

Specification for

Design and manufacture of water-tube steam generating plant (including superheaters, reheaters and steel tube economizers)

ICS 27.040

Committees responsible for this British Standard

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- Electricity Association
- Engineering Equipment and Materials Users' Association
- Health and Safety Executive
- Lloyd's Register of Shipping
- Power Generation Contractors' Association
- Safety Assessment Federation
- UK Steel Association

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Foreword

This British Standard, first published in 1943 under the title 'Water tube boilers and their integral superheaters' and subsequently revised in 1951, 1958, 1969, 1985, 1989, 1992 and 1998 has been prepared under the direction of the Engineering Sector Board. This edition supersedes the 1998 edition, which is withdrawn.

The 1969 edition incorporated the requirements of ISO/R 831 published by the International Organization for Standardization (ISO). The 1985 edition deviated from these requirements and incorporated a number of developments from BS 5500. Most of these developments, and particularly the adoption of stress tables incorporating a range of design lifetimes related to the creep rupture properties of the material, were subjects of amendments to the 1969 edition. Where design stresses in the creep range were significantly different from those hitherto used successfully, provision was made for alternative design stresses to be used in conjunction with a 'continued service' review at a specified fraction of the design life.

This approach recognized the limitations inherent in any simple design method for boiler components operating in the creep range and also provides a flexible basis that might be used in cases where the design stress values, which have been derived from ISO data, are significantly different from those used with success in the past.

Whilst recognizing that in practice the operating life of a component designed on a time-dependent basis might be extended beyond design lifetime, this aspect of plant operation was recognized as being under the control of the plant owner and therefore outside the scope of this standard. It is the subject of other British Standard publications.

The 1998 edition introduced many of the manufacturing practices contained in the draft European Standard currently being developed by CEN/TC 269.

These were:

- forming of drums, headers, and ends;
- forming and bending of tubes and associated procedure tests;
- manufacture of welded tubewalls;
- modification of seal welding requirements;
- modification of post weld heat treatment requirements;
- modification of production control test plate requirements;
- modification of hydrostatic pressure test requirements.

In addition, steel 91 tubes and pipes had been incorporated into table 2.1.2.

Safety valve clauses had been modified to remove ambiguities in the UK requirements.

This 1999 edition has been issued to incorporate several additional manufacturing and inspection requirements from the proposed European Standard on water-tube boilers, where they can be introduced without major changes and will be of an economic advantage to users of this British Standard.

They include:

- incorporation of EN materials;
- design considerations for welding;
- welder procedure and welder approval requirements;
- production control test plates; and
- non-destructive examinations.

This new edition should ease the introduction of the European Standard on water-tube boilers when it is published.

Because of the wide range of types of water-tube steam generating plant that may be designed and manufactured in accordance with this standard, general guidance has been given on some aspects with specific requirements being for agreement between the parties concerned according to the particular design and manufacturing details.

It has been assumed in the drafting of this British Standard that the execution of its provisions is entrusted to appropriately qualified and experienced people. It is recognized that clauses concerning inspection facilities and testing facilities which have traditionally been incorporated into boiler standards are a contractual matter. Nevertheless their importance is such that the purchaser in agreeing a contract should not overlook them. These clauses are reproduced in annex G.

Technical changes, made since the last amendment of this standard was published, are indicated by a sideline in the margin. Presentational changes have not been marked in this way.

The purchaser is recommended, as an aid to demonstrating the boiler supplier's capability of achieving the required quality level, to specify in the contract, that the manufacturer operates a quality system in compliance with the appropriate parts of BS EN ISO 9000.

This standard is included in the list of 'Standards significant to health and safety at work' published by the UK Health and Safety Executive (HSE) and is also referred to by HSE in giving guidance.

A British Standard does not purport to include all the necessary provisions of a contract. Users of British Standards are responsible for their correct application.

Compliance with a British Standard does not of itself confer immunity from legal obligations.

Summary of pages

This standard is kept up to date by the issue, from time to time, of replacement or additional pages. Each replacement or added page will carry a copyright date indicating its date of issue.

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Section 1. General

1.1 Scope

1.1.1 This British Standard specifies requirements for the design and the manufacture, including materials and design stresses, workmanship, inspection, testing, documentation and marking, valves, gauges and fittings of:

- a) water-tube steam generating plant subject to internal pressure; and
- b) parts of steam and water heating plant used in association with steam generating plant.

1.1.2 In addition to the definitive requirements, this standard also requires the items detailed in **1.5** to be documented. For compliance with this standard, both the definitive requirements and the documented items have to be satisfied.

1.1.3 This British Standard applies to the following types of plant.

- a) Water-tube boilers including their integral steel tube economizers and superheaters and all other parts connected to the boiler without the interposition of a shut-off valve with the exception of valves and fittings specified in section 7.

Water-tube boilers are divided, for the purposes of this British Standard, into the following three classes.

- 1) Natural circulation boilers, in which the circulation of the water through the tubes is a result of the thermo-syphonic head produced by heating.
- 2) Forced, assisted or controlled circulation boilers, in which a pump is used either entirely or partly to promote circulation of the water through the tubes.
- 3) Once-through boilers, in which the feed pump is used to force the flow of water and steam through the tubes.
- b) Steam reheaters (either integral with the boiler unit or independently fired) receiving steam which has passed through part of a turbine or other prime mover.
- c) Independently fired steam superheaters.
- d) Integral superheaters separated by a shut-off valve from the evaporative circuits.
- e) Steel tube economizers (either integral with the boiler or independently fired) separated by a shut-off valve from the boiler.

NOTE. For convenience, the term 'boiler' used throughout this standard includes the types of plant described in a) to e).

1.1.4 This standard applies to the pressure parts of boilers and superheaters (as described in **1.1.3a)** and d)) containing water and steam up to and including the valves separating these water and steam spaces from the steam pipes to other equipment, water supply pipes, drain pipes and the surrounding atmosphere (safety valves and vents).

In the case of reheaters (see **1.1.3b)**, this standard applies to pressure parts from the piping connection(s) at the inlet header to the piping connection(s) at the outlet header except that where a device is provided in reheat piping to blank off the reheater for hydraulic testing, this standard applies up to and including this device. In the case of independently fired superheaters (see **1.1.3c)**, and economizers (see **1.1.3e)**, the standard applies to the pressure parts containing steam and water respectively between the inlet and exit valves, including the respective valves and connections to drain pipes and to the atmosphere.

1.1.5 Section 4 of this standard (manufacture and workmanship) includes requirements applicable to the normal function of the boilermaker. Requirements for the quality of workmanship of plates, tubes, forgings and castings in the condition in which they are normally supplied to the boilermaker are included in the standards in the series specifying such materials.

1.1.6 This standard does not apply to brickwork or other settings or supports, or to insulation, or to air preheaters, mechanical stokers, fuel burning or ash disposal equipment, forced or induced draught equipment or their accessories. These items are matters for mutual agreement between the purchaser and the manufacturer.

NOTE. Where such items are required or contemplated, it is recommended that all particulars which could affect the boiler(s) should be disclosed by the purchaser to the manufacturer at the time of the enquiry.

1.1.7 This standard applies only to boilers manufactured under the survey of a competent engineering inspecting Authority or organization. The intent of this requirement may be regarded as satisfied where inspection is carried out by competent personnel of a separate engineering inspection department maintained by the purchaser of the boiler. An inspection department maintained by the manufacturer does not satisfy this requirement except:

- a) that specific responsibilities may be delegated at the discretion of the Inspecting Authority or organization; or
- b) in the case of boilers for the manufacturer's own use and not for resale.

This standard applies only to boilers made by manufacturers who can satisfy the Inspecting Authority or organization that they are competent and suitably equipped to fulfil the requirements specified.

In assessing the competence of a manufacturer, the Inspecting Authority or organization shall take into consideration any related nationally approved certification against the BS 5750 series of quality system standards.

NOTE. The titles of the publications referred to in this standard are listed on the last page.

1.2 Interpretation

If any ambiguity be found or doubt arise as to the meaning or effect of any part of this standard or as to whether anything ought to be done or omitted to be done in order that this standard should be complied with in full, the question shall be referred to the Boiler Technical Committee (PVE/2) of the British Standards Institution, whose interpretation of the requirements of this standard upon the matter at issue shall be given free of charge and shall be final and conclusive. Parties adopting this standard for the purposes of any contract shall be deemed to adopt this provision unless they expressly exclude it or else import an arbitration provision in terms extending to interpretation of this standard. However, this provision is limited to questions of interpretation and does not confer upon the committee any power, duty or authority to adjudicate upon the contractual rights or duties of any person under a contract except in so far as they may necessarily be affected by the interpretation arrived at by the committee.

Findings or rulings of the committee upon all enquiries, including matters of interpretation, which are of sufficient importance that both enquiries and replies be made public as soon as possible will be published in an enquiry-reply form for inclusion in the BS 1113 ring binder as Enquiry Cases. Their availability will be notified in *Standards Update*.

After taking into account any public comment thereon, Enquiry Cases may be incorporated, as appropriate, into the standard as amendments which will form part of the next convenient annual updating.

1.3 Definitions

For the purposes of this British Standard the following definitions apply.

1.3.1 purchaser

The individual or organization that buys the completed boiler from the manufacturer.

1.3.2 manufacture

The organization that designs, fabricates and erects the boiler in accordance with the purchaser's order. The design, fabrication and erection functions may be carried out by separate organizations.

1.3.3 Inspecting Authority

The body or association that checks that the design, materials and construction comply with this standard.

1.3.4 Regulating Authority

The authority in the country of installation that is legally charged with the enforcement of the requirements of the law and regulations of that country relating to boilers.

1.3.5 maximum permissible working pressure

For design purposes, one of the following:

- a) for natural or assisted circulation boilers and directly connected economizers, the highest set pressure of any safety valve mounted on the steam drum;
- b) for once-through boilers, the highest set pressure of any safety valve mounted on the final superheater outlet; or
- c) for steam reheaters, independently fired superheaters and economizers separated by a shut-off valve from the boiler, the highest pressure of any safety valve mounted on the steam or water outlet.

1.3.6 calculation pressure

For calculation purposes, either:

- a) except as defined in b), the maximum permissible working pressure as defined in **1.3.5**, increased where applicable to take account of the pressure drop and hydraulic head corresponding to the most severe conditions of operation; or
- b) for those components whose design stresses are time-dependent (see table 2.1.2 and **A.4**), the lowest set pressure of any superheater or reheater safety valve mounted on the steam outlet, increased to take account of the pressure drop corresponding to the most severe conditions of operation.

1.3.7 design temperature

The temperature to be used in the establishment of the minimum scantling thickness of pressure parts.

1.3.8 design lifetime

The operating lifetime to be used in the evaluation of design stresses for a component whose design temperature at calculation pressure is such that its design stress value is time-dependent.

1.3.9 continued service review

A review or series of reviews carried out to establish if a component, designed to the higher of alternative design stresses in the creep range is fit for continued service up to its design life.

1.3.10 design stress

The stress value, f , used in the equations in this standard for the calculation of the minimum scantling thickness of pressure parts.

1.3.11 pressure part

A component that contains pressurized water or steam or a mixture of the two.

1.3.12 drum

A cylindrical pressure part having a diameter sufficiently large to admit personnel, by design intent, for which purpose an access opening or branch is provided.

NOTE. A drum may have the function of a header, or additionally or alternatively other functions, such as to separate steam from water or to act as a reservoir for boiler water.

1.3.13 header

A pressure part whose principal purpose is to collect fluid from, or distribute fluid to, arrays of tubes directly connected to it.

1.3.14 tube

A tubular pressure part that is either exposed over much of its length to hot gases for purposes of heat transfer, or is directly butt welded to such a tubular pressure part.

NOTE. This definition includes a tube stub on a drum or header.

1.3.15 integral pipe

A tubular pressure part that is not a tube, drum or header.

NOTE 1. The term 'pipe' implies there is no design intent to effect heat transfer through the wall of the pressure part.

NOTE 2. The term 'integral' implies that the pressure part is connected directly to other pressure parts without any intervening valve.

NOTE 3. The term 'integral pipe', subject to note 2, includes those parts of pipes that carry a flow of steam or water during normal operation, those that carry intermittent flow, e.g. drain pipes, and those in which no flow occurs, e.g. instrument piping.

NOTE 4. The terms tube and pipe are used freely in British Standards for material and certain standards (e.g. BS 3602) state that they are interchangeable. The use of the term pipe or tube in any standard for material does not pre-empt the definitions given in 1.3.14 and 1.3.15 and likewise material described as pipe or tube may be used for any duty defined, if appropriate. The selective reference to British Standard materials in section 4 of this standard is intended for guidance based on the general usage of these standards.

1.4 Responsibilities

1.4.1 Responsibilities of the purchaser

The purchaser shall be responsible for furnishing the manufacturer with the information required by 1.5.1.

Where the Inspecting Authority is nominated by the purchaser, the purchaser shall be responsible for ensuring that any information which the manufacturer is required to supply, as specified in this standard, is made available to the Inspecting Authority.

Where necessary, it shall be the responsibility of the purchaser to ensure that the Inspecting Authority is acceptable to the Regulating Authority. This should be done at the time of the order.

1.4.2 Responsibilities of the manufacturer

The manufacturer shall be responsible for the completeness and accuracy of all design calculations and for compliance with all applicable requirements of this standard for the whole boiler. During fabrication, unexpected factors may arise which justify deviations from the specified requirements but which do not affect the safety as intended by this standard. Such deviations shall be submitted to the Inspecting Authority for approval and shall be recorded as part of the boiler dossier (see 6.1.3b).

In cases where the design, fabrication and erection functions are carried out by separate organizations, the manufacturer's responsibilities as laid down in this standard shall be discharged in a manner agreed between the parties concerned (see 1.5.2.2a).

Examinations carried out by the Inspecting Authority shall not absolve the manufacturer from his responsibility for compliance with the applicable requirements of this standard.

A certificate of conformity shall be provided by the manufacturer; see section 6.

1.4.3 Responsibilities of the Inspecting Authority

The Inspecting Authority shall be responsible for checking that the design of the boiler components complies with this standard and that all inspections and tests required during the manufacture of the boiler are carried out.

1.5 Information and requirements to be agreed and to be documented

1.5.1 Information to be supplied by the purchaser

The following information shall be supplied by the purchaser and shall be fully documented (see 6.1.3c). Both the definitive requirements specified throughout the standard and the documented items shall be satisfied before a claim of compliance with the standard can be made and verified.

- a) A specification of the normal working conditions of the boiler, together with details of any transient and/or adverse conditions under which the boiler is required to operate and of any special requirements for in-service inspection.
- b) The name of the Inspecting Authority to be commissioned by the purchaser; see 1.1.7.
- c) Any additional requirements not given in this standard (see 2.2.1.2a, 2.2.1.2b, 3.7.1.2 and 4.1.3).
- d) Any special statutory or other regulations with which the finished boiler is required to comply.
- e) The name of the Regulating Authority (if any).

1.5.2 Requirements to be agreed and documented

1.5.2.1 General

The items listed in 1.5.2.2 and 1.5.2.3 to be agreed, as appropriate, between the parties concerned, which are specified in the clauses referred to, shall be fully documented. Both the definitive requirements specified throughout this standard and the documented items shall be satisfied before a claim of compliance with this standard can be made and verified.

NOTE. Users of this standard should appreciate that a number of items of agreement, i.e. those listed under 1.5.2.3, cannot be agreed/documented at the time of placing the contract or order, for the boiler and do not necessarily apply in every case; they make provisions for individual agreements that may be necessary during the manufacturer's operations to deal with certain practical eventualities (see also 1.4.2). It is important to distinguish these items (1.5.2.3) from those listed under 1.5.2.2, which do need to be agreed and documented when the contract or order is placed.

1.5.2.2 Requirements to be agreed and documented at the contract or order stage

The following items, as appropriate, shall be agreed and documented (see 6.1.3d).

