

Specification for

Carbon and low alloy steel pressure vessels for primary circuits of nuclear reactors

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British Insurance (Atomic Energy) Committee*	Ministry of Labour (H.M. Factory Inspectorate)*
British Mechanical Engineering Federation*	Ministry of Power*
Electricity Council, the Generating Board and the Area Board in England and Wales*	Scientific Instrument Manufacturers' Association of Great Britain
Institution of Chemical Engineers	United Kingdom Atomic Energy Authority*
Institution of Electrical Engineers	

The Government departments and scientific and industrial organizations marked with an asterisk in the above list, together with the following, were directly represented on the committee entrusted with the preparation of this British Standard:

Associated Engineering Insurers (British Electricity) Management Committee	Ministry of Defence (Navy Dept.)
Associated Offices Technical Committee	Ministry of Technology — National Engineering Laboratory
Board of Trade — Marine Division	Tank and Industrial Plant Association
British Iron and Steel Federation	Water-tube Boilermakers' Association
British Welding Research Association	

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Foreword

This standard makes reference to the following British Standards:

BS 131, *Methods for notched bar tests — Part 2: The Charpy V-notch impact test.*

BS 427, *Method for Vickers hardness test.*

BS 709, *Methods of testing fusion welded joints and weld metal in steel.*

BS 759, *Valves, gauges and other safety fittings for application to land boilers and to piping installations for and in connection with land boilers.*

BS 1123, *Safety valves, gauges and other safety fittings for air receivers and compressed air installations.*

BS 1500, *Fusion welded pressure vessels for use in the chemical, petroleum and allied industries — Part 1: Carbon and low alloy steels.*

BS 1501, *Steels for fired and unfired pressure vessels. Plates.*

BS 1501-6, *Steels for use in the chemical, petroleum and allied industries.*

BS 1560, *Steel pipe flanges and flanged fittings (nominal sizes $\frac{1}{2}$ in to 24 in) for the petroleum industry.*

BS 1719, *Classification, coding and marking of covered electrodes for metal-arc welding.*

BS 1780, *Bourdon tube pressure and vacuum gauges.*

BS 1806, *Dimensions of toroidal sealing rings ("O" seals and their housings).*

BS 1966, *Domed ends for tanks and pressure vessels.*

BS 2600, *Radiographic examination of fusion welded butt joints in steel.*

BS 2633, *Class I metal-arc welding of steel pipelines and pipe assemblies for carrying fluids.*

BS 2704, *Calibration blocks and recommendations for their use in ultrasonic flaw detection.*

BS 2910, *General recommendations for the radiographic examination of fusion welded circumferential butt joints in steel pipes.*

BS 2915, *Domed metallic bursting discs and bursting disc assemblies.*

BS 3059, *Steel boiler and superheater tubes.*

BS 3601, *Steel pipes and tubes for pressure purposes. Carbon steel: ordinary duties.*

BS 3602, *Steel pipes and tubes for pressure purposes. Carbon steel: high duties.*

BS 3604, *Steel pipes and tubes for pressure purposes: Low and medium alloy steel.*

BS 3923, *Methods for ultrasonic examination of welds.*

BS 3971, *Image quality indicators for radiography and recommendations for their use.*

At the meeting in June, 1958 of Sub-committee 3 — Reactor Safety, of Technical Committee 85 — Nuclear Energy, of the International Organization for Standardization (ISO), a decision was taken to study aspects of reactor containment. To cover this work in the U.K. a committee was set up under the authority of the Nuclear Energy Industry Standards Committee, who decided that a study should be made at the same time of steel pressure vessels for nuclear power reactors. This study revealed that while current British practice for nuclear pressure vessels was based to some extent on BS 1500-1¹⁾, a considerable number of additional requirements covering design, construction and testing aspects was being imposed on the manufacturers.

In preparing this standard such additional requirements have been taken into consideration, together with any information available to the above ISO committee. Account has also been taken of the higher design stresses recently accepted by the ISO Boiler Committee.

¹⁾ BS 1500, "Fusion welded pressure vessels for use in the chemical, petroleum and allied industries", Part 1: "Carbon and low alloy steels".

At the same time it was desired to rationalize the approach to design and to this end Clause 3.3 has been introduced which requires detailed design appraisal in the light of the modes of failure relevant to the particular vessel. It is appreciated that this clause at present is not as detailed as might be considered desirable in order to achieve uniformity. However, the committee were of the opinion that the introduction of these principles in their present form served a useful purpose and, therefore, should not be delayed pending more detailed information becoming available.

In some cases, particularly for plain carbon steels at temperatures below 350 °C the nominal design stresses in Table 1 are lower than those used hitherto, based on 0.25 U.T.S. criterion. Although the adequacy of the criteria used in the past is in no way questioned, the new lower values have been given in Table 1 in order to retain a consistent basis of derivation over the whole temperature range based on the guaranteed mechanical properties now given in the associated material standards.

Where sections of this standard have been taken from other British Standards they have been repeated in full to avoid possible confusion. Some assistance has been obtained from consideration of other standards, notably ASME, and in particular the section of this standard dealing with flange design has largely been based on the relevant sections of ASME Section VIII. This was done because it was apparent that this method is almost universally used by designers in this particular field.

This standard is intended to cover the steel pressure vessels for all types of nuclear reactor and is based on the best current information available. While it is intended to keep it under frequent review it is realized that some of the requirements might be inapplicable to new developments, and in such cases scantlings, etc., should be the subject of agreement between the purchaser and manufacturer.

Unless otherwise defined in this standard, the terms used are consistent with those used in other British Standards or standard dictionaries and BS 2094²⁾, BS 499³⁾ and BS 3455⁴⁾ should be consulted for definitions not given.

NOTE Where metric equivalents are stated in parentheses the figures in British units are to be regarded as the standard. The metric equivalents are approximate. More accurate conversions should be based on the tables in BS 350, "Conversion factors and tables", Attention is also drawn to BS 2856, "Precise conversion of inch and metric sizes on engineering drawings".

Obsolescent (by Amendment No. 5)

The need for the continuance of this British Standard has been reviewed and it has been decided that it should be regarded as obsolescent. The standard will be withdrawn in due course.

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 document comprises a front cover, an inside front cover, pages i to vi, pages 1 to 128 and a back cover.

This standard has been updated (see copyright date) and may have had amendments incorporated. This will be indicated in the amendment table on the inside front cover.

²⁾ BS 2094, "Glossary of terms relating to iron and steel".

³⁾ BS 499, "Glossary of terms (with symbols) relating to the welding and cutting of metals".

⁴⁾ BS 3455, "Glossary of terms used in nuclear science".

Section 1. General

1.1 Scope

This British Standard covers the design, construction, inspection and testing of all primary pressure containing parts of nuclear reactors made of carbon or low alloy steel which fall within either of the categories given below:

Category A. Pressure vessels and components which are intended to contain radioactive materials and are subject to neutron irradiation such that the physical properties of the materials of construction may be affected during the projected life of the nuclear installation or those which, from biological considerations of radioactivity, will be inaccessible for comprehensive periodic examination and maintenance.

Category B. Pressure vessels and components which are intended to contain radioactive materials in service but which on reactor shut-down, will be accessible for comprehensive examination and maintenance at any time during the projected life of the nuclear installation.

Other components of the pressure circuit including such parts of ancillary circuits which can be isolated for inspection from the primary circuit at any time without having to shut the reactor down shall be in accordance with the appropriate British Standard.

This standard does not apply to pressure parts constructed of materials of qualities different from those specified and the use of such materials should be the subject of agreement between the purchaser and the manufacturer.

Where the words “purchaser” and “manufacturer” occur in the text they shall be taken to include representatives of the purchaser and manufacturer or inspectors mutually agreed upon.

1.2 Definitions

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

normal working pressure

the maximum pressure which will occur in the relevant part of the vessel or system, during normal operation

design pressure

a value or values specified by the purchaser (see 3.1.2) used in all calculations to determine the minimum thickness of the various component parts of the vessel

NOTE The design pressure shall be not less than 110 per cent of the normal working pressure.

maximum discharge pressure

the maximum pressure, which may be permitted to build up in the system under appropriate discharge conditions of pressure relief devices

NOTE This will be a maximum of 110 per cent of the design pressure of the various components (see Section 6).

maximum design temperature

the maximum estimated temperature which will occur in the relevant part of the vessel during normal operation due regard being given to the effects of nuclear heating, etc.

NOTE Where uncertainty exists in the estimation of nuclear heating effects it is recommended that a margin be agreed between purchaser and manufacturer to cover this.

minimum design temperature

the minimum temperature which will occur in the relevant part of the vessel during its operating life

nominal design stress

the stress value to be selected in accordance with 2.3 for use in the formulae in 3.2, Basic design

primary circuit

all parts of the pressure circuits containing coolant which is in contact with fuel elements

stress adjustment factor

an arbitrary factor which is used in certain cases to reduce the basic design stress

1.3 Classification

This standard is intended to cover the following categories of pressure vessels:

Category A. Pressure vessels and components which are intended to contain radioactive materials and are subject to neutron irradiation such that the physical properties of the materials of construction may be affected during the projected life of the nuclear installation or those which, from biological considerations of radioactivity, will be inaccessible for comprehensive periodic examination and maintenance.

Category B. Pressure vessels and components which are intended to contain radioactive materials in service but which on reactor shut-down will be accessible for comprehensive examination and maintenance at any time during the projected life of the nuclear installation. In this standard, where no distinction is made, the requirements apply to both categories.

1.4 Information to be supplied by the purchaser and manufacturer

1.4.1 In addition to information to be supplied in accordance with the detailed requirements of this standard, the purchaser shall specify to the manufacturer at the time of the enquiry:

- 1) The name of the inspecting authority representing the purchaser.
- 2) Any requirements in excess of the minimum requirements of this standard.
- 3) Any statutory or other regulations with which the vessel is required to comply.

1.4.2 In addition to information to be supplied in accordance with the detailed requirements of this standard, the manufacturer shall supply to the purchaser and/or the inspecting authority sufficiently detailed drawings to enable the compliance of the vessel to this standard to be established.

1.5 Agreed specification

A document shall be compiled by the manufacturer, covering the specifications for all the materials of construction of the vessel and the acceptance values for all the mechanical tests required by this standard, as agreed between him and the purchaser.

1.6 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 Nuclear Reactor Safety Technical Committee 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 by their contract they either expressly exclude it or else import an arbitration provision in terms extending to the interpretation of this standard. But this provision is limited to questions of interpretation and does not confer on the committee any power or jurisdiction to adjudicate upon the contractual rights or duties of any person under a contract except so far as they may necessarily be affected by the interpretation arrived at by the committee.

Findings or rulings of the committee upon matters of interpretation may be incorporated as “addenda” to the standard and if so incorporated shall be treated as amendments thereto.

Section 2. Materials

2.1 General

This Section covers the selection of materials and nominal design stresses for both Category A and B vessels without distinction, except where otherwise stated.

2.2 Selection of materials

2.2.1 The selection of the materials of construction for pressure containing parts and their integral attachments shall take into account the suitability of the material with regard to fabrication and all the modes of failure relevant to the conditions of operation of the vessel covered by the design. The manufacturer shall justify his choice of materials to meet the requirements to the satisfaction of the purchaser.

2.2.2 All the materials used in the manufacture of pressure parts of pressure vessels constructed to this standard shall be in accordance with the appropriate material specifications listed in Table 1A except as otherwise permitted in this Section.

The materials used in welding shall satisfy the requirements of Sections 4 and 5 of this standard. Attention is drawn to the fact that long term properties of weld metal may differ from those of the parent materials and this shall be taken into account where relevant.

Castings may be used for pressure parts subject to agreement between the purchaser and the manufacturer.

2.2.3 Other carbon and low alloy steels of specified chemical composition *may* be used for pressure parts of pressure vessels by express agreement between the contractor and the purchaser, provided that:

- 1) All the relevant mechanical properties of the material are available from a representative number of production casts.

These shall cover the same properties as those covered in the material specifications listed in Table 1. Information shall also be available to enable the nominal design stresses to be calculated in accordance with the requirements of **2.3** for the relevant design temperatures. Where the design temperature does not exceed 350 °C the creep rupture properties of the material need not be available.

