes referred to as 'half-pipe' coils. There are also some guide-lines in the American Boiler and Pressure Vessel Code, *ASME VIII Division 1 Appendix EE*, which provide some simple rules for the design of half-pipe jackets. Half-pipe or limpet coils are often used as an alternative to a jacketed vessel as a means of transmitting heating or cooling to the contents of a pressure vessel. Half-pipe coils are normally fabricated from pipe which is split, dressed and prepared for welding and then subsequently formed around the shell in a helical manner to provide a continuous channel through which the heating/cooling fluid as able to pass. Guidance is given in *clauses 3.11.4* and *4.3.5.4* (formerly *PD 5500 Enquiry Case 5500/126*) on the design and fabrication of limpet coils.

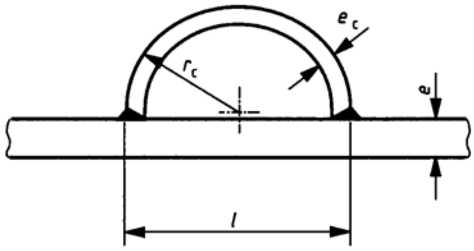


Figure 3.11-3 — Typical limpet coil

Figure 51.6 Typical limpit coil arrangement

When the cylinder to which the limpet coils are attached is subject to vacuum, the coils can be considered to contribute as light stiffeners. The total number of coils N is split into two or more groups N_1 to N_n as shown in *Figure 3.11-4*. The effective light stiffener is assumed to act at the centre of each group.

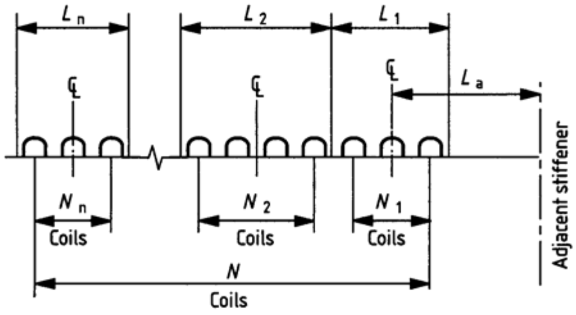


Figure 3.11-4 — Limpet coil arrangements

Figure 51.7 Limpit coil arrangement for use in stiffening for external pressure loading

This approach allows the use of the limpit coil as an effective stiffener against shell buckling.

51.2. 9 Welded joints and Manufacture and Workmanship

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PD 5500 has sections on both welded joint shape and form but little on the structural design for strength. There are some helpful comments on manufacture and workmanship including useful information on pre-heating for welding, stress relieving and shape measurements. This is not discussed in detail here.

51.2.10 Inspection and Testing

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The standard test pressure for a vessel section to PD 5500 shall be not less that p_t, where: $p_t = 1.25\left[p \dfrac{f_a}{f_t} \times \dfrac{t}{t-c} \right]$ where

f_a and f_t are the design strengths at ambient and temperature respectively, t is the shell thickness and c is the corrosion allowance. This allows the vessel to be tested at room temperature but factor in the changes in stress at high temperature and also when the vessel is corroded.

This equation applies to all levels of construction. Where the vessel to be tested comprises a number of **non-connected** parts (as in a heat exchanger) then each part is tested independently with the appropriate 'standard' test pressure. There are particular cases which require special consideration. These include glass lined vessels or those where coating could be damaged, vessels under external pressure and jacketed vessels. A full description of the limits of the ‘standard’ test pressure and the applicable test pressures for all other cases are given in *section 5.8.5.*

Where a component is subjected to an internal test pressure that is greater than 1.08 times the 'standard' test pressure then a design check must be carried out to ensure that the general membrane stress in that component during the test does not exceed 90% of the minimum specified yield or proof stress of the material.

For vertical vessels the effect of the pressure due to the static head of the test liquid must be considered. Tall vessels are normally tested horizontally in the shop, but may be required to be tested vertically at site. For large vessels the weight of the test liquid will be considerable (a cubic metre of water weighs 1 tonne), and may control the design of the supports.

51.2.11 Supports

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PD 5500 *Section 3.7* and *Annex G*, *Section G.3* are concerned with the supports for pressure vessels (and other fittings) which are carried by the shell or ends of a pressure vessel. The main design consideration is to understand the effect of these supports on the shell. No account is made of the support design. These can normally be designed by the usual structural methods. PD 5500 gives some general information about supports and attachments, but does not contain any calculation procedures. Supports produce local moments and membrane forces in the vessel wall and these are treated by applying the rules for Local Loads on Pressure Vessel Shells in PD 5500 *Annex G*, *Section G.2*, with the exception of saddle supports which will be dealt with in detail here. The assessment of stresses in saddle mounted vessels is covered by *Annex G*, *sub-section G.3.3*. A derivation of the equations and constants used in *sub-section G.3.3* is given in *PD 6497* [20].

When vessels are supported in the horizontal plane, they are subject to longitudinal bending moments and local shear forces due to the weight of the vessel and its contents. In addition to these, local stresses arise at the support or fitting. A vessel may be designed to be located in the vertical position in service, but most vessels, both horizontal and vertical, are constructed and transported in the horizontal position. Some vertical vessels may also have their hydraulic pressure test in the horizontal position.

In all cases, it is preferable to support the vessel at two profiles, equidistant from each end. If the vessel is very flexible, or is subject to external pressure, then ring supports may be required. However, under normal circumstances, saddle supports are the most commonly used method of supporting such vessels during fabrication, transportation and in service. It is noted that leg supports are used for small *L/r* ratios where the longitudinal stresses are small in comparison to the axial stresses in the shell due to pressure.

This arrangement means that the vessel acts like a beam in bending and this area requires to be considered at the design stage. Unlike the ring support, the shell is only supported over part of its circumference, typically 120° wraparound, and high stresses can occur at several areas in the plane of the saddle. Therefore, the twin saddle support problem is one of the more complex problems covered in *PD 5500* and is not discussed in detail herein.

The procedure, based on the work of Zick [21] and implemented into PD5500 by Tooth [22] can be broken down into four main design elements. When the vessel is situated on twin saddles, the following high stress ($\boldsymbol{f}$) areas must be investigated:

(i) longitudinal bending stresses at the vessel mid-span ie. $\boldsymbol{f_1}$ at the highest and $\boldsymbol{f_2}$ at the lowest sections

(ii) longitudinal stresses at the saddles i.e. $\boldsymbol{f_3}$ at the highest or equator and $\boldsymbol{f_4}$ at the lowest sections

(iii) tangential shearing stresses at the saddles i.e. $\boldsymbol{q}$ (and $\boldsymbol{q_e}$ when saddle is near the end)

(iv) circumferential stresses at the saddles i.e. $\boldsymbol{f_5}$ at the lowest and $\boldsymbol{f_6}$ at the horn sections.

Each of these stresses must be assessed against specific criteria depending on the influence of other loads and the possible failure mode (e.g. plastic deformation or buckling collapse).

Consideration must be given to establishing the bending moments at the mid-span and at the plane of the support. Thereafter a series of factors are derived which relate the location of the support with respect to the end. If the support is near the vessel end, then the shell remains circular and the full section is available to resist bending. If not, then only a partial section is available. The highest stress is often at the saddle horn, where the shell ‘bends’ over the support and has little or no resistance to radial deformation. Again, various factors are evaluated and the stress, f_6 determined. However the absolute value of f_6 should not exceed 1.25f. This is significantly different from the 3f secondary stress limit which should be used. This reduction is present due to the assumptions in the method and also that work by Tooth has shown the method to under predict stress levels when the support is ‘rigid’ i.e. like a concrete or very substantial support. For flexible saddles, the method provides reasonable agreement with experiments.