- a) The manner in which the manufacturer's responsibilities are to be discharged in cases where the design, fabrication and erection functions are carried out by separate organizations (see 1.4.2).
- b) Increased design stresses for certain alloy steels (see 2.2.8).
- c) The design lifetime, related to the creep rupture properties of the material, for each component whose design temperature is such that the design stress is time-dependent (see 2.2.8).
- d) Design stress values and the basis for extrapolation for design temperatures above those for which design stress values are available (see 2.2.7.1 and note b) and note 1c) to table A.1).
- e) The maximum values for forces and moments imposed by external piping at the boiler terminals for main steam, reheat steam and feedwater (see 3.1.2).
- f) If applicable, design parameters for wind and earthquake loads (see 3.1.2).

1.5.2.3 Requirements to be agreed and documented during the manufacturer's operations

The following items, as appropriate, shall be agreed and documented (see also 1.4.2 and 6.1.3j).

- a) The use, for pressure parts, of materials not covered by 2.1.2.1 (see 2.1.2.2) and whether elevated temperature yield point or proof stress properties related to tests in accordance with BS EN 10002-5 are not required for such materials (see 2.1.2.2.5).
- b) The heat treatment condition for materials not covered by 2.1.2.1, if an alternative to the condition appropriate to the nearest equivalent British Standard material is required (see 2.1.2.2.7).

c) The use of design stress values (f) determined by reference to 2.2.1.1 from the properties of the actual materials, where these properties have been proved by test to be higher than those given in the relevant material specification (see 2.2.1.1c).

d) The method of calculating the thicknesses of the cylindrical shells of drums and headers, for time-dependent (creep) conditions where the resultant thickness exceeds half the internal diameter (see 3.3.2.2b).

e) The scantlings for headers of rectangular section that are not straight (see 3.5.3).

f) Assembly tolerances for middle line alignments (see 4.2.5.1.1).

g) The welding materials to be used if these are not covered by British Standards (see 4.3.1.2b).

h) For post-weld heat treatment:

- 1) the heat treatment conditions if other than those given in table 4.5.4 (see 4.5.4.1.2a2)); or
- 2) waiver of post-weld heat treatment (see 4.5.4.1.2b).

i) Post-weld heat treatment of joints between any combinations of ferritic steels not included in table 4.5.4 (see 4.5.4.4.3).

j) The requirement for heat treatment of austenitic steel tubes or pipes that are hot or cold bent for integral piping (see 4.2.3.5), the bending temperature range to be used for hot bending, and the post-bend heat treatment procedures to be used (see 4.5.2.2.3).

k) Where necessary, acceptance, as fit for the intended purpose, of welds represented by charpy V-notch impact test specimens from boiler drum production control test plates that give impact values less than required by 5.5.7.3 (see 5.5.7.4b).

l) Where necessary, acceptance of computations if any tensile test specimen from the boiler drum production control test plates fails to comply with specified requirements (see 5.5.8).

m) Where necessary, remedial measures to be adopted if either of the bend retest specimens from the boiler drum production control test plates fails to comply with the specified requirements (see 5.5.8).

n) A reduction in the percentage rate of non-destructive examination for welds, where the manufacturer has demonstrated that the welds consistently comply with the standards of acceptance given in 5.9 (see 5.6.1.6).

o) Non-destructive examination before final stress relief, for the following weld types in carbon, carbon manganese type 620 and type 621 steels (see 5.6.1.9):

- i) butt welds in drums and headers;
- ii) branch, nozzle and stub welds;
- iii) attachment welds.

p) Where necessary, the use of a fitness for purpose analysis to assess the acceptability of particular flaws in excess of the acceptance levels specified in **5.9.3** to **5.9.5** (see **5.9.1b**2)).

q) The procedure to be used with regard to small pockets of trapped air which may remain after filling following venting or evacuation of air prior to hydraulic testing (see **3.1.5h**).

Section 2. Materials and design stresses

2.1 Materials

2.1.1 General

2.1.1.1 The selection of the materials of construction for pressure-containing parts of boilers and their integral attachments shall take into account the suitability of the material with regard to fabrication, e.g. cold forming, weldability and expanding, as appropriate, and the conditions under which they will eventually operate.

2.1.1.2 Welding material shall conform to 4.3.1.2.

2.1.1.3 Cast iron shall not be used in the construction of pressure parts, except for valves and fittings within the limits specified in BS 759 : Part 1 or BS 6759 : Part 1.

2.1.1.4 Studs, bolts and nuts shall be in accordance with the material specifications given in BS 4882.

2.1.1.5 Pipe fittings shall conform to BS 1640, BS 1965 : Part 1 or BS 3799, as appropriate.

2.1.1.6 Flanges shall conform to BS 10, BS 1560 : Section 3.1 or BS 4504 : Section 3.1.

2.1.2 Materials for pressure parts

2.1.2.1 General

With the exceptions of the materials covered by 2.1.1.2 to 2.1.1.6, all the materials used in the manufacture of pressure parts shall either:

- a) conform to the appropriate BS EN steels given in table 2.1.2; or
- b) conform to the appropriate British Standard steels given in table A.1; or
- c) be agreed between the manufacturer and the purchaser and conform to 2.1.2.2 (see 1.5.2.3a).

A combination of materials given in table 2.1.2 or table A.1 and corrosion resistant material is permitted for metallurgically bonded co-extruded tubes.

NOTE. The requirements of this British Standard have been drafted largely on the basis of experience with the materials specified in table 2.1.2 and table A.1; the selection of such materials should therefore be preferred. However, it is accepted that special circumstances may occur in which the selection of materials other than those covered by table 2.1.2 or table A.1 is necessary; the requirements of 2.1.2.2 have been specified to provide a basis for material selection in such circumstances.

2.1.2.2 Materials not covered by table 2.1.2 or table A.1

2.1.2.2.1 The material shall be covered by a written specification at least as comprehensive as the British Standards listed in table A.1 for the nearest equivalent material. For ferritic steels, the phosphorus and sulfur content shall not exceed 0.05 % (*m/m*) each in the ladle analysis.

NOTE. For ferritic steels intended for welding, the upper limit of the carbon range (in the ladle analysis) should not exceed 0.25 % (*m/m*). Steels with a higher carbon content that are intended for welding may be used subject to appropriate welding procedures and heat treatment.

2.1.2.2.2 The design stresses shall be determined in accordance with 2.2.

2.1.2.2.3 The deoxidation practice shall be appropriate to the type of steel ordered. Semi-killed steel shall be used only for plates, and seamless and welded tubes in carbon and carbon manganese steels with an upper limit of the specified tensile strength range of 640 N/mm² and with a thickness not exceeding 100 mm. Rimmed steel shall be used only for welded tubes in carbon and carbon manganese steel with an upper limit of the specified tensile strength range of 490 N/mm² and under service temperature conditions of 400 °C or less.

2.1.2.2.4 Mechanical properties at room temperature shall be determined by means of acceptance tests covering tensile strength, yield stress and elongation.

The specified minimum percentage elongation at fracture referred to a gauge length of $5.65\sqrt{s_0}$ ¹⁾ shall be appropriate to the type of steel, with a lower limit of 16 % for plates, 15 % for castings and 14 % for tubes and forgings.

The rate of testing and methods of acceptance testing shall be generally in alignment with the appropriate British Standards for similar product forms.

2.1.2.2.5 For all pressure parts, elevated temperature yield point or proof stress properties related to tests in accordance with BS EN 10002-5 shall be specified unless otherwise agreed between the manufacturer and Inspecting Authority (see 1.5.2.3a).

Where no elevated temperature yield point or proof stress property values exist or where the material cannot be related to another material for which values exist, property values shall be verified using the procedure described in BS 3920 or shall be established by the recognized standards of other countries.

2.1.2.2.6 Stress rupture properties shall be specified for materials which will be used in the creep range.

The manufacturer of the boiler shall be assured by the material supplier that the product supplied is capable of conforming to the specified properties by a statement that the manufacturing processes have remained equivalent to those for the steel for which the test results were obtained.

2.1.2.2.7 Materials shall be supplied in a heat treated condition appropriate to the nearest equivalent British Standard material unless otherwise agreed between the interested parties (see 1.5.2.3b).

NOTE. Plates for hot forming may be supplied in any suitable condition, e.g. hot finished, normalized, normalized and tempered.

Carbon and carbon manganese steel plates for cold forming shall be supplied in the normalized condition.

¹⁾ S_0 is the original cross-sectional area of the gauge length of the tensile test specimen (see BS EN 10002-1).

Low alloy steel plates for cold forming shall be supplied in the normalized and tempered condition except that, where metallurgically suitable and where post-weld heat treatment will suffice as the tempering treatment, plates supplied in the normalized condition shall be permitted.

Electric resistance welded or induction welded tubes shall be used in the as welded condition only where the specified upper limit of tensile strength does not exceed 540 N/mm².

2.1.2.2.8 In the case of withdrawn British Standard materials listed in this and previous issues of this standard, by agreement between the manufacturer and Inspecting Authority (see 1.5.2.3a), compliance with the requirements of 2.1.2.2.1 and 2.1.2.2.3 to 2.1.2.2.7 may be waived. The requirements of 2.1.2.2.2 for the determination of design stress shall however continue to apply.

NOTE. This waiver may be applied for a period of up to 3 years from the date of withdrawal of the material from listing in this standard.

2.1.3 Materials for non-pressure parts

Materials for supporting lugs, baffles, fins and similar non-pressure parts welded to boilers shall be of established identity and shall be compatible with the material to which they are attached.

2.2 Design stresses

2.2.1 General

2.2.1.1 The design stress f shall be determined in one of the following ways.

- Factors of safety shall be applied to the appropriate material property values of the BS EN steels listed in table 2.1.2 in the manner shown in 2.2.3, 2.2.4 and 2.2.5.
- In the case of existing British Standard steels, f shall be taken directly from table A.1.
- Where the materials used have time-independent properties proved by a test to be higher than those given in the relevant material specifications, the use of those higher properties to determine the design stress value f , at the design temperature, by application of the factors given in 2.2.3 and 2.2.4 shall be permitted subject to agreement between the manufacturer and the inspecting authority (see 1.5.2.3c).

NOTE 1. Material property values at intermediate temperatures and intermediate times to those given in the material specifications should be obtained by linear interpolation. Interpolated values should be rounded to two decimal places in accordance with BS 1957.

NOTE 2. The design stresses obtained in accordance with a) and b) above are derived on the basis of accepted principles and established practice and are intended for general application. However, it is recognized that, during fabrication, unexpected situations may arise where use may be made of the design stresses derived from the mechanical properties of the actual materials being used; the provisions of c) are intended to be used in such special circumstances rather than as a general rule.

2.2.1.2 In determining the design stress the following points shall be taken into account.

a) *Carbon and carbon manganese steels.* The following points shall be taken into account.

- The design stresses derived in accordance with 2.2.3, 2.2.4 and 2.2.5 or taken directly from table A.1 of annex A are intended for general use with the steels listed and acceptance tests on material heat treated with a complete component are not required, any reduction in properties of such steels due to post weld heat treatment being consistent with the overall benefit obtained by stress relief of the structure. A purchaser requiring such tests, or tests on samples subject to non-standard heat treatments, shall specify them in the supplementary specification together with appropriate acceptance criteria (see 1.5.1c).
- In designs where slight deformation is important or where post-weld heat treatment conditions other than those specified in table 4.5.4 are agreed (see 4.5.4.1.2a2)), materials that will conform to the properties in the material specification in the normalized plus simulated post weld heat treated condition shall be specified.

b) *Alloy steels.* The following points shall be taken into account.

- The time-independent design stresses derived in accordance with 2.2.3 and 2.2.4 or taken directly from table A.1 shall be used, provided that the proposed post weld heat treatment conditions are in accordance with 4.5.4 and table 4.5.4.
- Where post weld heat treatment conditions other than those specified in table 4.5.4 are agreed, (see 4.5.4.1.2a2)), the effect of any proposed heating or reheating during fabrication shall be determined in liaison with the material supplier before selecting the appropriate time-independent design stress. The time-independent properties in the material specification, or such properties determined from the selected design stress, shall be satisfied by the testing of material in the simulated heat treated condition.
- Acceptance tests on materials heat treated with the completed component are not required. A purchaser requiring such tests, or tests on samples subject to non-standard heat treatments, shall specify them together with the appropriate acceptance criteria (see 1.5.1c).

2.2.2 Terminology

f	is the design stress which shall be taken as the lesser of f_E and f_F .
f_E	is the design stress corresponding to the short time tensile strength characteristics of the material (time-independent).
f_F	is the design stress corresponding to the creep characteristics of the material (time-dependent).

R_e	is the minimum value of specified yield stress for the grade of steel concerned at room temperature. Where a material standard specifies minimum values of R_{eL} , or $R_{p0.2}$ ($R_{p1.0}$ for austenitic steels), these values are taken as corresponding to R_e .
R_{eL}	is the minimum value of the lower yield stress.
$R_{e(T)}$	corresponds to the minimum value of R_{eL} , or $R_{p0.2}$ ($R_{p1.0}$ for austenitic steels), specified for the grade of material concerned at a temperature T .
R_m	is the minimum tensile strength specified for the grade of material concerned at room temperature.
$R_{p0.2}$	is the minimum value of the 0.2 % proof stress.
$R_{p1.0}$	is the minimum value of the 1.0 % proof stress.
S_{Rt}	is the mean value of the stress required to produce rupture in time t at temperature T for the grade of steel in question.

2.2.3 Time-independent design stresses for carbon, carbon manganese and low alloy steels

2.2.3.1 Materials with specified elevated temperature values

- 1) For design temperatures up to and including 250 °C, use material property data at 250 °C.
- 2) For design temperatures of 250 °C and above:

$$f_E = \frac{R_e(T)}{1.5} \text{ or } \frac{R_m}{2.7}$$

whichever gives the lower value.

2.2.3.2 Materials without specified elevated temperature values

- 1) For design temperatures up to and including 250 °C use material property data at 250 °C.
- 2) For design temperatures of 250 °C and above:

$$f_E = \frac{R_e(T)}{1.6} \text{ or } \frac{R_m}{2.7}$$

whichever gives the lower value.

The values of $R_{e(T)}$ used above shall be taken as equal to those specified for otherwise similar materials having specified elevated temperature values.

2.2.4 Time-independent design stresses for austenitic steels

2.2.4.1 Materials with specified elevated temperature values

- 1) For design temperatures up to and including 250 °C use material property data at 250 °C.
- 2) For design temperatures of 250 °C and above:

$$f_E = \frac{R_e(T)}{1.35} \text{ or } \frac{R_m}{2.7}$$

whichever gives the lower value.

2.2.4.2 Materials without specified elevated temperature values

- 1) For design temperatures up to and including 250 °C use material property data at 250 °C.
- 2) For design temperatures of 250 °C and above:

$$f_E = \frac{R_e(T)}{1.45} \text{ or } \frac{R_m}{2.7}$$

whichever gives the lower value.

The values of $R_{e(T)}$ used above shall be taken as equal to those specified for otherwise similar materials having specified elevated temperature values.

2.2.5 Time-dependent design stresses

The time-dependent design stress is given by

$$f_F = \frac{S_{Rt}}{1.3}$$

NOTE. By agreement between the manufacturer and the purchaser (see 1.5.2.2b), time-dependent values at design temperature, for 10 CrMo 9-10, 11 CrMo 9-10 and 12 CrMo 9-10 materials only, may be increased by up to 10 %, subject to both of the following conditions being met:

- a) continued fitness for service reviews are instituted at two-thirds of the agreed design lifetime (see 2.2.8);
- b) the resulting values do not exceed the lower time-independent stress value at the same temperature, derived in accordance with 2.2.3 or 2.2.4.

2.2.6 Cast steels

The design stress for castings tested in accordance with BS 1504 test category I shall not be greater than 90 % of the values given in table A.1, and for those tested in accordance with BS 1504 test category II shall not be greater than 80 % of the said value. The acceptance level of both test categories shall be in accordance with 5.7.5.

2.2.7 Design temperature

2.2.7.1 General

The design temperature shall not be less than the highest of:

- a) the maximum mean wall temperature of the pressure part, calculated by the manufacturer. In the case of pressure parts carrying superheated steam the factors specified in 2.2.7.8 shall be taken into account;
- b) the design temperatures specified in 2.2.7.2 to 2.2.7.7.