- 2) The design stresses for the material shall be derived in accordance with the requirements of **2.3**.

2.2.4 The materials of construction of non-pressure parts integrally welded to pressure containing parts, shall be so chosen as not to impair the integrity of the pressure part and for Category A components shall comply with **2.2.2** and **2.2.3**.

2.2.5 The materials of construction of components other than pressure containing parts or welded attachments thereto, but which are intended by design to contribute to the integrity of the pressure vessel, shall conform to an approved specification.

2.2.6 Corrosion-resistant linings shall not be included in the computation for the required wall thickness, except in the case of clad steels as may be agreed between manufacturer and purchaser.

2.3 Nominal design stresses

2.3.1 The following criteria have been used to obtain the nominal design stresses given in Table 1 and shall be used to calculate the corresponding values for materials not covered by these tables but otherwise meeting the requirements of **2.2.3**.

$$\left. \begin{array}{l} \text{Nominal design stress} \\ \text{or} \\ \text{or} \end{array} \right\} \begin{array}{l} \frac{R_{20}}{2.7} \\ = \frac{E_t}{1.5} \\ = \frac{S_r}{1.6} \end{array} \text{ whichever is least}$$

where R_{20} = minimum value of U.T.S. specified for the material at room temperature;

E_t = minimum value of yield point (0.2 per cent proof stress) for the material at design temperature or the nearest higher temperature for which it can be determined;

S_r = the average stress for the material to produce rupture in 100 000 hours at design temperature, or if the width of the end of the scatter band of results exceeds ± 20 per cent of the average value, S_r shall be taken as 1.25 times the lowest stress to produce rupture in 100 000 hours at design temperature.

In the special case of plain carbon steels, the use of which is well established by past experience, a nominal design stress of 0.25 U.T.S. at room temperature may be used in the range of design temperatures 0 °C – 350 °C by agreement between the manufacturer and the purchaser (see Foreword).

2.3.2 The following criteria shall be used to calculate the nominal design stresses for bolting materials meeting the requirements of this Section.

One third of the specified minimum yield strength at room temperature

or

One third of the 0.2 per cent proof stress at the design temperature, whichever is lower.

Credit may be taken for the enhancement of properties resulting from heat treatment provided that it can be shown by the manufacturer that such heat treatment remains effective under the design conditions of the bolting.

2.4 Modulus of elasticity

Table 2 gives values of E for the materials listed in Table 1.

Table 1A — Permissible materials (See 2.2.2.)

Form	Material	Specification
Plate	Carbon steel (semi-killed)	BS 1501-151 ^a
	Carbon steel (silicon-killed)	BS 1501-161 ^a
	Carbon manganese steel (semi-killed)	BS 1501-211 ^a
	Carbon manganese steel (semi-killed, niobium treated)	BS 1501-213 ^a
	Carbon manganese steel (silicon-killed)	BS 1501-221 ^a
	Carbon manganese steel (silicon-killed, aluminium treated)	BS 1501-224 ^a
	$\frac{1}{2}$ per cent Chromium $\frac{1}{2}$ per cent molybdenum steel	BS 1501-620A ^b
	$\frac{3}{4}$ per cent Chromium $\frac{1}{2}$ per cent molybdenum steel	BS 1501-620B ^b
Bars and Sections		BS 1502 ^c
Forgings	Carbon steel (silicon-killed)	BS 1503-161 ^b
	Carbon manganese steel (silicon-killed)	BS 1503-221 ^b
	1 per cent Chromium $\frac{1}{2}$ per cent molybdenum steel	BS 1503-620 ^b
Pipes and Tubes	Carbon steel — ordinary duties: hot finished	BS 5301-HFS ^d
	cold drawn	BS 5301-CDS ^d
	elec. resist. welded	BS 3601-ERW ^d
	— high duties: hot finished	BS 3602-HFS ^e
	cold drawn	BS 3602-CDS ^e
	elec. resist. welded	BS 3602-ERW ^e
	Low and medium alloy steel	BS 3604 ^f
	1 per cent Chromium $\frac{1}{2}$ per cent molybdenum steel:	
	hot finished	BS 3604-HF620 ^f
	cold drawn	BS 3604-CD620 ^f
	$1\frac{1}{4}$ per cent Chromium $\frac{1}{2}$ per cent molybdenum steel:	
	hot finished	BS 3604-HF621 ^f
	cold drawn	BS 3604-CD621 ^f
	$\frac{1}{2}$ per cent Chromium $\frac{1}{2}$ per cent molybdenum $\frac{1}{4}$ per cent vanadium steel:	
	hot finished	BS 3604-HF660 ^f
	cold drawn	BS 3604-CD660 ^f
Tubes		BS 3059 ^g

^a BS 1501, "Steels for fired and unfired pressure vessels. Plates".

^b BS 1501-6, "Steels for use in the chemical, petroleum and allied industries".

^c BS 1502, "Steels for fired and unfired pressure vessels. Sections and bars".

^d BS 3601, "Steel pipes and tubes for pressure purposes. Carbon steel: ordinary duties".

^e BS 3602, "Steel pipes and tubes for pressure purposes. Carbon steel: high duties".

^f BS 3603, "Steel pipes and tubes for pressure purposes. Low and medium alloy steel".

^g BS 3059, "Steel boiler and superheater tubes" (Metric units).

Table 1B — Nominal design stresses for plates

BS ref.	Nominal tensile design stress (lbf/in ² × 1 000) at temperature °C															Notes
	100	150	200	250	300	350	400	425	450	475	500	525	550	575	600	
BS 1501 ^a																Use at temperatures above 350 °C subject to special agreement between manufacturer and purchaser.
151-Gr 23A	13.74	13.74	13.74	13.74	12.37	11.34	10.65	10.30	8.25	5.50	3.13					
26A	15.54	15.54	15.54	15.54	13.96	12.80	12.02	11.55	8.25	5.50	3.13					
28A	16.71	16.71	16.71	16.71	15.05	13.79	12.96	11.55	8.25	5.50	3.13					
BS 1501 ^a																Up to and including 1 1/4 in thick. Above 350 °C subject to special agreement between manufacturer and purchaser.
151-Gr 23B	16.72	16.28	15.53	14.19	12.40	11.80	11.36	11.20	8.25	5.50	3.13					
26B	18.99	18.36	17.62	16.14	14.05	13.30	12.85	12.70	8.94	5.81	3.13					
28B	20.45	19.85	18.96	17.33	15.09	14.35	15.90	12.70	8.94	5.81	3.13					
BS 1501 ^a																Over 1 1/4 in, up to and including 2 1/2 in thick. Use at temperatures above 350 °C subject to special agreement between manufacturer and purchaser
151-Gr 23B	15.84	15.23	14.64	13.75	12.40	11.80	11.36	11.20	3.25	5.50	3.13					
26B	17.92	17.17	16.58	15.53	14.05	13.30	12.85	12.70	8.94	5.81	3.13					
28B	19.25	18.51	17.77	16.73	15.09	14.35	13.90	12.70	8.94	5.81	3.13					
BS 1501 ^a																
161-Gr 23A	13.74	13.74	13.74	13.74	12.37	11.34	10.65	10.30	8.25	5.50	3.13					
26A	15.54	15.54	15.54	15.54	13.96	12.80	12.02	11.55	8.25	5.50	3.13					
28A	16.71	16.71	16.71	16.71	15.05	13.79	12.96	11.55	8.25	5.50	3.13					
BS 1501 ^a																Up to and including 1 1/4 in thick.
161-Gr 23B	16.72	16.28	15.53	14.19	12.40	11.80	11.36	11.20	8.25	5.50	3.13					
26B	18.99	18.36	17.62	16.14	14.05	13.30	12.85	12.70	8.94	5.81	3.13					
28B	20.45	19.65	18.96	17.33	15.09	14.35	13.90	12.70	8.94	5.81	3.13					
BS 1501 ^a																Over 1 1/4 in, up to and including 2 1/2 in thick. For over 2 1/2 in thick see material specification.
161-Gr 23B	15.84	15.23	14.64	13.75	12.40	11.90	11.36	11.20	8.25	5.50	3.13					
26B	17.92	17.17	16.58	15.53	14.05	13.30	12.85	12.70	8.94	5.81	3.13					
28B	19.25	18.51	17.77	16.73	15.09	14.35	13.90	12.70	8.94	5.81	3.13					
BS 1501 ^a																Use at temperatures above 350 °C subject to special agreement between manufacturer and purchaser.
211-Gr 26A	15.54	15.54	15.54	15.54	13.96	12.80	12.02	11.64	8.94	5.81	3.13					
28A	16.71	16.71	16.71	16.71	15.05	13.79	12.96	12.54	8.94	5.81	3.13					
30A	17.94	17.94	17.94	17.94	16.10	14.75	13.85	12.70	8.94	5.81	3.13					
32A	19.12	19.12	19.12	19.12	17.15	15.75	14.80	12.70	8.94	5.81	3.13					
BS 1501 ^a																Up to and including 1 1/4 in thick. Use at temperatures above 350 °C subject to special agreement between manufacturer and purchaser.
211-Gr 26B	19.96	19.56	19.12	18.07	15.38	14.20	13.60	12.70	8.94	5.81	3.13					
28B	21.35	21.05	20.60	19.42	16.97	15.24	14.64	12.70	8.94	5.81	3.13					
30B	22.85	22.55	22.10	20.75	17.78	16.29	15.68	12.70	8.94	5.81	3.13					
32B	24.35	24.05	23.60	22.25	18.98	17.47	16.73	12.70	8.94	5.81	3.13					
BS 1501 ^a																Over 1 1/4 in, up to and including 2 1/2 in thick. For over 2 1/2 in thick see material specification. Use at temperatures above 350 °C subject to special agreement between manufacturer and purchaser.
211-Gr 26B	18.81	18.36	18.21	17.32	15.38	14.20	13.60	12.70	8.94	5.81	3.13					
28B	20.31	19.85	19.56	18.67	16.57	15.24	14.64	12.70	8.94	5.81	3.13					
30B	21.80	21.20	20.90	20.00	17.78	16.29	15.68	12.70	8.94	5.81	3.13					
32B	23.15	22.70	22.40	21.35	18.98	17.47	16.73	12.70	8.94	5.81	3.13					

