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BRITISH STANDARD 1500 : 1958

# FUSION WELDED PRESSURE VESSELS

FOR USE IN THE

CHEMICAL, PETROLEUM AND

ALLIED INDUSTRIES

Part 1. Carbon and Low Alloy Steels

BRITISH STANDARDS INSTITUTION

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CHEMICAL, PETROLEUM AND

ALLIED INDUSTRIES

**Part 1. Carbon and Low Alloy Steels**

**B.S. 1500 : 1958**

Price 40/- net

BRITISH STANDARDS INSTITUTION

INCORPORATED BY ROYAL CHARTER

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THIS BRITISH STANDARD, having been approved by the Chemical Engineering Industry Standard Committee and endorsed by the Chairman of the Engineering Divisional Council, was published under the authority of the General Council on 20th November, 1958.

First issued as a Provisional Standard in November, 1949.
First revision November, 1958.

The Institution desires to call attention to the fact that this British Standard does not purport to include all the necessary provisions of a contract.

In order to keep abreast of progress in the industries concerned, British Standards are subject to periodical review. Suggestions for improvements will be recorded and in due course brought to the notice of the committees charged with the revision of the standards to which they refer.

A complete list of British Standards, numbering over 3000, indexed and cross-indexed for reference, together with an abstract of each standard, will be found in the Institution's Yearbook price 15s.

*British Standards are revised, when necessary, by the issue either of amendment slips or of revised editions. It is important that users of British Standards should ascertain that they are in possession of the latest amendments or editions.*

## CO-OPERATING ORGANIZATIONS

The Chemical Engineering Industry Standards Committee under whose supervision this British Standard was prepared consists of representatives of the following Government departments and scientific and industrial organizations:

*Association of British Chemical Manufacturers
Association of Consulting Engineers (Incorporated)
Board of Trade
*British Chemical Plant Manufacturers' Association
Coke Oven Managers' Association
*Engineering Equipment Users' Association
Gas Council
Glass Manufacturers' Federation
Institute of Metal Finishing
*Institute of Petroleum
*Institution of Chemical Engineers
Institution of Gas Engineers
*Institution of Mechanical Engineers
Institution of Structural Engineers
*Society of Chemical Industry (Chemical Engineering Group)

The scientific and industrial organizations marked with an asterisk in the above list, together with the following, nominated members to serve upon the committee entrusted with the preparation of this British Standard:

Associated Offices Technical Committee
Association of Shell Boilermakers
British Iron and Steel Federation
British Welding Research Association
D.S.I.R.—Mechanical Engineering Research Laboratory
Engineer Surveyors' Association
Imperial College of Science and Technology
Institute of Refrigeration
Institute of Welding
Lloyd's Register of Shipping
Ministry of Labour and National Service
Ministry of Supply
Ministry of Transport and Civil Aviation
Oil Companies' Materials Committee
Individual manufacturers

The following B.S.I. references relate to the work on this standard:
Committee reference CHE 19/-/1 Draft for comment: 1949 provisional edition of B.S. 1500.



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BRITISH STANDARD FOR

# FUSION WELDED PRESSURE VESSELS

## FOR USE IN THE CHEMICAL, PETROLEUM AND ALLIED INDUSTRIES

### Part 1. Carbon and Low Alloy Steels

## FOREWORD

Part 1 of this British Standard supersedes the 1949 edition, which was issued as a provisional standard. In preparing this standard, account has been taken of all the comments received on the 1949 provisional standard.

The industry which makes the greatest use of pressure vessels (other than steam boilers) is undoubtedly the chemical industry in its many branches—petroleum, soap, paper, rubber, plastics, etc. There are also many subsidiary 'chemical units' in general industrial plants, such as impregnators in electric cable manufacture and vacuum stills for dry cleaning in laundries.

The petroleum branch operates on a very large scale and, appropriately, the special activities of this branch have received consideration during the formulation of the requirements for pressure vessels. In the preparation of this standard, therefore, full consideration has been given not only to British practice but also to the experience of the American petroleum industry, and acknowledgement is made to the American Petroleum Institute and American Society of Mechanical Engineers.

It is recognized that the design of pressure vessels, particularly in the chemical industry, is the subject of continuous development, and it is intended to keep the standard under review in order to incorporate such modifications as are found desirable as a result of progress.

Part 2 and Part 3 of this standard are under consideration, namely:

Part 2. Alloy steel pressure vessels.

Part 3. Non-ferrous pressure vessels.

**When it is required to convert the figures in this standard from British units into metric units it is recommended that the conversion factors and the tables of conversion contained in B.S. 350, 'Conversion factors and tables', be used. Attention is also drawn to B.S. 2856, 'Precise conversion of inch and metric sizes on engineering drawings'.**

# Section One: General

## A. SCOPE

Part I of this British Standard covers the design, construction and inspection of fusion-welded pressure vessels in carbon and low alloy steels for use in the chemical, petroleum and allied industries. It does not necessarily apply to pressure vessels covered by existing British Standards. Vessels in which the internal pressure is due solely to the static head of liquid are outside the scope of this standard, and provision for lightly loaded breathing devices has not been made. Also excluded from this standard are vessels in which the calculated stress in the welded seam at the thinnest point is less than 2500 lb/sq. in. at 350°F.

The design of vessels lined with vitreous enamel requires special consideration and is not covered by this standard.

The term 'pressure vessel' as used in this standard also includes branches up to the point of connection, by bolting, screwing or welding, to the connecting piping.

This standard is intended to serve as a guide for agreement between purchaser and manufacturer. Some of the requirements are not obligatory and nothing in this standard is intended to contravene any provision of the Factories Act, 1937, or of any regulations made thereunder, or any other statutory requirements.

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.

## B. DEFINITIONS

For the purpose of this British Standard the following definitions shall apply:

*Safe working pressure*—the maximum gauge pressure, at the coincident metal temperature, that is permitted for the vessel when in operation.

*Design pressure*—a value used in the formulae of Section Three to determine the minimum thicknesses of the various component parts of the vessel. In the case of



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vessels subject to vacuum conditions or external pressure the design pressure is the maximum differential pressure between opposite sides of the shell or other element of the vessel under consideration. The design pressure should not be less than the maximum pressure reached during operation and should include a suitable margin to allow for possible fluctuation of pressure during operation.

*Design temperature*—the maximum operating temperature. This should include a margin for possible temperature fluctuations.

*Design stress*—the stress to be used in the formulae of Section Three is selected from Table 2 for the design temperature.

NOTE. It is recognized that vessels designed and fabricated in accordance with this standard will contain regions at which the localized and secondary stresses are appreciably higher than those tabulated in this standard. In so far as it is practicable, design rules for details have been formulated to hold such stresses at safe levels consistent with experience.

*Welded joint factor*—the ratio of an arbitrary strength of the welded joint to the strength of the plates welded, expressed as a decimal.

*Ligament efficiency*—the ratio of an arbitrary strength of a ligament to the strength of the unpierced plate, expressed as a decimal.

*Stress relief*—the heat treatment of a vessel at predetermined temperatures. (See Section Four D.)

*Fusion welding*—see B.S. 499* for definitions of welding processes and types of joints.

* B.S. 499, 'Glossary of terms (with symbols) relating to the welding and cutting of metals'.

## C. CLASSIFICATION

### CLASSIFICATION

1. *a. General.* The pressure vessels covered by this Standard shall conform to one of the classes shown in Table I. See also Figs. 1 and 2.

Each class of construction provides for the use, in design, of a welded joint factor associated with the quality control, inspection and testing prescribed for that class.

The joint factors ordinarily to be used in design are shown in Table 1. It is customary, however, in certain sections of industry, to assign joint factors of 0·85 for stress relieved vessels and 0·80 for non-stress relieved vessels which comply in all respects with Class II requirements and, in addition, have been subjected to spot radiography of the main seams. The use of these higher joint factors is permitted only when agreed between the purchaser and the manufacturer.

The details of inspection and testing are given in Section Five, and requirements for the application of and procedure for (i) stress relief by heat treatment and (ii) radiography are given in Section Four D and Section Five C respectively.

*b. Vessels for low temperature service.* Special consideration should be given to the choice of materials in the case of vessels designed to operate at temperatures below 32°F (0°C). (See Appendix C.)