NOTE. The design temperature should not exceed the upper temperature for which properties are available. Where extrapolation of the available values is required, the extrapolation method should be agreed between the manufacturer and the purchaser (see 1.5.2.2d).

2.2.7.2 Drums and headers

Where drums and headers are not heated by hot gases or are adequately protected by insulation from hot gases, the design temperature shall be equal to the maximum temperature of the internal fluid.

NOTE. A covering of refractory or insulating material that may be liable to become dislodged is not considered to be adequate protection.

Where drums and headers are heated by hot gases, the design temperature shall be taken as not less than 25 °C in excess of the maximum temperature of the internal fluid.

2.2.7.3 Boiler tubes

The design temperature shall be taken as not less than:

- a) 50 °C for tubes subject to radiant heat from the combustion chamber;
- b) 25 °C for other tubes;

in excess of the saturated steam temperature at the calculation pressure.

2.2.7.4 Superheater and reheater tubes

The design temperature shall be taken as not less than:

- a) 50 °C for tubes subject to radiant heat from the combustion chamber;
- b) 35 °C for other tubes;

in excess of the steam temperature expected in the part being considered.

2.2.7.5 Economizer tubes

The design temperature shall be taken as not less than 25 °C in excess of the maximum temperature of the internal fluid.

2.2.7.6 Integral pipes and those parts of tubes not heated by gases

The design temperature shall be taken as the maximum temperature of the fluid inside the pipe or tube.

2.2.7.7 Parts beyond the final superheater header(s)

Where, because of design arrangements, factors such as inequalities in gas temperature and flow occurring in any cross section of the gas path cannot have any sustained effect on the steam temperature, the design temperature shall be the rated steam temperature at the superheater outlet, provided that in service:

- a) the average temperature during any one year of operation does not exceed the design temperature; and
- b) for systems having a rated temperature of 380 °C and below, fluctuations in temperature do not exceed the design temperature by more than 10 %; or
- c) for systems having a rated temperature above 380 °C:
 - 1) normal fluctuations in temperature do not exceed the rated temperature by more than 8 °C; and
 - 2) abnormal fluctuations in temperature do not exceed the rated temperature by more than 20 °C for a maximum of 400 h in any one year or 30 °C for a maximum of 100 h in any one year or 40 °C for a maximum of 60 h in any one year.

Where the maximum temperature will exceed these limits, the design temperature shall be increased by the amount of the excess.

NOTE. The limits in a), b) and c) represent realistic figures for modern control systems and boilers meeting normal load demands. It is the responsibility of the purchaser to advise the manufacturer, before design of the plant is commenced, of any circumstances that could prevent these limits being complied with (see 1.5.1a); it is the responsibility of the purchaser also to ensure that the boiler is operated within these or any other agreed limits.

2.2.7.8 Additional factors

2.2.7.8.1 In determining the maximum temperature of superheated steam and the related pressure part mean wall temperature, the following factors shall be taken into account:

- a) variation in the steam flow, from the mean value, in individual tubes of a tube bank, produced by entry and exit conditions and by manufacturing tolerances on thickness and diameter;
- b) variation in the heat input to the tube bank under consideration, resulting from deviation in the performance of preceding heating surfaces and deviation from ideal combustion conditions;
- c) variation in steam inlet temperature resulting from deviation in gas temperature and flow occurring in any cross section of the gas path.

2.2.7.8.2 For drums, headers and parts beyond the superheater header(s), where the expected steam temperature is not in excess of 425 °C in the part under consideration, an addition of 15 °C shall be deemed to cover the factors listed in 2.2.7.8.1.

2.2.7.8.3 For superheater and reheater tubes where, in the part under consideration:

- a) the tube is not subjected to direct radiant heat from the combustion chamber;
- b) the expected steam temperature is not in excess of 425 °C;

the factors listed in 2.2.7.8.1 shall be deemed to be included in the requirements of 2.2.7.4.

2.2.8 Design lifetime

A design lifetime related to the creep rupture properties of the material shall be agreed between the manufacturer and the purchaser for each component whose design temperature is such that the design stress is time-dependent (see 1.5.2.2c).

NOTE 1. it is not necessarily intended that one lifetime be used to cover all components; therefore replaceable items may be designed for lifetimes shorter than the general life expectancy of the boiler.

NOTE 2. The basis of the design in this standard is that no component remains in service after its operational hours have exceeded the design lifetime. However, experience has demonstrated that components designed to these rules can be expected to exceed their design life and reference may be made to PD 6510 for procedures for assessing whether the life could be extended.

Components of certain alloy materials designed using increased time-dependent stress values (see 2.2.5 or note 4 of table A.1) shall be identified on the drawings and listed by the manufacturer. Continued service reviews shall be instituted at not later than two-thirds of the design lifetime.

Table 2.1.2 List of materials covered by BS EN material standards

Product form	BS EN standard	Material description	Grade	Heat treatment ¹⁾	Thickness mm		Steel group according to EN ISO/TR 15608
					min.	max.	
Plate and strip	BS EN 10028-2	Elevated temperature properties	P235GH	N	0	150	1.1
Plate and strip	BS EN 10028-2	Elevated temperature properties	P265GH	N	0	150	1.1
Plate and strip	BS EN 10028-2	Elevated temperature properties	P295GH	N	0	150	1.2
Plate and strip	BS EN 10028-2	Elevated temperature properties	P355GH	N	0	150	1.2
Plate and strip	BS EN 10028-2	Elevated temperature properties	16Mo3	N	0	150	1.2
Plate and strip	BS EN 10028-2	Elevated temperature properties	13CrMo4-5 (1)	NT	0	60	5.1
Plate and strip	BS EN 10028-2	Elevated temperature properties	13CrMo4-5 (2)	NT Q	60	100	5.1
Plate and strip	BS EN 10028-2	Elevated temperature properties	13CrMo4-5 (3)	Q	100	150	5.1
Plate and strip	BS EN 10028-2	Elevated temperature properties	10CrMo9-10 (1)	NT	0	60	5.2
Plate and strip	BS EN 10028-2	Elevated temperature properties	10CrMo9-10 (2)	NT Q	60	100	5.2
Plate and strip	BS EN 10028-2	Elevated temperature properties	10CrMo9-10 (3)	Q	100	150	5.2
Plate and strip	BS EN 10028-2	Elevated temperature properties	11CrMo9-10 (1)	NT Q	0	60	5.2
Plate and strip	BS EN 10028-2	Elevated temperature properties	11CrMo9-10 (2)	Q	60	100	5.2
Plate and strip	BS EN 10028-3	Fine grain steel, normalized	P275N	N	0	150	1.1
Plate and strip	BS EN 10028-3	Fine grain steel, normalized	P275NH	N	0	150	1.1
Plate and strip	BS EN 10028-3	Fine grain steel, normalized	P355N	N	0	150	1.2
Plate and strip	BS EN 10028-3	Fine grain steel, normalized	P355NH	N	0	150	1.2
Plate and strip	BS EN 10028-3	Fine grain steel, normalized	P460NH	N	0	150	2.1
Tubes, seamless	prEN 10216-1	Room temperature properties	P195TR2	N	0	60	1.1
Tubes, seamless	prEN 10216-1	Room temperature properties	P235TR2	N	0	60	1.1
Tubes, seamless	prEN 10216-1	Room temperature properties	P265TR2	N	0	60	1.1
Tubes, seamless	prEN 10216-2	Elevated temperature properties	P195GH	N	0	16	1.1
Tubes, seamless	prEN 10216-2	Elevated temperature properties	P235GH	N	0	60	1.1
Tubes, seamless	prEN 10216-2	Elevated temperature properties	P265GH	N	0	60	1.1
Tubes, seamless	prEN 10216-2	Elevated temperature properties	16Mo3	N	0	60	1.2
Tubes, seamless	prEN 10216-2	Elevated temperature properties	10CrMo5-5	NT	0	60	5.1
Tubes, seamless	prEN 10216-2	Elevated temperature properties	13CrMo4-5	NT	0	60	5.1
Tubes, seamless	prEN 10216-2	Elevated temperature properties	10CrMo9-10	NT	0	60	5.2
Tubes, seamless	prEN 10216-2	Elevated temperature properties	11CrMo9-10	QT	0	60	5.2

Table 2.1.2 List of materials covered by BS EN material standards (continued)

Product form	BS EN standard	Material description	Grade	Heat treatment ⁽¹⁾	Thickness mm		Steel group according to EN ISO/TR 15608
					min.	max.	
Tubes, seamless	prEN 10216-2	Elevated temperature properties	15NiCuMoNb5-6-4	NT	0	80	2.1
Tubes, seamless	prEN 10216-2	Elevated temperature properties	X11CrMo5+NT1	NT	0	100	5.3
Tubes, seamless	prEN 10216-2	Elevated temperature properties	X11CrMo5+NT2	NT	0	100	5.3
Tubes, seamless	prEN 10216-2	Elevated temperature properties	X11CrMo9-1+NT	NT	0	60	5.4
Tubes, seamless	prEN 10216-2	Elevated temperature properties	X10CrMoVNb9-1	NT	0	100	6.4
Tubes, seamless	prEN 10216-2	Elevated temperature properties	X20CrMoNiV11-1	NT	0	100	6.4
Tubes, seamless	prEN 10216-3	Fine grain steel	P275NL1	N	0	100	1.1
Tubes, seamless	prEN 10216-3	Fine grain steel	P275NL2	N	0	100	1.1
Tubes, seamless	prEN 10216-3	Fine grain steel	P355NH	N	0	100	1.2
Tubes, seamless	prEN 10216-3	Fine grain steel	P460NH	N	0	100	2.1
Tubes, seamless	prEN 10216-5	Stainless steel					
Tubes, welded	prEN 10217-1	Room temperature properties	P195TR2	N	0	40	1.1
Tubes, welded	prEN 10217-1	Room temperature properties	P235TR2	N	0	40	1.1
Tubes, welded	prEN 10217-1	Room temperature properties	P265TR2	N	0	40	1.1
Tubes, welded	prEN 10217-2	Elevated temperature properties	PH195GH	N	0	16	1.1
Tubes, welded	prEN 10217-2	Elevated temperature properties	PH235GH	N	0	16	1.1
Tubes, welded	prEN 10217-2	Elevated temperature properties	PH265GH	N	0	16	1.1
Tubes, welded	prEN 10217-2	Elevated temperature properties	16Mo3	N	0	16	1.2
Tubes, welded	prEN 10217-3	Alloy fine grain steel	P275NL1	N	0	40	1.1
Tubes, welded	prEN 10217-3	Alloy fine grain steel	P275NL2	N	0	40	1.1
Tubes, welded	prEN 10217-3	Alloy fine grain steel	P355NH	N	0	40	1.2
Tubes, welded	prEN 10217-3	Alloy fine grain steel	P460NH	N	0	40	2.1
Tubes, welded	prEN 10217-5	Stainless steel					
Forging	prEN 10222-2	Elevated temperature properties	P245GH (1)	A	0	35	1.1
Forging	prEN 10222-2	Elevated temperature properties	P245GH (2)	N NT QT	35	160	1.1
Forging	prEN 10222-2	Elevated temperature properties	P280GH (1)	N	0	35	1.2
Forging	prEN 10222-2	Elevated temperature properties	P280GH (2)	NT QT	35	160	1.2
Forging	prEN 10222-2	Elevated temperature properties	P305GH (1)	N	0	35	1.2

Table 2.1.2 List of materials covered by BS EN material standards (continued)

Product form	BS EN standard	Material description	Grade	Heat treatment ¹⁾	Thickness mm		Steel group according to EN ISO/TR 15608
					min.	max.	
Forging	prEN 10222-2	Elevated temperature properties	P305GH (2)	NT	35	160	1.2
Forging	prEN 10222-2	Elevated temperature properties	P305GH (3)	QT	0	70	1.2
Forging	prEN 10222-2	Elevated temperature properties	16Mo3 (1)	N	0	35	1.2
Forging	prEN 10222-2	Elevated temperature properties	16Mo3 (2)	QT	35	100	1.2
Forging	prEN 10222-2	Elevated temperature properties	16Mo3 (3)	QT	100	500	1.2
Forging	prEN 10222-2	Elevated temperature properties	13CrMo4-5 (1)	NT	0	70	5.1
Forging	prEN 10222-2	Elevated temperature properties	13CrMo4-5 (2)	NT QT	70	500	5.1
Forging	prEN 10222-2	Elevated temperature properties	14MoV6-3	NT QT	0	500	6.1
Forging	prEN 10222-2	Elevated temperature properties	11CrMo9-10 (1)	NT	0	200	5.2
Forging	prEN 10222-2	Elevated temperature properties	11CrMo9-10 (2)	NT QT	200	500	5.2
Forging	prEN 10222-2	Elevated temperature properties	X16CrMo5-1 (1)	A	0	300	5.3
Forging	prEN 10222-2	Elevated temperature properties	X16CrMo5-1 (2)	NT	0	300	5.3
Forging	prEN 10222-2	Elevated temperature properties	X10CrMoVNb9-1	NT	0	130	6.4
Forging	prEN 10222-2	Elevated temperature properties	X20CrMoV11-1	QT	0	330	6.4
Forging	BS EN 10222-4	Fine grain, high proof strength	P285NH	N	0	70	1.2
Forging	BS EN 10222-4	Fine grain, high proof strength	P285QH	QT	70	400	1.2
Forging	BS EN 10222-4	Fine grain, high proof strength	P355NH	N	0	70	1.2
Forging	BS EN 10222-4	Fine grain, high proof strength	P355QH	QT	70	400	1.2
Forging	BS EN 10222-4	Fine grain, high proof strength	P420NH	N	0	70	2.1
Forging	BS EN 10222-4	Fine grain, high proof strength	P420QH	QT	70	400	2.1
Forging	prEN 10222-5	Stainless steel					
¹⁾ A	annealed	NT					
	normalized and tempered	Q					
N	normalized	QT					
	quenched and tempered	QT					



Section 3. Design

3.1 General

3.1.1 The thickness and other dimensions of pressure parts, sufficient to withstand the calculation pressure at the design temperature sustained (where relevant) for the design life time, shall be determined in accordance with this section.

NOTE 1. In determining the minimum permissible thickness in millimetres the calculated value may be rounded down to the next lowest single decimal place.

The design for loadings arising from the following shall also be determined in accordance with this section:

- a) the bending of a drum or header as a beam under self-weight and imposed loads;
- b) local support loads on drums;
- c) thermally-induced forces and moments within or arising from systems of integral piping; and
- d) local loading of tubes by structural attachments.

NOTE 2. The purpose of this section is to give specific design rules for common forms of loading to which boiler pressure parts are subject and general rules on how other loadings are to be considered. It does not give specific design methods for loadings other than those described in a) to d).

The design rules are adequate for boilers of established construction, installed and operated in accordance with the manufacturer's instructions.

In such cases, thermally-induced forces and moments within or arising from properly designed and erected tube systems do not require analysis.

Determination of the dimensions of pressure parts when any of the following conditions are present requires special consideration outside the scope of this standard:

- 1) abnormally corrosive products of combustion;
- 2) highly pressurized products of combustion;
- 3) poor feedwater;
- 4) abnormally rapid or frequent changes of pressure or temperature.

3.1.2 The strengths of the pressure parts shall be such as to withstand the following normally present loads:

- a) the weight of all pressure parts, contents, components depending on them and any superimposed slag, ash or dust;
- b) loads caused by pressure differentials over the flue gas containment envelope;
- c) loads arising at connections between the boiler system and other plant. The maximum values for forces and moments imposed by external piping at the boiler terminals for main steam, reheat steam and feedwater shall be agreed between manufacturer and purchaser (see 1.5.2.2e).

If applicable, the pressure parts shall adequately withstand wind and earthquake loads. Design parameters for such loads shall be agreed between the manufacturer and the purchaser (see 1.5.2.2f).

3.1.3 The results of any stress calculations carried out for loadings not explicitly covered by equations in this section shall be assessed using the criteria given in annexes A and B of BS 5500 : 1997.

Transient, uncontrolled pressure differentials over the flue gas containment envelope shall be treated as wind loads not coincident with true wind loads, when using annex B of BS 5500 : 1997.