If the value of the stress f_6 calculated using the above method is less than 1.25f then this will provide adequate safety margin for static loading conditions, however this value of f_6 is not appropriate for use in a fatigue assessment [23]. A procedure is given in *sub-section G.3.3.2.8* for the determination of a maximum stress, that when added to the stress due to pressure, can be used in a fatigue analysis to *Annex C.*

51.2.12 Local Loads

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Nozzles and attachments such as legs, brackets and trunnions are often subjected to applied forces and moments. These loads produce local stresses in the vessel at the edge of the nozzle or attachment. *Annex G.2* contains methods for calculating these stresses for local loads on either cylindrical or spherical shells. Stresses due to pressure must also be considered. The calculated stresses due to the various loadings are combined and assessed in accordance with the requirements of *Annex A.*

The calculation methods presented in PD 5500 *Annex G.2* for the analysis of cylindrical and spherical shells subject to local loads have different approaches. Whilst they are both **design-by-analysis** methods, i.e. choose geometry and loading and solve a stress problem, they are inherently different in their approaches. A detailed discussion of the theory and development of the calculation methods in *Annex G.2* is contained in the text, 'Pressure Vessel Design - Concepts and principles’ by Spence and Tooth [24].

New alternative methods are also given in *sub-section G.2.7*. These are based on the methods used in the new European standard for unfired pressure vessels *EN 13445*, and allow the maximum permissible loads and moments on nozzles in cylindrical and spherical shells to be calculated.

Local Loads on Cylindrical Shells

The method for local loading on cylindrical shells is based on the solution for a uniform radial line load represented as a Fourier series acting on a thin cylindrical shell. In order to obtain the solution for a rectangular radially loaded area, such as a support bracket, stress and deflection results for the line load were integrated around the circumference of the shell and plotted as *Figures G.2.2-6* to *G.2.2-9*. Plotting in this manner minimised the number of charts required for the stress analysis.

Externally applied moment loading is represented by two equal and opposite rectangular radially loaded areas. This representation allows the original integrated line load data to be used to evaluate stresses for moment loading. Since the moment load is represented by two rectangular areas which, depending on the actual patch size, may be quite close to one another. If the two patches interact, the stresses resulting at the outer edge of one patch may be affected by the die-out length on the second loaded area. Additional data is provided in *Annex G* to allow this interaction and overall die-out to be assessed.

Once the stresses from each load component are evaluated, they must be combined and added to the stresses caused by pressure loading. *Annex G* provides the appropriate equations to evaluate the stresses in a cylindrical shell due to pressure loading only. Stresses are combined in accordance with the **Quadrant Method** and assessed according to the limitations of *Annex A*.

Other shapes of loaded area, such as circular or elliptical, are represented by evaluating the equivalent loaded rectangular or square area. Limitations on the vessel/attachment geometry are also given since the method may be unreliable outside these limits.

The method is slightly different from that contained in WRC Bulletin 107 [25] where the loading is represented explicitly by a double Fourier series. The representation of a moment loading is by a triangular distribution rather than the two equal and opposite radial loads in PD 5500. However, the results obtained by each method have been shown to be very similar [26].

Local Loads on Spherical Shells

Methods are provided for calculating stresses and deflections due to externally applied radial loads and moments applied to spherical shells.

If the attachment is rigid, stresses and deflections are evaluated using equations and graphs as detailed in *sub-section G.2.4*. Each stress resultant is evaluated in turn and the resulting stresses are then combined with pressure stresses.

Sub-section G.2.5 is based on the work of Leckie and Penny [27] and deals with the evaluation of the maximum principal stress which can occur at a nozzle/sphere attachment due to the application of internal pressure, thrust, external moment and shear force. The method covers both flush and protruding nozzles, however, although the original theory assumes semi-infinite nozzle lengths a minimum length of protrusion equal to $\sqrt{2rt}$ is required if the attachment is to be regarded as protruding. Otherwise, the attachment must be considered flush. If the vessel is subjected to cyclic loading through the attachment, then the ***shakedown load*** may be of interest. By keeping the cyclic loadings within the shakedown limit the method ensures that, after initial plastic deformation, further deformation will be in the elastic region, i.e. the vessel has 'shaken down' to purely elastic behaviour. *Sub-section G.2.6* provides the shakedown factors for various loads and the interaction of these under combined load conditions. This method is based on the 'maximum shear stress' theory.

51.2.13 Design for Fatigue

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Fatigue assessment of pressure vessels is normally applied as a checking procedure – after the vessel has been fully designed in terms of dimensions and details. The objective normally is to estimate the fatigue service life for comparison with the notional or explicitly desired fatigue life. Consideration needs to be given in fatigue checking to three main topic areas, namely, the load variation, the stress concentration, and the fatigue properties of the material. All fatigue checking procedures direct attention to these three areas, although they may deal with the problem in different ways. Fatigue properties are generally quoted in terms of constant amplitude stress or strain cycling, whereas real structures are normally subject to variable amplitude patterns – e.g. start-up to full pressure, followed by thermal loading, with superimposed mechanical vibrations, etc. The effect of various numbers of cycles at different amplitudes in succession is normally assessed using a *cumulative damage* law, the most common being Miner's Rule. In this method the 'damage' caused by n_i cycles of a particular amplitude S_{ai} is accounted for by the term n_i/N_i, where N_i is the number of cycles of constant amplitude at S_{ai} which will produce failure. The damage therefore sums up as;

$$\sum \frac{n_i}{N_i} = \frac{n_1}{N_1} + \frac{n_2}{N_2} + ...etc$$

and failure occurs when the sum reaches unity. Use of this law implies that the order of application of the loads does not affect the result and that there are no time-dependent effects. Some codes have also introduced a safety factor by limiting damage summation to 0.6 or some other fraction.

Annex C describes the current mandatory approach to fatigue checking in *PD 5500*.

The main reason for developing the approach in the revised *Annex C* was that the philosophy on which the previous method, developed from the ASME method, was based was inappropriate for weld areas in vessels where fatigue failures are mostly likely. The assessment of **load variation** is virtually the same in both approaches, the main differences being in the philosophies used to assess and link the **stress concentration** effects and the **fatigue strength/life property curves** (hereinafter called '*S/N curves*').

In the **ASME approach** only one S/N curve was given to cover all metallic materials used in the vessel, aside from bolts. This curve was based on data obtained by testing flush-ground welds under strain control (hence without significant stress concentration effect) and the design curve was set four standard deviations below the mean of the test results to give a safety margin (equivalent to a factor of 2.2 on stress and 15 on life). The curve was expressed in terms of stress amplitude S_a (being half the stress range) and in the units used in the Standard, the linear part can be represented by the equation:

$$S_a^{3.5} N = 1.1 \times 10^{12}$$

which is typical of the log-log form for fatigue curves.