## TABLE 1. CLASSIFICATION OF PRESSURE VESSELS

| | Class I | Class II | | Class III | |
|---|---|---|---|---|---|
| Design | To drawings approved by the purchaser. | To drawings approved by the purchaser. Maximum thickness of shell plate 1½ in. after corrosion allowance is added. | | To drawings approved by the purchaser. Maximum thickness of shell plate ⅝in. before corrosion allowance is added. | |
| Materials | Any materials prescribed in Section Two. | Restricted to the carbon steels prescribed in Section Two. | | Restricted to the carbon steels prescribed in Section Two. | |
| Control of quality<br>(1) Materials | (1) Inspection and tests at the steelmaker's works. | (1) Inspection and tests at the steelmaker's works. | | (1) Mill Certificates (or other proof of quality) of each plate, etc. | |
| (2) During fabrication | (2) *a.* Identification of plates and other components at the manufacturer's works.<br>*b.* Inspection of plates after forming and of plate edges prepared for welding.<br>*c.* Assembly and alignment of vessel sections and branches prior to welding.<br>*d.* Inspection during progress of welding including second side welding grooves after preparation by chipping or machining.<br>*e.* Inspection of main seams after dressing flush with plates.<br>*f.* Identification and stamping of weld test plates.<br>*g.* Calibration and dimensional check on completion. | (2) *a.* Identification of plates and other components at the manufacturer's works.<br>*b.* Inspection of plates after forming and of plate edges prepared for welding.<br>*c.* Inspection of assembly, including alignment of main seams, prior to welding.<br>*d.* Inspection during progress of welding including:<br>(i) second side welding grooves after preparation by chipping or machining, or<br>(ii) spot radiography of each main seam where second side welding groove is not prepared by chipping or machining in such a manner that all unsoundness at the root of the first side weld is removed.<br>*e.* Identification and stamping of weld test plates.<br>*f.* Calibration and dimensional check on completion. | | (2) *a.* Identification of plates and other components at the manufacturer's works.<br>*b.* Inspection of assembly, including alignment of main seams, prior to welding.<br>*c.* Inspection during progress of welding including:<br>(i) second side welding grooves after preparation by chipping or machining, or<br>(ii) spot radiography of each main seam where second side welding groove is not prepared by chipping or machining in such a manner that all unsoundness at the root of the first side weld is removed.<br>*d.* Calibration and dimensional check on completion. | |
| (3) Mechanical tests | (3) Mechanical tests on a test plate provided at one end of each longitudinal seam:<br>*a.* All-weld metal tensile test.<br>*b.* Bend test, outer surface in tension.<br>*c.* Bend test, inner surface in tension.<br>*d.* Notched bar impact test.<br>*e.* Micro- and macro-examination. | (3) Where stipulated by the purchaser, mechanical tests on a test plate provided at one end of each longitudinal seam:<br>*a.* Transverse joint tensile test.<br>*b.* Bend test, outer surface in tension.<br>*c.* Bend test, inner surface in tension.<br>*d.* Nick break test. | | (3) Bend tests of welded test pieces may be called for at purchaser's option. | |
| (4) Radiography | (4) Radiographic examination of full length of main seams. | (4) Special cases for which radiography is prescribed are covered in Section Five, C 'Radiography', Clause 1: Application. | | | |
| (5) Stress relief | (5) Heat treatment for stress relief on completion of welding. | (5) Conditions for the application of stress relief are prescribed in Section Four, D. | | | |
| (6) Pressure test | (6) Hydraulic pressure test. | (6) Hydraulic pressure test. | | (6) Hydraulic pressure test. | |
| Weld joint factor | 0·95 | Stress relieved 0·80 | Not stress relieved 0·70 | 0·60 | |
| | | NOTE. Attention is drawn to Clause 1 of this section which provides for higher values than those above for joint factors in appropriate circumstances. | | | |
| Identification markings | 'CLASS I' | 'CLASS II S.R.' | 'CLASS II' | 'CLASS III S.R.' (For 'Class III' vessels which have been heat-treated on completion of all welding operations) | 'CLASS III' |



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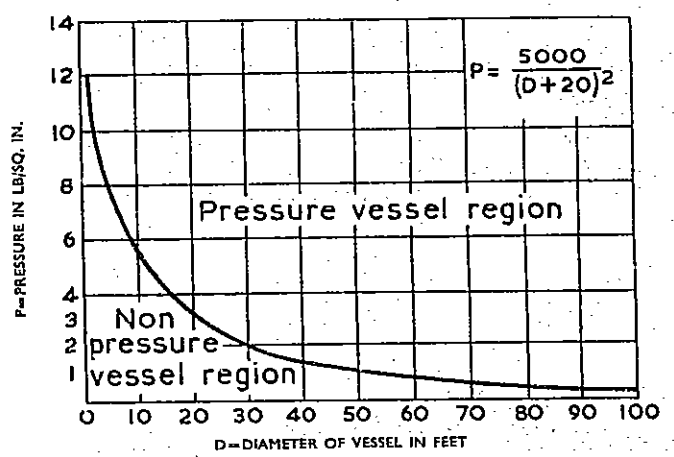


Fig. 1. Limits for pressure vessels covered by the requirements of the code

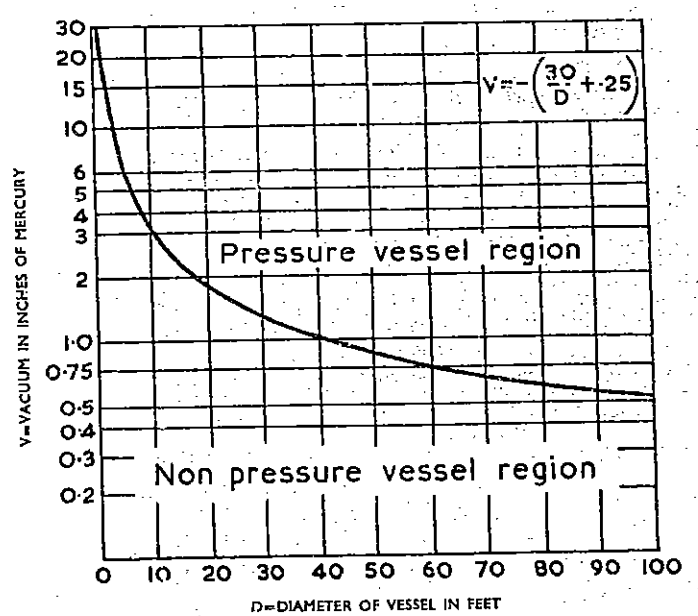


Fig. 2. Limits for vacuum vessels covered by the requirements of the code





# Section Two: Materials and Design Stresses

## A. MATERIALS

### GENERAL

1. *a.* All materials used in the manufacture of pressure parts of pressure vessels constructed to this Standard shall be in accordance with the appropriate material specifications referred to in this section.

*b.* Materials for which test certificate are not available may be used for supporting lugs and skirts, baffles and other similar non-pressure parts of a vessel constructed in accordance with this Standard, provided that such material is otherwise suitable for the purpose.

### MATERIALS FOR LOW TEMPERATURE SERVICE

2. Special consideration should be given to the choice of materials for vessels designed to operate at temperatures below 32°F (0°C). (See Appendix C.)

### VESSELS CONSTRUCTED OF STANDARD PIPES AND TUBES

3. Vessels may be made of standard pipes and tubes provided the design stresses given in Table 2 are not exceeded.

### BOLTS AND NUTS

4. Bolts and nuts shall be screwed with Unified threads in accordance with B.S. 1580*, or British Standard Whitworth (B.S.W.) or British Standard Fine (B.S.F.) threads in accordance with B.S. 84†.

*a.* Bolts and nuts shall comply with one of the following standards:

B.S. 916. Black bolts, screws and nuts.

B.S. 1083. Precision hexagon bolts, screws, nuts and plain washers.

B.S. 1750. Bolting for the petroleum industry.

B.S. 1768. Unified precision hexagon bolts, screws, nuts and plain washers. Normal series.

B.S. 1769. Unified black hexagon bolts, screws, nuts and plain washers. Heavy series.

B.S. 2708. Unified black hexagon bolts, screws, nuts and plain washers. Normal series.

*b.* Bolts and nuts for bolted flanges operating at temperatures in excess of 450°F or pressures in excess of 250 lb/sq. in. shall be manufactured from steels chosen from the grades specified in Table 9 of Section Three M.

## B. DESIGN STRESSES

### DESIGN STRESSES

1. The design stresses in Table 2 are maximum values to be used in conjunction with the formulae given in this Standard and do not make any allowance for corrosion or the efficiency of welded joints or ligaments between openings. (See also Section Three, E, Clauses 2 and 3.)

### SHEAR STRESSES

2. The maximum shear stress shall not exceed the values given in Table 2 multiplied by 0·8.

### BEARING STRESSES

3. The maximum bearing stress shall not exceed the values given in Table 2 multiplied by 1·7.

NOTE. If it is desired to keep a vessel in service after the corrosion allowance (see Section Three, B) has wasted away, the maximum working pressure shall be reduced to ensure that the allowable working stress is not exceeded.

* B.S. 1580, Unified screw threads.
† B.S. 84, Parallel screw threads of Whitworth form.



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## TABLE 2a. DESIGN STRESSES (TENSILE) FOR FERROUS PLATES, BARS AND SECTIONS

| Material | *Reference B.S. number | Tensile strength (tons/sq. in.) | Notes | Permissible design stresses at the design temperature of the metal (lb/sq. in.) | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | up to 300°F (149°C) | 500°F (260°C) | 550°F (288°C) | 600°F (315°C) | 650°F (342°C) | 700°F (371°C) | 750°F (399°C) | 800°F (427°C) | 850°F (454°C) | 900°F (482°C) | 925°F (495°C) | 950°F (510°C) | 975°F (523°C) | 1 000°F (538°C) |
| Carbon steel | 1501–101 | 26–32 | (1) | 14 500 | — | — | — | — | — | — | — | — | — | — | — | — | — |
| | 1501–151 | | | | | | | | | | | | | | | | |
| | Grade A | 24–28 | (2) | 13 400 | 13 400 | 13 400 | 13 400 | 13 400 | 13 200 | 12 000 | — | — | — | — | — | — | — |
| | Grade B | 26–30 | (2) | 14 500 | 14 500 | 14 500 | 14 500 | 14 500 | 14 300 | 12 700 | — | — | — | — | — | — | — |
| | Grade C | 28–32 | (2) | 15 700 | 15 700 | 15 700 | 15 700 | 15 700 | 15 200 | 13 400 | — | — | — | — | — | — | — |
| Carbon steel (silicon killed) | 1501–161 | | | | | | | | | | | | | | | | |
| | Grade A | 24–28 | (2) | 13 400 | 13 400 | 13 400 | 13 400 | 13 400 | 13 200 | 12 000 | 10 500 | 8 500 | 6 300 | 4 800† | 3 350† | — | — |
| | Grade B | 26–30 | (2) | 14 500 | 14 500 | 14 500 | 14 500 | 14 500 | 14 200 | 12 700 | 10 900 | 8 700 | 6 300 | 4 800† | 3 350† | — | — |
| | Grade C | 28–32 | (2) | 15 700 | 15 700 | 15 700 | 15 700 | 15 700 | 15 200 | 13 400 | 11 300 | 8 900 | 6 300 | 4 800† | 3 350† | — | — |
| Carbon-manganese steel | 1501–221 | 30–36 | (2) (3) | 16 800 | 16 800 | 16 800 | 16 800 | 16 800 | 16 600 | 15 000 | 12 100 | 9 200 | 6 400 | 4 900† | 3 350† | — | — |
| | | 33–39 | (2) (4) | 18 500 | 18 500 | 18 500 | 18 500 | 18 500 | 18 100 | 16 000 | 12 700 | 9 500 | 6 400 | 4 900† | 3 350† | — | — |
| | | 35–41 | (2) (5) | 19 600 | 19 600 | 19 600 | 19 600 | 19 600 | 19 100 | 16 700 | 13 100 | 9 700 | 6 400 | 4 900† | 3 350† | — | — |
| Carbon-molybdenum steel | 1501–240 | 27–33 | (2) | 15 100 | 15 100 | 15 100 | 15 100 | 15 100 | 15 100 | 15 100 | 14 600 | 13 800 | 13 000 | 12 200† | 9 000† | 5 200† | 3 800† |