Table 1B — Nominal design stresses for plates

BS ref.	Nominal tensile design stress (lb/in² × 1 000) at temperature °C															Notes
	100	150	200	250	300	350	400	425	450	475	500	525	550	575	600	
BS 1501 ^a																Use at temperatures above 350 °C subject to special agreement between manufacturer and purchaser.
213-Gr 28A	16.71	16.71	16.71	16.71	15.05	13.79	12.96	10.20	7.12	4.50	2.31					
30A	17.94	17.94	17.94	17.94	16.10	14.75	13.75	10.20	7.12	4.50	2.31					
32A	19.12	19.12	19.12	19.12	17.15	15.75	13.75	10.20	7.12	4.50	2.31					
BS 1501 ^a																Up to and including 2½ in thick. For over 2½ in thick see material specification. Use at temperatures above 350 °C subject to special agreement between manufacturer and purchaser.
213-Gr 28B	23.25	23.25	23.25	21.38	18.66	17.31	13.75	10.20	7.12	4.50	2.31					
30B	24.90	24.90	24.90	22.85	20.00	18.66	13.75	10.20	7.12	4.50	2.31					
32B	26.55	26.55	26.55	24.35	21.20	19.86	13.75	10.20	7.12	4.50	2.31					
BS 1501 ^a																
221-Gr 26A	15.54	15.54	15.54	15.54	13.96	12.80	12.02	11.64	8.94	5.81	3.13					
28A	16.71	16.71	16.71	16.71	15.05	13.79	12.96	12.54	8.94	5.81	3.13					
30A	17.94	17.94	17.94	17.94	16.10	14.75	13.85	12.70	8.94	5.81	3.13					
32A	19.12	19.12	19.12	19.12	17.15	15.75	14.80	12.70	8.94	5.81	3.13					
BS 1501 ^a																Up to and including 1¼ in thick.
221-Gr 26B	19.96	19.54	19.12	18.07	15.38	14.20	13.60	12.70	8.94	5.81	3.13					
28B	21.35	21.05	20.60	19.42	16.57	15.24	14.64	12.70	8.94	5.81	3.13					
30B	22.85	22.55	22.10	20.75	17.78	16.29	15.68	12.70	8.94	5.81	3.13					
32B	24.35	24.05	23.60	22.25	18.98	17.47	16.73	12.70	8.94	5.81	3.13					
BS 1501 ^a																Over 1¼ in, up to including 2½ in thick. For over 2½ in thick see material specification.
221-Gr 26B	18.81	18.36	18.21	17.32	15.38	14.20	13.60	12.70	8.94	5.81	3.13					
28B	20.31	19.85	19.56	18.67	16.57	15.24	14.64	12.70	8.94	5.81	3.13					
30B	21.80	21.20	20.90	20.00	17.78	16.29	15.68	12.70	8.94	5.81	3.13					
32B	23.15	22.70	22.40	21.35	18.98	17.47	16.73	12.70	8.94	5.81	3.13					
BS 1501 ^a																
224-Gr 26A	15.54	15.54	15.54	15.54	13.96	12.80	12.02	11.13	7.88	4.16	2.94					
28A	16.71	16.71	16.71	16.71	15.05	13.79	12.96	11.13	7.88	4.16	2.94					
30B	17.94	17.94	17.94	17.94	16.10	14.75	13.85	11.13	7.88	4.16	2.94					
32A	19.12	19.12	19.12	19.12	17.15	15.75	14.80	11.13	7.88	4.16	2.94					
BS 1501 ^a																Up to and including 2½ in thick. For over 2½ in thick see material specification.
224-Gr 26B	20.75	19.56	18.52	16.73	14.94	14.04	13.15	11.13	7.88	4.16	2.94					
28B	22.40	21.05	20.00	17.93	16.13	15.09	14.20	11.13	7.88	4.16	2.94					
30B	24.05	22.70	21.35	19.27	17.33	16.13	15.25	11.13	7.88	4.16	2.94					
32B	25.53	24.30	22.85	20.60	18.37	17.18	16.29	11.13	7.88	4.16	2.94					
BS 1501 ^b																
620-GR A	15.25	15.25	15.25	15.25	14.55	14.55	14.55	14.43	14.30	13.45	11.90	8.25	5.15	3.50		
B	17.00	17.00	17.00	17.00	15.70	15.70	15.10	14.70	14.30	13.45	11.90	8.25	5.15	3.50		

NOTE 1 The nominal design stress values are based on the heat treatment limitations given in the appropriate material specifications. In some cases heat treatment outside these limits may be required. Where this is the case it is essential that due consideration be given to ensure that the properties of the material are not affected to a degree which is unacceptable within the criteria given in 3.3.

NOTE 2 The nominal design stresses for grades “A” of BS 1501 have been derived from typical values.

^a BS 1501, “Steels for fired and unfired pressure vessels. Plates”.

^b BS 1501-6, “Steels for use in the chemical, petroleum and allied industries”.

Table 1C — Nominal design stresses for bars and sections

Specification	Nominal tensile design stress (lbf/in ² × 1 000) at temperature °C																	Notes
	100	150	200	250	300	350	400	425	450	475	500	525	550	575	600	625	650	
BS 1502																		

The range of properties for use in conjunction with Section 2 is not yet available. When BS 1502 is published the appropriate values will be inserted in this table. Until BS 1502 is published nominal design stress values shall be agreed between the manufacturer and the purchaser.

Table 1D — Nominal design stresses for forgings

Specification	Nominal tensile design stress (lbf/in ² × 1 000) at temperature °C																	Notes
	100	150	200	250	300	350	400	425	450	475	500	525	550	575	600	625	650	
BS 1503																		

The range of properties for use in conjunction with Section 2 is not given in BS 1503 at present. BS 1503 is being revised to incorporate these properties and when published, the appropriate values will be inserted in this table. Until the revision is published nominal design stress values shall be agreed between the manufacturer and the purchaser.

Table 1E — Nominal design stresses for pipes

BS ref.	Nominal tensile design stress (lbf/in ² × 1 000) at temperature °C															Notes
	100	150	200	250	300	350	400	425	450	475	500	525	550	575	600	
BS 3601 ^a																Category “B” components only
HFS.22	13.10	13.10	13.10	13.10												
HFS.27	16.10	16.10	16.10	16.10												
CDS.22	13.10	13.10	13.10	13.10												
CDS.27	16.10	16.10	16.10	16.10												
ERW.22	13.10	13.10	13.10	13.10												
ERW.27	16.10	16.10	16.10	16.10												
BS 3602 ^b																Not to be manufactured of rimming steel.
HFS.23	16.14	16.14	16.14	16.14	13.30	11.65										
CDS.23	16.14	16.14	16.14	16.14	13.30	11.65										
ERW.23	16.14	16.14	16.14	16.14	13.30	11.65										
HFS.27	17.33	17.33	17.33	17.33	15.24	13.30										
CDS.27	17.33	17.33	17.33	17.33	15.24	13.30										
ERW.27	17.33	17.33	17.33	17.33	15.24	13.30										
BS 3602 ^b																Not to be manufactured of oxygen blown steel. Use at temperatures above 425 °C subject to special agreement between manufacturer and purchaser.
HFS.23H	16.14	16.14	16.14	16.14	13.30	11.65	11.05	8.40	6.13	4.43	3.13					
CDS.23H	16.14	16.14	16.14	16.14	13.30	11.65	11.05	8.40	6.13	4.43	3.13					
ERW.23H	16.14	16.14	16.14	16.14	13.30	11.65	11.05	8.40	6.13	4.43	3.13					
HFS.27H	17.33	17.33	17.33	17.33	15.24	13.30	11.85	8.40	6.13	4.43	3.13					
CDS.27H	17.33	17.33	17.33	17.33	15.24	13.30	11.85	8.40	6.13	4.43	3.13					
ERW.27H	17.33	17.33	17.33	17.33	15.24	13.30	11.85	8.40	6.13	4.43	3.13					
BS 3602 ^b																Not to be manufactured of oxygen blown steel.
HFS.23HS	16.14	16.14	16.14	16.14	13.30	11.65	11.05	8.40	6.13	4.43	3.13					
CDS.23HS	16.14	16.14	16.14	16.14	13.30	11.65	11.05	8.40	6.13	4.43	3.13					
HFS.27HS	17.33	17.33	17.33	17.33	15.24	13.30	11.85	8.40	6.13	4.43	3.13					
CDS.27HS	17.33	17.33	17.33	17.33	15.24	13.30	11.85	8.40	6.13	4.43	3.13					
ERW.27HS	17.33	17.33	17.33	17.33	15.24	13.30	11.85	8.40	6.13	4.43	3.13					

Table 1E — Nominal design stresses for pipes

BS ref.	Nominal tensile design stress (lb/in ² × 1 000) at temperature °C															Notes
	100	150	200	250	300	350	400	425	450	475	500	525	550	575	600	
BS 3604 ^c																
CD.620	16.50	16.50	16.50	16.50	15.10	15.10	15.10	14.70	14.30	15.45	11.90	8.25	5.15	3.50		
HF.620	16.50	16.50	16.50	16.50	15.10	15.10	15.10	14.70	14.30	15.25	11.90	8.25	5.15	3.50		
CD.621	16.50	16.50	16.50	16.50	15.10	15.10	15.10	14.70	14.30	13.45	11.90	8.25	5.15	3.50		
HF.621	16.50	16.50	16.50	16.50	15.10	15.10	15.10	14.70	14.30	13.45	11.90	8.25	5.15	3.50		
CD.660	21.50	21.50	21.50	21.50	20.00	18.25	16.50	15.75	14.50	13.50	12.25	11.00	9.95	7.00	4.20	
HF.660	21.50	21.50	21.50	21.50	20.00	18.25	16.50	15.75	14.50	13.50	12.25	11.00	9.95	7.00	4.20	
NOTE 1 The nominal design stress values are based on the heat treatment limitations given in the appropriate material specifications. In some cases heat treatment outside these limits may be required. Where this is the case it is essential that due consideration be given to ensure that the properties of the material are not affected to a degree which is unacceptable within the criteria given in 3.3. NOTE 2 The nominal design stresses have been derived from typical values for pipes to BS 3601, BS 3604 (all values) and BS 3602. ^a BS 3601, “Steel pipes and tubes for pressure purposes. Carbon steel: ordinary duties”. ^b BS 3602, “Steel pipes and tubes for pressure purposes. Carbon steel: high duties”. ^c BS 3604, “Steel pipes and tubes for pressure purposes: Low and medium alloy steel”.																

Table 1F — Nominal design stresses for tubes

BS Specification	Grade	Nominal design stress (10 ³ lbf/in ²) at temperature °C											Notes
		100	150	200	250	300	350	400	450	500	550	600	
BS 3059	STEEL												
PART 1	33	16.60	15.20	13.80	12.55	11.30	10.35						C/STEEL 33/19
PART 2	33	17.00	15.90	14.90	13.84	11.75	10.90	10.33	6.33	2.94			C/STEEL 33/19
PART 2	45	22.70	21.70	20.75	19.80	17.35	15.35	14.60	7.70	3.38			C/STEEL 45/25
PART 2	620	22.00	21.35	20.70	20.00	19.30	18.65	17.15	16.40	13.45	4.51		1 Cr 1/2 Mo 45/24
PART 2	622/42	21.00	19.60	18.00	16.50	14.80	13.27	12.80	12.50	12.15	7.57	3.38	2 1/4 Cr 1 Mo 42/24
PART 2	622/50	24.75	24.00	23.25	22.35	21.60	20.80	19.90	18.94	15.30	7.57	3.38	2 1/4 Cr 1 Mo 50/27

Table 2 — Values of *E*

Temperature °C	Value of <i>E</i> lbf/in ² × 10 ⁶
0	30
20	29.6
100	28.66
200	27.53
300	26.0
400	24.66
500	23.33
600	22

Intermediate values may be obtained by linear interpolation.

Section 3. Design

3.1 General

3.1.1 This Section covers the design of Category A and B vessels without distinction unless otherwise stated.

3.1.2 The purchaser shall issue to the manufacturer a design specification for vessels covered by this standard which shall specify the normal working conditions of the vessel together with such combinations of fault or transient conditions as the vessel is required to withstand.

3.1.3 For Category A vessels the manufacturer shall submit to the purchaser a design report containing sufficient information to show that the design is in accordance with the requirements of this Section. Where the vessel is provided with devices to maintain it at a temperature lower than that of its contents and where failure of such devices could give rise to higher temperatures, account shall be taken of the possible consequences of such a failure.

The design report for Category B vessels need not be submitted but the manufacturer shall submit such calculations in **3.2** as may be requested by the purchaser.

The design report should be approved on behalf of the manufacturer by a Chartered Engineer with adequate experience of the design of pressure vessels.

In case of dispute regarding suitability of the design for pressure loading the manufacturer shall have the option of demonstrating the suitability of the vessel by extending the scope of the pressure test (see **5.10**) in a manner to be agreed with the purchaser.

3.1.4 Clause **3.2**, “*Basic design*”, covers the determination of the preliminary scantlings of shells, ends, openings, flanges and tube plates in relation to:

- 1) internal and external pressure loading;
- 2) weight of vessel and contents where simple design formulae can be given.

Forms of construction other than those specifically covered in **3.2** are permitted provided that it is demonstrated to the satisfaction of the purchaser by sound analytical methods, experimental techniques or relevant past experience that they meet the requirements of **3.3**.

3.1.5 Clause **3.3**, “*Final design appraisal*”, lists additional factors which shall be taken into account where relevant. Various modes of failure which have to be considered in relation to these are also listed.

3.2 Basic design

3.2.1 Shells subject to internal pressure. The following formulae for calculating the thickness of shells subject to internal pressure are based on the mean diameter membrane theory and do not apply to vessels with a thickness exceeding 25 per cent of the internal diameter.