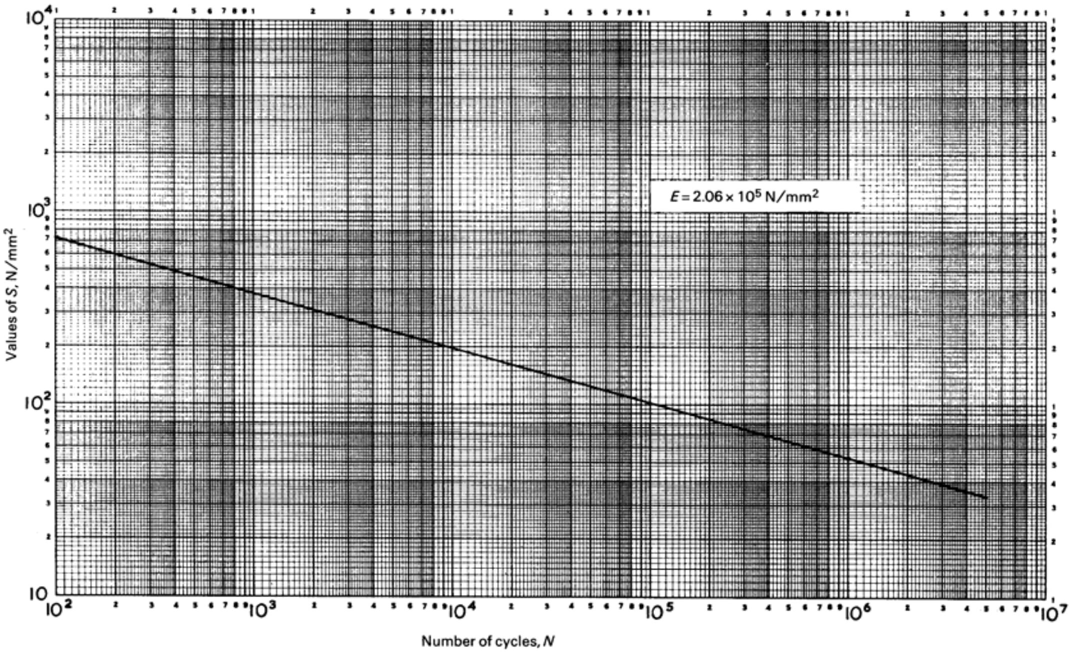

Figure C.2.1 Design fatigue curve for steels for temperatures up to and including 375 °C and for aluminium alloys for temperatures up to and including 100 °C (see C.2.3.4)

Figure 51.8 ASME based/old BS 5500 fatigue design curve (SOURCE: Figure C.2.1 of BS 5500:1994 edition)

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The peak stress amplitude in the region of the vessel to be assessed was then calculated in terms of the nominal stress amplitude times a universal stress concentration factor of 2.5 for any detail other than a smoothly dressed weld or a threaded member.

The following table illustrates the different bases of the **ASME** and the current *Annex C*.

Basis	**ASME/old BS 5500**	*Annex C*
Fatigue Data	'Single' S-N curve for all materials	Several S-N curves for different welds
Relevant Stress	Max stress intensity range ÷ 2	Max normal stress perpendicular to crack direction
Stress Level	Gross and local effects to be included	Gross effects to be included
Weld Detail Consideration	SCF = 2.5 **MIN** (Butt or fillet)	Included in all S-N curves

Table 51.1 Comparison of the bases of ASME and PD5500 fatigue methods (Source: Table C.1 Annex C of PD 5500:2003 edition)

The complexity of stress concentration factors in welds and the difficulty of estimating these properly, are recognized in the *Annex C* approach by including the stress concentration effect of the weld geometry in the family of S/N curves provided for different geometries (see *Annex C, Figure C.3*). The curves do not however include the effect of 'discontinuity stress concentrations' such as would occur at a shell/head junction.

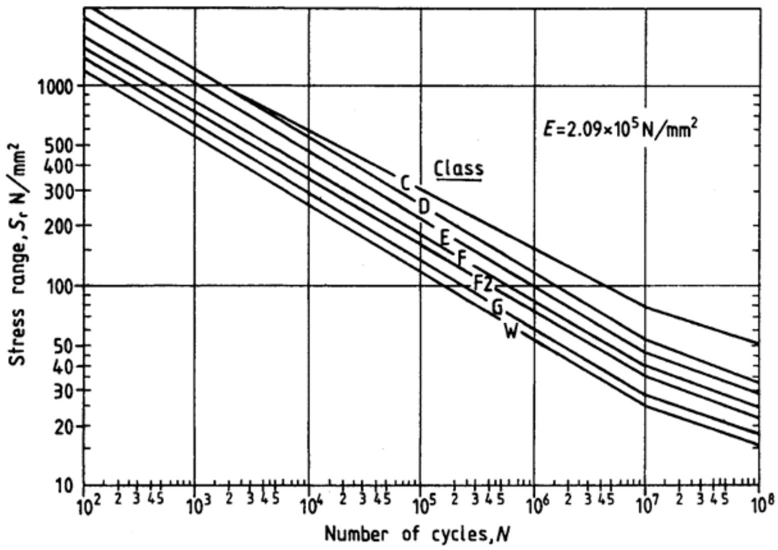

Figure C.3 Fatigue design *S–N* curves for weld details applicable to ferritic steels up to and including 350 °C, austenitic stainless steels up to and including 430 °C and aluminium alloys up to and including 100 °C

Figure 51.9 Family of fatigue design curves from Annex C of PD 5500 **(SOURCE: Figure C.3 of PD 5500:2003 edition)**

The *Annex C* associates each curve with pictures of typical welded details (*Table C.2*) which have been found through fatigue testing programmes to have the indicated strengths. The designer is not given an SCF therefore, but assesses the stress concentration effect by selecting the detail which corresponds best to the one he is checking (there is a considerable art in this and a fair amount of scope for different interpretation). The fatigue curves are therefore cast in terms of the **nominal** Stress Range S_r (Note $S_r = 2 \times S_a$). The S/N curves are also given in equation form, using the constants m and A given in *Table C.1* of the *Annex C*.

$$S_r^m N = A$$

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Table C.1 Details of fatigue design curves					
Class	**Constants of S–N curve**				**Stress range at $N = 10^7$ cycles**
	for $N < 10^7$ cycles		**for $N > 10^7$ cycles**		
	m	A[1)]	m	A[1)]	N/mm^2
C[2)]	3.5	4.22×10^{13}	5.5	2.55×10^{17}	78
D	3	1.52×10^{12}	5	4.18×10^{15}	53
E	3	1.04×10^{12}	5	2.29×10^{15}	47
F	3	6.33×10^{11}	5	1.02×10^{15}	40
F2	3	4.31×10^{11}	5	5.25×10^{14}	35
G	3	2.50×10^{11}	5	2.05×10^{14}	29
W	3	1.58×10^{11}	5	9.77×10^{13}	25

[1)] for $E = 2.09 \times 10^5$ N/mm^2.
[2)] if $S_r > 766$ N/mm^2 or $N < 3380$ cycles, use class D curve.

ASME stresses and Stress Concentration factors

The ASME is based on a single S-N curve originating from uni-axial tests on flush-ground butt weld specimens. The relevant stress in this procedure was the ***'Alternating Stress Intensity'*** denoted as S_{alt}, and defined as: $S_{alt} = 0.5 S_r$ where S_r is the absolute magnitude of the stress intensity. It is noted that the stress intensity is the maximum absolute value of the stress differences of the three principal stresses. It is worth noting that the values of the principal stresses may change throughout a load cycle and therefore, the designer should ensure that the maximum stress differences are considered with respect to time over the whole cycle. The above procedure relates primarily to the case where the principal stress direction remains constant over the load cycle. In those cases where the principal stress directions change, the range of fluctuation should be determined from the stress differences in order to find the full algebraic range. It may be necessary to try various points in time to find the one which results in the largest value of the alternating stress intensity.