Intermediate values may be obtained by linear interpolation.

NOTE 1. This material is suitable for the construction of Class III vessels only.

NOTE 2. Consideration should be given to the effect of notch brittleness in vessels designed to operate below 32°F. Attention is drawn to Appendix C and B.S. 1510, ' Steels for use in the chemical, petroleum and allied industries (Low temperature supplementary requirements to B.S. 1501–B.S. 1506) '.

NOTE 3. For plates up to and including 1 in. thick.

NOTE 4. For plates above 1 in. up to and including 3 in. thick.

NOTE 5. For plates over 3 in. thick.

* See B.S. 1501, ' Carbon and alloy steel plates, sections and bars for use in the chemical, petroleum and allied industries '.

† Atmospheric scaling at these temperatures should be taken into account.







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## TABLE 2b. DESIGN STRESSES (TENSILE) FOR FERROUS FORGINGS

| Material | *Reference B.S. number | Tensile strength (tons/sq. in.) | Note | Permissible design stresses at the design temperature of the metal (lb/sq. in.) | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | Up to 300°F (149°C) | 500°F (260°C) | 550°F (288°C) | 600°F (315°C) | 650°F (342°C) | 700°F (371°C) | 750°F (399°C) | 800°F (427°C) | 850°F (454°C) | 900°F (482°C) |
| Carbon steel (silicon killed) | 1503–161 | | | | | | | | | | | | |
| | Grade A | 24–28 | (1) | 13 400 | 13 400 | 13 400 | 13 400 | 13 400 | 13 200 | 12 000 | 10 500 | 8 500 | 6 300 |
| | Grade B | 28–32 | (1) | 15 700 | 15 700 | 15 700 | 15 700 | 15 700 | 15 200 | 13 400 | 11 300 | 8 900 | 6 300 |
| | Grade C | 32–36 | (1) | 18 000 | 18 000 | 18 000 | 18 000 | 18 000 | 17 200 | 14 800 | 12 100 | 9 300 | 6 300 |
| Carbon-manganese steel | 1503–221 | 32–38 | (1) | 18 000 | 18 000 | 18 000 | 18 000 | 18 000 | 17 700 | 15 700 | 12 500 | 9 400 | 6 400 |
| Carbon-molybdenum steel | 1503–240 | | | | | | | | | | | | |
| | Grade A | 26–33 | (1) | 14 600 | 14 600 | 14 600 | 14 600 | 14 600 | 14 600 | 14 600 | 14 200 | 13 400 | 12 600 |
| | Grade B | 32 min. | (1) | 18 000 | 18 000 | 18 000 | 18 000 | 18 000 | 18 000 | 17 000 | 16 500 | 15 800 | 14 500 |

| Material | 925°F (495°C) | 950°F (510°C) | 975°F (523°C) | 1 000°F (538°C) |
|---|---|---|---|---|
| Carbon steel (silicon killed) | | | | |
| Grade A | 4 800† | 3 350† | — | — |
| Grade B | 4 800† | 3 350† | — | — |
| Grade C | 4 800† | 3 350† | — | — |
| Carbon-manganese steel | 4 900† | 3 350† | — | — |
| Carbon-molybdenum steel | | | | |
| Grade A | 11 800† | 9 000† | 5 200† | 3 800† |
| Grade B | 12 600† | 8 700† | 4 700† | 3 600† |

Intermediate values may be obtained by linear interpolation.

NOTE 1. Consideration should be given to the effect of notch brittleness in the case of vessels designed to operate below 32°F. Attention is drawn to Appendix C and B.S. 1510, 'Steels for use in the chemical, petroleum and allied industries (Low temperature supplementary requirements to B.S. 1501–B.S. 1506)'.

* See B.S. 1503, 'Carbon and alloy steel forgings for use in the chemical, petroleum and allied industries'.

† Atmospheric scaling at these temperatures should be taken into account.







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# TABLE 2c. DESIGN STRESSES (TENSILE) FOR FERROUS CASTINGS

| Material | *Reference B.S. number | Tensile strength (tons/sq. in.) | Notes | Permissible design stresses at the design temperature of the metal (lb/sq. in.) | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | Up to 300°F (149°C) | 500°F (260°C) | 550°F (288°C) | 600°F (315°C) | 650°F (342°C) | 700°F (371°C) | 750°F (399°C) | 800°F (427°C) | 850°F (454°C) | 900°F (482°C) | 925°F (495°C) | 950°F (510°C) | 975°F (525°C) | 1 000°F (538°C) |
| Carbon steel | 1504-101 | | | | | | | | | | | | | | | | |
| | Grade A | 26–32 | (1) (2) | 11 600 | 11 600 | 11 600 | 11 600 | 11 600 | 11 100 | 10 200 | 8 700 | 7 000 | 5 000 | 3 900† | 2 700† | — | — |
| | Grade B | 28–35 | (1) (2) | 12 500 | 12 500 | 12 500 | 12 500 | 12 500 | 12 200 | 10 700 | 9 000 | 7 100 | 5 000 | 3 900† | 2 700† | — | — |
| | Grade C | 35–40 | (1) (2) | 15 700 | 15 700 | 15 700 | 15 700 | 15 700 | 15 000 | 12 700 | 10 200 | 7 700 | 5 000 | 3 900† | 2 700† | — | — |
| Carbon steel (for parts under pressure) | 1504-161 | | | | | | | | | | | | | | | | |
| | Grade A | 28 min. | (1) (2) | 12 500 | 12 500 | 12 500 | 12 500 | 12 500 | 12 200 | 10 700 | 9 000 | 7 100 | 5 000 | 3 900† | 2 700† | — | — |
| | Grade B | 31 min. | (1) (2) | 13 900 | 13 900 | 13 900 | 13 900 | 13 900 | 13 400 | 11 600 | 9 500 | 7 400 | 5 000 | 3 900† | 2 700† | — | — |
| ½ per cent molybdenum steel | 1504-240 | 30 min. | (1) (2) | 13 400 | 13 400 | 13 400 | 13 400 | 13 400 | 13 400 | 13 100 | 12 600 | 12 000 | 11 200 | 10 100† | 7 000† | 3 750† | 2 400† |

Intermediate values may be obtained by linear interpolation.

NOTE 1. Consideration should be given to the effect of notch brittleness in vessels designed to operate below 32°F.

NOTE 2. The stress values tabulated above apply to castings inspected in accordance with the minimum requirements of the corresponding standards. A casting quality factor of 0·80 has been included.

A casting quality factor of 0·90 may be allowed and the tabulated stresses increased by 12½ per cent provided additional inspection to the following extent is carried out:

(i) Each casting is radiographically examined at all critical locations and found free from harmful defects, or the castings are machined to such an extent that all critical sections are exposed for the full thickness as in the case of tube plates drilled with holes spaced not further apart than the wall thickness of the casting;

or (ii) *a.* at least three pilot castings representing the first batch of five castings from a new or altered design are sectioned or radiographically examined at all critical locations and found free from harmful defects. Changes in moulding technique or practice shall be considered as an alteration to design and the first castings produced by any foundry from any pattern shall be considered as a new design even where satisfactory castings have been produced elsewhere from the same pattern;

*b.* one casting selected at random from each subsequent batch of five castings shall be similarly examined;

*c.* all castings other than those which are radiographically examined shall be examined at all critical locations using a magnetic particle, or penetrant fluid procedure, or by grinding, or machining, and etching;

*d.* all castings shall be thoroughly examined on all surfaces, particularly those exposed by machining operations.

(iii) Castings found to be defective shall be rejected or repaired to the purchaser's satisfaction. If repairs by welding are carried out the castings shall subsequently be stress-relieved or heat-treated as agreed between purchaser and manufacturer. Repaired castings shall be re-examined in accordance with (i) or (ii) above and shall be shown to be free from harmful defects.

(iv) Castings to which a quality factor of 0·90 is applied shall be clearly and permanently marked with an approved symbol after being examined as above and found satisfactory.

* See B.S. 1504, 'Carbon and alloy steel castings for use in the chemical, petroleum and allied industries'.

† Atmospheric scaling at these temperatures should be taken into account.