For the purposes of the calculations in **3.2.1.1** to **3.2.1.3**, the following symbols apply:

- t = minimum thickness in inches of shell;
- p = design pressure, lbf/in²;
- D_i = inside diameter in inches of shell before the internal corrosion allowance has been added;
- D_o = outside diameter in inches of shell before the external corrosion allowance has been added;
- f = nominal design stress, lbf/in² from Table 1;
- J = a stress adjustment factor of:
 - 0.95 for Category A vessels
 - 1.00 for all other vessels;
- c = an allowance for corrosion or erosion (see **3.3.2.4**);
- E = modulus of elasticity, lbf/in² of the material at the design temperature from Table 2.

3.2.1.1 Cylindrical shells. The minimum thickness of shells shall be determined by equation (1) or (2):

$$t = \frac{pD_i}{2fJ - p} + c \quad (1)$$

$$t = \frac{pD_o}{2fJ + p} + c \quad (2)$$

3.2.1.2 Spherical shells. The minimum thickness of shells shall be determined by equation (3) or (4):

$$t = \frac{pD_i}{4fJ - p} + c \quad (3)$$

$$t = \frac{pD_o}{4fJ + p} + c \quad (4)$$

3.2.1.3 Combined loadings

a) *Cylindrical shells. Vertical.* For vertical shells the maximum axial membrane stress resulting from the combined effects of internal pressure, and the total weight of the vessel, attachments and contents shall be calculated in accordance with equations (5) and (6):

i) Under operating conditions:

$$f \geq \left[\frac{pD_o}{4(t-c)} + \frac{4M}{\pi D_o^2(t-c)} - \frac{W}{\pi D_o(t-c)} \right] \frac{1}{J} \quad (5)$$

NOTE Negative values denote compressive stresses.

ii) Under conditions of atmospheric internal pressure:

$$f \geq \left[\frac{4M}{\pi D_o^2(t-c)} + \frac{W}{\pi D_o(t-c)} \right] \frac{1}{J} \quad (6)$$

where:

M = bending moment at the cross section under consideration due to the weight W , (lbf in);

W = total weight at the cross section under consideration (lb).

The value of maximum axial stress obtained from equation (5) or equation (6) shall not exceed the design stress at design temperature shown in Table 1 and in no case shall the compressive stress, indicated by equation (6), or a negative value of equation (5), exceed the value of

$$0.125E \left(\frac{t-c}{D_o} \right) \text{ lbf/in}^2.$$

b) *Cylindrical shells. Horizontal.* For horizontal shells supported at two or more points in their length, the maximum axial membrane stress resulting from internal pressure and bending moment due to the weight of the vessel and its contents, shall be calculated as follows:

i) Under operating conditions:

$$f \geq \left[\frac{pD_o}{4(t-c)} + \frac{4M}{\pi D_o^2(t-c)} \right] \frac{1}{J} \quad (7)$$

ii) Under conditions of atmospheric internal pressure:

$$f \geq \left[\frac{4M}{\pi D_o^2(t-c)} \right] \frac{1}{J} \quad (8)$$

The value of maximum axial stress obtained from equation (7) or equation (8) shall not exceed the design stress at design temperatures shown in Table 1B, and in no case shall the compressive stress obtained from equation (8) exceed the value of

$$0.125E \left(\frac{t-c}{D_o} \right) \text{ lbf/in}^2.$$

3.2.2 Shells subject to external pressure. The rules given herein apply to the design of cylindrical shells with or without stiffening rings and spherical shells, for which the chart, Figure 1, shall be used to determine the minimum shell thickness. The value of the design pressure to be used in calculations shall be not less than the maximum pressure differential that may occur between the outside and inside of the shell in question. The possible influence of steam or vapour condensation should be considered. Superimposed plate stresses resulting from local supports (see 3.3.1), should be compensated for, preferably by increasing the shell plate thickness at the supports.

For the purposes of the calculations in 3.2.2.1 to 3.2.2.3 the following symbols apply:

- t = minimum thickness of cylindrical or spherical shell, inches;
- p = Design external pressure, lbf/in²;
- D_o = outside diameter of cylindrical or spherical shell, inches;
- D_s = diameter of the neutral axis of the stiffener, inches;
- R_o = external radius of spherical vessel, inches;
- L = effective length of cylindrical shell, inches; (see Figure 2 to Figure 6);
- f = nominal design stress, lbf/in² from Table 3;
- E = modulus of elasticity, lbf/in² of the material at the design temperature from Table 2;
- I = moment of inertia, in⁴, of stiffening rings;
- c = an allowance for corrosion or erosion (see 3.3.2.4).

3.2.2.1 Cylindrical shells. The minimum thickness of a cylindrical shell under external pressure shall be not less than that determined by the following procedure using the graph in Figure 1.

Assume a value for the corroded thickness $(t - c)$ and determine the ratios $\frac{L}{D_o}$ and $\frac{D_o}{(t - c)}$. Enter the chart,

Figure 1, at the horizontal scale value equal to $\frac{D_o}{(t - c)}$ and move vertically to the sloping line representing the value of $\frac{L}{D_o}$.

Move horizontally to the left and read off the value of factor A on the vertical scale. Enter Table 3 with the value of A , read off the design stress corresponding to the design temperature and enter this stress in the expression $p = \frac{2f(t - c)}{D_o}$. If the value of p so obtained is less than the design value, the chosen value of t

must be increased and the chart re-entered with a new value of $\frac{D_o}{(t - c)}$ such that the required design pressure is obtained.

NOTE The above procedure is based on the assumption that the value of L is already established. Examples in Appendix D of BS 1500, Part 1 illustrate the procedure to be adopted in the case of an existing vessel when it is the value of L that is required.

3.2.2.2 Cylindrical shells with stiffening rings subject to external pressure

a) Intermediate stiffening rings composed of structural shapes welded continuously to the inside or outside of the shell shall be such that together with a permissible length of shell on each side of the stiffener (see Figure 7) the moment of inertia I of the combined section is not less than:

$$I = \frac{0.18 D_o p L D_s^2}{E} \quad (9)$$

The permissible length of shell (L_s) which may be regarded as effectively contributing to the moment of inertia of the stiffening section shall be calculated as follows:

$$L_s = 0.75 \sqrt{(t - c) D_o} \quad (10)$$

but in no case shall L_s be taken greater than L .

- b) Stiffening rings not welded to the shell but fitted inside the vessel in close contact with the shell and provided with adequate lateral support shall have a moment of inertia I not less than that required by equation (9). In this case no support by the shell plate shall be taken into account.
- c) Stiffening rings shall extend completely around the circumference of the shell and any joints shall be so designed as to develop the full stiffness of the ring.
- d) Stiffening rings arranged with local spaces between the shell and ring, as shown in Figure 7, shall be the subject of special consideration, but in no case shall the length of unsupported shell plate exceed the value given in Figure 8.

3.2.2.3 Spherical shells. The minimum thickness of a spherical shell under external pressure shall be not less than that determined by the following procedure using the graph in Figure 1.

Assume a value for corroded thickness $(t - c)$ and determine the ratio $\frac{R_o}{(t - c)}$. Enter the chart, Figure 1, at

the horizontal scale value equal to $\frac{R_o}{(t - c)}$ and move vertically to the sphere line. Move horizontally to the

left and read off the value of Factor A on the vertical scale. Enter Table 3 with this value of A , read off the design stress corresponding to the design temperature and enter this stress in the expression

$p = \frac{2f(t - c)}{R_o}$. If the value of p so obtained is less than the design value, the chosen value of t must be

increased and the chart re-entered with a new value of $\frac{R_o}{(t - c)}$ such that the required design pressure is obtained.

3.2.2.4 Combined loading

a) Cylindrical shells.

- i) *Vertical.* For vertical shells subject to external pressure the maximum axial membrane compressive stress, resulting from the combined effects of external pressure and the total weight of the vessel, attachments and contents, shall be calculated in accordance with equation (11)

$$f \geq \left[\frac{pD_o}{4(t - c)} + \frac{4M}{\pi D_o^2(t - c)} + \frac{W}{\pi D_o(t - c)} \right] \frac{1}{J} \quad (11)$$

where:

p = nominal design external pressure, lbf/in²;

M = bending moment at the cross section under consideration due to the weight W , (lbf ft);

W = total weight at the cross section under consideration (lb).

The value of maximum axial stress obtained from equation (11) shall not exceed the design stress at design temperature taken from Table 1, and in no case shall it exceed the value of

$$0.125E \left(\frac{t - c}{D_o} \right) \text{ lbf/in}^2.$$

- ii) *Horizontal.* For horizontal shells supported at two or more points on their length, the maximum axial membrane compressive stress resulting from external pressure and bending moment due to the weight of the vessel and its contents, shall be calculated as follows:

$$f \geq \left[\frac{pD_o}{4(t - c)} + \frac{4M}{\pi D_o^2(t - c)} \right] \frac{1}{J} \quad (12)$$

The value of maximum axial stress obtained from equation (12) shall not exceed the design stress at design temperature taken from Table 1, and in no case shall it exceed the value of

$$0.125E \frac{(t - c)}{D_o} \text{ lbf/in}^2.$$

3.2.3 Protective linings. Linings shall not be included in the required thickness, except in the case of clad steels as may be agreed between manufacturer and purchaser.

3.2.4 Flat ends and covers

3.2.4.1 General

a) Flat ends and covers shall be designed in accordance with the formulae in 3.2.4.2.

b) Figure 9 shows typical flat ends and flat bolted covers, with approved methods of attachment to the shell.

NOTE Other types of flat end may be used by agreement between manufacturer and purchaser but any significant discontinuity stresses arising therefrom should receive special consideration and the value of K used in equation 13b be chosen to limit these stresses to an acceptable figure. In general such values of K should be of the order of 3 or 4.

For the purposes of the calculations in 3.2.4.2 the following symbols apply:

t = minimum thickness of end in inches; anywhere within the gasket inside diameter;

p = design pressure, lbf/in² (internal or external);

D = diameter in inches of the plate measured as indicated in Figure 9;

r = knuckle radius, in inches, as indicated in Figure 9;

f = nominal design stress, lbf/in², from Table 1;

c = an allowance for corrosion or erosion (see 3.3.2.4);

K = a constant to be used in equation 13b only which depends upon method of attachment to shell and is given in Column 4 of Figure 9c, Figure 9d and Figure 9e.

t_1 = minimum thickness of end, in inches, anywhere within the cover outside diameter;

G = diameter, in inches, at location of gasket reaction (see 3.2.9.4);

W = design bolt load, lbf, for bolting-up condition [see 3.2.9.5 c)];

h_G = radial distance from gasket load reaction to the bolt circle, inches.

3.2.4.2 Determination of thickness. The minimum thickness of flat ends and covers shall be calculated from equations (13a), (13b), (13c) and (13d) according to type (see Figure 9).