There are various features of a vessel and the weld which will reduce the fatigue life of the component. These are dealt with by multiplying the nominal stress by an appropriate '**stress concentration factor**' (SCF) or '**fatigue reduction factor**' in calculating the peak stress. Stress concentration factors are not given explicitly but made reference to various well known publications in its bibliography.

An SCF of one was assumed for a dressed smooth butt weld. A stress concentration factor of '***at least 2.5***' was used for the toe of an as-welded butt or fillet weld. For a contour dressed fillet weld, the stress concentration would be dependent on the local geometry and suitable SCF values had to be obtained from technical literature [28].

PD 5500 stresses

In *Annex C*, the fatigue assessment is based on the '**primary plus secondary stress**' category. 'Direct stress' is used rather than the stress intensity which is used elsewhere in PD 5500. The full stress range is used, regardless of applied or mean stress since the design S-N curves provided, take into account the effects of peak and residual stresses.

However, the fatigue curves for bolting do not take account of stress concentrations in the bolt and the stress range should include a suitable concentration factor – see *Annex C, clause C.3.3.4*.

When the directions of the principal stresses remain fixed, then S_r is the maximum range through which any of the principal stresses changes:

$$\begin{array}{l} f_{1\max} - f_{1\min} \\ f_{2\max} - f_{2\min} \\ f_{3\max} - f_{3\min} \end{array}$$

where f_1, f_2 and f_3 are the three principal stresses. Usually f_3 is rarely relevant and can often be ignored.

If the principal stress directions change, then the three direct and three shear components must be determined and thereafter, for each stress component, the algebraic difference between the stresses must be found. From this, the principal stresses can be found and S_r is the greatest of these principal stresses.

From stresses in weld metal, S_r is the maximum range of stress across the effective weld throat, calculated as the load carried by the weld divided by the throat area. This therefore assumes that no load is carried by bearing between either of the two adjoining components.

For stress cycling in weld metal due to a single application and removal of load: $S_r = \sqrt{\sigma^2 + \tau^2}$

where σ is the direct stress on the weld throat and τ is the shear stress on the weld throat.

For complex cycling conditions, it is preferable to evaluate the vector difference of all pairs of extreme load conditions. It is always safe to assume, $S_r = \sqrt{(\sigma_{1\,max} - \sigma_{1\,min})^2 + (\tau_{1\,max} - \tau_{1\,min})^2 + (\tau_{2\,max} - \tau_{2\,min})^2}$

where τ_1 and τ_2 are the components of shear stress.

In general, in arriving at the primary plus secondary stresses required for use in *Annex C*, it is necessary to take account of structural discontinuities including the following:

- discontinuities such as cylinder to end junctions, changes in thickness and welded-on rings
- deviations from the intended shape, including ovality, peaking and mismatched welds
- temperature gradients

Methods in PD 5500 allow the required stresses to be evaluated for many geometries or at least allow a **conservative estimate** to be made.

51.2.14 Design-by-Analysis

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In the early 1960's the ASME Pressure Vessel and Boiler Code Committee, having recognised with considerable foresight, the advantages to be gained from **detailed stress analysis**, introduced the so-called '***design-by-analysis'*** route for nuclear pressure vessel design. The rules formulated in the design by analysis approach were based upon concepts derived from plasticity theory in an attempt to avoid the consequences of possible bursting or ratcheting failure and fatigue.

However, the effort in determining whether a vessel would fail by any of these mechanisms, was often considerable and expensive and only undertaken by experienced persons. Nevertheless, it was realised that there were many more circumstances where benefit could be gained if detailed stress analysis were performed, essentially by proving a design by carrying out a proper stress analysis.

This approach required an elastic analysis of the vessel or component (although it did not restrict design to elastic analysis alone) and the subsequent classification of the calculated **'shell type'** stresses into certain categories – **primary, secondary and peak** – to which different design allowables could be applied. The method of calculation was not specified, although it was clear at the time that shell discontinuity analysis was seen as the main method of analysis.

The dominant problem in **'design-by-analysis'** is not usually in carrying out the analysis (at least nowadays) but is rather that of the categorisation of the resulting stresses. The rules governing this are not precise, but experience and common practice coupled with the use of thin shell calculations has allowed some degree of reliability to be introduced into the design process. However, if the elastic analysis is performed using more detailed modern continuum finite element calculations then the categorisation of the stresses and the extraction of shell type through wall membrane and bending stresses becomes fraught with difficulty, even though some feel a finite element solution **must** be better.

The PD 5500 code committee essentially recognised the advantages of this design philosophy and subsequently incorporated it (with minor notational changes and with reference to the UK material specifications) into PD 5500 as ***Annex A.***

Essentially design-by-analysis is based on the use of the results of elastic stress analysis. When the *ASME* Code was introduced, the writers were specifically thinking of general vessel stress analysis based on shell discontinuity analysis, or on specific analysis for specific components. Current modern day practice would be to use advanced finite element analysis. Unfortunately the pressure vessel codes do not really address the use of such methods, which can and do lead to various problems of interpretation.

A full discussion of Annex A is not presented fully herein rather the outcomes are noted.

The main failure mechanisms which ***PD 5500*** addresses are those of

- **Gross plasticity**: - large, obvious bulges in the shell

- **Incremental Collapse** (or ratcheting): - collapse by repeated load cycling which increments the amount of plasticity through the thickness

- **Buckling**: - general collapse of the shell into a number of modes, normally attributed to compressive loads and applied external pressure

- **Fatigue**: - repeated load cycling, either mechanical or thermal which continually impairs the material and induces a material breakdown.

Once an analysis has been carried out, there follows the process of assigning the resulting stresses into specific categories depending on the nature and source of the stress and its location and influence on adjacent components.

Different limits are applied to stress categories, as shown in *Figure A.1 of Annex A.* **Primary stresses** are limited for gross deformation. **Primary plus secondary stresses** are limited by the shakedown limit. Summarising the rules and their limitations gives

- Primary membrane stresses are limited by ***f***

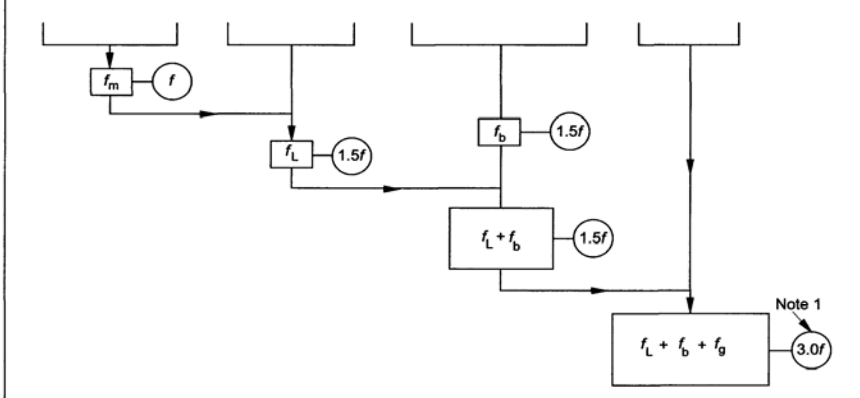
- Local primary membrane by ***1.5f***

- Primary membrane plus bending by ***1.5f***

- Primary membrane plus bending plus secondary stress by ***3f***

Typical cases of stress classification are given in *Table A.1* of *Annex A*. A hopper diagram is also provided which acts as an aid to the combination of the stress components and shows the allowable limits of stress intensity for these groups.