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# TABLE 2d. DESIGN STRESSES (TENSILE) FOR FERROUS PIPES

(FOR USE AS SHELLS, BRANCH PIPES AND SIMILAR PARTS OF PRESSURE VESSELS)

| Material | *Reference B.S. number | Minimum tensile strength (tons/sq. in.) | Note | Permissible design stresses at the design temperature of the metal (lb/sq. in.) | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | Up to 300°F (149°C) | 500°F (260°C) | 550°F (288°C) | 600°F (315°C) | 650°F (342°C) | 700°F (371°C) | 750°F (399°C) | 800°F (427°C) | 850°F (454°C) | 900°F (482°C) | 925°F (495°C) | 950°F (510°C) | 975°F (525°C) | 1 000°F (538°C) | 1 050°F (566°C) | 1 100°F (593°C) | 1 150°F (620°C) | 1 200°F (649°C) |
| Mild steel, seamless | 1507–101 | 22 | (1) | 12 300 | 12 300 | — | — | — | — | — | — | — | — | — | — | — | — | — | — | — | — |
| Mild steel, seamless | 1507–151 | 23 | (1) | 12 900 | 12 900 | 12 900 | 12 900 | 12 900 | 12 800 | 11 800 | 10 300 | 8 400 | 6 300 | 4 900† | 3 350† | — | — | — | — | — | — |
| Mild steel, cold drawn, seamless | 1507–171 | 23 | (1) | 12 900 | 12 900 | 12 900 | 12 900 | 12 900 | 12 800 | 11 800 | 10 300 | 8 400 | 6 300 | 4 900† | 3 350† | — | — | — | — | — | — |
| ½ per cent molybdenum steel, seamless | 1507–240 | | | | | | | | | | | | | | | | | | | | |
| | Grade A | 25 | (1) | 14 000 | 14 000 | 14 000 | 14 000 | 14 000 | 14 000 | 14 000 | 13 700 | 13 300 | 12 400 | 12 000† | 9 000† | 5 200† | 3 800† | — | — | — | — |
| | Grade B | 31 | (1) | 17 400 | 17 400 | 17 400 | 17 400 | 17 400 | 17 400 | 16 800 | 16 100 | 15 400 | 14 500 | 12 600† | 8 700† | 4 700† | 3 000† | — | — | — | — |
| 1 per cent chromium molybdenum steel, seamless | 1507–621 | 25 | (1) | 14 000 | 14 000 | 14 000 | 14 000 | 14 000 | 14 000 | 14 000 | 14 800 | 13 800 | 12 400 | 11 200 | 9 800 | 8 400† | 7 000† | 4 500† | — | — | — |

Intermediate values may be obtained by linear interpolation.

The stress values given above apply only for pipes used as parts of pressure vessels with simple loading. They do *not* apply to fabricated pipework.

NOTE 1. Consideration should be given to the effect of notch brittleness in vessels designed to operate below 32°F.

* See B.S. 1507, 'Steel pipes for use in the chemical, petroleum and allied industries'.

† Atmospheric scaling at these temperatures should be taken into account.







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# TABLE 2e. DESIGN STRESSES (TENSILE) FOR FERROUS TUBES

(FOR USE IN PRESSURE VESSELS SUCH AS CONDENSERS, HEAT EXCHANGERS AND THE LIKE)

| Material | *Reference B.S. number | Minimum tensile strength (tons/sq. in.) | Note | Permissible design stresses at the design temperature of the metal (lb/sq. in.) | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | Up to 300°F (149°C) | 500°F (260°C) | 550°F (288°C) | 600°F (315°C) | 650°F (342°C) | 700°F (371°C) | 750°F (399°C) | 800°F (427°C) | 850°F (454°C) | 900°F (482°C) | 925°F (495°C) | 950°F (510°C) | 975°F (525°C) | 1 000°F (538°C) | 1 050°F (566°C) |
| Mild steel, hot finished, seamless | 1508–151 | 20 | (1) | 11 200 | 11 200 | 11 200 | 11 200 | 11 200 | 10 800 | 10 100 | 9 000 | 7 500 | 6 000 | — | — | — | — | — |
| Mild steel, cold drawn, seamless | 1503–171 | 20 | (1) | 11 200 | 11 200 | 11 200 | 11 200 | 11 200 | 10 800 | 10 100 | 9 000 | 7 500 | 6 000 | — | — | — | — | — |
| Mild steel electric resistance welded | 1654<br>1678 | 20<br>20 | — | 11 200 | 11 200 | 11 200 | 11 200 | 11 200 | 10 800 | 10 100 | 9 000 | 7 500 | 6 000 | — | — | — | — | — |
| ½ per cent molybdenum steel, seamless | 1508–240 | 25 | (1) | 14 000 | 14 000 | 14 000 | 14 000 | 14 000 | 14 000 | 14 000 | 13 700 | 13 300 | 12 500 | 11 500 | 9 000 | 5 200 | 3 800 | — |
| 1 per cent chromium molybdenum steel, seamless | 1508–621 | 25 | (1) | 14 000 | 14 000 | 14 000 | 14 000 | 14 000 | 14 000 | 14 000 | 14 000 | 13 800 | 12 400 | 11 200 | 9 800 | 8 400 | 7 000 | 4 500 |

Intermediate values may be obtained by linear interpolation.

The stress values given above apply only for tubes forming an integral part of a pressure vessel.

NOTE 1. Consideration should be given to the effect of notch brittleness in vessels designed to operate below 32°F.

* See B.S. 1508, 'Steel tubes for use in the chemical, petroleum and allied industries'.
* See B.S. 1654, 'Electrically welded mild steel boiler and superheater tubes'.
* See B.S. 1678, 'Cold drawn electrically welded mild steel boiler and superheater tubes'.







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## TABLE 3. VALUE OF 'E' (1,000,000 LB/SQ. IN.) FOR FERROUS MATERIALS

| Material | Permissible design values at design temperature of the metal (lb/sq. in. × $10^6$) | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 150°F (66°C) | 300°F (145°C) | 450°F (232°C) | 600°F (316°C) | 700°F (371°C) | 750°F (399°C) | 800°F (427°C) | 850°F (454°C) | 900°F (487°C) | 950°F (510°C) | 1 000°F (538°C) | 1 050°F (566°C) | 1 100°F (593°C) | 1 150°F (621°C) | 1 200°F (649°C) |
| Carbon steel | 29·5 | 28·8 | 27·8 | 26·5 | 25·5 | 25·0 | 24·3 | 23·3 | 22·0 | 20·5 | — | — | — | — | — |
| Carbon manganese steel | 29·5 | 28·8 | 27·8 | 26·6 | 25·5 | 25·0 | 24·3 | 23·3 | 22·0 | 20·5 | — | — | — | — | — |
| Carbon molybdenum steel | 29·5 | 28·8 | 27·8 | 26·5 | 25·5 | 25·0 | 24·3 | 23·3 | 22·0 | 20·5 | — | — | — | — | — |

Intermediate values may be obtained by linear interpolation.

The above values are for purposes of design and shall not form part of the material specifications. The figures given were chosen as an average of a large number of experiments and the experimental error was such as to require a tolerance of from ± 1 per cent at room temperature to ± 7 per cent at 1100°F.







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# Section Three : Design

## A. GENERAL

Vessels covered by this standard shall be designed for the most severe combination of operating conditions which may be experienced in normal operation.

### DESIGN THICKNESS

1. *a. General.* In the clauses that follow, methods are given for calculating, for the various parts of pressure vessels, the thicknesses required to withstand whichever of the design loads have to be considered.

It must be emphasized that all such thicknesses are minima, and may require to be increased to allow for other factors affecting the actual use of the vessels.

For example the thickness may require to be increased from that sufficient as far as pressure considerations are concerned in order to give the vessel sufficient rigidity for practical use.

*b. Castings.* In particular, the calculated thicknesses of castings shall be increased by an amount sufficient to cover any unevenness of thickness and shall be not less than the minimum thickness necessary to ensure sound castings.

### DESIGN TEMPERATURE

2. *a. General.* The temperature to be used when determining the safe working stress should be the maximum temperature it is possible for the material to attain under the operating conditions.

Where vessels are subjected to alternate heating and cooling, provision shall be made in the design to permit expansion or contraction to avoid severe thermal stresses.

*b. Thermal stresses.* Temperature gradients between the inner and outer surfaces of the wall of a pressure vessel give rise to thermal stresses which only require to be evaluated in special cases. Guidance on the significance of the calculation of thermal stresses is given in Appendix B.

*c. Low temperature vessels.* Special consideration should be given to vessels designed to operate at temperatures below 32°F (0°C). (See Appendix C.)

### DESIGN PRESSURE

3. The normal operating pressure is determined by the technical requirements of the process to be used.

The design pressure is usually in excess of this to avoid unnecessary blowing of safety devices, and varies according to the use of safety valves or rupture discs. (See Section Six A.)

Where the maximum pressure is limited by means of pressure-relieving devices it is recommended that there should be sufficient margin between the design pressure and the normal operating pressure to avoid frequent blowing of the safety device. For normal spring-loaded safety valves this is usually 10 per cent of the operating pressure.

Rupture discs are highly stressed in service; it is therefore necessary, in order to avoid premature failure due to the effects of creep and fatigue, to allow a substantial margin (of the order of 50 per cent of the operating pressure but depending upon the particular conditions) between the normal operating pressure of the vessel and the designed bursting pressure of the disc. It should be noted that the bursting pressure of rupture discs is materially influenced by temperature.

Vessels subject to vacuum only should be designed for the maximum negative pressure to which the vessel may be subjected in service. Except in those cases where a vacuum break valve or a hydraulic seal is provided, it is recommended that such vessels be designed for a full negative pressure of 14·7 lb/sq. in.