3.1.4 The cyclic loadings to which boilers are normally subject are such that fatigue analysis of pressure parts made from materials listed in table 2.1.2 is unnecessary. When pressure parts are to be subject to abnormal cyclic loading, the assessment of the need for a fatigue analysis and any resulting analysis shall be in accordance with annex C of BS 5500 : 1997.

3.1.5 Cognizance shall be taken of the following requirements in sections 4 and 5:

- a) branches and nozzles (see 4.3.1.1.6 and 4.3.1.1.7);
- b) welds between dissimilar materials (see 4.3.2.5.4 and 4.5.4.6);
- c) the distance of nozzles or branches from locally heat treated butt welds (see 4.5.1.1.3);
- d) the length of branches to which pipes are to be butt welded and the weld stress relieved (see 4.5.1.1.4);
- e) design of joints subject to non-destructive examination (see 5.6.1.7);
- f) identification of load-bearing attachments (see 5.6.1.7, 5.8.2.3, 5.8.3.3 and 5.8.4.3.1);
- g) test category for pipes and tubes (see 5.7.4);

NOTE. Pipes and tubes conforming to test category 2 of BS 3059 : Part 2, BS 3602 : Part 1, BS 3604 : Part 1 and BS 3605 conform to this standard. Selection of pipes and tubes to test category 1 of these standards is at the option of the designer, taking into account the service conditions of the boiler.

- h) venting or evacuating of air prior to hydraulic testing (see 5.10.4.2). Small pockets of trapped air may remain after filling and the procedure with regard to these shall be the subject of agreement between the Inspecting Authority and the manufacturer (see 1.5.2.3r).

3.2 Metal wastage

NOTE. For the purpose of design in accordance with this section, metal wastage includes oxidation, corrosion, erosion and abrasion.

3.2.1 Internal wastage is normally small and shall not be considered for boilers operated with water in accordance with BS 2486.

NOTE. Suitable means should be provided within the boiler design to enable regular water analysis to be safely undertaken in order to ensure the water quality remains within acceptable limits.

3.2.2 External wastage of pressure parts not exposed to flue gases is normally small and the thickness determined by this section shall be adequate without further addition.

Tubes exposed to flue gases experience wastage to a varying extent. If the boiler design data indicate that wastage will be significant, the tubes shall be increased in thickness (see 3.7.1.2). It is permitted to provide for wastage by means of metallurgically bonded co-extruded tubes.

NOTE. In such cases it may be necessary for the manufacturer and the purchaser to agree on a design life for components designed to time-independent stresses as well as for those for time-dependent stresses (see 2.2.4).

For tubes designed to time-dependent stresses, integration of the effects of creep and wastage shall be permitted so that failure is predicted at a time not less than the design lifetime even though the tube thickness over the latter part of its life is less than that required by equations 31 and 32 (see 3.7.1.1).

3.2.3 Unless exceptional conditions of chemical concentration are expected to occur, only austenitic materials shall be considered liable to the effect of stress corrosion. The phenomenon shall be considered adequately dealt with in the design if austenitic materials are not used to carry steam having at full boiler load an enthalpy less than 2900 kJ/kg or that corresponding to 425 °C, whichever is less.

Where exceptional chemical concentrations are predicted, consideration shall be given to the materials and heat treatment of components.

NOTE. It is not possible to provide compensation for stress corrosion by increasing the thickness of components.

3.3 Cylindrical shells of drums and headers

NOTE. This clause specifies the design rules for drums and headers. An example of the design of a boiler drum is given in annex B.

3.3.1 Shell thickness

3.3.1.1 The thickness of the shells of drums and headers shall be the greatest of those required by the following:

- a minimum of 9.5 mm for headers 300 mm and above in outside diameter and a minimum of 6 mm for headers below 300 mm in outside diameter;
- the requirements of 3.3.2 and 3.3.3;
- the requirements of 3.3.4, 3.3.5 and 3.3.6;
- for those parts of shells drilled for tube expansions, such thickness as is required to provide a parallel belt of tube seat not less than 10 mm wide.

3.3.1.2 All welding of the main seams of boilers constructed in accordance with this standard shall be accounted as having a joint efficiency of 100 %.

3.3.2 Basic calculation

3.3.2.1 Except where the conditions described in 3.3.2.2 apply, the minimum thickness t (in mm) of a cylindrical shell shall be determined by either of the following equations:

$$t = \frac{pD_i}{2f\eta - p} \quad (1)$$

or

$$t = \frac{pD_o}{2f\eta + p} \quad (2)$$

where

- p is the calculation pressure (in N/mm²);
- D_i is the nominal inside diameter of the shell (in mm) (see 3.3.2.3 and 3.3.2.4);
- D_o is the nominal outside diameter of the shell (in mm) (see 3.3.2.3 and 3.3.2.4);
- f is the design stress (in N/mm²) (see 2.2);
- η is the minimum efficiency of longitudinal, diagonal and circumferential ligaments, as determined in 3.3.3.

3.3.2.2 The minimum thickness determined in accordance with 3.3.2.1 shall not be applicable in the following cases.

- For time-independent (non-creep) conditions where the resultant thickness calculated using equation (1) or (2) exceeds one-quarter of the internal diameter. In this case, the thickness shall also be determined from either of the following equations and compared with the thickness determined from equation (1).

$$t = \frac{D_i}{2} \left[\sqrt{\left\{ \frac{1.5f\eta + p(1 - \eta)}{1.5f\eta - p(1 + \eta)} \right\}} - 1 \right] \quad (3a)$$

$$t = \frac{D_o}{2} \left[1 - \sqrt{\left\{ \frac{1.5f\eta - p(1 + \eta)}{1.5f\eta + p(1 - \eta)} \right\}} \right] \quad (3b)$$

The required thickness shall be not less than the greater of the thicknesses determined from equation (1) or (3a), or (2) or (3b).

- For time-dependent (creep) conditions where the resultant thickness exceeds one-half of the internal diameter. In such cases, the method of calculating the thickness shall be subject to agreement between the purchaser and the manufacturer (see 1.5.2.3d).

3.3.2.3 Where a shell has tube and wrapper plates of different thicknesses and/or different radii, twice the respective radii shall be used for D_i or D_o , in equation (1), (2), (3a) or (3b).

3.3.2.4 Manufacturing tolerances in shell inside or outside diameters shall not be taken into account in calculating shell thickness.

3.3.3 Ligament efficiency

3.3.3.1 Limits of application

Where openings such as tube holes are drilled in a definite pattern, the efficiency of the ligaments shall be determined by the methods given in **3.3.3.2** to **3.3.3.7**, provided that the diameter of the largest hole in the group does not exceed that permitted by the following.

a) The maximum diameter d (in mm) of any opening designed by this method shall be obtained from the following equation:

$$d = 8.08 \{(1 - K) D_o t\}^{1/3} \quad (4)$$

where

D_o is the nominal outside diameter of the shell (in mm);

t is the actual thickness of the shell (in mm);

K is the factor given by:

$$K = \frac{p D_o}{1.82 f t} \text{ (but not exceeding 0.99)}$$

where

p is the calculation pressure (in N/mm²);

f is the design stress (in N/mm²) (see **2.2**).

For elliptical or obround²⁾ holes, d shall refer to the mean of the major or minor axes.

b) No opening designed by this method shall exceed 200 mm diameter.

Openings larger than those permitted in a) and b) shall be compensated in accordance with **3.4**.

3.3.3.2 Equivalent hole diameter

Where a drum or header is drilled for tube stubs fitted by welding either in line or in staggered formation, the equivalent diameter of the holes d_e (in mm) shall be taken as:

$$d_e = d_a - \frac{A}{t} \quad (5)$$

where

d_a is the actual diameter of the hole (in mm);

t is the thickness of the shell (in mm);

A is the compensating area provided by each tube stub and its welding fillets (in mm²).

The compensating area A shall be measured in a plane through the axis of the tube stub parallel to the longitudinal axis of the drum or header and shall be calculated as follows (see figure **3.3.3.2a**).

a) The sectional area of the stub, in excess of that required by equation (31) or (32) for the minimum tube thickness disregarding the minimum thickness given in table 3.7.1.1, from the interior surface of the shell up to a distance b from the outer surface of the shell (see note).

b) Plus the sectional area of the stub projecting inside the shell within a distance b from the inner surface of the shell.

c) Plus the sectional area of the welding fillets inside and outside the shell.

For a) and b) above

$$b = \sqrt{d_a t_a};$$

where

t_a is the actual thickness of the tube stub (in mm).

NOTE. In the case of a set-on tube stub (see figure 3.3.3.2b), the compensation extends from the outer surface of the shell only.

Where the material of the tube stub has an allowable stress lower than that of the shell, the compensating sectional area of the stub shall be multiplied by the ratio:

$$\frac{\text{Design stress of stub at design temperature}}{\text{Design stress of shell at design temperature}}$$

No credit shall be taken for the additional strength of tube stub material having a higher stress value than that of the shell plate.

3.3.3.3 Regular drilling

Where the distance between centres of adjacent tube holes is constant:

$$\eta = \frac{s - d}{s} \quad (6)$$

where

s is the pitch of tube holes (in mm);

d is either the mean effective diameter of the tube holes after allowing for any serrations, counter-boring or recessing, or the equivalent diameter (d_e) after allowing for the compensating effect of the tube stub (in mm) (see **3.3.3.2**).

²⁾ An obround opening is, for the purposes of this standard, to be understood as an opening or hole that is neither circular nor elliptical; it is formed by the blending of two radii to give a rounded opening/hole, the major and minor axes of which are different.

3.3.3.4 Irregular drilling

Where the distance between centres of adjacent tube holes is not constant (see figure 3.3.3.4):

$$\eta = \frac{s_1 + s_2 - 2d}{s_1 + s_2} \quad (7)$$

where

- s_1 is the shorter of any two adjacent pitches (in mm);
- s_2 is the longer of the two adjacent pitches (in mm);
- d is either the mean effective diameter of the tube holes after allowing for any serrations, counter-boring or recessing, or the equivalent diameter (d_e) after allowing for the compensating effect of the tube stub (in mm) (see 3.3.3.2).

When applying equation (7), the double pitch ($s_1 + s_2$) chosen shall be that which makes η a minimum, and in no case shall s_2 be taken as greater than twice s_1 .

3.3.3.5 Diagonal ligaments

3.3.3.5.1 Where bending stresses due to weight are negligible (otherwise refer to 3.3.4) and tube holes are arranged either along a diagonal line with respect to the longitudinal axis as shown in figure 3.3.3.5-1, or in a regular pattern as shown in figure 3.3.3.5-2, the efficiency η to be applied in equation (1) shall be obtained from the series of curves given in figure 3.3.3.5-3 with the ratio b/a as abscissa and the ratio $(2a - d)/2a$ or d/a as parameter;

where

- a and b are measured as shown in figures 3.3.3.5-1 and 3.3.3.5-2 on the median line of the plate;
- d is either the mean effective diameter of the tube holes after allowing for any serrations, counter-boring or recessing, or the equivalent diameter (d_e) after allowing for the compensating effect of the tube stub (in mm) (see 3.3.3.2).

NOTE. The data in figure 3.3.3.5-3 are based on the following:

$$\eta = \frac{2}{A + B + \sqrt{(A - B)^2 + 4C^2}}$$

where

$$A = \frac{\cos^2 \alpha + 1}{2 \left(1 - \frac{d \cos \alpha}{a} \right)}$$

$$B = \frac{1}{2} \left(1 - \frac{d \cos \alpha}{a} \right) (\sin^2 \alpha + 1)$$

$$C = \frac{\sin \alpha \cos \alpha}{2 \left(1 - \frac{d \cos \alpha}{a} \right)}$$

$$\cos \alpha = \frac{1}{\sqrt{\left(1 + \frac{b^2}{a^2} \right)}}$$

$$\sin \alpha = \frac{1}{\sqrt{\left(1 + \frac{a^2}{b^2} \right)}}$$

where

α is the angle of centre line of cylinder to centre line of diagonal holes.

3.3.3.5.2 Where there is regular staggered spacing of tube holes as shown in figure 3.3.3.5-4 the smallest value of the efficiency of all the ligament (longitudinal, circumferential and diagonal) shall be obtained from figure 3.3.3.5-5 with

the ratio S_c/S_l , as abscissa and the ratio $\frac{S_l - d}{S_l}$ or d/a

as parameter:

where

- d is either the mean effective diameter of the tube holes after allowing for any serrations, counter-boring or recessing, or the equivalent diameter (d_e) after allowing for the compensating effect of the tube stub (in mm) (see 3.3.3.2);
- S_c is twice the distance (b) between axial rows of holes measured on the median line of the plate (in mm);
- S_l is twice the distance (a) between circumferential rows of holes (in mm).

3.3.3.6 Irregular drilling not in a straight line

For holes placed longitudinally along a drum or header which do not come in line, equation (7) in 3.3.3.4 shall apply except that an equivalent longitudinal width of the diagonal ligament shall be used. An equivalent longitudinal width shall be that width which gives, using equation (6), the same efficiency as would be obtained using figure 3.3.3.5-3 for the diagonal ligament in question.

3.3.3.7 Circumferential ligaments

3.3.3.7.1 Where bending stresses due to weight are negligible (otherwise, refer to 3.3.4) the efficiency of ligaments between holes measured on a circumferential line shall not be used to calculate the thickness of a cylindrical shell provided that the efficiency of the circumferential ligaments calculated in accordance with equations (6) and (7) is not less than one-half of that of the longitudinal ligaments.

3.3.3.7.2 Where the circumferential pitch between tube holes measured on the mean of the external and internal drum or header diameters is such that the circumferential ligament efficiency determined by equations (6) and (7) is less than one-half of the ligament efficiency on the longitudinal axis, then η in equations (1) to (3b) shall be taken as twice the circumferential efficiency.

3.3.4 Combined stresses in drum or header shells

3.3.4.1 General

Notwithstanding the thickness of shells calculated in accordance with 3.3.2 and 3.3.3, calculations shall be made to ensure that in no case does the average ligament stress resulting from the combination of stresses arising from internal pressure, the self-weight of the drum or header and its contents, and all externally applied loads, exceed the design stress f (see 2.2) in any line of holes whether longitudinal, circumferential or diagonal.

The average ligament stress in ligaments between holes on a circumferential line shall be the sum of the longitudinal stresses calculated in accordance with 3.3.4.2 and 3.3.4.3. The average ligament stress in ligaments between holes on a diagonal line shall be calculated in accordance with 3.3.4.4.

3.3.4.2 Longitudinal direct pressure stress

The maximum direct longitudinal stress f_d (in N/mm²) due to internal pressure on the drum or header ends shall be calculated as follows:

$$f_d = \frac{pD_i^2}{1.273A} \quad (8)$$

where

- p is the calculation pressure (in N/mm²);
- D_i is the nominal inside diameter of the drum or header (in mm);
- A is the net cross-sectional area of the drum or header shell plate taken through the tube holes in a plane at right angles to its axis (in mm²).

When a hole in the shell is wholly or partly reinforced by the attaching nozzle the hole size shall be reduced accordingly (see 3.3.3.2).

Nozzles designed in accordance with 3.4 shall be considered as wholly reinforced.

3.3.4.3 Longitudinal bending stress

3.3.4.3.1 The resultant bending moment M_R (in N-mm) at any section shall be the algebraic sum of the resultant bending moments due to the eccentricity of the end pressure M_E and due to externally applied loads M_W , i.e.

$$M_R = M_E + M_W \quad (9)$$

The resultant bending moment M_E (in N-mm) due to eccentricity of end pressure shall be calculated from:

$$M_E = \frac{p D_i^2 e}{1.273} \quad (10)$$

where

- p is the calculation pressure (in N/mm²);
- D_i is the nominal inside diameter of the drum or header (in mm);
- e is the eccentricity of the net cross section (see figure B.3).

The resultant bending moment M_W (in N-mm) due to externally applied loads shall be calculated by treating the shell as a beam carrying the externally applied loads including its own weight and that of the contents under working conditions. M_W is the bending moment at the section subjected to the greatest bending from this cause.