For types Figure 9

$$\text{a and b:} \quad t = \left(\frac{D}{2} - r \right) \sqrt{\frac{p}{f}} + \frac{pr}{2f} + c \quad (13a)$$

$$\text{For type Figure 9c:} \quad t = D \sqrt{\frac{p}{Kf}} + c \quad (13b)$$

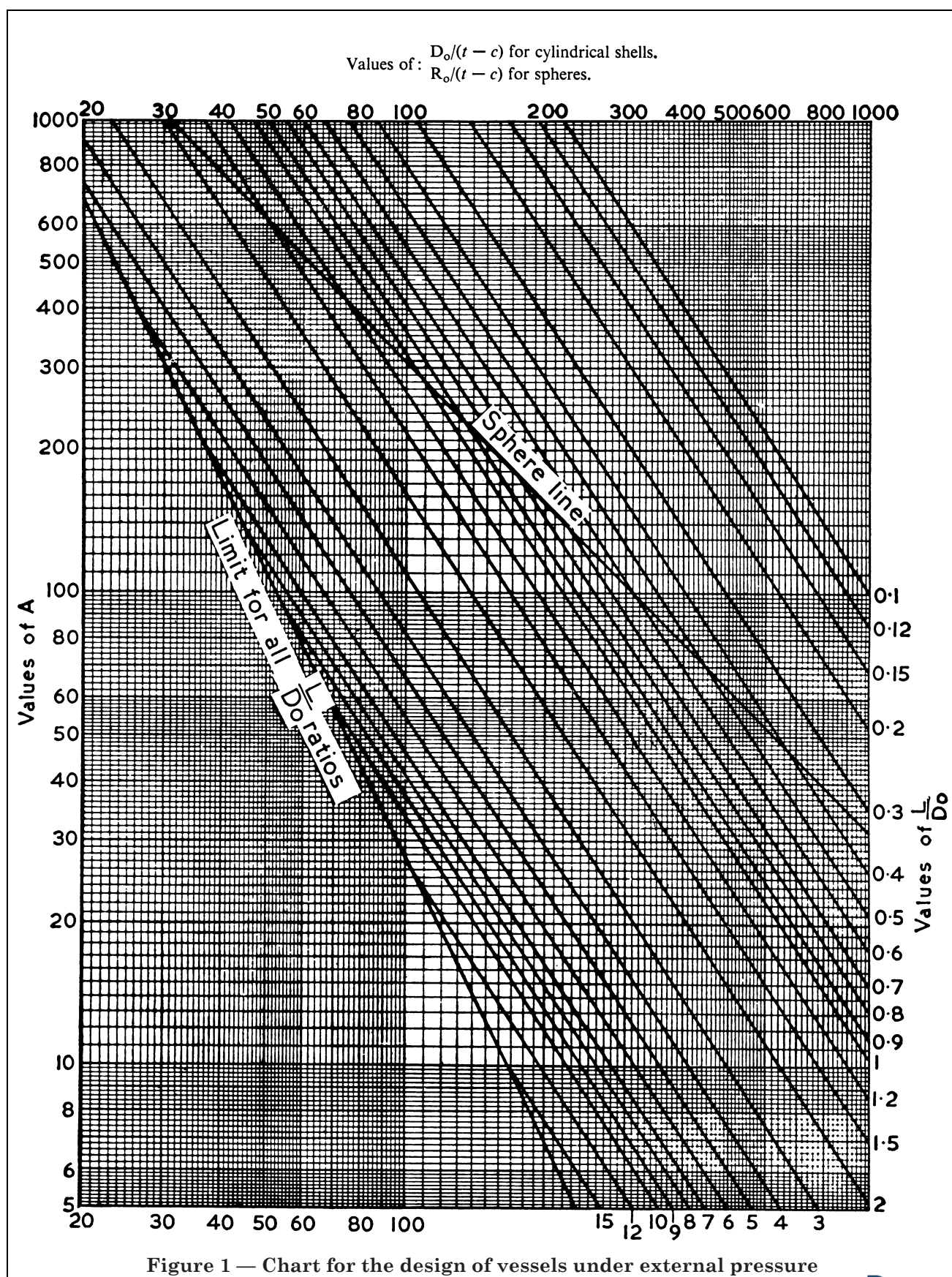
$$\text{For types Figure 9d and Figure 9e:} \quad t = G \sqrt{\frac{p}{Kf}} + c \quad (13c)$$

$$\text{and} \quad t_1 = \sqrt{\frac{1.909Wh_G}{Gf}} + c \quad (13d)$$

Table 3 — Values of f (lbf/in²) for vessels subject to external pressure

Design temp. °F	Values of factor A from Figure 1														
	Up to 100	120	140	160	180	200	250	300	400	500	600	700	800	900	1 000
	Design stress lbf/in ²														
(21 °C) 70	30.4	3 600	4 200	4 800	5 400	6 000	7 000	8 000	9 500	10 600	11 400	12 000	12 200	12 400	12 500
(38 °C) 100	29.8A	3 600	4 200	4 750	5 350	5 950	6 950	7 950	9 450	10 550	11 350	11 900	12 100	12 300	12 400
(91 °C) 200	29.3A	3 500	4 100	4 700	5 300	5 850	6 850	7 850	9 300	10 350	11 000	11 500	11 750	11 900	12 000
(149 °C) 300	28.8A	3 450	4 050	4 600	5 200	5 750	6 700	7 700	9 100	10 100	10 650	11 050	11 300	11 500	11 600
(204 °C) 400	28.2A	3 400	3 950	4 500	5 100	5 650	6 600	7 500	8 850	9 800	10 300	10 650	10 850	11 000	11 100
(260 °C) 500	27.4A	3 300	3 850	4 400	4 950	5 500	6 400	7 250	8 500	9 300	9 800	10 050	10 250	10 400	10 500
(315 °C) 600	26.5A	3 200	3 700	4 250	4 800	5 300	6 200	6 950	8 000	8 800	9 200	9 400	9 600	9 700	9 800
(371 °C) 700	25.5A	3 050	3 550	4 100	4 600	5 100	5 900	6 600	7 500	8 100	8 400	8 600	8 800	8 900	9 000
(427 °C) 800	24.3A	2 900	3 400	3 900	4 400	4 850	5 400	6 000	6 800	7 200	7 500	7 600	7 700	7 800	7 900
(482 °C) 900	22A	2 600	3 000	3 400	3 800	4 100	4 500	4 700	4 850	4 900	5 000	5 000	5 000	5 000	5 000

Intermediate values may be interpolated.



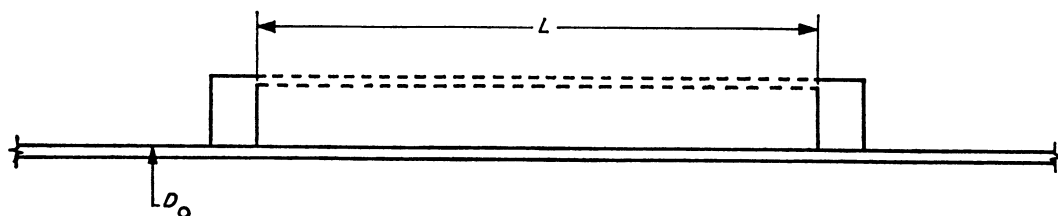


Figure 2

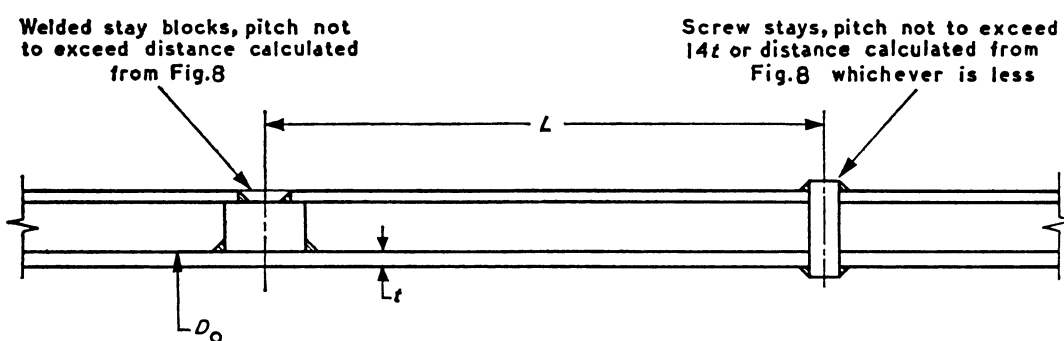


Figure 3

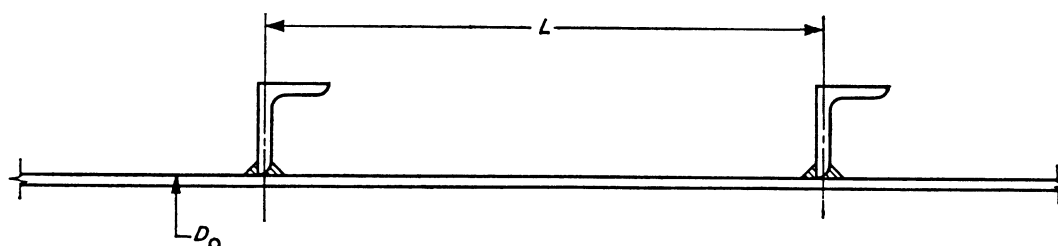


Figure 4

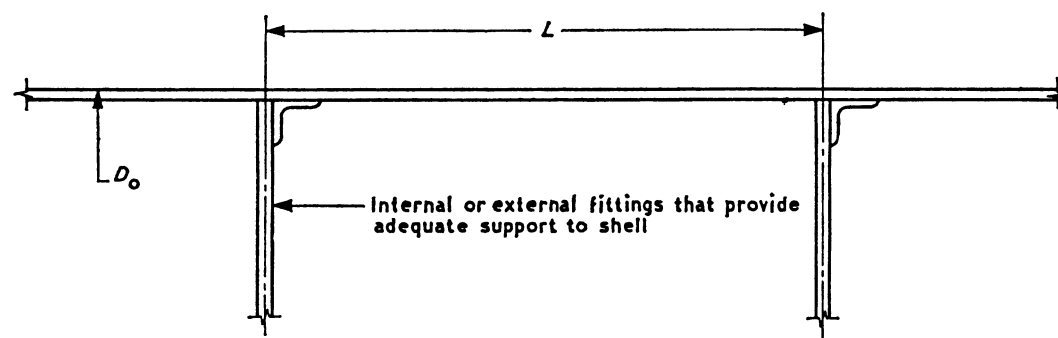
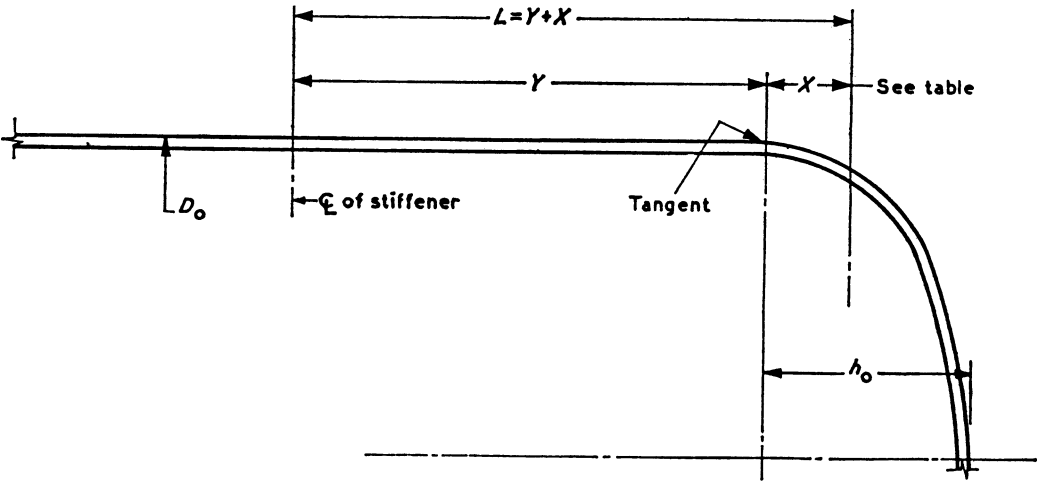


Figure 5

Figure 2–Figure 5 — Effective length of cylinders subject to external pressure



h_o/D_o	X/D_o
0.169	0.07
0.20	0.08
0.25	0.10
0.30	0.12
0.40	0.16
0.50	0.20

NOTE Allowance “X” to be made for each end of vessel.

Figure 6 — Effective length of cylinders subject to external pressure

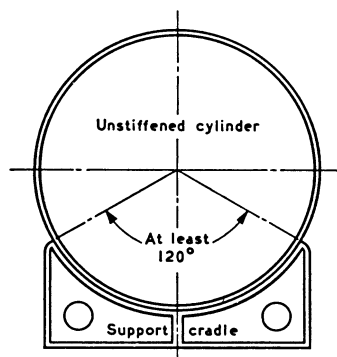
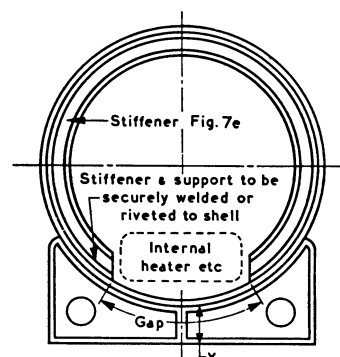


Figure 7a



This type of construction when the gap is greater than length of arc to Fig. 8. Moment of inertia of support at 'X' not to be less than I of stiffener.