### LOADINGS

4. *a.* The loadings to be considered in designing a vessel shall include the following:

(i) Internal and external operating pressure.

(ii) Static head of contained fluid.

(iii) The weight of vessel and normal contents at operating temperature, or, if such a load is probable, the weight of vessel full of water at atmospheric temperature, except that in this case the total stress under test conditions shall not exceed 1·5 times the tabulated value.

(iv) Superimposed and wind loads.

*b.* Special consideration may require to be given to the effect of the following:

(i) Local stresses due to supporting lugs, ring girders, saddles, internal structures or connecting piping.

(ii) Shock loads due to water hammer or surging of vessel contents.

(iii) Bending moments caused by eccentricity of the centre of working pressure relative to the neutral axis of the vessel.

(iv) Forces due to temperature differences, including the effects of differential expansion.

(v) Forces caused by the method of supporting the vessel during transit or erection.

### NUMBER OF WELDED SEAMS

5. Whenever practicable the cylindrical portion of a vessel shall be made from a single plate. When the dimensions are such that this is not possible the minimum number of plates shall be used. Where the cylindrical portion is made in two or more courses the longitudinal seams of each course shall be completed before commencing the adjoining circumferential seam(s) and where practicable the longitudinal seams of adjacent courses shall be staggered.





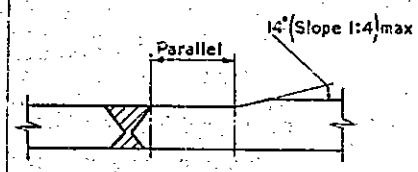
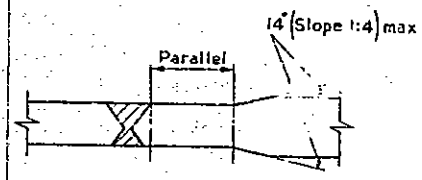
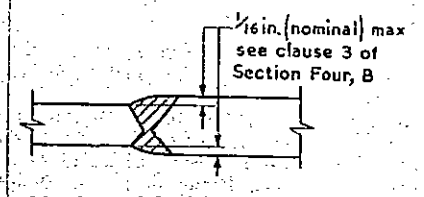
| Fig. | Weld connection | Preferred minimum length of parallel portion from edge of weld preparation | |
|---|---|---|---|
| | | Fully radiographed seams | Other seams |
| 3 a |   Taper may be inside vessel or outside. | in. | ½ in. |
| 3 b |   | 1¼ in. | ½ in. |
| 3 c |   May be used for joints between plates differing in thickness by not more than ⅛ in. (nominal dimensions). | — | — |

Fig. 3. Butt welds in plates of unequal thickness (See Figs. 4 and 5 for weld preparations).

### BUTT WELDS BETWEEN PLATES OF UNEQUAL THICKNESS

6. Where a butt welded joint is required between plates of different thicknesses the thicker plate shall be reduced in thickness by machining to provide a parallel portion so that the welded joint is made between material of the same thickness.

The width of the parallel portion shall be not less than 1¼ in. when the butt weld is to be radiographically examined, or not less than ½ in. when the seam is not to be radiographed, and the junction between the section of reduced thickness and the unmachined part of the plate shall consist of a tapered portion having a slope not greater than 1 in 4. (See Fig. 3.)

### PLATES WELDED PRIOR TO FORMING

7. The welded seams in plates welded together in the flat shall be demonstrated to be of the appropriate quality for the joint factor involved. For example, the welded seams in the plates to be incorporated in Class I welded pressure vessels shall be of ' Class I ' quality (i.e. in accordance with the requirements for ' Class I ' vessels as regards mechanical properties and freedom from discontinuities as revealed by radiographic examination). Similarly, the butt welds in the plates incorporated in domed ends shall be of ' Class I ' quality if the design joint factor used is appropriate to this class.



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# B. CORROSION, EROSION AND PROTECTION

## GENERAL

1. Whenever the word corrosion is used in this standard it shall be taken to mean corrosion, oxidation, scaling, abrasion, erosion and all other forms of wastage.

It is impossible to lay down definite precautionary rules to safeguard against the effects of corrosion, owing to the complex nature of corrosion itself, which may exist in many forms:

(i) Chemical attack, where the metal is dissolved by the reagents; it may be general over the whole surface, or localized (causing pitting), or a combination of the two.

(ii) Rusting, caused by the combined action of moisture and air.

(iii) Erosion corrosion, where a reagent, otherwise innocuous, flows over the surface at a velocity greater than some critical value.

(iv) High temperature oxidation (scaling).

Designers should give careful consideration to the effect which corrosion may have upon the useful life of the vessel. When in doubt, corrosion tests should be undertaken; these should be carried out on the actual metal (including welds) or combination of metals under exposure to the actual chemicals used in service. It is very dangerous to assume that the major constituent of a mixture of chemicals is the active agent as, in many cases, small traces of impurities exert an accelerating or inhibiting effect out of all proportion to the amount of impurity present. Fluid temperatures and velocities should be equivalent to those met with in operation. Corrosion tests should be continued for a sufficiently lengthy period to determine the trend of any change in the rate of corrosion with respect to time.

Corrosion may occur on *both* sides of the wall of the vessel, particularly with vessels heated by hot gases of combustion.

Pressure vessels may be classified in relation to corrosion under one of the following groups:

*a. Corrosion rate predictable.* Vessels in which corrosion rates may be definitely established by reason of accurate knowledge of the chemical characteristics of whatever substances they are to contain.

*b. Corrosion rate unpredictable.* Vessels in which corrosion rates are either variable throughout the vessel or indeterminate in magnitude.

*c. Corrosion rate negligible.* Vessels in which corrosion rates are known to be negligible.

## ADDITIONAL THICKNESS TO ALLOW FOR CORROSION

2. The allowances adopted shall be adequate to cover the total amount of corrosion anticipated on either or both surfaces of the vessel.

In cases where corrosion may occur, additional metal thickness over and above that required for the design conditions should be provided, at least equal to the expected corrosion loss during the desired life of the vessel. It is recommended that in all such cases a minimum corrosion allowance of 1/16 in. should be provided unless a protective lining is employed.

Where the corrosion effects are negligible, no excess thickness need be provided.

## LININGS

3. *a.* Vessels may be fully or partially lined with corrosion resistant material. Such linings may be loose, intermittently welded to the vessel base material or provided as clad steel, i.e. corrosion resistant material integrally bonded to the base material.

Provided linings are designed so as to exclude contact between the corrosive agent and the vessel base material no corrosion allowance need be provided against internal wastage.

The design of linings should take into account the effects of differential thermal expansion.

*b. Wear plates.* Where severe conditions of erosion and abrasion arise it is recommended that local protective or 'wear' plates, of an easily renewable type, should be fitted directly in the path of the impinging material.

# C. TYPES OF WELDED JOINTS

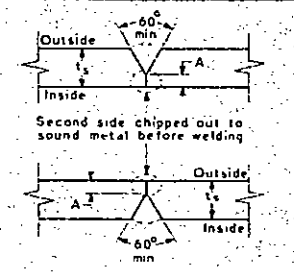
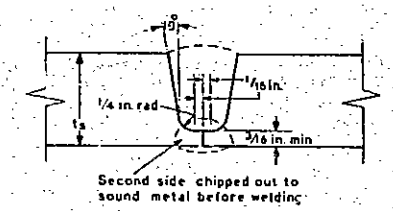
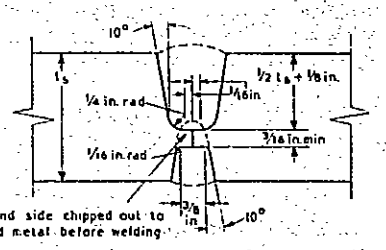
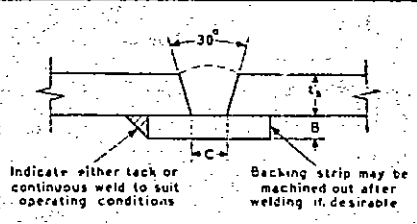
The types of welded joints to be used on vessels covered by this standard shall be in accordance with Figs. 4, 5 and 6. Weld preparations in accordance with Figs. 4*a*, 4*b* and 4*c* may be arranged so that 'first side' welding is carried out from either the inside or the outside of the vessel.

The requirements of this standard are applicable to welded joints made by the metal-arc process with covered electrodes or by any other electric arc process in which the arc stream and deposited weld metal are shielded from atmospheric contamination.