3.3.4.3.2 The stress due to bending f_b (in N/mm²) shall be calculated as follows:

$$f_b = \frac{M_R Y}{I_A} \quad (11)$$

where

M_R is the resultant bending moment (in N-mm) (see 3.3.4.3.1);

Y is the distance from the neutral axis of the net cross section to the extreme mid point of the shell thickness (in mm);

I_A is the second moment of area of the net cross section through the tube holes taken about its neutral axis (in mm⁴) (see also 3.3.4.3.3).

If a welded nozzle has been designed to compensate fully for the removal of drum material (i.e. when $\eta = 1.0$) the nozzle and its opening shall be assumed to be absent in calculating the longitudinal bending stress. If the nozzle is designed to compensate only for part of the material removed from the drum, the second moment of area of the drum section shall be calculated assuming the size of the hole to be reduced by the amount of compensation provided by the nozzle (see 3.3.3.2).

3.3.4.3.3 In calculating the longitudinal stress in a drum supported on stools or slings and connected to a second drum by a bank of tubes (so arranged as to form substantial struts or ties between the drums), an increase in the value of the second moment of area I_A (in mm⁴) used in 3.3.4.3.2 shall be permitted to take account of the support received from the second drum as follows:

$$I_A = I_b + SI_c$$

where

I_b is the second moment of area of the supported drum (in mm⁴);

I_c is the second moment of area of the second drum (in mm⁴);

S is a factor given by:

$$S = 1 - \frac{a^2}{240} \quad (12)$$

where

a is the angle between the vertical and the line joining the centres of the two drums (in degrees);

where a is equal to or greater than 15.5°, S shall be taken as 0.

In no case shall the actual value of l_A used in equation (11) be taken as more than 1.33 times the second moment of area of the supported drum l_b .

NOTE. In this context, 'bank of tubes' means four or more rows of tubes extending over at least three-quarters of the drum length between supports and pitched longitudinally at not greater than an average pitch of four tube diameters.

3.3.4.4 Evaluation of stresses

Where bending stresses due to weight, externally applied loads and eccentricity of end pressure are not negligible, the stress on ligaments which are on a line which is at an angle a to the longitudinal axis of the drum, as shown in figure 3.3.4.4, shall be evaluated as follows:

Let the circumferential stress on gross area of longitudinal section σ_1 (in N/mm^2) be:

$$\sigma_1 = p \frac{(D_i + t)}{2t}$$

where

- p is the calculation pressure (in N/mm^2);
- D_i is the nominal inside diameter of the shell (see 3.3.2.3 and 3.3.2.4) (in mm);
- t is the minimum thickness of the shell before allowance for metal wastage (see equation (1)) (in mm).

Let the longitudinal stress on gross area of circumferential section σ_2 (in N/mm^2) be:

$$\sigma_2 = \eta_2 (f_d + f_b)$$

where

- f_d is the stress due to pressure end load (see equation (8)) (in N/mm^2);
- f_b is the bending stress due to eccentricity of pressure end load added algebraically to bending stress due to weight loading calculated in accordance with equation (11) taking Y as the distance from the neutral axis of the net cross section to the mid point of the inclined ligament (in N/mm^2);
- η_2 is the ligament efficiency on a circumferential section at the point where f_d and f_b act.

Where the ligament under consideration is between holes of equal size, the actual ligament efficiency (η_3) on the diagonal line shall be given by:

$$\eta_3 = 1 - \frac{d \cos a}{a}$$

Where holes are of unequal size, the ligament efficiency (η_3) on the diagonal line shall be given by:

$$\eta_3 = 1 - \frac{(d_1 + d_2) \cos a}{2a}$$

where the symbols are as shown on figure 3.3.4.4.

The average direct stress A_1 (in N/mm^2) on the ligament shall be given by:

$$A_1 = \frac{1}{\eta_3} \left\{ \frac{\sigma_1 + \sigma_2}{2} + \frac{(\sigma_1 - \sigma_2) \cos 2a}{2} \right\}$$

The average transverse stress B_1 (in N/mm^2) on the ligament shall be given by:

$$B_1 = \eta_3 \left\{ \frac{\sigma_1 + \sigma_2}{2} - \frac{(\sigma_1 - \sigma_2) \cos 2a}{2} \right\}$$

The average shear stress C_1 (in N/mm^2) on the ligament shall be given by:

$$C_1 = \frac{(\sigma_1 - \sigma_2) \sin 2a}{2\eta_3}$$

The stress f (in N/mm^2) on the ligament shall be given by:

$$f = \frac{1}{2} [A_1 + B_1 + \sqrt{A_1^2 - B_1^2 + 4C_1^2}] \quad (13)$$

and this shall not exceed the design stress (see 2.2).

3.3.5 Boiler drum supports

Where large diameter but relatively thin drums are carrying heavyloads on stools or slings, equation (14) shall be used as a guide to assess whether precise calculations are needed to determine local stresses:

$$0.78 \left(\frac{W}{St} \right) \sqrt{\left(\frac{R}{t} \right)} \leq f \quad (14)$$

where

- R is the mean radius of the drum shell (in mm);
- t is the thickness of the drum shell (in mm);
- W is the load on the stool or sling (in N);
- S is the horizontal length of the stool or sling at right angles to axis of the drum but in any case not greater than the chord described by 120° (in mm);
- f is the design stress at the design temperature (in N/mm^2).

Where application of this equation suggests that the design stress is exceeded, reference shall be made to the appropriate methods of analysis given in annex G of BS 5500: 1997.

3.3.6 Other stresses in cylindrical shells

In addition to the requirements of 3.3.2 and 3.3.4, which cover general membrane stresses in the shell, and 3.3.5, which covers local stresses at supports, any other local stresses shall also be considered. Local stresses shall not be included when applying 3.3.4. Local stresses at nozzles carrying significant load shall be accounted for as required by 3.4.4.1.

3.4 Openings and branches in cylindrical shells and integral pipes

NOTE. This clause specifies the design rules for openings and branches in cylindrical shells. Recommendations on the forms of connections to be used for branches are given in annex C, which additionally covers studded and welded sockets.

3.4.1 General

3.4.1.1 The amount of compensation to be provided at an opening for which the ligament efficiency method of **3.3.3** cannot be applied shall be not less than that specified in **3.4.1.2** to **3.4.1.1**.

3.4.1.2 Design charts are referred to in **3.4.4.3** that are based on approximate analysis considering internal pressure loading only, but the effects of other loads shall be taken into account.

NOTE 1. The basis on which the design charts (figures 3.4.4-1 to 3.4.4-3) are founded is outlined in annex D.

NOTE 2. The effects of loading other than that from internal pressure may be taken into account by the selection of an appropriate value for the factor C (see **3.4.2** and **3.4.4**).

Similarly, the effects of attachments and discontinuities in the proximity of the openings shall be taken into account and in no case shall branch connections be less than the thickness specified in **3.4.7**. Where material having a lower design stress than that of the shell is taken as compensation, its effective area shall be assumed to be reduced in the ratio of the allowable design stresses at the design temperature, except that in no case shall a material having an allowable stress less than 60 % of the shell material be used. Material having a design stress up to 50 % greater than that for the shell may be used but no credit shall be taken for this in calculating the necessary effective area of such compensation.

All branch connections, nozzles and openings not covered in **3.4.3** shall be designed on the basis of special analysis, experimental evidence, or tests to the satisfaction of the purchaser. The alternative method given in annex E shall be permitted where experience has shown it to be satisfactory.

NOTE 3. All dimensions exclude any wastage allowances.

3.4.2 Notation

For the purposes of the calculations in **3.4.3** to **3.4.6** the following symbols apply (see also figures 3.4.2-1, 3.4.2-2, 3.4.4-1 to 3.4.4-6, 3.4.4-9 to 3.4.4-11 and 3.4.5-2):

A_s , A_n and A_t are cross-sectional areas used in the calculation of compensation for adjacent branches (see figure 3.4.5-2)

C is a factor applied to T_r/T

D is the mean diameter of the cylindrical shell

d is the bore of an opening not provided with a branch, or the mean diameter of a branch; in the case of non-circular openings (see **3.4.4.6**).

d_A is the average value of d for any two adjacent openings being considered

f_n is the design strength of the rim or forging

f_s is the design stress of the shell

g is the arrangement factor from figure 3.4.5-1

H is the distance along the shell within which the shell thickening is assumed to contribute to the reinforcement of an opening

h is the distance along the branch within which branch thickening is assumed to contribute to the reinforcement of an opening

P is the pitch measured between the centrelines of two openings along mid-thickness of shell

T is the shell thickness as calculated using either equations in **3.3.2** with $\eta = 1.0$ or equations (31) or (32) in **3.7.1**

T_r is the total thickness of the shell as required throughout **3.4**

T'_r is the modified thickness of the shell (see figures 3.4.4-7 and 3.4.4-8)

t is the branch thickness as calculated using the branch design stress by the equations for pressure loading only in **3.7.1** for a plain cylinder

t_r is the total thickness of the branch as required throughout **3.4**

t'_r is the modified thickness of the branch (see figures 3.4.4-7 and 3.4.4-8)

Y is a factor from figure 3.4.4-2

$$\rho = \frac{d}{D} \sqrt{\left(\frac{D}{2T_r}\right)}$$

3.4.3 Application

The rules specified in **3.4.4** to **3.4.9** shall be regarded as valid for the design of circular, elliptical and obround³⁾ openings and nozzles (including oblique nozzles) arranged singly or in groups, in cylindrical shells, provided that the mean diameter to thickness ratio of the shell D/T satisfies $D/T \leq 100$ and provided that the following conditions are satisfied.

a) *Openings and nozzles normal to the shell.* The ratio of the major to minor axes of the opening does not exceed 2.

b) *Oblique nozzles.* The nozzle is of circular cross section and the angle between the axis of the nozzle and a line normal to the shell surface does not exceed 50°.

c) *Protruding nozzles.* The mean diameter of the branch does not exceed one-third of the mean diameter of the shell.

³⁾ See footnote to **3.3.3.1** for an explanation of 'obround'.

d) *All nozzles.* The nozzle connection is made by fully penetrating welds.

NOTE. This requirement therefore excludes connections similar to those shown in figure C.10 and C.11 and the fillet welded connections shown in figure C.20.

Nozzle connections made with other than fully penetrating welds shall be considered as openings not fitted with branch pieces (see **3.4.4.2**), the opening being that formed in the shell before the branch is attached.

NOTE. In cases where the design stress is time-dependent, the requirements of this clause should give adequate margins against creep rupture. However, for parts made from austenitic materials, it may be desirable to check that the end of life deformation is acceptable.

3.4.4 Design of isolated openings and branch connections

3.4.4.1 General

The thickness of shell, T_r , shall be not less than T and the thickness of a branch connection, t_r , shall be not less than t . Where external loads are negligible, the factor C shall be taken as not more than 1.1.

Where branch or nozzle connections are attached to a piping system designed with due allowance for expansion thrusts, etc. (e.g. conforming to the flexibility requirements of BS 806) C shall be taken as not more than 1.0.

Other loadings shall be evaluated in accordance with annex G of BS 5500 : 1997 and the thickness of shell or branch increased if necessary, such as by selection of a value of C less than 1.0.

For vessels operating in the creep range, $C \leq 1$.

3.4.4.2 Openings not fitted with branches

The value $t_r/T_r = 0$ shall be taken in figure 3.4.4-1 and the factor $C = 1.1$ used to obtain the shell thickness T_r . An iteration shall be performed using a revised value of ρ until the assumed and required values of T_r are equal.

3.4.4.3 Openings fitted with branches

Branch connections shall be permitted to be reinforced by means of an increase in shell thickness or nozzle thickness or by a combination of increases. In some cases, e.g. where external pipework loads are significant, or in order to meet the branch thickness limit specified in **3.4.4.4c**, thickening of the shell may be necessary. This will need to be considered at an early stage in the design process. The following design charts shall apply:

- figure 3.4.4-1, for flush nozzles with $d/D < 0.3$;
- figure 3.4.4-2, for flush nozzles with $d/D > 0.2$;
- figure 3.4.4-3, for protruding nozzles with $d/D < 0.33$.

A value of shell thickness $T_r \geq T$ shall be chosen, and an estimate of the branch thickness $t_r \geq t$ made so that the mean branch diameter d and hence d/D and

(where necessary) ρ can be calculated. Using an appropriate value of C from **3.4.4.1**, the appropriate design chart shall be entered to give the required value of t_r/T_r (figures 3.4.4-1 and 3.4.4-3) or Y (figure 3.4.4-2) from which t_r shall be determined. In the case of flush nozzles in cylindrical vessels where $0.2 < d/D < 0.3$, it is necessary to derive a value t_{r1} from figure 3.4.4-1 and t_{r2} from figure 3.4.4-2 and interpolate so that:

$$t_r = t_{r1} + 10(d/D - 0.2)(t_{r2} - t_{r1})$$

If the value of t_r obtained differs from the estimate a revised value shall be used and the calculation repeated until convergence is achieved. Figures 3.4.4-1, 3.4.4-2 and 3.4.4-3 are provided for ease of application in manual calculations. Definitive thicknesses shall be obtained from the tabulated data in tables 3.4.4a and 3.4.4b. Where it is required to determine t_r/T_r for CT_r/T and ρ values that are not explicitly tabulated, the following procedure shall be used.

- 1) For the nearest listed ρ value below that required, find the nearest tabulated CT_r/T values bounding the exact value above and below. Interpolate linearly to obtain t_r/T_r corresponding to the required value of CT_r/T .
- 2) Similarly obtain t_r/T_r for the exact CT_r/T value for the nearest tabulated ρ value greater than that required.
- 3) Finally interpolate linearly to obtain t_r/T_r for the exact value of ρ .

Extrapolation of the data beyond the limits presented is not allowed. No reinforcing of the branch is required if the chosen CT_r/T value for the vessel is greater than CT_r/T , as read from the relevant figure, corresponding to $t_r/T_r = 0$ and for the relevant value of ρ . In this case the thickness of the branch is determined by **3.4.7**.

3.4.4.4 Limits on reinforcement

The extent of reinforcement shall be as follows.

- a) Where reinforcement is provided by uniform branch thickening, the thickness t_r of the branch and protrusion, if fitted, shall not be reduced within a distance $h = \sqrt{dt_r}$ measured from the relevant surface of the shell of thickness T_r (see figures 3.4.4-4, 3.4.4-5 and 3.4.4-6).

It is permissible to modify the distribution of reinforcement so as to concentrate the material close to the opening in the shell. For flush nozzles one-half of the total cross-sectional area of the branch walls falling within the distance h as calculated and measured above shall be not less than ht_r (see figure 3.4.4-8). For protruding nozzles, the corresponding cross-sectional area shall be not less than $2ht_r$ and the reinforcement shall be (approximately) equally disposed about the shell mid-thickness (see figure 3.4.4-7). The value of h is established for the uniform branch thickening case and is not recalculated if t_r is varied.

b) Where reinforcement is provided by uniform shell thickening, the thickness T_r of the shell shall not be reduced within a distance H measured from the outer surface of the branch piece of thickness t_r or the bore of the opening if no branch is fitted (see figures 3.4.4-6, 3.4.4-9 and 3.4.4-10) where H is the smaller of $H = d/2$ or $H = \sqrt{DT_r}$.

It is permissible for reinforcing material to be concentrated close to the opening. In this case one-half of the total cross-sectional area taken between the outermost extremity of the dimension H on one side of the nozzle and a similar point on the opposite side of the nozzle, but excluding any area included in a), shall be not less than $T_r (H + t_r)$. The value of H is to be calculated and measured as above (see figures 3.4.4-7 and 3.4.4-8) and once established is not recalculated.

c) The required thickness of the branch t_r (except for studded pads, see 3.4.8) shall be less than $(2 - d/D) T_r$. Where the distribution of the reinforcement has been concentrated close to the opening as permitted in a) and b) the modified thicknesses t'_r and T'_r (see figures 3.4.4-7 and 3.4.4-8) shall be substituted in this relationship as appropriate.

d) The transitions between sections of shell of different thicknesses shall be achieved by means of smooth tapers no steeper than 1 : 4. The transitions between sections of branch or nozzle connections of different thicknesses shall be smoothly tapered.