Figure 7b

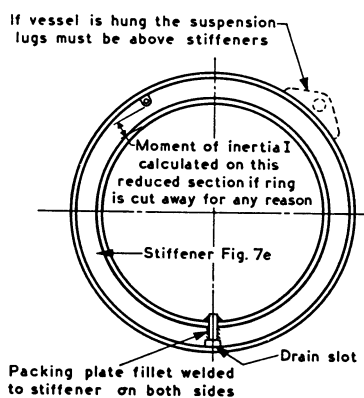


Figure 7c

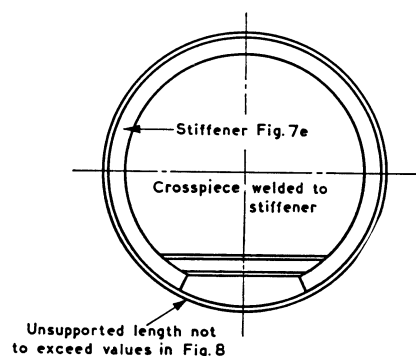


Figure 7d

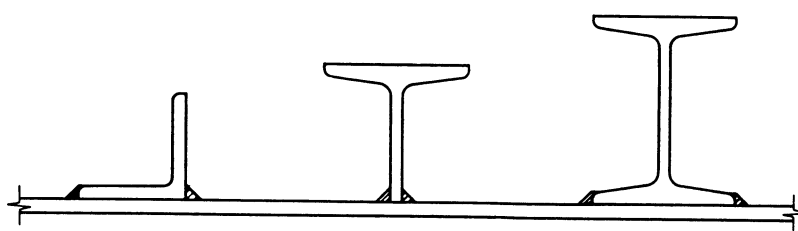


Figure 7e — Typical stiffeners continuously welded to shell

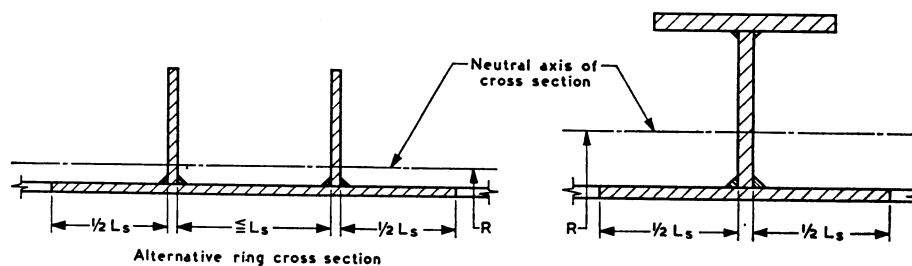


Figure 7f

Figure 7 — Stiffening rings for cylindrical vessels subject to external pressure

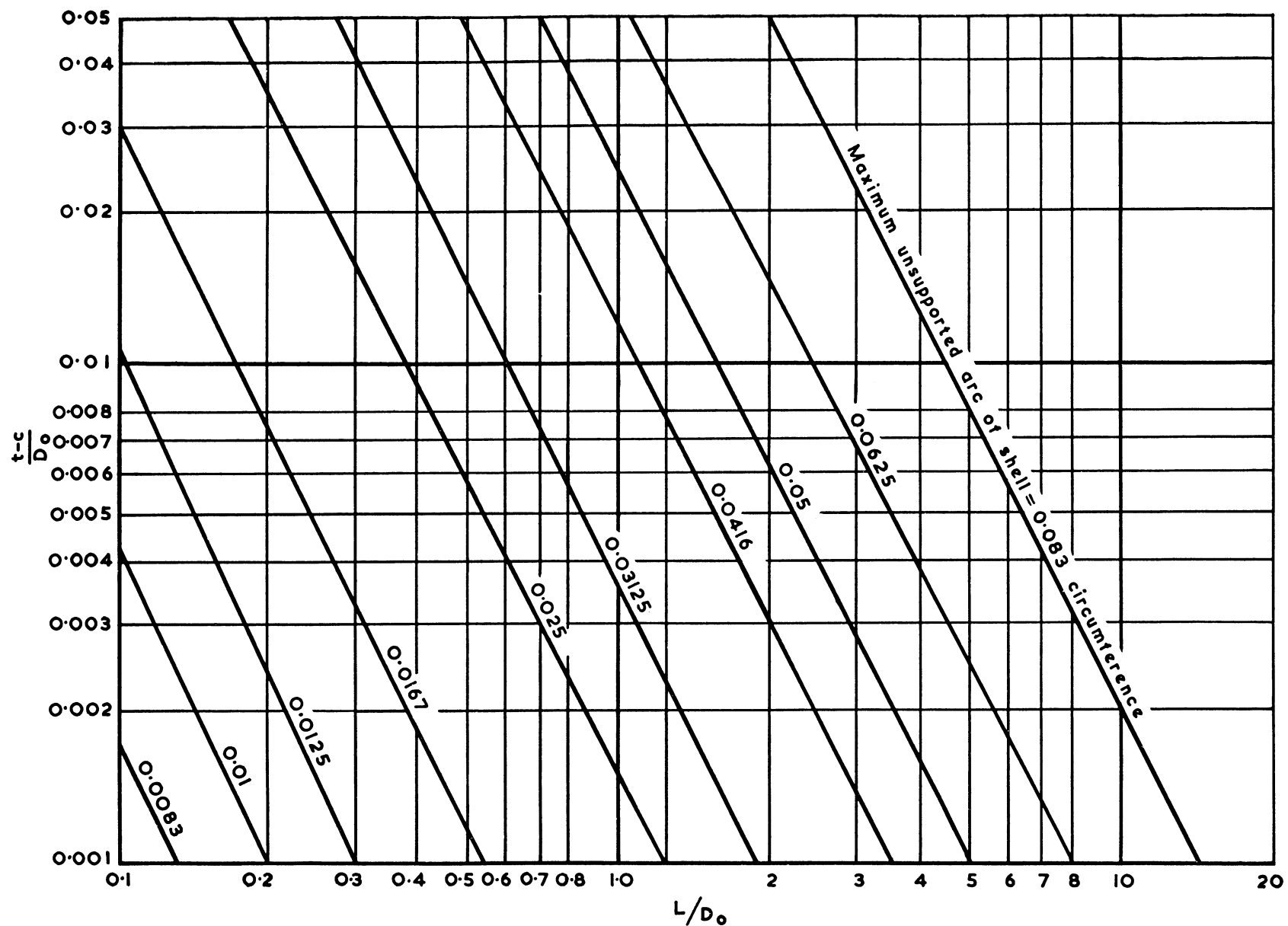


Figure 8 — Maximum permissible arc of shell unsupported by stiffening rings, etc.

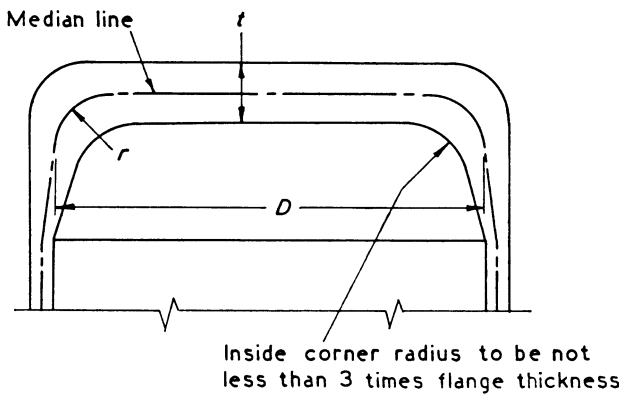
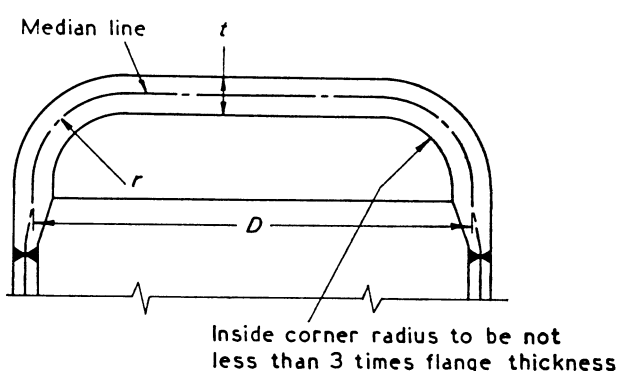
Figure	Type
Figure 9a	
Figure 9b	

Figure 9 — Typical flat ends and covers (*continued*)

Figure	Type	Limitations	K
Figure 9c		Gasket shall be full width of flange	6.0
Figure 9d		$K = \frac{1}{\left[0.30 + \frac{1.5 \times W_{m1} \times h_G}{H \times G} \right]}$ W_{m1} = minimum required bolt load for operating conditions, lbf; h_G = radial distance from gasket load reaction to the bolt circle, inches; H = total hydrostatic force, lbf; G = diameter at location of gasket reaction, inches.	
Figure 9e			

Figure 9 — Typical flat ends and covers (concluded)

3.2.4.3 Fixings of flat bolted covers. Where bolted covers and ends as in Figure 9c, Figure 9d and Figure 9e are used, the mating flanges and bolt sizes shall comply with 3.2.9.

3.2.5 Conical ends

3.2.5.1 General. Conical ends and conical reducing sections as shown in Figure 10 shall be designed in accordance with the formulae in 3.2.5.2.

For the purposes of the calculations in 3.2.5.2 the following symbols apply:

t	= minimum thickness, in inches, of conical section or end;
p	= design pressure, lbf/in ² ;
f	= nominal design stress, lbf/in ² , as listed in Table 1;
J	= a stress adjustment factor of 0.95 for Category A vessels and 1.00 for all other vessels;
D_c	= inside diameter, in inches, of conical section or end at the position under consideration before corrosion allowance is added (see Figure 10);
D_o	= outside diameter, in inches, of conical section or end (see Figure 10);
r_i	= inside radius of transition knuckle, in inches, which shall be taken as 0.01 D_c in the case of conical sections without knuckle transition;
c	= an allowance for corrosion or erosion (see 3.3.2.4);
$\alpha, \alpha_1, \alpha_2$	= angle of slope of conical section (at the point under consideration) to the vessel axis (see Figure 10);
Ψ	= difference between angle of slope of two adjoining conical sections (see Figure 10);
K	= a factor, taking into account the stress in the knuckle (see Table 4);
L	= distance, in inches, from knuckle or junction within which meridional stresses determine the required thickness (see Figure 10),

$$= 0.5 \sqrt{\frac{D_o(t - c)}{\cos \psi}}$$

3.2.5.2 Conical ends subject to internal pressure

a) *Design method.* The stresses in the knuckle or in the circumferential seam at the wide end of the cone acting in a meridional direction are predominantly bending stresses and the stresses acting in a circumferential direction are predominantly membrane stresses. The design takes into account each of these effects in turn in b) and c) below. The greater of the two wall thicknesses so derived governs.

For shallow cones, the thickness may be determined by the method in d), even though the resulting wall thicknesses may be less than those determined under b) and c).

b) *Thickness of knuckle or conical section at junction.* The thickness of cylinder and conical section within a distance L from the junction shall be determined by:

$$t = \frac{pD_oK}{2fJ} + c \quad (14a)$$

If the distance of a circumferential seam from the knuckle or junction is less than L then J shall be taken as 1.00 otherwise J shall be the factor appropriate to the circumferential seam.

c) *Thickness of conical section away from junction.* The thickness of those parts of conical sections not less than a distance L away from the junction with a cylinder or other conical sections shall be determined by:

$$t = \left(\frac{pD_c}{2fJ - p} \right) \left(\frac{1}{\cos \alpha} \right) + c \quad (14b)$$

d) *Shallow conical sections.* The thickness of conical sections having an angle of inclination to the vessel of more than 75° shall be determined as for a flat end as prescribed in 3.2.4.