It is intended that the relevant provisions of this standard shall also apply to the single run, heavy run and inert gas shielded arc processes and that welded vessels manufactured by those processes shall comply with this standard, except those in which divergence is necessary solely because of special requirements essential for the most efficient utilization of such process.





| Fig. | Joint | Name | Application |
|---|---|---|---|
| 4 a | (i)   (ii) For small diameter vessels. | Double-welded butt joint with single 'V'. | Longitudinal and circumferential butt welds in plates not less than 3/16 in. thick and not more than 3/4 in. thick.<br>The 'V' should be on the inside of small diameter vessels as shown in (ii) opposite.<br>A—1/16 in. where $t_s$ is less than 3/8 in.<br>A—1/8 in. where $t_s$ is 3/8 in. or over.<br>See Fig. 3 for details of parallel and taper portions where plates are of unequal thickness. |
| 4 b |   Weld dimensions are minima. | Double-welded butt joint with single 'U'. | Longitudinal and circumferential butt welds in plates where the thickness is greater than 3/4 in.<br>See Fig. 3 for details of parallel and taper portions where plates are of unequal thickness. |
| 4 c |   Weld dimensions are minima. | Double-welded butt joint with double 'U'. | Longitudinal and circumferential butt welds where the thickness is greater than 3/4 in.<br>See Fig. 3 for details of parallel and taper portions where plates are of unequal thickness. |
| 4 d |   Weld dimensions are minima. | Single-welded butt joint with backing strip. | Longitudinal and circumferential butt welds where the second side is inaccessible for welding.<br>See Fig. 3 for details of parallel and taper portions where plates are of unequal thickness. |

| Plate thickness $t_s$ | B | C |
|---|---|---|
| Up to 5/16 in. | 3/16 in. | 5/16 in. |
| 5/16 in. to 1/2 in. | 1/4 in. | 3/8 in. |
| Over 1/2 in. | 3/8 in. | 3/8 in. |

Fig. 4. Typical weld preparations for butt welds



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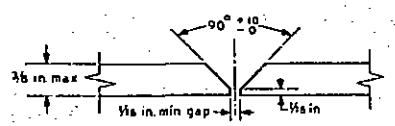
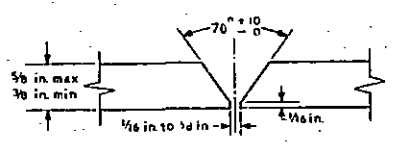
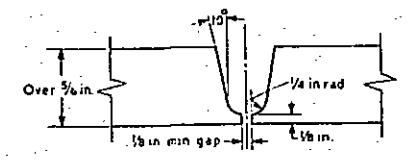
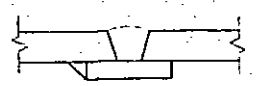
| Fig. | Joint | Name | Application |
|---|---|---|---|
| 5 a |   | Single-welded butt joint with 'V' groove, without backing strip. | Butt welds in plates not exceeding 3/8 in. thickness. |
| 5 b |   | Single-welded butt joint with 'V' groove, without backing strip. | Butt welds in plates having a thickness not less than 3/8 in. and not greater than 5/8 in. |
| 5 c |   Weld dimensions are minima. | Single-welded butt joint with 'U' groove, without backing strip. | Butt welds in plates having a thickness greater than 5/8 in. |
| 5 d (4 d) |  See Fig. 4d for dimensions. | Single-welded butt joint with backing strip. (As Fig. 4d.) | Butt welds in all thicknesses of plate. |

Fig. 5. Typical weld preparations for circumferential welds where the second side is inaccessible for welding (See also Fig. 4d)

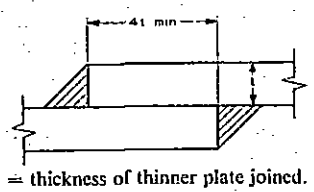
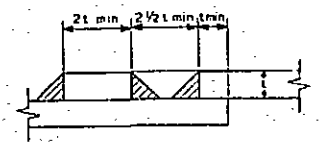
| Fig. | Joint | Name | Application | |
|---|---|---|---|---|
| | | | Longitudinal | Circumferential |
| 6 a |   $t$ = thickness of thinner plate joined. | Double full-fillet lap joint | Not allowed | Not exceeding 5/8 in. plate for Class III vessels only |
| 6 b |   $t$ = thickness of thinner plate joined. | Single full-fillet lap joint with plug weld | Not allowed | Not exceeding 9/10 in. plate. Plugs shall be proportioned to take 20 per cent of total load. For Class III vessels only |

Fig. 6. Typical lap welded circumferential joints





# D. LIGAMENT EFFICIENCY

## GENERAL

1. Where a shell is drilled with multiple holes its strength is reduced in proportion to the amount of metal removed, and according to the disposition of the holes. The 'ligament efficiency' shall then be obtained from the formulae set out below.

## SYMBOLS

2. The following symbols are used in the various formulae. Fig. 7 shows the meaning of the various pitch symbols used:

$J$ = ligament efficiency (expressed as a fraction).

$P$ = distance, in inches, between centres of adjacent holes when the pitch is equal.

$P_1$ = the shorter of any two adjacent pitches, in inches, for unequal pitching.

$P_2$ = the longer pitch of two adjacent pitches, in inches, for unequal pitching.

$P_3$ = the diagonal pitch of tube holes, measured in inches, on the mean of the external and internal diameter of the shell.

$P_c$ = the circumferential pitch of the tube holes, measured in inches, on the mean of the external and internal diameters of the shell. Where tubes are arranged in one staggered row only, as in Fig. 7d, $P_c$ should be taken as twice the value of $P_s$.

$P_s$ = distance between the lines of centres for tubes arranged in one staggered row, as Fig. 7d.

$d$ = the mean effective diameter, in inches, of tube holes, allowing for any serrations, counter boring or recessing.

$K$ = the ratio of $\frac{\text{longitudinal stress on line AB}}{\text{circumferential stress on line CD}}$.

(See Fig. 7c.)

## LIGAMENT EFFICIENCY FOR HOLES IN LINE

3. *a. Equal pitch.* Where the distance between the centres of holes is constant, as in Fig. 7a:

$$J = \frac{P - d}{P} \qquad \text{Equation (1)}$$

*b. Unequal pitch.* Where the distance between centres of holes is unequal, as in Fig. 7b:

$$J = \frac{P_1 + P_2 - 2d}{P_1 + P_2} \qquad \text{Equation (2)}$$

When applying the above formulae the double pitch $(P_1 + P_2)$ chosen shall be that which makes $J$ a minimum, and in no case shall $P_2$ be taken as greater than $1{\cdot}3P_1$.

*c. Circumferential pitch.* The circumferential pitch between parallel rows of tube holes, measured on the mean of the external and internal shell diameters, shall be such that the circumferential ligament efficiency is equal to, or greater than, half the ligament efficiency on the longitudinal axis, unless the circumferential stresses are reduced so as to prevent the longitudinal stress exceeding that permitted by Table 2.

## LIGAMENT EFFICIENCY FOR STAGGERED HOLES

4. For staggered holes as shown in Figs. 7c, 7d and 7e, reference should be made to B.S. 1113, 'Water-tube boilers and their integral superheaters'.



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Fig. 7a

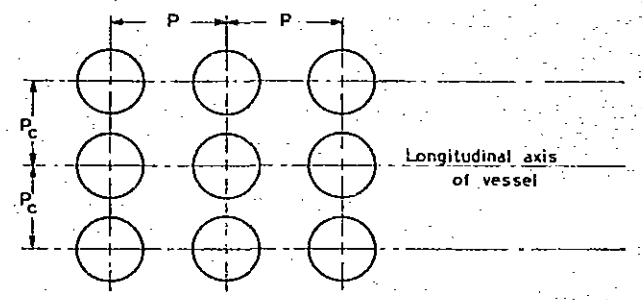


Fig. 7b

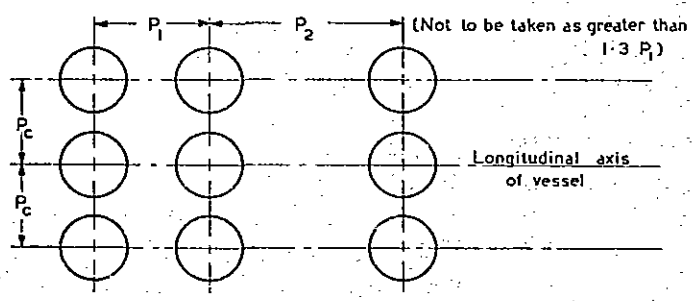


Fig. 7c

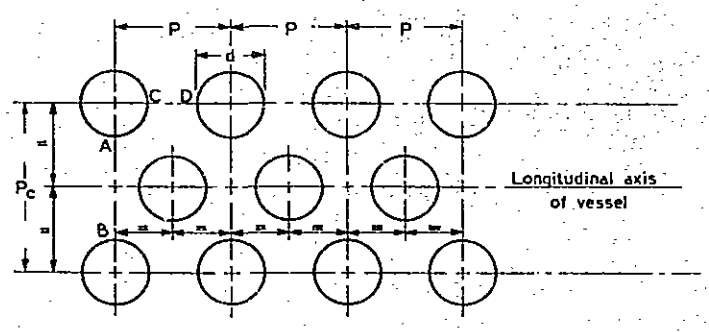


Fig. 7d

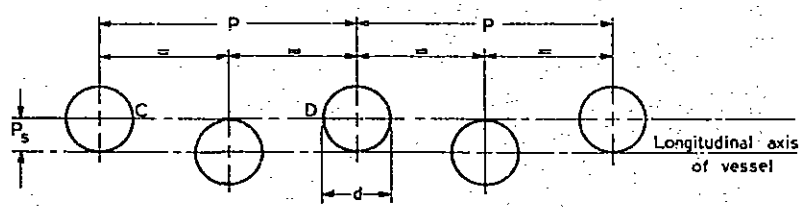


Fig. 7e

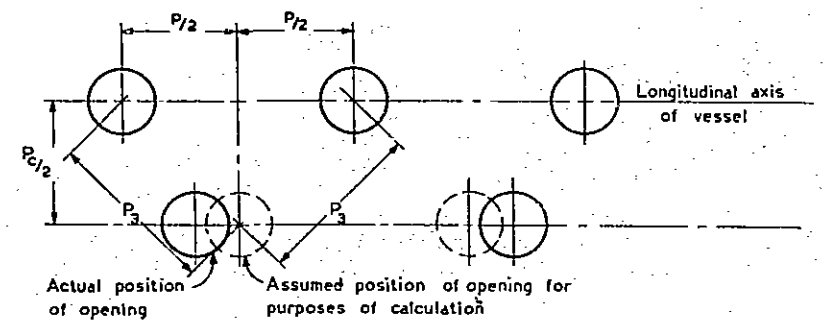


Fig. 7. Symbols, etc., for ligament efficiency





# E. CYLINDRICAL SHELLS

## GENERAL

1. *a.* The thickness shall be not less than that calculated by the following formulae and shall be increased if necessary to meet the requirements of Section Three A.