Table 3.4.4a Design values of t_r/T_r for flush nozzles in cylindrical shells ($0 < d/D < 0.3$)

See also figure 3.4.4-1

ρ	2.0	1.8	1.6	1.4	1.2	1.0	0.9	0.8	0.7	0.6	0.5	0.4	0.3	0.2	0.1
CT_r/T															
3.1	0.04														
3.0	0.11														
2.9	0.15	0.06													
2.8	0.19	0.13													
2.7	0.23	0.17	0.08												
2.6	0.27	0.22	0.13												
2.5	0.31	0.26	0.18	0.06											
2.4	0.35	0.30	0.22	0.12											
2.3	0.40	0.35	0.27	0.18	0.07										
2.2	0.45	0.39	0.32	0.23	0.15										
2.1	0.50	0.45	0.38	0.30	0.22	0.07									
2.0	0.56	0.50	0.44	0.36	0.27	0.16	0.06								
1.9	0.63	0.56	0.49	0.42	0.33	0.23	0.14	0.04							
1.8	0.70	0.63	0.56	0.48	0.39	0.28	0.21	0.14	0.03						
1.7	0.80	0.71	0.63	0.54	0.44	0.33	0.26	0.21	0.12						
1.6	0.92	0.82	0.72	0.62	0.51	0.38	0.32	0.27	0.20	0.11					
1.5	1.06	0.95	0.83	0.72	0.59	0.46	0.38	0.32	0.26	0.20	0.09				
1.4	1.23	1.11	0.98	0.84	0.70	0.55	0.46	0.39	0.33	0.27	0.19	0.03			
1.3	1.44	1.30	1.17	1.02	0.84	0.67	0.57	0.50	0.41	0.34	0.26	0.16			
1.2	1.77	1.60	1.42	1.26	1.05	0.85	0.72	0.61	0.50	0.42	0.34	0.26	0.14		
1.1			1.81	1.57	1.33	1.10	0.95	0.79	0.65	0.53	0.43	0.35	0.25	0.10	
1.0				1.96	1.68	1.42	1.22	1.04	0.82	0.66	0.54	0.45	0.34	0.22	0.10

Table 3.4.4b Design values of t_r/T_r for protruding nozzles in cylindrical shells ($0 < d/D < 0.33$)															
See also figure 3.4.4-3															
ρ	2.0	1.8	1.6	1.4	1.2	1.0	0.9	0.8	0.7	0.6	0.5	0.4	0.3	0.2	0.1
CT_r/T															
3.2															
3.1	0.04														
3.0	0.11														
2.9	0.15	0.06													
2.8	0.19	0.13													
2.7	0.22	0.17	0.07												
2.6	0.25	0.21	0.13												
2.5	0.29	0.24	0.17	0.06											
2.4	0.33	0.28	0.21	0.13											
2.3	0.36	0.31	0.25	0.18	0.04										
2.2	0.40	0.35	0.29	0.22	0.13										
2.1	0.45	0.39	0.33	0.26	0.18	0.00									
2.0	0.49	0.44	0.37	0.30	0.23	0.12									
1.9	0.55	0.49	0.42	0.35	0.28	0.19	0.11								
1.8	0.61	0.54	0.46	0.39	0.32	0.25	0.20	0.10							
1.7	0.68	0.61	0.52	0.45	0.37	0.30	0.26	0.19	0.07						
1.6	0.75	0.68	0.59	0.51	0.42	0.35	0.31	0.26	0.17	0.04					
1.5	0.85	0.77	0.68	0.58	0.49	0.40	0.36	0.31	0.25	0.15	0.02				
1.4	0.95	0.88	0.79	0.68	0.58	0.47	0.43	0.37	0.32	0.24	0.15	0.01			
1.3	1.08	1.00	0.91	0.80	0.68	0.56	0.51	0.45	0.39	0.33	0.25	0.14			
1.2	1.22	1.14	1.04	0.93	0.81	0.68	0.61	0.55	0.48	0.41	0.34	0.25	0.14		
1.1	1.37	1.29	1.19	1.08	0.96	0.82	0.75	0.67	0.58	0.51	0.43	0.35	0.25	0.09	
1.0	1.53	1.44	1.35	1.24	1.13	0.99	0.91	0.81	0.71	0.62	0.52	0.44	0.34	0.21	0.06

Table 3.4.4c Design values of CT_r/T at $t_r/T_r = 0$ for nozzles in cylindrical shells															
Type of nozzle	ρ														
	2.0	1.8	1.6	1.4	1.2	1.0	0.9	0.8	0.7	0.6	0.5	0.4	0.3	0.2	0.1
Flush nozzle	3.14	2.94	2.76	2.54	2.36	2.15	2.04	1.94	1.82	1.70	1.56	1.42	1.29	1.16	1.05
Protruding nozzle	3.14	2.94	2.76	2.54	2.32	2.10	1.99	1.87	1.74	1.63	1.51	1.41	1.29	1.16	1.04

3.4.4.5 Rim reinforcements

The design of rim reinforcements (see figure 3.4.4-11) shall be derived from that for branch connections in the following manner.

Proceeding as specified in 3.4.4.3, a combination of T_r and t_r shall be determined using figure 3.4.4-3 if the rim is to be protruding (see figure 3.4.4-11a) and figure 3.4.4-1 for a flush rim (see figure 3.4.4-11b). The rim shall be designed to a smooth profile, concentrating material near the edge of the opening, such that one-half of the total cross-sectional area falling within the outermost extremities of the dimensions h and H as calculated and measured in 3.4.4.4a and b) is not less than the following:

- a) $|2ht_r + T(H + t_r)| (f_s/f_n)$ if the rim protrudes
- b) $|ht_r + T(H + t_r)| (f_s/f_n)$ if the rim is flush.

For the protruding rim, the cross-sectional area shall be equally disposed about the shell mid-thickness.

NOTE. The cross-sectional area of the rim required in this derivation will vary depending on the particular combination of T_r and t_r . A trial procedure using different combinations of T_r and t_r may be employed to establish the minimum area required.

3.4.4.6 Obround⁴⁾, elliptical and oblique openings and nozzles

Non circular openings and oblique nozzles shall be treated according to the rules for circular openings and nozzles normal to the shell except the following.

- a) For a cylindrical shell with the minor axis or the opening parallel to the axis of the shell, where a branch is not provided, d is the dimension measured on the chord across the minor axis of the opening and, where a branch is attached, d is the corresponding dimension of the opening plus the nozzle thickness.

For all other cases, where a branch is not provided, d is the dimension measured across the major axis of the opening and where a branch is attached, d is the corresponding dimension of the opening plus the nozzle thickness.

- b) In determining dimensions H and h the value of d shall be as given in a).
- c) In the case of multiple openings, the value of d_A used in 3.4.5 shall be determined using d as given in a).

3.4.4.7 Dissimilar materials

Where the branch and shell are not of equal design strength and the design strength of the branch f_n is within the range $0.6f_s$ to $1.5f_s$, the following shall apply:

if $0.6f_s < f_n < f_s$ the calculated branch thickness (disregarding here the requirement of 3.4.7) shall be increased to:

$$t_{rs} \text{ where } t_{rs} \geq t_r \frac{f_s}{f_n}$$

A check shall be made to ensure that t_{rs} meets the minimum thickness requirements of 3.4.7.

NOTE 1. Where the thickness t_r is modified in accordance with this subclause, it is not necessary to recalculate the mean branch diameter d , and iterate further. Neither is it necessary to recalculate h .

NOTE 2. If $f_n > f_s$ no reduction in t_r is allowed.

NOTE 3. Where reinforcement is concentrated near the opening, see also figures 3.4.4-7 and 3.4.4-8.

NOTE 4. For forged nozzle inserts, the procedure in 3.4.4.5 includes the necessary thickening correction factor.

3.4.5 Design of groups of openings and branches

A group of openings or branches is defined as a group in which the shortest pitch between any two branches or openings is less than $d_A + 3\sqrt{DT}$.

The compensation required shall be calculated as follows.

- a) The reinforcement required shall be calculated as for the isolated case assuming that the full width H is available.

- b) Where the shortest pitch between adjacent openings is such that $P < d_A + 2\sqrt{DT}$, the reinforcement calculated in a) shall be increased if required so that, for each ligament (see figures 3.4.5-1 and 3.4.5-2) $A_s + A_n - A_t \geq gPT$.

- c) Where the shortest pitch between adjacent openings lies between $P_1 = d_A + 2\sqrt{DT}$ and $P_2 = d_A + 3\sqrt{DT}$, the reinforcement required shall be determined by linear interpolation as follows, using the dimensions determined in a).

- 1) The compensation ratio K for pitch P_2 shall be calculated, using the reinforcement required in a), as follows:

$$K = \frac{A_s + A_n - A_t}{gP_2T} \quad (15)$$

(K shall be not greater than 1)

- 2) The compensation ratio K_2 required at the actual pitch P shall be:

$$K_2 = 1 - \frac{(P - P_1)}{(P_2 - P_1)} (1 - K) \quad (16)$$

- 3) The dimensions calculated in a) shall be increased if required so that, for each ligament, $A_s + A_n - A_t \geq K_2gPT$.

- d) The final thickness of the branch t_r shall be less than $(2 - d/D) T_r$.

- e) The transitions between sections of shell or between section of branch or nozzle connections of different thickness shall be achieved by means of a smooth taper.

⁴⁾ See footnote to 3.3.3.1 for an explanation of 'obround'.

3.4.6 Reinforcing pads

3.4.6.1 General

Reinforcing pads shall not be used where there is a possibility of severe corrosion or oxidation or of large temperature gradients across the thickness of the shell.

3.4.6.2 Pressure considerations

When reinforcing pads are used for reinforcement of penetrations or openings, the following conditions shall be observed.

- a) The ratio d/D of the branch diameter to the cylinder diameter shall not be greater than the following unless the adequacy of the design is demonstrated by experience or by a hydraulic proof test in accordance with 3.8.
 - 1) $\frac{1}{2}$ for double-sided pads;
 - 2) $\frac{1}{4}$ for single-sided pads.
- b) The width of the pad shall not be less than $H/2$, where H is the distance along the shell within which the shell thickening is assumed to contribute to the reinforcement of an opening.
- c) The thickness of a pad shall not exceed 40 mm or the as-built shell thickness, whichever is the lesser.
- d) The thickness of the pad shall not be less than $T/4$, where T is the shell thickness.
- e) The amount of compensation to be provided shall be equal to the amount which would have been necessary had the compensation been integral.
- f) The weld preparations shall be as shown in either figure C.18 or figure C.19.

3.4.6.3 Non-pressure considerations

Conditions a) to d) in 3.4.6.2 do not apply to reinforcing pads which are used to limit the local stresses due to mechanical loads on branches, supports or mountings. However, the maximum thickness of a pad which can be counted as effective reinforcement of a branch for pressure loading shall be limited to the value given in 3.4.6.2c.

If the thickness of the reinforcing pad, t_1 , is greater than the drum shell thickness, t , its size ($d_o \times d_x$), shall be such that the design leg length of the attachment welds to the drum shell does not exceed the drum thickness, t (see G.2 of BS 5500 : 1997).

3.4.7 Branches

The design of branches shall be governed by the following four considerations.

- a) The ability to withstand design pressure; for this purpose the minimum thickness of the branch shall be calculated in accordance with 3.7.1.
- b) Compensation requirements for openings in the main pressure part, which shall be determined, for cylindrical shells, in accordance with 3.4.4 or 3.4.5.
- c) The ability to withstand superimposed loading due to connected pipework or fittings; the requirements of 3.4.4.1 shall be taken into account.
- d) Suitability for the recommended forms of branch to shell attachment welds shown in annex C.

The minimum thickness of the branch shall be the greatest of the values required by a) to d) and shall not be less than $0.015d_o + 3.2$ mm, where d_o is the outside diameter of the branch (in mm). The requirements of this subclause shall not apply to tube stubs, which shall be designed as tubes.

3.4.8 Studded connections

The attachment of connections by studding only shall be permitted where the design temperature does not exceed 400 °C. Where the thickness of the drum or header is inadequate to permit direct attachment, pads or saddles shall be secured to the pressure part by welding (see figure C.20). Where required by this section, reinforcement shall be incorporated. Stud and nuts shall be of a material suitable for the operating conditions (see BS 4882).

3.4.9 Screwed connections into shell

The attachment of connections by screwing into the shell shall be permitted only if the following are complied with.

- a) The outside diameter of the connection does not exceed 60.3 mm.
- b) The calculation pressure does not exceed 4 N/mm².
- c) The design temperature does not exceed 400 °C.
- d) Seal welding shall be applied if the design temperature is greater than 220 °C or the calculation pressure is greater than 2 N/mm².
- e) The steel of the screwed component does not contain more than 0.25 % (m/m) carbon.

The design of connections shall conform to 3.4.7; the minimum thickness shall be measured at the root of the thread.

The main pressure part shall be designed as having an uncompensated opening with a diameter equal to that of the root of the thread in the hole.

3.4.10 Screwed socket connections

Screwed socket connections (see figure C.21) shall be permitted only if the conditions given in 3.7.7.5 are complied with.

The main pressure part shall be designed as having an uncompensated opening.

3.4.11 Socket welded connections

Socket welded connections (see figure C.21) shall be permitted only if the conditions given in 3.7.7.4 are complied with.

The main pressure part shall be designed as having an uncompensated opening.

NOTE. To achieve the required throat dimension the leg lengths of the fillet weld shown in BS 2633 and BS 4677 will need to be increased.

3.5 Headers of rectangular section

3.5.1 Thickness of straight headers

3.5.1.1 To determine the thickness of headers of rectangular section with solid walls, the thickness required at the centre of the sides, at all the lines of holes and at the corners shall be considered. The required thickness shall be the greatest value so determined. In the case of staggered holes, the effect of the off-set shall be investigated by considering the diagonal ligaments (see figure 3.5.1-1).

The minimum thickness t (in mm) of the shell before allowance for wastage shall be given by the following equation:

$$t = \frac{Pn}{2f\eta} + \sqrt{\left(\frac{4YP}{f\eta'}\right)} \quad (17)$$

where

- P is the design pressure (in N/mm²);
- f is the design stress (in N/mm²) (see 2.2);
- Y is a coefficient given by equations (18), (19), (20) or (21);
- n is one-half of the internal width of the wall perpendicular to that referred to in the calculation (in mm);
- η is the ligament efficiency referring to the membrane stresses and is calculated by the use of the equation (22) or (23);
- η' is the ligament efficiency referring to the bending stresses and is calculated by the use of the equations (24), (25), (26) or (27).

3.5.1.2 The coefficient Y to be used in determining the thickness required at the centre of the side with internal width $2m$ shall be determined as follows:

$$Y = \frac{1}{3} \left(\frac{m^3 + n^3}{m + n} \right) - \frac{1}{2} m^2 \quad (18)$$

where

- m is one-half of the internal width of the wall referred to in the calculation (in mm) (see figure 3.5.1-2);
- n is one-half of the internal width of the wall perpendicular to that referred to in the calculation (in mm).

The coefficient Y to be used in determining the thickness required at a line of holes parallel to the longitudinal axis of the header on the wall of width $2m$ shall be determined as follows:

$$Y = \frac{1}{3} \left(\frac{m^3 + n^3}{m + n} \right) - \frac{m^2 - c^2}{2} \quad (19)$$

where

- c is the distance from the centre of the holes to the centre line of the wall (in mm) (see figure 3.5.1-1a and b).

To check the effect of the off-set on a staggered hole arrangement where the holes are position equidistantly from the centre line of the wall, the coefficient Y shall be calculated as follows:

$$Y = \cos a \left\{ \frac{1}{3} \left(\frac{m^3 + n^3}{m + n} \right) - \frac{m^2}{2} \right\} \quad (20)$$

where

- a is the angle indicated in figure 3.5.1-1b.

NOTE. If the value of Y is negative, the sign does not have to be considered.

The coefficient Y to be used in determining the thickness required due to the stresses at the corners, shall be determined as follows:

$$Y = \frac{1}{3} \left(\frac{m^3 + n^3}{m + n} \right) \quad (21)$$

NOTE. In this case the value of η and η' in equation (17) will be equal to 1.0.