3.2.5.3 Conical ends subject to external pressure. For a conical end or conical section (frustum) under external pressure, whether the end is of seamless or butt welded construction, the general provisions of b), c) and d) above are applicable, except that the thickness shall be not less than as prescribed below:

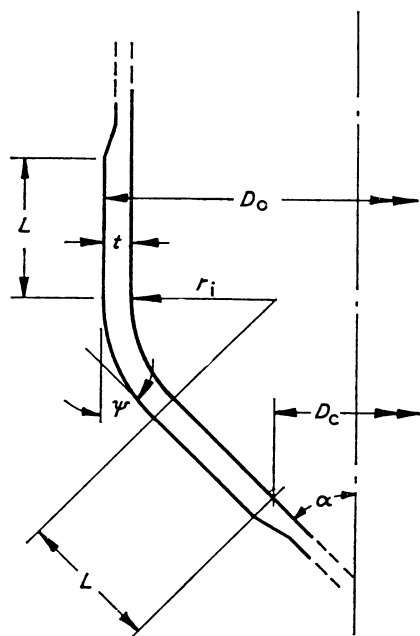
a) The thickness of a conical end or conical section under external pressure, when the angle of inclination of the conical section (at the point under consideration) to the vessel axis is not more than 75° , shall be made equal to the required thickness of a cylindrical shell, in which the diameter is $D_o/\cos \alpha$ and the effective length is equal to the slant height of the cone or conical section or slant height between the effective stiffening rings, whichever is less.

b) The thickness of conical ends having an angle of inclination to the vessel axis of more than 75° shall be determined as for a flat cover as prescribed in 3.2.4.

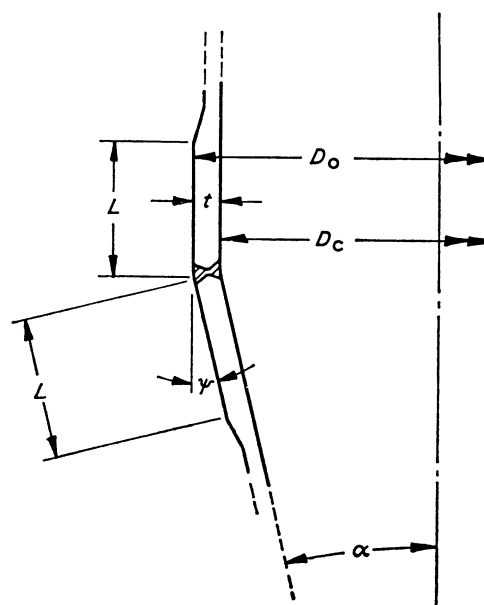
3.2.5.4 Constructional details. Connections between cylindrical shells and conical sections and ends shall preferably be by means of a knuckle transition radius. Typical permitted details are shown in Figure 10. Alternatively conical sections and ends may be butt welded to cylinders without a knuckle radius when the change in angle of slope ψ between the two sections under consideration does not exceed 30° .

Table 4 — Values of K as a function of ψ and r_i/D_o

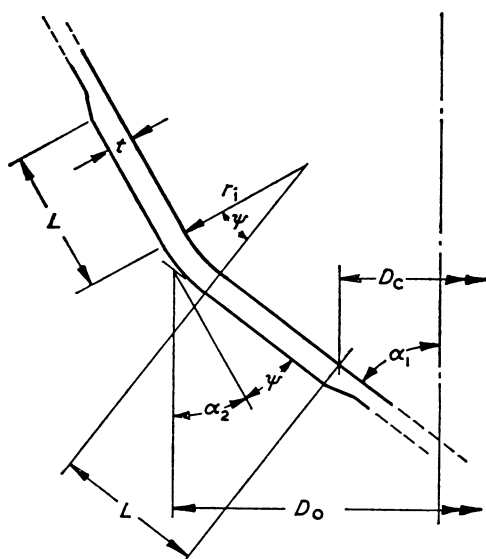
ψ	Values of K for r_i/D_o ratios of											
	0.01	0.02	0.03	0.04	0.06	0.08	0.10	0.15	0.20	0.30	0.40	0.50
10°	0.70	0.65	0.60	0.60	0.55	0.55	0.55	0.55	0.55	0.55	0.55	0.55
20°	1.00	0.90	0.85	0.80	0.70	0.65	0.60	0.55	0.55	0.55	0.55	0.55
30°	1.35	1.20	1.10	1.00	0.90	0.85	0.80	0.70	0.65	0.55	0.55	0.55
45°	2.05	1.85	1.65	1.50	1.30	1.20	1.10	0.95	0.90	0.70	0.55	0.55
60°	3.20	1.85	2.55	2.35	2.00	1.75	1.60	1.40	1.25	1.00	0.70	0.55
75°	6.80	5.85	5.35	4.75	3.85	3.50	3.15	2.70	2.40	1.55	1.00	0.55



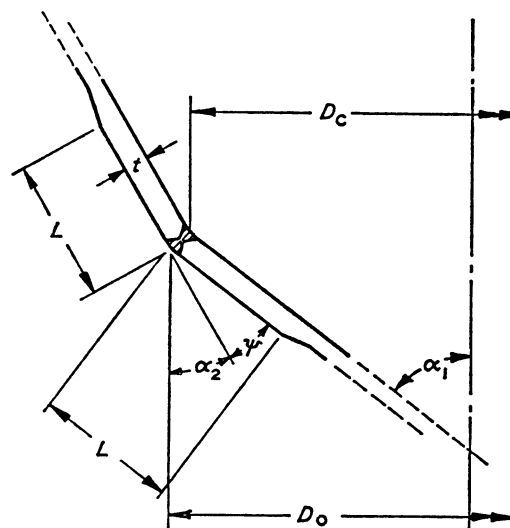
a. Cone/cylinder with knuckle



b. Cone/cylinder without knuckle



c. Cone/cone with knuckle



d. Cone/cone without knuckle

Figure 10

3.2.6 Domed ends

3.2.6.1 General. Domed ends of hemispherical, semi-ellipsoidal or dished shape shall be designed in accordance with the formulae in **3.2.6.2**.

a) *Limitations of shape.* End plates shall conform to one of the following shapes (see BS 1966⁵⁾):

- 1) Hemispherical (see Figure 11a).
- 2) Semi-ellipsoidal (see Figure 11b).

The ratio of major axis to minor axis shall be not greater than 2.6 : 1.

3) Torispherical (dished and flanged) (see Figure 11c).

- i) The inside radius of dishing R_i shall be not greater than the outside diameter D_o .
- ii) The inside corner radius r_i shall be not less than 10 per cent of the inside diameter, nor less than $3t$.
- iii) The external height h_o shall be not less than $0.18 D_o$.

b) *Symbols.*

t = minimum thickness in inches of end;

c = allowance for corrosion or erosion (see **3.3.2.4**);

p = design pressure, lbf/in²;

D_i = inside diameter, in inches, of end before the internal corrosion allowance has been added;

D_o = outside diameter, in inches, of end before the external corrosion allowance has been added;

h_E = effective outside height, in inches, of end before the external corrosion allowance has been added when

$$h_E = h_o \text{ or } D_o^2/4R_o \text{ or } \sqrt{D_o r_o}/2 \text{ whichever is smallest;}$$

h_i = inside height, in inches, of end before the internal corrosion allowance has been added;

h_o = outside height, in inches, of end before the external corrosion allowance has been added;

R_i = inside radius, in inches, of crown before the internal corrosion allowance has been added;

R_o = outside radius, in inches, of crown before the external corrosion allowance has been added;

r_i = inside radius, in inches, of corner before the internal corrosion allowance has been added;

r_o = outside radius, in inches, of corner before the external corrosion allowance has been added;

f = nominal design stress, lbf/in², from Table 1;

J = stress adjustment factor of 0.95 for Category A vessels and 1.00 for all other vessels;

K = a factor depending upon the ratio $\frac{h_E}{D_o}$ and $\frac{t}{D_o}$ obtained from Figure 12;

S_f = length of straight flange after edge preparation. The thickness of the length S_f shall be at least equal to that required for a cylinder of equal diameter.

⁵⁾ BS 1966, "Domed ends for tanks and pressure vessels"

3.2.6.2 Thickness of domed ends concave to pressure. The thickness of the ends shall be determined by equation (15)

$$t = \frac{pD_o K}{2fJ} + c \quad (15)$$

NOTE The external height of dishing h_o may for ends of partially spherical form may be determined from equation (16).

$$h_o = R_o - \sqrt{\left(R_o - \frac{D_o}{2}\right) \times \left(R_o + \frac{D_o}{2} - 2r_o\right)} \quad (16)$$

3.2.6.3 Thickness of domed ends convex to pressure. The symbols and values listed in 3.2.6.1 above may be used in the design of domed ends convex to pressure.

a) *Spherically dished ends.* The thickness of domed ends dished to partially spherical form shall be the greater of the thicknesses determined by the following procedures:

- i) The thickness of an equivalent sphere, having a radius R_o equal to the outside crown radius of the end, determined in accordance with 3.2.2.3.
- ii) The thickness $(t - c)$, as determined for a domed end under an equivalent internal pressure equal to 1.3 times the external pressure.

b) *Ellipsoidal ends.* The thickness of domed ends of true semi-ellipsoidal form shall be the greater of the thicknesses determined by the following procedures:

- i) The thickness of an equivalent sphere determined in accordance with 3.2.2.1. The value of R_o shall be taken as equal to the outside diameter of the end multiplied by a factor taken from the following table:

h_o/D_o	=	0.167	0.178	0.192	0.208	0.227	0.25
Factor	=	1.36	1.27	1.18	1.08	0.99	0.90
h_o/D_o	=	0.278	0.313	0.357	0.417	0.50	
Factor	=	0.81	0.73	0.65	0.57	0.50	

- ii) The thickness $(t - c)$ as determined for a domed end under an equivalent internal pressure equal to 1.3 times the external pressure.