*b.* For supports and stiffeners, see Section Three O.

## SHELLS SUBJECT TO INTERNAL PRESSURE

2. *a.* Symbols. The following symbols and values are used in the design of cylindrical and spherical vessels subject to internal pressure:

$t$ = minimum thickness in inches of shell plates.

$p$ = design pressure in lb/sq. in.

$D_i$ = 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.

$f$ = design stress in lb/sq. in. from Table 2.

$J$ = joint factor as shown in Table 1 or the ligament efficiency as given in Section Three D, whichever is applicable.

$c$ = a corrosion allowance as agreed between purchaser and manufacturer but not less than as recommended in Section Three B.

$E$ = modulus of elasticity, lb/sq. in. of the material at the operating temperature from Table 3.

*b.* Cylindrical shells. The thickness of shells shall be determined by Equation (3) or (4):

$$t = \frac{pD_i}{2fJ - p} + c \qquad \text{Equation (3)}$$

or

$$p = \frac{2fJ(t-c)}{D_i + (t-c)}$$

$$t = \frac{pD_o}{2fJ + p} + c \qquad \text{Equation (4)}$$

or

$$p = \frac{2fJ(t-c)}{D_o - t + c}$$

*c.* Spherical shells. The thickness of shells shall be determined by Equation (5) or (6):

$$t = \frac{pD_i}{4fJ - p} + c \qquad \text{Equation (5)}$$

or

$$p = \frac{4fJ(t-c)}{D_i + (t-c)}$$

$$t = \frac{pD_o}{4fJ + p} + c \qquad \text{Equation (6)}$$

or

$$p = \frac{4fJ(t-c)}{D_o - t + c}$$

*d.* Combined loadings. (i) VERTICAL VESSELS. For vertical vessels the maximum direct axial stress resulting from the combined effects of internal pressure, bending moment due to wind loads* or other causes, and the total weight of the vessel, attachments and contents shall be calculated in accordance with Equations (7) and (8):

(1) *Under operating conditions:*

$$f = \frac{pD_o}{4(t-c)} + \frac{48\,M}{\pi D_o^2 (t-c)} - \frac{W}{\pi D_o (t-c)} \qquad \text{Equation (7)}$$

NOTE. Negative values denote compressive stresses.

(2) *Under conditions of atmospheric internal pressure:*

$$f = \left[\frac{48\,M}{\pi D_o^2 (t-c)} + \frac{W}{\pi D_o (t-c)}\right] \qquad \text{Equation (8)}$$

where:

$M$ = bending moment at point under consideration in pounds feet.

$W$ = total weight at point under consideration in pounds.

The value of maximum axial stress obtained from Equation (7) or Equation (8) shall not exceed the design stress at operating temperature shown in Table 2 and in no case shall the compressive stress, indicated by Equation (8), or a negative value of Equation (7), exceed the value of

$$0{\cdot}125\,E\left(\frac{t-c}{D_o}\right) \text{ lb/sq. in.}$$

(ii) HORIZONTAL VESSELS. For horizontal vessels supported at two or more points on their length, the maximum direct axial stress resulting from internal pressure and bending moment due to the weight of the vessel and its contents, shall be calculated as follows:

(1) *Under operating conditions:*

$$f = \frac{pD_o}{4(t-c)} + \frac{48\,M}{\pi D_o^2 (t-c)} \qquad \text{Equation (9)}$$

(2) *Under conditions of atmospheric internal pressure:*

$$f = \frac{48\,M}{\pi D_o^2 (t-c)} \qquad \text{Equation (10)}$$

The value of maximum axial stress obtained from Equation (9) or Equation (10) shall not exceed the design stress at operating temperatures shown in Table 2, and in no case shall the compressive stress obtained from Equation (10) exceed the value of

$$0{\cdot}125\,E\left(\frac{t-c}{D_o}\right) \text{ lb/sq. in.}$$

## SHELLS SUBJECT TO EXTERNAL PRESSURE

3. *a. Symbols.* The following symbols and values are used in the design of cylindrical and spherical vessels subject to external pressure:

$t$ = minimum thickness of cylindrical or spherical shell, inches.

$p$ = design external pressure, lb/sq. in.

* B.S. 449, 'The use of structural steel in buildings' gives recommendations for the calculation of wind loads on structures.



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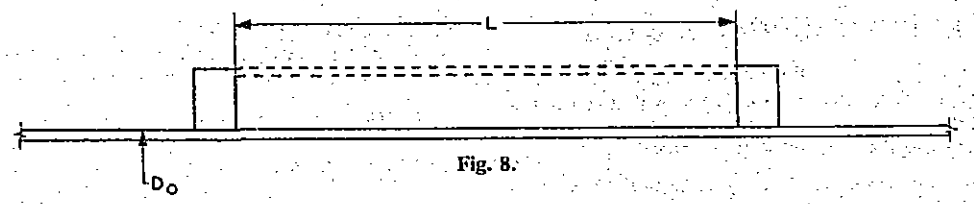



Fig. 8.

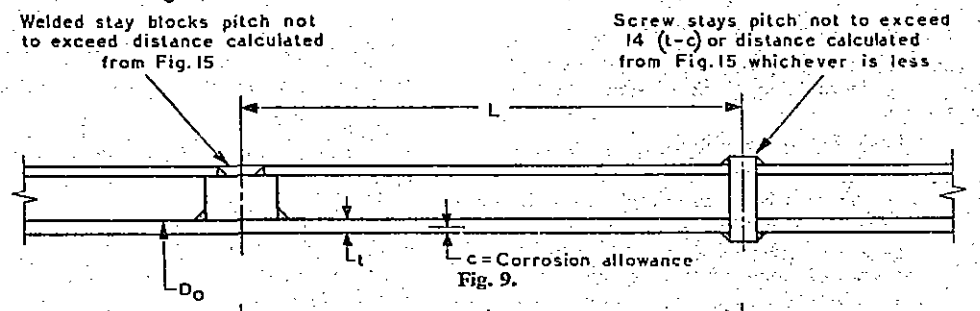



Fig. 9.

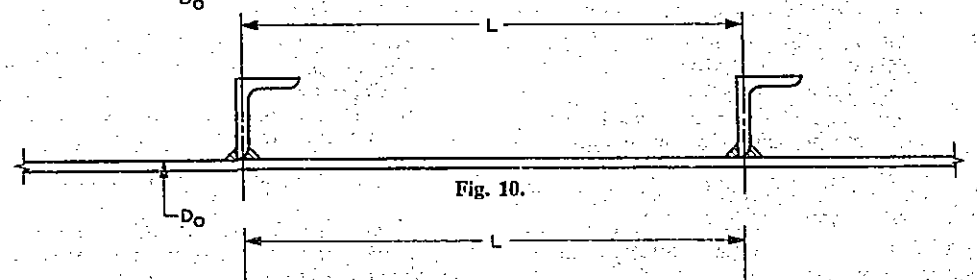



Fig. 10.

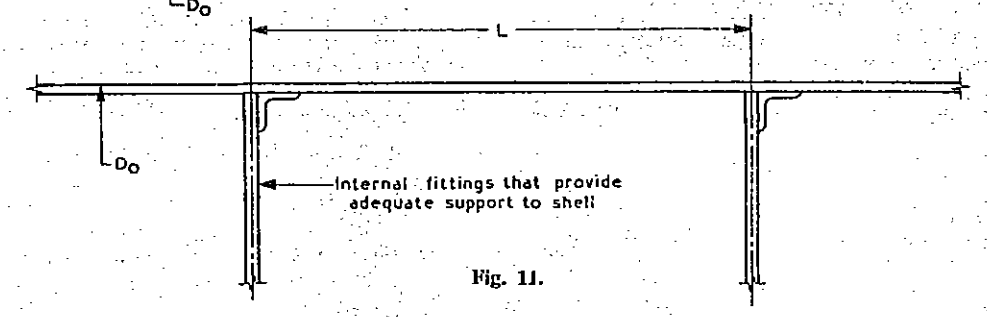
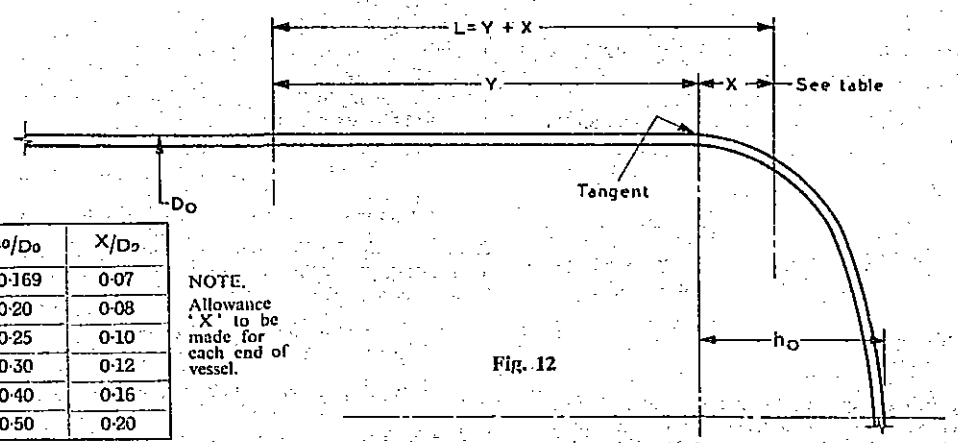



Fig. 11.




| $h_o/D_o$ | $X/D_o$ |
|---|---|
| 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.