3.5.1.3 The ligament efficiency η in the case of a line of holes parallel to the longitudinal axis of the header shall be determined as follows:

$$\eta = \frac{p - d}{p} \quad (22)$$

(See figure 3.5.1-1a and b.)

The ligament efficiency η to be considered in the calculation relating to the diagonals shall be determined as follows:

$$\eta = \frac{p_1 - d}{p_1} \quad (23)$$

(See figure 3.5.1-1b.)

The ligament efficiency η' in the case of a line of holes parallel to the longitudinal axis of the header shall be determined as follows:

$$\eta' = \frac{p - d}{p}, \text{ when } d < 0.6m \quad (24)$$

or

$$\eta' = \frac{p - 0.6m}{p}, \text{ when } d \geq 0.6m \quad (25)$$

The ligament efficiency η' to be considered in the calculation relating to the diagonals shall be determined as follows:

$$\eta' = \frac{p_1 - d}{p_1}, \text{ when } d < 0.6m \quad (26)$$

or

$$\eta' = \frac{p_1 - 0.6m}{p_1}, \text{ when } d \geq 0.6m \quad (27)$$

In the case of elliptical or obround⁵⁾ holes, the value of d used in the equations for η and η' shall be the inside dimension of the hole measured parallel to the longitudinal axis of the header.

For evaluation of the two limiting values of d in equations (24) or (25) and (26) or (27), the value of d shall be the inside dimension of the hole measured perpendicular to the longitudinal axis of the header.

3.5.2 Corner radius of straight headers

To avoid excessive discontinuity stresses at the corners, the following condition shall be satisfied.

$$r \geq \frac{1}{3} e \geq 8 \text{ mm}$$

where

- r is the internal corner radius (in mm)
(see figure 3.5.1-2);
- e is the mean of the nominal thicknesses of the two sides (in mm).

In any case r shall not be taken as less than 8 mm.

3.5.3 Headers of rectangular section which are not straight

The scantlings of rectangular section headers of the curved, staggered, sinuous or corrugated type shall be subject to agreement between the manufacturer, the purchaser and the Inspecting Authority (see 1.5.2.3e). Where sufficient experience of previous satisfactory service of similar headers cannot be shown, the suitability of the headers shall be proved in accordance with 3.8.

3.6 Ends of drums and headers

3.6.1 Dished ends

3.6.1.1 Shape

Torispherical and semi-ellipsoidal unstayed drum ends, dished from plate, having pressure on the concave side shall conform to the following:

a) *Torispherical ends* (see figure 3.6.1-1a).

The internal radius of dishing, R_1 shall not be greater than D_o .

The internal corner radius r_1 shall not be less than 10 % of D_o , nor less than $3t$.

The external height H shall not be less than $0.18D_o$.

b) *Semi-ellipsoidal ends* (see figure 3.6.1-1b).

The external height H shall not be less than $0.2D_o$.

For a) and b) above

D_o is the outside diameter of the parallel portion of the end (in mm);

t is the thickness of the end plate (in mm).

H shall be measured from the commencement of curvature (see figure 3.6.1-1b).

NOTE. No special shape requirements are specified for hemispherical unstayed drum ends.

3.6.1.2 Thickness

3.6.1.2.1 Subject to the limitations given in 3.6.1.1, the thickness of torispherical, semi-ellipsoidal and hemispherical ends shall be calculated from the following equation:

$$t = \frac{pD_o K}{2f} \quad (28)$$

where

t is the minimum thickness after dishing (in mm);

p is the calculation pressure (in N/mm²);

D_o is the outside diameter of the domed end (in mm);

K is a shape factor (see 3.6.1.2.2);

f is the design stress (in N/mm²) (see 2.2).

The minimum thickness (t) shall in no circumstances be less than 9.5 mm.

For ends which are butt welded to the drum shell, the thickness of the edge of the skirt for connection to the shell shall not be less than the thickness of a seamless, unpierced shell, determined by equation (1).

NOTE 1. The external height of dishing H may, for ends of torispherical form, be determined as follows (see figure 3.6.1-1a):

$$H = R_o - \sqrt{\{ (R_o - D_o/2) \times (R_o + D_o/2 - 2r_o) \}}$$

where R_o and r_o are as shown in figure 3.6.1-1.

NOTE 2. For deep ends with low values of K , and particularly for hemispherical ends, the limitation of the flange edge thickness may determine the end thickness and overrule the thickness resulting from equation (28).

⁵⁾ See footnote to 3.3.3.1 for an explanation of 'obround'.

3.6.1.2.2 The shape factor K to be used in equation (28) shall be obtained from the curves in figure 3.6.1-2 and depends on the ratio of height to diameter H/D_o .

NOTE 1. The curve drawn with a full line in the series provides the factor K for plain (i.e. unpierced) ends. Where the value of H/D_o is lower than 0.25, the value of K depends on the ratio of thickness to diameter t/D_o as well as on the ratio H/D_o and a trial calculation may be necessary to arrive at the correct value of K .

NOTE 2. Figure 3.6.1-2 takes account of the highest stresses occurring in ends whose dishing radius R_i is not greater, and whose corner radius r_i is not less, than those of semi-ellipsoidal ends of the same height H . If, in exceptional circumstances, very shallow ends are used having a large internal radius R_i or a small internal radius r_i , it is recommended that instead of using the actual height H when reading K from figure 3.6.1-2, an equivalent height:

$$H_E = \frac{D_o^2}{4R_o}$$

or

$$H_E = \sqrt{\left(\frac{D_o r_o}{2}\right)}$$

whichever is the smaller, be used, if H_E is smaller than H .

3.6.1.3 Openings in dished ends

3.6.1.3.1 Unreinforced openings

Openings in dished ends shall be either circular, elliptical or obround⁶⁾.

The curves drawn with broken lines in figure 3.6.1-2 provide values of K that shall be used in equation (28) for ends with unreinforced openings (e.g. manholes or tube holes). The selection of the correct curve depends on the value:

$$\frac{d}{\sqrt{(D_o t)}}$$

where

d is the diameter of the largest opening in the end plate (in the case of an elliptical or obround⁶⁾ opening, the major axis of the opening) (in mm);

t is the minimum calculated thickness, after dishing (in mm);

D_o is the outside diameter of dished end (in mm).

Trial calculation shall be made in order to select the correct curve.

In any case the following shall be satisfied:

- t/D_o shall not exceed 0.1;
- d/D_o shall not exceed 0.7.

Figure 3.6.1-2 shows that for values of H/D_o less than 0.25, certain of the broken line curves fall below the full curve t/D_o . In this region when the stresses in the knuckle are more severe than those in the crown, K shall be chosen as for a plain head and an uncompensated opening in the crown shall be permitted.

NOTE 1. The size of such an uncompensated opening may be determined as in the following example.

For an end with $H = 190$ mm and $D_o = 1000$ mm:

$$H/D_o = 0.19$$

$$K = 1.56$$

and

$$d/\sqrt{(D_o t)} = 1.1.$$

If the thickness calculated from equation (28) is 10 mm, then $d = 110$ mm.

These requirements apply equally to flanged openings and to unflanged openings simply cut in the plate of an end. No reduction shall be made in end-plate thickness on account of flanging.

Where openings are flanged, the radius r_m of the flanging shall not be less than 25 mm (see figures 3.6.1-1 and 3.6.1-3).

NOTE 2. The thickness of the flanged portion may be less than the calculated thickness t .

Unreinforced and flanged openings in dished ends shall be so arranged that the distance from the edge of the hole to the outside edge of the plate and the distance between openings are not less than those shown in figure 3.6.1-3.

NOTE 3. In figure 3.6.1-3, d_2 is equal to the diameter of the smaller hole.

3.6.1.3.2 Reinforced openings

Where it is desired to use an opening on a dished end of less thickness than would be required by the application of 3.6.1.3.1, reinforcement of the end shall be provided.

NOTE 1. Reinforcement may consist of a ring and/or standpipe welded into the hole (see figure 3.6.1-4). Alternatively, forged reinforcement may be used.

Added reinforcing material shall be considered as effective reinforcement within the following limits:

- the effective width l_1 of reinforcement shall not exceed $\sqrt{(2R_i t)}$ or $d_o/2$, whichever is the smaller,
- the effective length l_2 of a reinforcing ring shall not exceed $\sqrt{(d_o l_t)}$.

⁶⁾ See footnote to 3.3.3.1 for an explanation of 'obround'.

For a) and b) above

- R_i is the internal radius of the spherical part of a torispherical end or, for a semi-ellipsoidal end, the internal radius of the meridian of the ellipse at the centre of the opening (in mm);
- t is the actual thickness of the end (in mm);
- t_t is the actual thickness of the ring (in mm);
- d_o is the external diameter of the ring (in mm).

NOTE 2. The dimensions l_1 and l_2 are shown shaded in figure 3.6.1-4.

NOTE 3. The shape factor K for a dished end having a reinforced opening may be read from figure 3.6.1-2 using the value obtained from:

$$\frac{d_o - A/t}{\sqrt{(D_o t)}} \text{ instead of from } \frac{d}{\sqrt{(D_o t)}}$$

where

- A is the effective cross-sectional area of reinforcement (in mm²).

When required for the determination of K , the value of A (see note 3) shall be taken as twice the shaded area shown in figure 3.6.1-4.

The shaded area shown in figure 3.6.1-4 shall be calculated as follows.

- 1) Calculate the sectional area of reinforcement both inside and outside the end plate within the length l_1 .
- 2) Add to it the full sectional area of that part of the stem of the nozzle which projects inside the end plate up to the distance l_2 .
- 3) Add to it the full sectional area of that part of the stem of the nozzle which projects outside the internal surface of the end plate up to the distance l_2 and deduct from it the sectional area which the stem would have if its thickness were calculated in accordance with equations (31) or (32) disregarding the minimum thickness from table 3.7.1.1.

If the material of the ring has an allowable stress lower than that of the end plate, then the effective cross section A shall be reduced below that calculated in proportion to the difference in the allowable stresses for the materials. As in **3.6.1.3.1**, trial calculation shall be made in order to select the correct curve.

3.6.1.3.3 Thickness of internally fitted doors

The minimum calculated thickness of doors of flat plate construction (i.e. unstiffened) made from one plate shall be not less than that determined by the following equations:

$$t = \sqrt{\left[\frac{0.35pd^2 + W}{f} \right]} \text{ for circular doors, and}$$

$$t = \sqrt{\left[\frac{0.35p(2 - a/b)a^2 + W}{f} \right]} \text{ for ellipsoidal doors}$$

where

- t is the minimum calculated thickness of the flat door at or near the centre (in mm);
- p is the calculation pressure (in N/mm²);
- d is the diameter of the opening to which the door is fitted, if round (in mm);
- a is the minor axis of the opening to which the door is fitted, if an ellipse (in mm);
- b is the major axis of the opening to which the door is fitted, if an ellipse (in mm);
- W is the full load capacity of one stud (effective stud area $\times$ design stress value at design temperature) (in N);
- f is the maximum allowable stress in the plate at the design temperature (in N/mm²).

NOTE. A design stress value of 50 N/mm² may be used for carbon steel bolts of grade 4.6 or equivalent for design temperatures not exceeding 300 °C. For other bolting materials and greater temperatures refer to BS 5500 for the allowable stresses when deriving W .

3.6.2 Flat ends of headers

3.6.2.1 Unpierced flat ends

The minimum thickness of an unstayed flat end without opening shall be that given by the following equations:

$$\text{circular ends: } t_e = Cd_i \sqrt{\left(\frac{p}{f} \right)} \quad (29)$$

$$\text{rectangular ends: } t_e = CZa \sqrt{\left(\frac{p}{f} \right)} \quad (30)$$

where

- t_e is the minimum thickness of the end plate (in mm);
- d_i is the internal diameter of the cylindrical shell (in mm);
- a is the least width between walls of a rectangular header (in mm);
- p is the calculation pressure (in N/mm²);
- f is the design stress (N/mm²) (see **2.2**);
- C is a constant, not less than 0.41, depending on the method of end attachment, derived from:
 - a) figure 3.6.2.1-1 for ends as shown in figure 3.6.2.1-1a and b for which shell and end materials are not restricted;
 - b) figure 3.6.2.1-2 for ends as shown in this figure for which shell and end materials are restricted to carbon and carbon manganese steels up to type 460 (see table 2.1.2);
- Z is the coefficient for rectangular ends as given in figure 3.6.2.1-3;
- b is the greatest width between walls of a rectangular header (in mm).

3.6.2.2 Pierced flat ends

Where flat end plates are pierced concentrically or eccentrically, the thickness required by 3.6.2.1 shall be increased by 15 %. Where an end detail as shown in figure 3.6.2.1-2 is employed, care shall be taken that any eccentrically positioned hole does not encroach into the attachment weld.

3.6.3 Branches and connections

The requirements of 3.4.7 to 3.4.9 shall be complied with in so far as they are applicable to the ends of drums and headers.

Where branch or nozzle connections are attached to a piping system designed with due allowance for expansion thrusts, etc. (e.g. conforming to the flexibility requirements of BS 806), the thickness determined for the head or end in 3.6.1 or 3.6.2 shall be considered sufficient.

3.7 Tubes and integral pipes

3.7.1 Thickness of straight tubes and integral pipes

3.7.1.1 The minimum thickness of straight tubes, including drum or header connections or tube stubs, and integral pipes shall be determined by either of the following equations:

$$t = \frac{pd_o}{2f + p} \quad (31)$$

or

$$t = \frac{pd_i}{2f - p} \quad (32)$$

where

t is the minimum thickness of the straight tube or pipe exclusive of any wastage allowance (in mm);

p is the calculation pressure (in N/mm²);

d_o is the outside diameter of the tube or pipe (in mm);

d_i is the inside diameter of the tube or pipe (in mm);

f is the design stress (in N/mm²) (see 2.2).

The thickness t , plus any wastage allowance (see 3.7.1.2), shall however in no case be less than the minimum given in table 3.7.1.1.

NOTE 1. The value of thickness so determined is the minimum thickness of straight tubes and further provision will be needed for minus thickness tolerances where necessary.

NOTE 2. Diameters are subject to manufacturing tolerances which are not used in the calculation.

Table 3.7.1.1 Minimum thickness of straight tubes

Dimensions in millimetres	
Nominal outside diameter	Minimum thickness
Not exceeding 38	1.7
Exceeding 38 not exceeding 51	2.2
Exceeding 51 not exceeding 70	2.4
Exceeding 70 not exceeding 76	2.6
Exceeding 76 not exceeding 95	3.0
Exceeding 95 not exceeding 102	3.3
Exceeding 102 not exceeding 127	3.5

3.7.1.2 In all cases where reduction of the wall thickness of the tubes or pipes is expected as a result of surface wastage, an increase in thickness sufficient for the design life of the tubes and pipes shall be provided over that required by the other design requirements of this clause. This allowance shall be determined by the manufacturer unless otherwise specified by the purchaser (see 1.5.1c and 3.2.2).

3.7.2 Tube bends

Where tubes are to be bent, the requirements of 4.2.4 shall be taken into account at the design stage.

3.7.3 Integral pipe bends

3.7.3.1 Where integral pipes are bent the minimum thickness (t_b) of a straight pipe, from which a pipe bent to a radius given in table 3.7.3 is to be made, shall be determined from equation (33) or (34) except where it can be demonstrated that the use of a thickness less than t_b would not reduce the thickness below t_a at any point after bending.

For pipes 219.1 mm outside diameter and below, and for pipes above 219.1 mm outside diameter bent to the radii given in column 2 of table 3.7.3, t_b shall be given by:

$$t_b = 1.125 t \quad (33)$$

For pipes above 219.1 mm outside diameter where t is 32 mm or more bent to the radii given in column 3 of table 3.7.3, t_b shall be given by:

$$t_b = 1.1 t$$

The value of t_b is the minimum thickness and provision shall be made for minus tolerances.

NOTE. Manufacturing considerations may make it necessary for pipes thicker than this minimum to be used.

Segmental and cut-and-shut bends shall not be used.