The peak component of stress only needs to be included if a fatigue assessment other than *Annex C* is used, in which case it is added to primary membrane and bending and secondary stresses. It is the component of stress left over from the averaging and linearisation process. The peak stress category, as defined in *A.3.4.2.5* **includes** the contribution of all primary, secondary and peak stresses.

Stress category	Primary			Secondary
	General	Local membrane	Bending	
Description (for examples see Table A.1)	Average primary stress across solid section. Excludes discontinuities and concentrations. Produced only by mechanical loads.	Average primary stress across any solid section. Considers discontinuities but not concentrations. Produced only by mechanical loads, by definition includes f_m in those cases where it is present.	Component of primary stress proportional to distance from centroid of solid section. Excludes discontinuities and concentrations. Produced only by mechanical loads.	Self-equilibrating stress necessary to satisfy continuity of structure. Occurs at structural discontinuities. Can be caused by mechanical load or by differential thermal expansion. Excludes local stress concentrations (see Note 2).
Symbol (see Note 3)	f_m	f_L	f_b	f_g
Combination of stress components and allowable limits of stress intensities				

NOTE 1 This limitation applies to the range of stress intensity. When the secondary stress is due to a temperature excursion at the point at which the stresses are being analysed, the value of f is to be taken as the average of the f values for the highest and the lowest temperature of the metal during the transient. When part or all of the secondary stress is due to mechanical load, the value of f is to be taken as the f value for the highest temperature of the metal during the transient.

NOTE 2 The stresses in category f_g are those parts of the total stress which are produced by thermal gradients, structural discontinuities, etc., and do not include primary stresses which may also exist at the same point. It should be noted, however, that a detailed stress analysis frequently gives the combination of primary and secondary stresses directly and, when appropriate, this calculated value represents the total of f_m (or f_L) + f_b + f_g and not f_g alone.

NOTE 3 The symbols f_m, f_L, f_b and f_g do not represent single quantities but rather sets of six quantities representing the six stress components.

Figure A.1 — Stress categories and limits of stress intensity

Figure 51.10 Annex A Stress categories and limits of stress intensity - Hopper diagram **(SOURCE: Figure A.1 of PD 5500:2003 edition)***Specific Criteria for Limited Application*

The criteria of PD 5500 *A.3.3.1* through *A.3.3.3* provide stress intensity limits for elastically calculated stresses adjacent to attachments, supports, nozzles and openings. These components are normally subjected to combined effects from pressure and from externally applied loads.

These apply to attachments and supports, under certain load restrictions, i.e. the load being distributed over and area with less that 120° circumferential encompassment. Assuming this restriction is satisfied, the concentrating effects can be ignored and conventional shell pressure stresses can be used as the basis for the design.

For this case, the membrane stress intensity is limited to ***1.2f***, and the membrane plus bending stress intensity is limited to ***2f***. Reference needs to be made to *Sec. A.3.4.1* for terminology. For nozzles and openings the maximum stress intensity can be found from *Annex G*, in a certain geometry range (*G.2.3.5.2 (a)* for cylindrical shells or *G.2.5.2* for spherical shells). This is limited, for membrane and bending stress only, to ***2.25f***. Additional stress limits are also provided for localised buckling (but the reasoning for this is not entirely clear!)Where shear stress is present alone, it shall not exceed ***0.5f***. The maximum permissible bearing stress shall not exceed ***1.5f***.

51.2.15 Tubesheets

Section 3.9 of PD 5500 provides a comprehensive treatment of the design of the tubesheets for a range of styles of heat exchanger. This is a somewhat complex subject and is not treated fully herein. Further guidance on the topic can be found in PD6550 Part 4 [30].

51.3 EN 13445

EN 13445 was developed as a harmonised standard for use with the Pressure Equipment Directive and is intended to cover the Essential Safety Requirements of the PED. Use of the CEN standard is not mandatory in the PED, but vessels designed, manufactured and tested in accordance with the CEN standard will have an automatic presumption of conformity with the essential safety requirements of the Directive.

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The major difference from other pressure vessel standards and codes is that there is **(or should be)** no reference in the standard to the responsibilities of the parties - the Purchaser, Manufacturer and Independent Inspection Authority. This aspect is covered by the Directive.

There are some safety matters that the Directive says must be considered (e.g. venting and draining, external fire, etc.), not all of which are mentioned in the standard.

The general layout of the standard and the philosophy behind it are similar to the UK approach in PD 5500:2000. There is naturally some common material since BS 5500 was always a well regarded and well maintained standard, but there are beneficial new methods for dished ends, flanges, tubesheets, expansion bellows and vessels of rectangular section. In recent years there has been a tendency for EN 13445 and PD 5500 to converge as new methods or improvements to existing methods in the standard have been incorporated into PD 5500.

Various safety factors are provided by the Directive and are copied over verbatim. Actually they were firstly agreed by the code writers at an early stage and then transferred into the Directive. Now they are in the PED they can only be modified with great difficulty.

The standard comprises the following parts:

Part 1: General

Part 2: Materials

Part 3: Design

Part 4: Manufacture

Part 5: Inspection and testing

Part 6: Additional requirements for design and fabrication of pressure vessels and vessel parts constructed of spheroidal graphite cast iron.

Each part contains various annexes, including an Annex Z which lists the clauses of the standard addressing the essential safety requirements and other provisions of the PED. (There is a similar Annex Z in PD 5500). It is noted in common with other CEN standards, EN 13445 uses a comma for the decimal point.

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51.3.1 Part 1: General

This part gives the scope of the standard and contains some general definitions, responsibilities of the manufacturer, and requirements for symbols and units.

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51.3.2 Part 2: Materials

This part gives general requirements for materials and lists the various CEN standards for materials. It has several annexes containing specific requirements, including those for prevention of brittle fracture (similar to Annex D in PD 5500).

There are **no** tables of design stresses. The evaluation of design stresses from the properties of the material is covered in Part 3.

Carbon steels, low alloy steels and austenitic stainless steels are included, but non-ferrous materials are not yet covered. Creep is also not covered in the standard, but work is in hand.

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51.3.3 Part 3: Design

The first six sections of Part 3 deal with general matters, such as definitions, symbols and abbreviations, basic design criteria and design stresses.

Section 3: Definitions provides clarity in some areas that is an improvement over PD 5500. Distinctions are made between design pressure and calculation pressure, and between design temperature and calculation temperature. These are particularly relevant for heat exchangers. A distinction is also made between required thickness, analysis thickness and nominal thickness (now adopted by PD 5500).

Section 5: Basic Design Criteria includes information on corrosion, loadings, design methods, thickness, weld joint coefficients and design of welded joints.

General. The requirements are applicable when:

a) The materials and welds are not subject to localised corrosion in the presence of products which the vessel is to contain, and

b) The design is outside the creep range.

Note: This will be changed when the section on creep design is prepared.

Corrosion. There is some clear guidance given for cases where reduction of wall thickness is possible as a result of surface corrosion. **Note**: The minimum corrosion allowance of 1 mm in the 1999 draft has been removed.