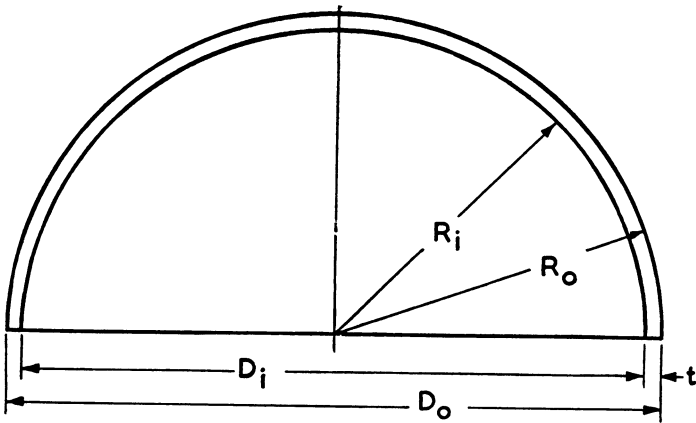


Figure 11a

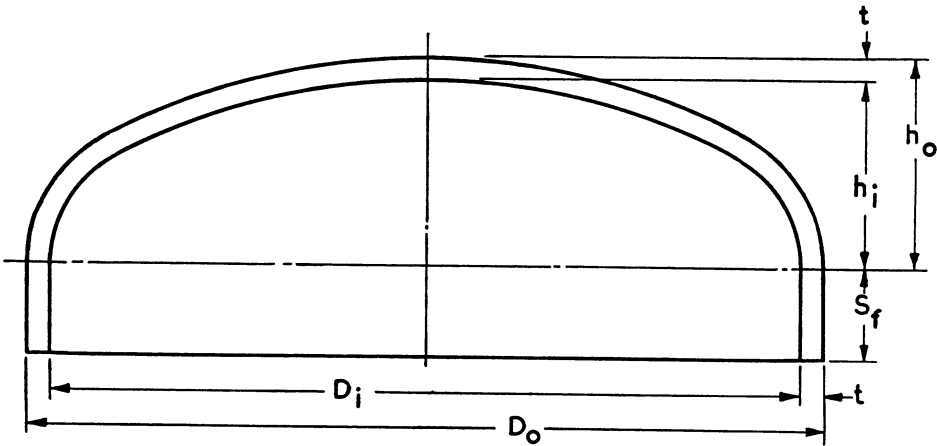


Figure 11b

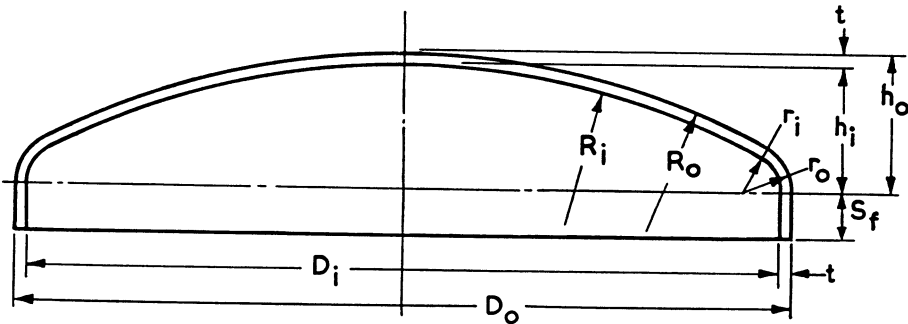
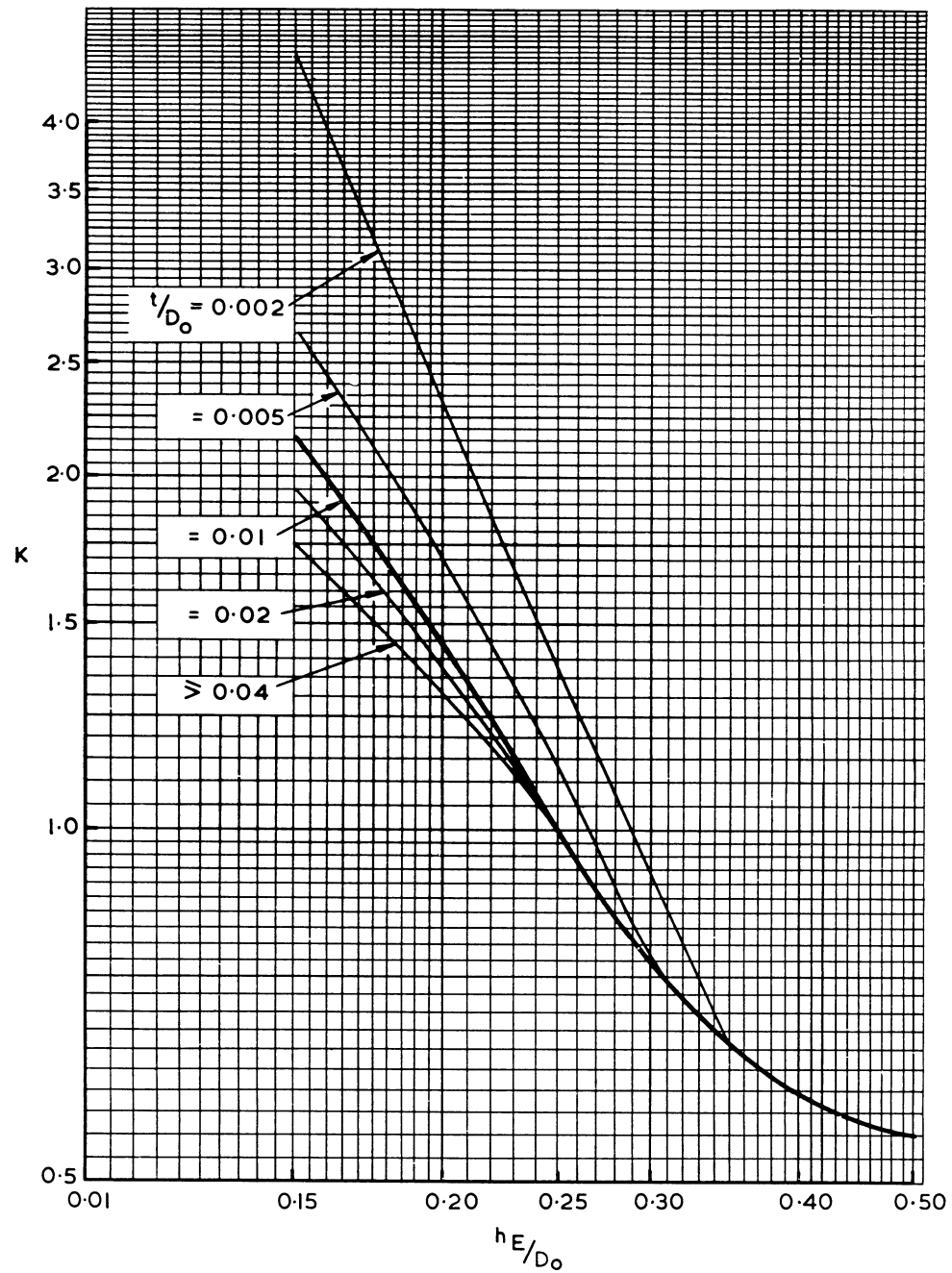


Figure 11c

Figure 11 — Domed ends

Figure 12 — Shape factor K for domed ends

3.2.7 Spherically domed and bolted ends conforming to Figure 13

3.2.7.1 General. Except as specified below for bolted ends conforming to Figure 13 conical and domed and bolted ends shall be designed by treating the domed end and the bolted flange as two separate components in compliance with the relevant sections of this standard.

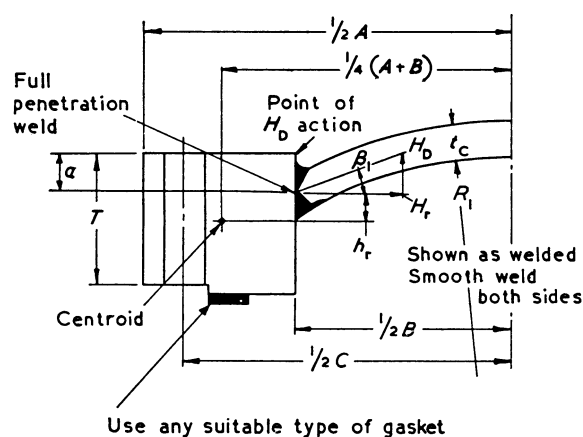


Figure 13 — Spherically domed and bolted end

The method of determining the thickness of the flange ring involves assessing the final thickness in order to arrive at the location of the centroid and hence the value of h_r and is thus a “trial and error” calculation.

For the purposes of the calculations in 3.2.7.2 and 3.2.7.3 the following symbols apply:

- β_1 = angle between tangent to domed crown section at its edge and a plane parallel to flange face;
 a = axial distance from the mid-surface of the crown section at edge to the back face of the flange ring, inches (governed by weld dimensions);
 K_1 = bolt pitch correction factor

$$= \sqrt{\frac{\text{bolt spacing}}{2(\text{bolt diameter}) + T}}$$

- T = minimum flange ring thickness, inches in the fully corroded condition;
 T_{fo} = flange thickness required for operating condition, inches;
 T_{fA} = flange thickness required for bolting-up condition, inches;
 H_r = radial component of the membrane force developed in the spherical crown section, acting at the edge, lb;
 h_r = axial distance from the mid-surface of the crown section at edge to the centroid of the flange ring cross section, inches;
 R_1 = inside radius of curvature of spherical crown section (inches) in the fully corroded condition;
 t_c = minimum thickness of spherical crown section;
 f = nominal design stress for the material of the spherical crown section at the design temperature from Table 1;
 J = stress adjustment factor: 0.95 for Category A components and 1.00 for all other components.

For definition of other symbols, see 3.2.9.4.

3.2.7.2 Subject to internal pressure (convex to pressure)

a) *Crown section.* The minimum thickness of the spherical crown section shall be

$$t_c = \frac{5PR_1}{6Jf} + c \quad (17)$$

b) *Flange ring.* The bolting area required, the bolt loads and the gasket width check shall be calculated as specified in **3.2.9 Bolted flanged connections**.

c) *Flange ring thickness.* The minimum thickness of the flange ring T shall be the greater of T_{fo} or T_{fa} as determined below but shall be not less than twice the crown thickness i.e. $T \geq 2t_c$.

i) *Operating condition:*

$$T_{fo} = F + \sqrt{F^2 + J_o} \quad (18)$$

$$\text{where } F = \frac{pB\sqrt{4R_1^2 - B^2}}{8S_{FO}(A - B)}$$

$$J_o = \frac{M_{op}K_1(A + B)}{BS_{FO}(A - B)}$$

$$M_{op} = H_D h_D + H_G h_G + H_T h_T + H_r h_r$$

$$H_r = H_D \cot \beta_1$$

$$h_r = 0.5 T \text{ (assumed)} - a$$

NOTE The product $H_r h_r$ may be negative if the sense of this moment is opposite to the moments $H_D h_D$, $H_G h_G$ and $H_T h_T$.

ii) *Bolting up condition:*

$$T_{fa} = \sqrt{\frac{K_1(A + B)M_{atm}}{(A - B)BS_{FA}}} \quad (19)$$

3.2.7.3 Subject to external pressure (convex to pressure)

a) *Crown section.* The minimum thickness of the spherical crown section shall be determined in accordance with **3.2.6.3 Thickness of domed ends convex to pressure**.

b) *Flange ring.* The thickness of the flange ring shall be determined in accordance with the rules for internal pressure in **3.2.7.2** except that

$$M_{op} = H_D(h_D - h_G) + H_T(h_T - h_G) - H_r h_r.$$

NOTE The gasket should be checked against excessive deformation under the action of the bolt load and the external pressure thrust.

3.2.8 Openings and branch connections

3.2.8.1 General. The amount of compensation to be provided at an opening, from the point of view of internal pressure, shall be not less than that specified in this section. The effect of any other loads shall be taken into account in accordance with the requirements of **3.3**. Similarly the effects of attachments and discontinuities in the proximity of the openings shall be taken into account.

NOTE 1 The basis on which these requirements are founded is outlined in Appendix A.

NOTE 2 All dimensions exclude corrosion allowance.

For the purposes of the calculations in **3.2.8.2** to **3.2.8.5**, the following symbols apply (see also Figure 16–Figure 28):

D = mean diameter of spherical or cylindrical section of shell, inches,

= $2R_c \sec \alpha$ for a conical section of a shell (see Figure 16);

T = shell thickness as calculated by the formulae for pressure loading only, omitting c , in **3.2** for a plain spherical, cylindrical or conical shell section, inches;

- R_c = radius of conical shell section at opening, inches (see Figure 16);
 α = one half apex angle of cone, degrees (see Figure 16);
 d = bore of opening not provided with a branch, or mean diameter of branch, inches;
 t = branch thickness as calculated by the formulae for pressure loading only, omitting c , in 3.2, for a plain cylinder, inches;
 T_r = total thickness of shell as required by this section for single openings, inches;
 t_r = total thickness of branch as required by this section for single openings, inches;
 T_f = final thickness of shell between the openings considered, inches;
 t_f = final thickness of branch, inches;
 P = pitch measured between centre lines of two openings along mid thickness of shell, inches;
 d_A = average value of d for any two adjacent openings being considered, inches;
 A_s = cross-sectional area of shell between bores of the two adjacent openings considered, square inches (see Figure 28);
 A_n = half the total cross-sectional area of the two nozzles, taken within the appropriate distance $0.64\sqrt{dt_f}$ from the shell in the plane containing the pitch and nozzle axis, together with half the total cross-sectional area of the weld metal lying outside the surfaces of the shell and branch, square inches (see Figure 28);
 A_t = sum of the two products $0.64t\sqrt{dt_f}$ for the two openings considered, square inches (see Figure 28);
 g = arrangement factor from Figure 27;
 H = minimum width of thickened shell at thickness T_r , inches;
 h = minimum length of thickened branch at thickness t_r , inches.

3.2.8.2 Application. The following rules shall be used for the design of circular and obround openings and nozzles (including oblique nozzles), arranged singly or in groups, in spherical, cylindrical and conical shells provided that:

a) *Spherical shells.*

i) *Openings and nozzles normal to shell surface.*

- A. The major axis of the opening does not exceed one half of the diameter of the shell.
 B. The ratio of the major to minor axes of the opening does not exceed 2.

Oblique nozzles. The nozzle is of circular cross section complying with a) i) A above and the angle between the axis of the nozzle and a line normal to the shell surface does not exceed 50° .

b) *Cylindrical and conical shells.*

i) *Openings and nozzles normal to the shell.*

- A. The ratio of the axis of the opening measured circumferentially along the shell to the shell diameter does not exceed $1/3$.
 B. The ratio of the major to minor axes of the opening does not exceed 2.

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

c) *Other configurations.* All nozzles and openings not covered above shall be designed on the basis of special analysis, experimental evidence, operating experience or tests to the satisfaction of the purchaser.