3.7.3.2 Pipes conforming to BS 3601 shall not be bent to radii less than those given in table 3.7.3. Other pipes of a thickness determined as in **3.7.3.1** shall not be bent to radii less than those given in table 3.7.3 unless:

- a) it can be demonstrated that the use of this thickness will not reduce the thickness at any point after bending to below t , determined from equations (31) or (32);
- b) for piping not less than 76.1 mm outside diameter where the design stress is time-dependent and the radius is less than three times the inside diameter, it can be additionally demonstrated that the thickness t_i (in mm) at the intrados of the bend is not less than that resulting from the following:

$$t_i \geq t \frac{2R - r}{2R - 2r}$$

where

- t is the thickness derived from equations (31) and (32) (in mm);
- R is the radius of the bend (in mm);
- r is the mean radius of pipe (in mm).

NOTE. In general, it will be necessary to increase the thickness above that determined by **3.7.3.1** in order to comply with this subclause.

3.7.3.3 There is a minimum thickness for each size of pipe, dependent on bending procedure, below which the allowance for thinning will be exceeded, and in such cases the radius given in table 3.7.3 shall be increased where necessary to ensure that the thickness is not below t at any point after bending.

3.7.3.4 For pipes rolled to a specified inside diameter, the minimum bending radius and corresponding thinning allowance shall be that applicable to the nearest outside diameter given in table 3.7.3 greater than that resulting from the specified inside diameter plus $2t_b$.

Table 3.7.3 Minimum bending radii for pipes of thickness determined by 3.7.3.1		
Dimensions in millimetres		
Outside diameter	Radii measured to centre line of pipe	
	$t_b = 1.125 t$ all thicknesses	$t_b = 1.1 t$ $t_b = 35 \text{ mm or above}$
26.9	65	
33.7	75	
42.4	100	
48.3	115	
60.3	150	
76.1	190	
88.9	230	
101.6	265	
114.3	305	
139.7	380	
168.3	460	
193.7	630	
219.1	710	
244.5	810	1140
273.0	1020	1270
323.9	1220	1520
355.6	1500	1780
406.4	1730	2030
457.0	2030	2280

3.7.4 Flexibility of integral piping systems

3.7.4.1 General

The pipes shall be arranged so that the system is sufficiently flexible to absorb the whole of its own expansion and that of the connecting equipment without exceeding the requirements of 4.11 of BS 806 : 1993.

Account shall also be taken of end reactions in the design of branches to drums, headers, etc., in accordance with 3.4.4 or 3.6.3 as appropriate.

3.7.4.2 Analysis

A flexibility analysis shall be used if there is any doubt as to the ability of the system to satisfy the specified requirements. The analysis shall be in accordance with 4.11 of BS 806 : 1993.

The designer shall be responsible for performing any necessary analysis, unless the system conforms to one of the following criteria.

- The piping system duplicates a successfully operating installation or replaces a system with a satisfactory service record.
- The piping system can be adjudged adequate by comparison with previously analysed systems.

NOTE. The pipes may be prestressed to reduce end effects under hot conditions of working and also to reduce hot stress levels. (For recommended methods, reference should be made to BS 806.)

3.7.4.3 Misalignment at site butt welds

To avoid excessive stresses at the pipe bends, consideration shall be given to the permissible limits of pull-up to correct for misalignment at the final welds. Where template pipes are not provided, the cold alignment tolerances shall be assessed (refer to 3.7.4.2) and specified.

3.7.5 Structural attachments to tubes

3.7.5.1 Where structural attachments such as lugs and brackets are welded to tubes for load-carrying purposes, the material of the attachment and of the weld shall be suitable for the environment and compatible with the tube material.

NOTE. Figure 3.7.5.1 is intended to illustrate typical details for the clarification of design requirements. Attention is drawn to the requirements of BS 2633 concerning the distance between a butt weld and an attachment weld.

3.7.5.2 The thickness of the attachment section, measured in the tube circumferential direction, shall not exceed one-quarter of the tube diameter at the point of attachment to the tube. The length of the attachment measured along the axis of the tube shall either conform to 3.7.5.3, or be determined by stress analysis conforming to the criteria of annex A of BS 5500 : 1997.

3.7.5.3 The intensity of radial loading shall not exceed that given by figure 3.7.5.3 for tensile or compressive attachment loading:

where

- q is the greatest intensity of radial load (in N/mm);
- f is the design stress of tube material as used in 3.7.1 (in N/mm²);
- t is the minimum tube thickness, i.e. nominal thickness less negative tolerance (in mm);
- d_o is the outside diameter of tube (in mm);
- a is the angle subtended by attachment at tube centre (see figure 3.7.5.1.) (in degrees).

3.7.5.4 For an attachment welded longitudinally along a tube, the greatest occurring tensile and compressive radial load intensities shall be calculated from the following equation:

$$q = \frac{R}{L} \pm \frac{6We}{L^2} \quad (35)$$

where

- W is the total load carried by attachment (in N);
- e is the eccentricity of line of action of load W about the line of attachment to the tube (in mm);
- R is the radial component of the load W (in N);
- L is the length of attachment (in mm).

NOTE. Where eccentric loading occurs, it may be necessary to check both compressive and tensile radial load intensities, which occur at opposite ends of the attachment. W , R and q are positive when tensile, and negative when compressive.

3.7.5.5 In calculating the strength of welds for structural attachments the allowable design stress shall correspond to that of the weaker of the tube or structural attachment, multiplied by the following weld reduction factors:

- 0.7 for fillet welds;
- 0.75 for partial penetration welds (with or without superimposed fillet welds);
- 1.0 for full penetration.

The loaded area of the weld shall be taken as the throat thickness $\times$ the length of weld. For the purpose of stress calculations the throat thickness shall be taken as:

- 0.7 of the leg length for fillet welds;
- the depth of groove for partial penetration welds;
- the thickness of the structural attachment for full penetration welds.

For compound welds the throat thickness shall be the sum of the constituent parts.

3.7.6 Fitting and jointing of tubes

3.7.6.1 Fitting of tubes to drums or headers shall be in accordance with **4.3.2.3** or **4.4.4**.

3.7.6.2 Jointing of tubes shall be by butt welds in accordance with **4.3.2.5**. Socket welds, sleeve welds, flanges and screwed joints shall not be used.

3.7.7 Joints in integral piping

3.7.7.1 Integral pipes shall be joined by welded or integral flanges, butt welds, socket welds, screwed, or screwed and seal welded joints. Screwed flanges and sleeve welds shall not be used. Flanges shall not be used where the bolts would be exposed to products of combustion.

3.7.7.2 The flanges and bolting for ordinary bolted flange joints shall be in accordance with BS 4504 : Section 3.1 or BS 1560 : Section 3.1 for flanges which are dimensioned in millimetres, or in accordance with BS 10 for flanges dimensioned in inches. For welded-on flanges, the weld preparations and flange types shall be selected according to BS 806. Gasket selection (e.g. material and dimensions) shall be such that the gasket factor and seating stresses are consistent with the flanges and bolting.

NOTE. Special joints and special types of flange may be used provided that they are shown to be suitable for the design conditions.

3.7.7.3 Butt welded joints shall be in accordance with **4.3.2.5**.

3.7.7.4 Socket weld joints shall not be used in service where fatigue, severe erosion or crevice corrosion is expected to occur, or for pipes exceeding 60.3 mm outside diameter.

The thickness of socket weld fittings shall be in accordance with **3.7.1.1** but shall not be less than 1.25 times the nominal thickness of the pipe. Leg lengths of the fillet weld shall be as shown in figure 3.7.7.4 so as to achieve the throat dimension not less than the nominal thickness of the pipe (t_n). The material shall be compatible with the associated piping.

Socket weld fittings shall be of forged steel. The dimensions and clearances of the socket end shall be in accordance with BS 3799. The ratings of BS 3799 shall not be exceeded.

The welding of socket weld joints shall be in accordance with **4.3.2.5**.

3.7.7.5 Screwed or screwed and seal welded joints shall not be used in service where fatigue, severe erosion, crevice corrosion or shock is expected to occur, or for pipes exceeding 60.3 mm outside diameter. The screwing shall have a taper thread, and the steel component shall not contain more than 0.25 % (m/m) carbon if it is to be seal welded.

The calculation pressure shall not exceed 4 N/mm² and the design temperature shall not exceed 400 °C, but seal welding shall be applied if the calculation pressure is greater than 2 N/mm² or the design temperature is greater than 220 °C.

3.8 Pressure parts of irregular shape

Where pressure parts are of such irregular shape that it is impracticable to design their scantlings by the application of formulae, the suitability of their construction shall be determined by a proof hydraulic test of a prototype. The component design shall be considered suitable if the proof test pressure is achieved without significant permanent deformation of the prototype.

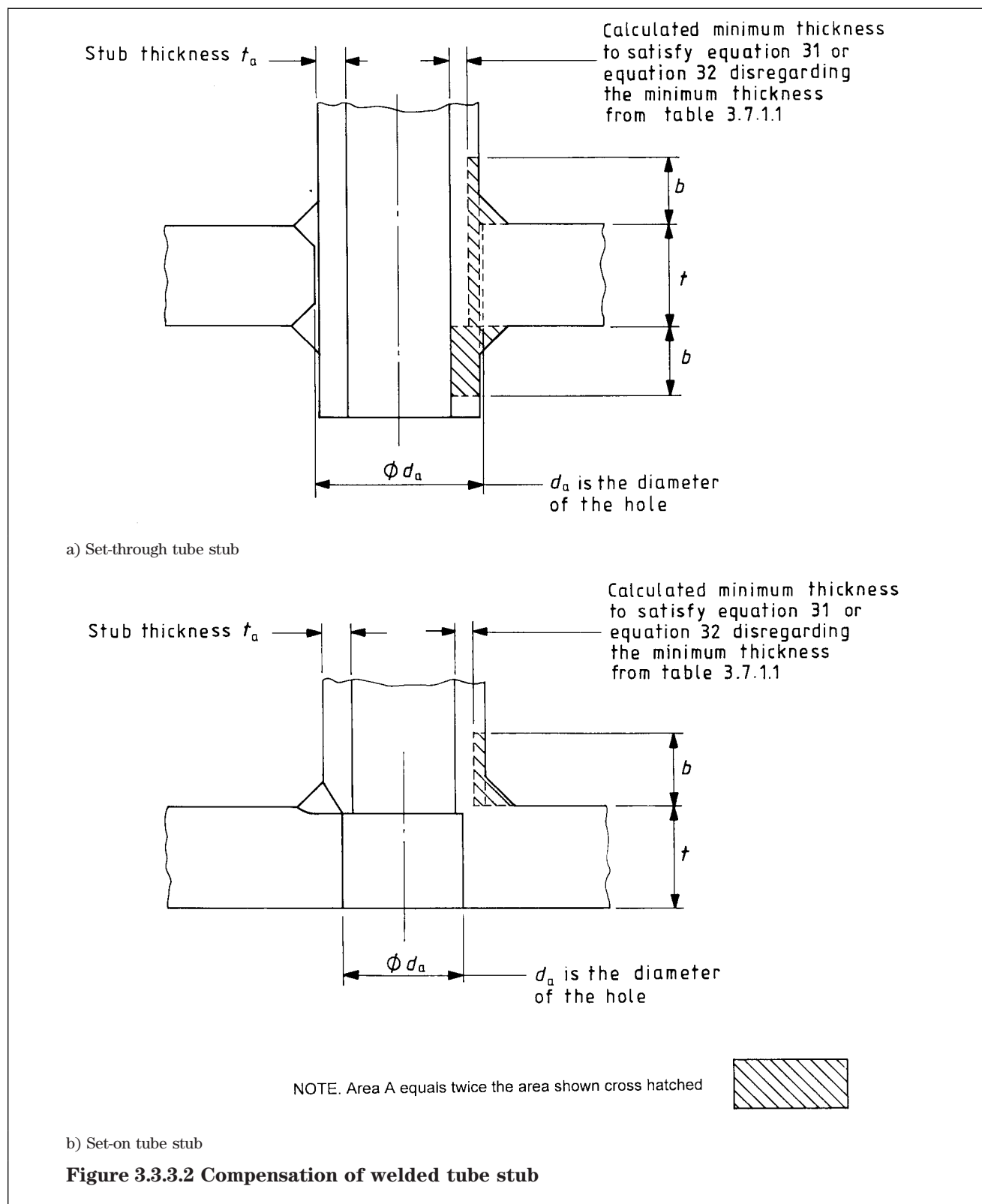
The proof test pressure P_t (in N/mm²) shall not be less than that given by the following equation:

$$P_t = 0.9p \times \frac{\text{yield stress at test temperature}}{\text{design stress } f} \quad (36)$$

where

p is the calculation pressure (in N/mm²).

NOTE. No requirements for pressure parts of irregular shape are specified for steels other than ferritic steels.



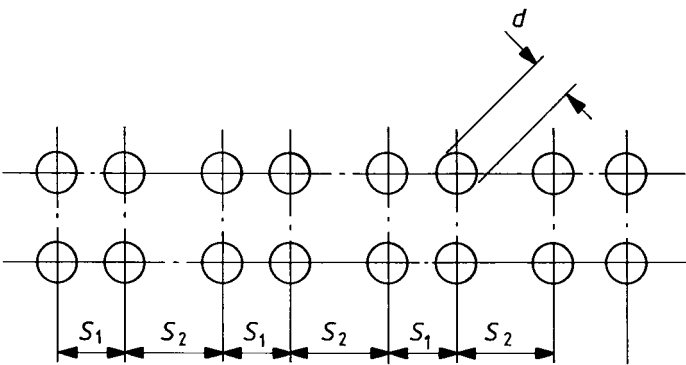


Figure 3.3.3.4 Irregular drilling

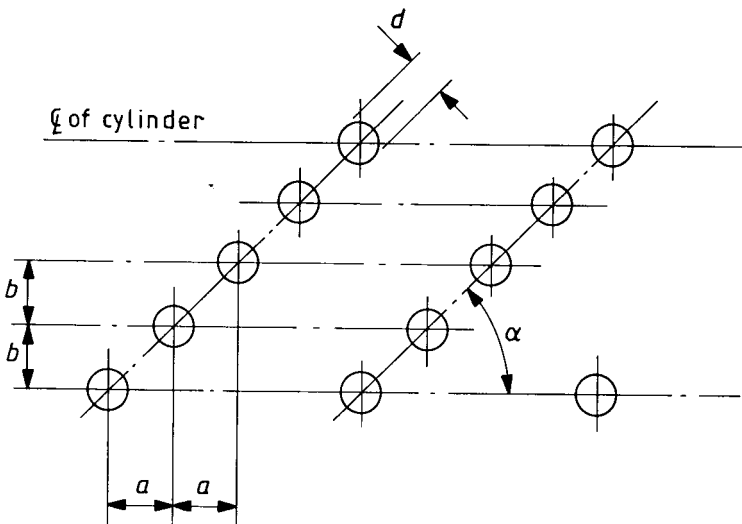


Figure 3.3.3.5-1 Spacing of holes on a diagonal line

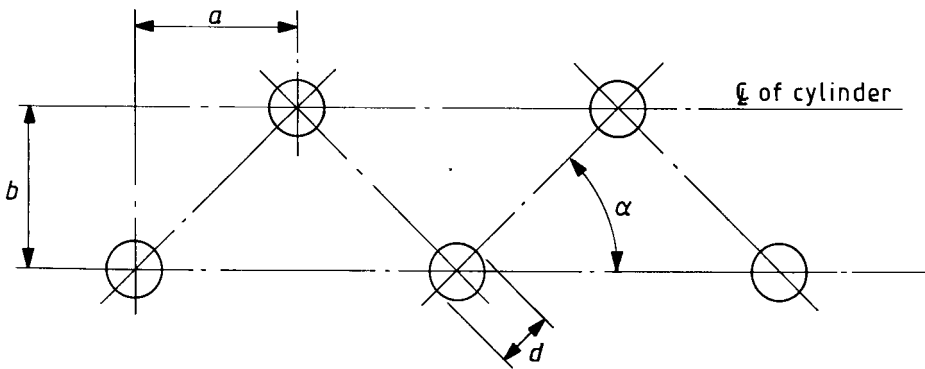


Figure 3.3.3.5-2 Regular saw-tooth pattern of holes