Load cases. A familiar looking list of loadings is provided, which must be taken into account in the design of a vessel. In EN 13445 these are defined as **actions**.

a) internal and/or external pressure;
b) maximum static head of contained fluid under operating conditions;
c) weight of the vessel;
d) maximum weight of contents under operating conditions;
e) weight of water under hydraulic pressure test conditions;
f) wind, snow and ice loading;
g) earthquake loading;
h) other loads supported by or reacting on the vessel, including loads during transport and installation.

When necessary, consideration shall be given to the effect of the following loads in cases where it is not possible to demonstrate the adequacy of the proposed design e.g. by comparison with the behaviour of other vessels:

i) stresses caused by supporting lugs, ring, girders, saddles, internal structures or connecting piping or intentional offsets of median lines on adjacent components;
j) shock loads caused by water hammer or surging of the vessel contents;
k) bending moments caused by eccentricity of the centre of the working pressure relative to the neutral axis of the vessel;
l) stresses caused by temperature differences including transient conditions and by differences in coefficients of thermal expansion;
m) stresses caused by fluctuations of pressure and temperature and external loads applied to the vessel;
n) stresses caused by decomposition of unstable fluids.

Classification of load cases identifies three classes:

a) normal operating load case
b) exceptional load cases (e.g. internal explosion)
c) testing load cases.

Note: Higher nominal design stresses may be used for exceptional load cases.

Failure modes considered in this Part (i.e. Part 3 Design) specifies five failure modes considered:

a) gross plastic deformation (GPD);
b) plastic instability (burst);
c) elastic or plastic instability (buckling);
d) progressive deformation (PD);
e) fatigue.

Design Methods provides the all important design philosophy. In addition to the ***design by rule*** methods given in section 5, two alternative methods can be used – ***design by analysis*** (DBA) which is covered in Annexes B and C, and ***experimental techniques***. Very little guidance is currently given on the experimental techniques approach.

The rules in Part 3 provide satisfactory designs for vessels where the number of full pressure cycles does not exceed 500.

Weld Joint Coefficient. A major difference between EN 13445 and PD 5500 is the use of weld joint coefficients which depend on the extent of NDE applied to the governing welded joints. The level of NDE is determined by the ***testing group*** (similar to construction category in PD 5500) – see Table 6.6.1-1 in Part 5 of EN 13445.

Table 6.6.1-1 — Testing groups for steel pressure vessels

Requirements	Testing group [a]						
	1		2		3		4 [b,j]
	1a	1b	2a	2b	3a	3b	
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.

Table 51.3 Nominal design stresses: (Source: Table 6 -1 of EN13445:2002 edition.

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Although weld joint coefficients are not used in PD 5500, for construction category 3 lower design stresses are used to compensate for the lack of NDT, which is a similar concept to a joint coefficient.

The weld joint coefficients are similar to those in ASME VIII Div. 1.

- For testing group 1 (100% NDE) weld joint coefficient $z = 1$.
- For testing group 2 (10% to 100% NDE) weld joint coefficient $z = 1$.

Initially the UK were concerned that it would not be possible to use a joint factor of 1 with partial NDE (category 2 in PD 5500). However testing group 2 allows just that for well established automatic welding.

- For testing group 3 (10% or 25% NDE depending on material) weld joint coefficient $z = 0.85$.
- For testing group 4 (0% NDE) weld joint coefficient $z = 0.7$. However, for testing group 4 the nominal design stress is also multiplied by 0.9 for normal design conditions.

A single testing group shall normally be applied to an entire vessel, but combinations of testing groups 1, 2 and 3 are permitted, subject to certain limitations.

Joggle joints, permanent backing strips and lap joints are dealt with in 5.7.4.

Section 6. Nominal Design Stresses contains rules for evaluating allowable stresses for design (similar to section 2.3.3 in PD 5500). These are summarised in Table 51.3 Nominal design stresses for pressure parts other than bolts.

Table 51.2 Testing groups for steel pressure vessels (Source: Table 6.6.1-1 of EN13445:2002 edition)

	Normal operating load cases [a) b)]	**Testing and exceptional load cases** [b)]
Steels other than austenitic	$f = \min\left(\frac{R_{p0.2/t}}{1.5}; \frac{R_{m/20}}{2.4}\right)$	$f_{test} = \min\left(\frac{R_{p0.2/t\,test}}{1.05}\right)$
Austenitic steels $A \geq 30$ %	$f = \left(\frac{R_{p1.0/t}}{1.5}\right)$	$f_{test} = \left(\frac{R_{p1.0/t\,test}}{1.05}\right)$
Austenitic steels $A \geq 35$ %	$f = \left(\frac{R_{p1.0/t}}{1.5}\right)$ or $\min\left(\frac{R_{p1.0/t}}{1.2}; \frac{R_{m/t}}{3}\right)$	$f_{test} = \max\left(\frac{R_{p1.0/t_{test}}}{1.05}; \frac{R_{m/t_{test}}}{2}\right)$
Steel castings	$f = \min\left(\frac{R_{p0.2/t}}{1.9}; \frac{R_{m/20}}{3}\right)$	$f_{test} = \left(\frac{R_{p0.2/t\,test}}{1.33}\right)$
a)	For testing category 4 the nominal stress shall be multiplied by 0,9	
b)	Yield strength R_{eH} may be used in lieu of $R_{p0,2}$	

Design stresses for non-austenitic steels are similar to PD 5500, except that the factor for UTS is 2.4 rather than 2.35.

Design stresses for austenitic stainless steels depend on the minimum rupture elongation A. For $30\% < A \leq 35\%$ the design stress is limited only by **1.0% proof strength/1.5** at the design temperature. For $A > 35\%$ the design stress may go up to **1.0% proof strength/1.2** with a long stop of **UTS/3** at temperature. If the material standard does not provide the UTS at temperature then this option is not available, and the design stress is limited to **1.0% proof strength/1.5**.

Design stresses are provided for the hydrotest and exceptional load cases. **Note**: the safety factor is **1.05** compared with **1/0.9** (= 1.11) in PD 5500.

Section 7. Shells under internal pressure contains rules for cylinders, spheres, dished ends and cones, and they are similar to those in PD 5500.

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Cylindrical shells. The rules are based on thin shell theory, and the equations are the same as those in PD 5500, except for the weld joint coefficient. Design stresses in PD 5500 and EN 13445 for most carbon steel materials are usually controlled by yield strength divided by 1.5. For testing groups 1 and 2 the weld joint coefficient $z = 1$, and required thickness will be the same for PD 5500 and EN 13445.

For testing group 3 (10% or 25% NDE depending on material) the weld joint coefficient $z = 0.85$, and the required thickness to EN 13445 will be approximately 18% greater than the required thickness to PD 5500 (no reduction in allowable stress in PD 5500 for 10% radiography).

For testing group 4 (0% NDE) the weld joint coefficient $z = 0.7$ and nominal design stress is multiplied by 0.9 (see clause 6.1.3). The design stress in PD 5500 for Category 3 shells is limited to UTS/5 for carbon steels, which is approximately half the design stress for Category 1 or 2 for grade 430 carbon steel. Hence the required thickness to EN 13445 will be approximately 20% less than the required thickness to PD 5500.

For stainless steels the design stress for Category 3 shells is approximately 70% of the design stress for Category 1 or 2, so the required thicknesses to EN 13445 and PD 5500 will be similar.

Dished ends. The required thickness is the greatest of three calculated values. One value is the required thickness to limit membrane stress in the central part using the spherical shell formula. The second value is the required thickness of the knuckle to avoid axisymmetric yielding (based on parametric FE studies by Kalnins and Updike [11]). The third value is the required thickness of the knuckle to avoid plastic buckling (based on the 1986 paper by Galletly). The minimum thickness is $0.001D_e$ compared with $0.002D_e$ in PD 5500.

For Kloepper and Korbbogen type dished ends rules are also given for nozzles in the knuckle region of the head (not permitted in PD 5500).

The charts, Figs 51.11a-c show a comparison between EN 13445, PD 5500 and ASME VIII Division 1 for various dished ends [26].

Note: The curves for carbon steel materials to ASME VIII Division 1 include an adjustment to incorporate the effect of the lower allowable stresses to ASME.

Conical shells. The procedure in EN 13445 for the conical shells is basically the same as that in the current edition of PD 5500, but with the inclusion of the weld joint coefficient *z*. The procedure for the reinforcement of cone to cylinder junctions is a limit analysis method and originates from the TGL Standards of the former East Germany [12]. This method is now included in PD 5500. Cones may be provided with a knuckle at the large end, but this is **not** mandatory for half apex angles greater than 30° (unlike ASME VIII Div.1), and there is no minimum knuckle radius for cones with knuckles.

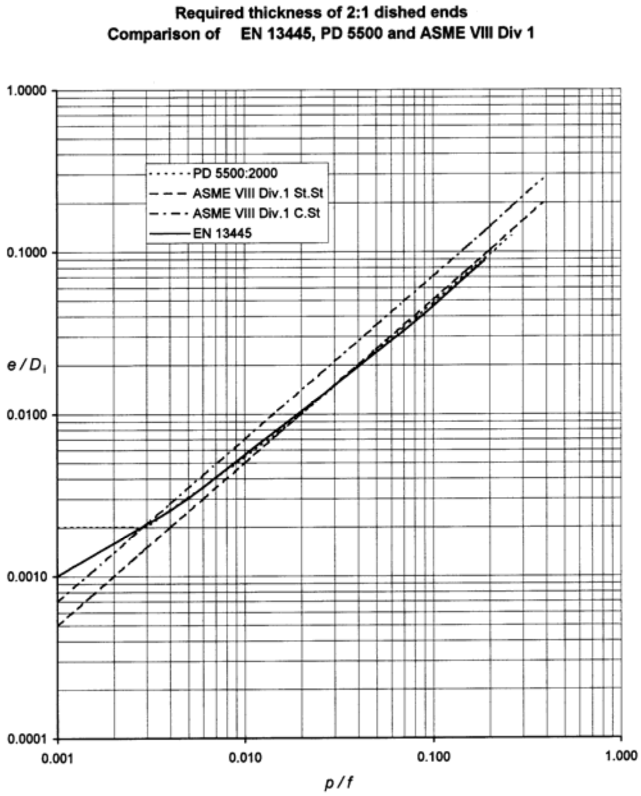

Figure 51.11a Comparison of dished end thicknesses for 2:1 ellipsoidal form (SOURCE: Ref 32)

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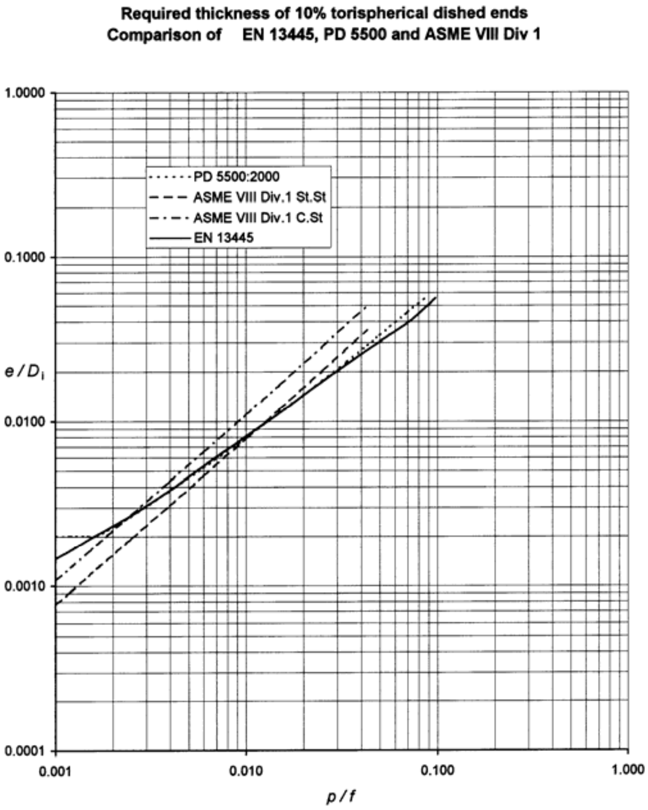

Figure 51.11b Comparison of dished end thicknesses for 10% torispherical form (SOURCE: Ref 32)

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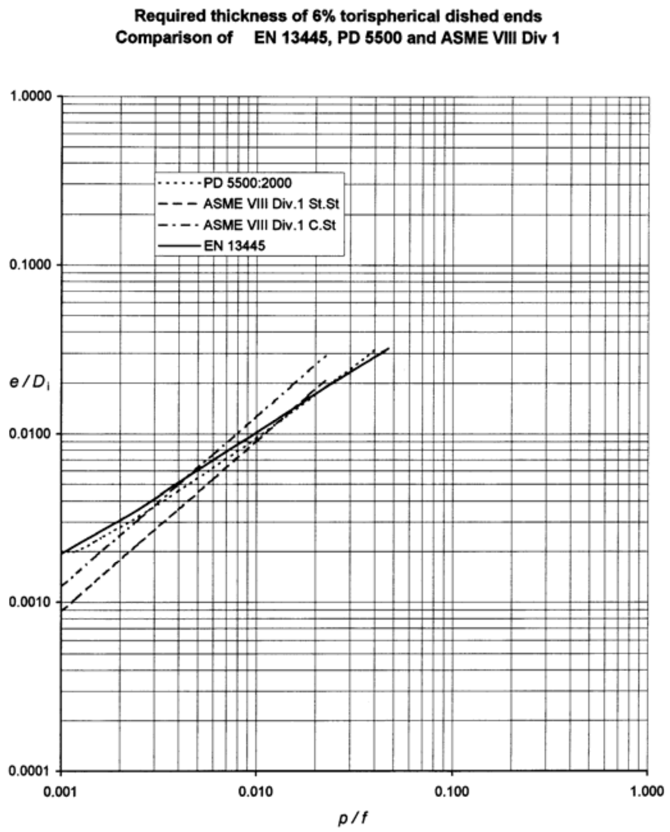

Figure 51.11c Comparison of dished end thicknesses for 6% torispherical form (SOURCE: Ref 32)

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Section 8. Shells under external pressure is based on PD 5500 section 3.6, except that the elastic limit is different (higher for carbon steels and lower for stainless steels). A new method for calculating L_e is provided (now included in PD 5500 sub-section 3.6.2.3.1.1, formerly enquiry case 5500/116). A procedure is given for determining an increased circularity tolerance when there is excess thickness available in the shell.

Section 9. Openings in shells uses the European pressure area method. Here the force due to pressure acting over an area inside the vessel must be balanced by the design stress of the material multiplied by the area available for reinforcement. A single equation applies to all geometries. This method is now included in PD 5500 section 3.5.4.9

Specific rules are given for openings close to a discontinuity, such as a flange or a cone to cylinder junction. Similar rules have now been incorporated into PD 5500 section 3.5.4.10 (formerly enquiry case 5500/130).

Section 10. Flat ends is similar to PD 5500. The method for assessing nozzles and openings in flat ends has now been incorporated into PD 5500 section 3.5.5.3.1(b).

Section 11. Flanges is a rewritten version of the method in PD 5500, which in turn is based on the ASME method, originally the Taylor Forge "Modern Flange Design" method published in 1937. The gasket data is unchanged and still includes CAF, and has no data on more modern alternatives.

There is an alternative method given in Annex G which is based on EN 1591 for piping flanges [29] and comes from the former East Germany. The method takes account of the geometry of the mating flange or flat end and covers non-pressure loads and thermal expansion effects. A flange bolted to a flat end will give different results from those for two identical mating flanges. Elastic analysis is used to obtain the bolt load at ambient so that the designer can ensure sufficient bolt load at the operating condition. Scatter of bolt load at bolting up is a major consideration. A limit analysis is used to check loadings in the flange.

The calculations require much iteration, and even changing the flange thickness means that the whole calculation has to be restarted.

It is worth noting that the alternative rules are most appropriate when;

a) thermal cycling is important,

b) bolt stress is controlled by use of a defined tightening procedure,

c) there are significant additional loadings (forces or moments),

d) leak tightness is of special importance.

The alternative rules do not apply to joints where over compression of the gasket is prevented by contact of either the flanges or a spacer ring, e.g. spiral wound joints.

Section 12. Bolted domed ends is similar to PD 5500 section 3.5.6.

Section 13. Heat exchanger tubesheets provides rules for U-tube, fixed tubesheet and floating head heat exchanger tubesheets. The method follows the traditional approach of calculating a stress and comparing it with an allowable value, but goes into more detail than previous methods. Support from the shell and/or channel can be optimised, but the stresses in the shell and channel must be considered. Bolt loads for fixed tubesheet exchangers are covered by simply specifying a lower allowable stress.

The safety factor on tube buckling is only 1.1 as it is not considered to be ‘fatal’.

There is an alternative method given in Annex J which uses limit analysis and comes from the former East Germany. For fixed tubesheets without bellows it is also necessary to calculate stresses and apply a fatigue failure criterion.

The method is built around the limit analysis of the tubesheet as an axisymmetric flat plate. Limit analysis of pressure vessel components is often difficult, or nearly impossible, but the circular flat plate is relatively easy. This leads to equations that would be relatively simple but for the complexity of allowing correctly for the untubed annulus.

Much of the procedure is devoted to establishing the range of moments available or imposed on the edge of the tubesheet. Another factor to be considered is whether the tubes can carry, by themselves, the local pressure difference across the tubesheet. This is liable to be a problem with high pressure on the tube side and tubes poorly supported against buckling. For fixed tubesheets the ability of the shell to take the axial load from the channel is also considered.

Section 14. Expansion bellows covers both thick and thin walled bellows. The rules are based on EJMA Standards with additional parts taken from AD-Merkblatter, ASME VIII Division 1, ASME B31.3, CODAP and Stoomwezen. The methods cover internal pressure, external pressure and fatigue assessment, and some useful additional information is given in Annex K.

Section 15. Pressure vessels of rectangular section provides rules for unreinforced vessels, with or without a central stay, and reinforced vessels where stiffeners are attached to the outside of the vessel. The methods are similar to those in ASME VIII Div. 1, Appendix 13. Allowance is made for perforated plates by means of a ligament efficiency. Procedures are also included for calculating the required reinforcement for openings in rectangular vessels.

Section 16. Non-pressure loads covers the following topics:

Local loads on nozzles in spherical and cylindrical shells;

Line loads and lifting eyes;

Horizontal vessels on saddle and ring supports;

Vertical vessels on bracket supports, legs, skirts and ring supports;

Global loads.

One major difference between the methods for local loads on nozzles in EN 13445 and those given in WRC 107, WRC 297 or PD 5500 Annex G, is that in EN 13445 local loads are assessed by comparing them with maximum allowable loads based on a limit load analysis in addition to calculating stresses and comparing these with allowable stresses.

A major advantage for the designer is that there are considerably fewer charts and tables in EN 13445 compared with the other methods, and equations are given for each of the curves for use in computer programs or spreadsheets. PD 5500 Annex G, section G.2.8 (formerly enquiry case 5500/122) now includes alternative methods for analysing local loads on nozzles in cylindrical and spherical shells based on EN 13445.

The method for line loads comes from East Germany. Simple allowable load equations are combined with a bending limit stress which allows for pressure and global loads. A procedure is given for the evaluation of allowable loads on lifting eyes (lifting lugs).

The rules for saddle supports are based on a limit load analysis and are quite different from those in PD 5500 (which are based on the Zick method). The method in EN 13445 requires a geometry configuration to be given and a limit load evaluated. This requires various formulae to be calculated and the evaluation of several factors from graphs.

For vertical vessels EN 13445 contains design methods for:

- vertical vessels on bracket supports (section 16.10)
- vertical vessels on leg supports (section 16.11)
- vertical vessels with skirts (section 16.12)
- vertical vessels with ring supports (section 16.13)

Sections 16.10 and 16.11 only cover assessment of support loadings in the shell or head, and do not cover the design of the actual brackets or legs.

Section 16.12 includes design procedures for skirts, including the effects of skirt openings and skirt to vessel attachment.

Section 16.13 includes procedures for the design of ring supports.

Section 17. Simplified assessment of fatigue life gives rules for simplified fatigue assessment for pressure loading only.

Section 18. Detailed assessment of fatigue life provides rules for detailed fatigue assessment for pressure vessels and components subject to stress fluctuations.

Both fatigue sections are based on the method in PD 5500 Annex C with additional refinements.

Ten separate fatigue design curves are provided for different weld categories, compared with seven in PD 5500. In addition, a fatigue curve is given for unwelded material, together with corresponding rules.

51.3.4 Part 4: Manufacture

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This part covers requirements for material traceability and marking, manufacturing tolerances, acceptable weld details, welding, NDE personnel, production testing, post weld heat treatment and repairs.

51.3.5 Part 5: Inspection and Testing

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This part covers requirements for design documentation, inspection and testing during fabrication, final assessment, marking and declaration of conformity with the standard, and files to be compiled (records).

For testing groups 1, 2 and 3 the standard hydraulic test pressure is the higher of:

$$p_t = 1.25\, p_d \frac{f_a}{f_t} \quad or \quad p_t = 1.43\, p_d$$

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51.3.6 New Developments

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There have been several minor revisions to EN 13445 to correct various errors in the standard, and there is a considerable programme of work to cover areas not yet included in the current edition, including:

- Aluminium vessels;
- Reinforced and toroidal bellows;
- Experimental design methods;
- Creep;
- Austenitic nodular cast irons;
- Stiffened flat walls.

A new edition of EN 13445 is planned for 2006. Further guidance on the use of EN13445 can be found in Ref 32. Further background in formation on PD 5500 can be found in Ref 33.

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A document entitled *EN 13445 "Unfired pressure vessels", Background to the rules in Part3: Design* [34] has been produced by Guy Baylac and Danielle Koplewicz. This is available for downloading from the EN 13445 help desk web site at www.unm.fr/en/general/en13445/.

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