Fig. 12

Effective length of cylinders subject to external pressure





$D_o$ = outside diameter of cylindrical or spherical shell, inches.

$R_o$ = external radius of spherical vessel, inches.

$L$ = effective length of cylindrical shell, inches. (See Figs. 8 to 12.)

$f$ = design stress, lb/sq. in. from Table 4.

$E$ = modulus of elasticity, lb/sq. in. of the material at the operating temperature from Table 3.

$I$ = moment of inertia, in.$^4$, of stiffening rings.

$c$ = a corrosion allowance as agreed between purchaser and manufacturer but not less than that recommended in Section Three B.

The rules given herein apply to the design of cylindrical and spherical shells with or without stiffening rings, for which the chart, Fig. 13, 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 Section Three A, Clause 4*b* (i) ), should be compensated for, preferably by increasing the shell plate thickness at the supports.

*b. Cylindrical shells.* Referring to Fig. 13, the thickness of a cylindrical shell under external pressure, whether it is a seamless shell or has butt welded joints, shall be not less than that determined by the following procedure:

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, Fig. 13, 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 4 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 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 illustrate the procedure to be adopted in the case of an existing vessel when it is the value of $L$ that is required.

*c. Spherical shells.* Referring to Fig. 13, the thickness of a spherical shell under external pressure, whether the shell is seamless or has butt-welded joints, shall be not less than that determined by the following procedure:

Assume a value for corroded thickness $(t - c)$ and determine the ratio $\frac{R_o}{(t-c)}$. Enter the chart, Fig. 13, 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 4 with this value of $A$, read off the design stress corresponding to the design temperature and enter

TABLE 4. DESIGN STRESSES LB/SQ. IN. FOR VESSELS SUBJECT TO EXTERNAL PRESSURE

| Design temp. °F | Values of factor $A$ from Fig. 13 | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Up to 100 | 120 | 140 | 160 | 180 | 200 | 250 | 300 | 400 | 500 | 600 | 700 | 800 | 900 | 1 000 |
| | Design stress lb/sq. in. | | | | | | | | | | | | | | |
| 70 | 30A | 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 |
| 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 |
| 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 |
| 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 |
| 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 |
| 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 |
| 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 |
| 700 | 25·5A | 3 050 | 3 550 | 4 000 | 4 600 | 5 100 | 5 900 | 6 600 | 7 500 | 8 100 | 8 400 | 8 600 | 8 800 | 8 900 | 9 000 |
| 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 |
| 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.



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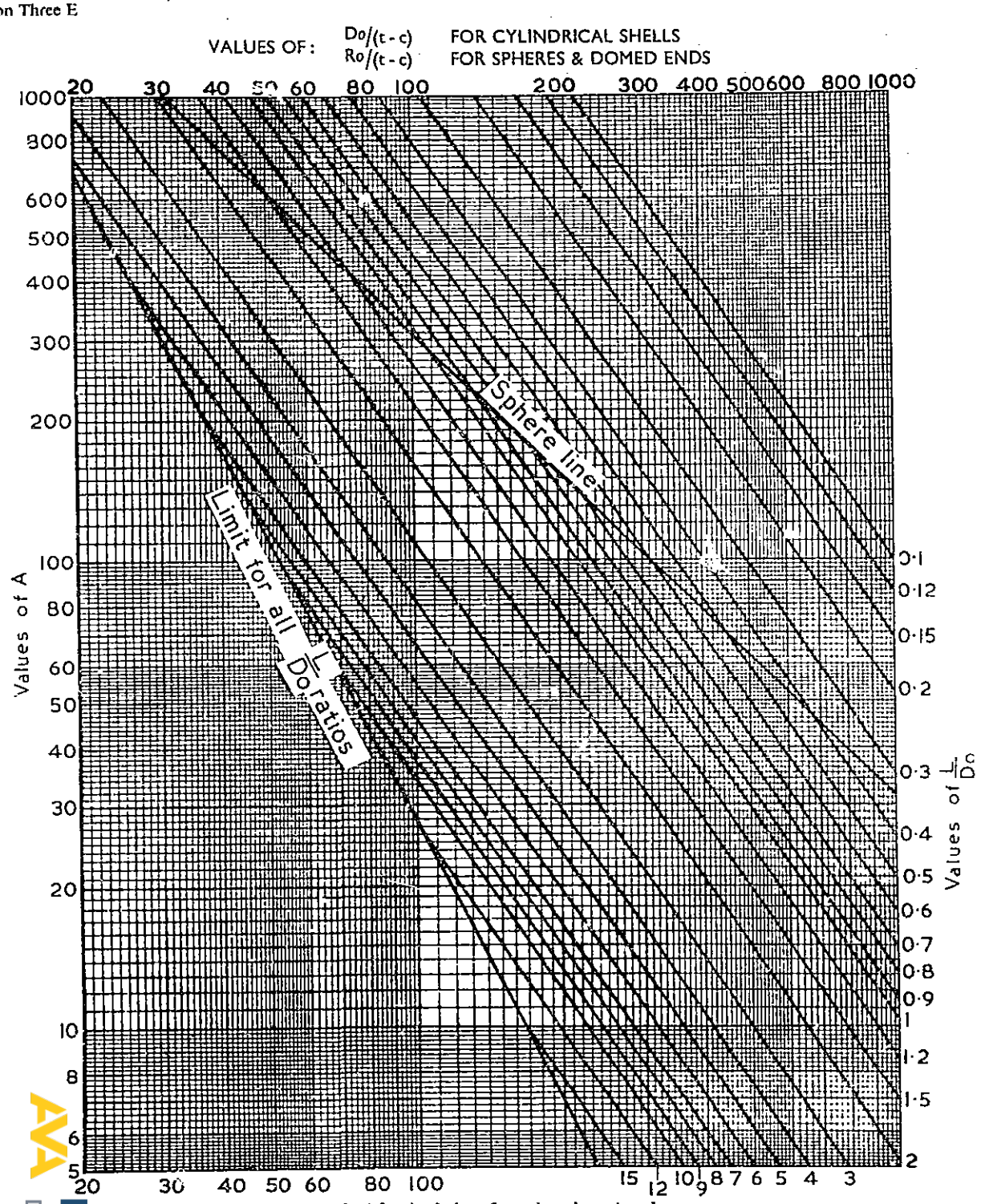



Fig. 13. Chart for the design of vessels under external pressure





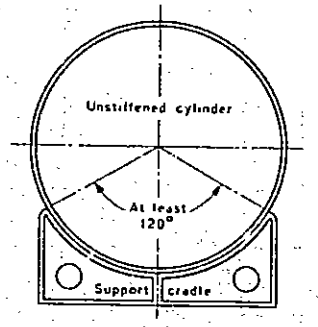
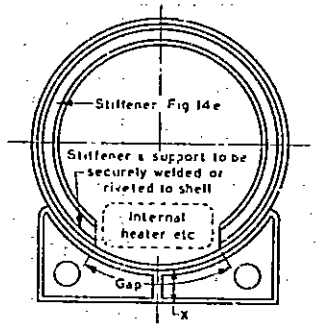



Fig. 14a.




This type of construction when the gap is greater than length of arc to Fig. 15. Moment of inertia of support at 'X' not to be less than I of stiffener.

Fig. 14b.

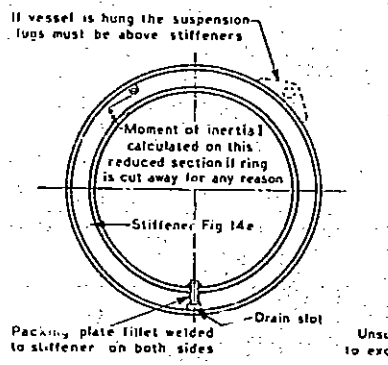



Fig. 14c.

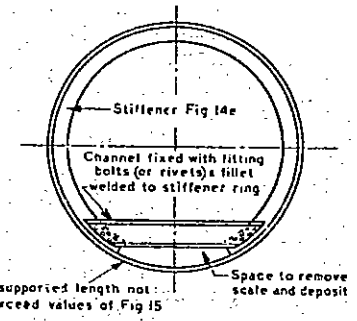



Fig. 14d.

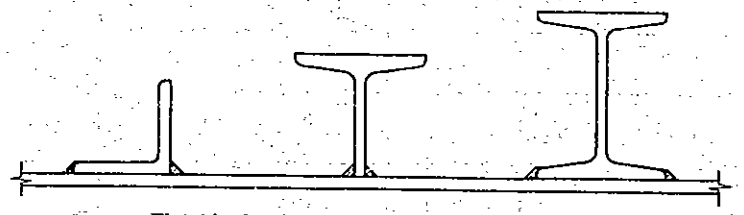


Fig. 14e. Typical stiffeners solid welded to shell.

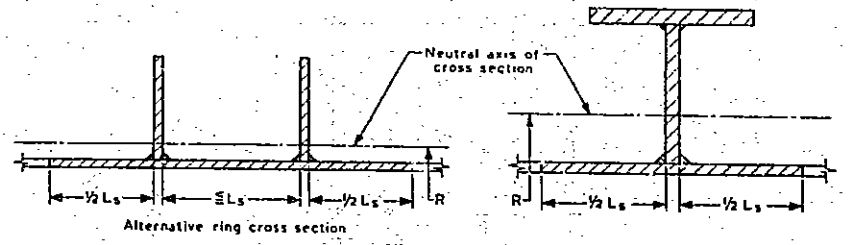



Fig. 14f.

Fig. 14. Stiffening rings for cylindrical vessels subject to external pressure



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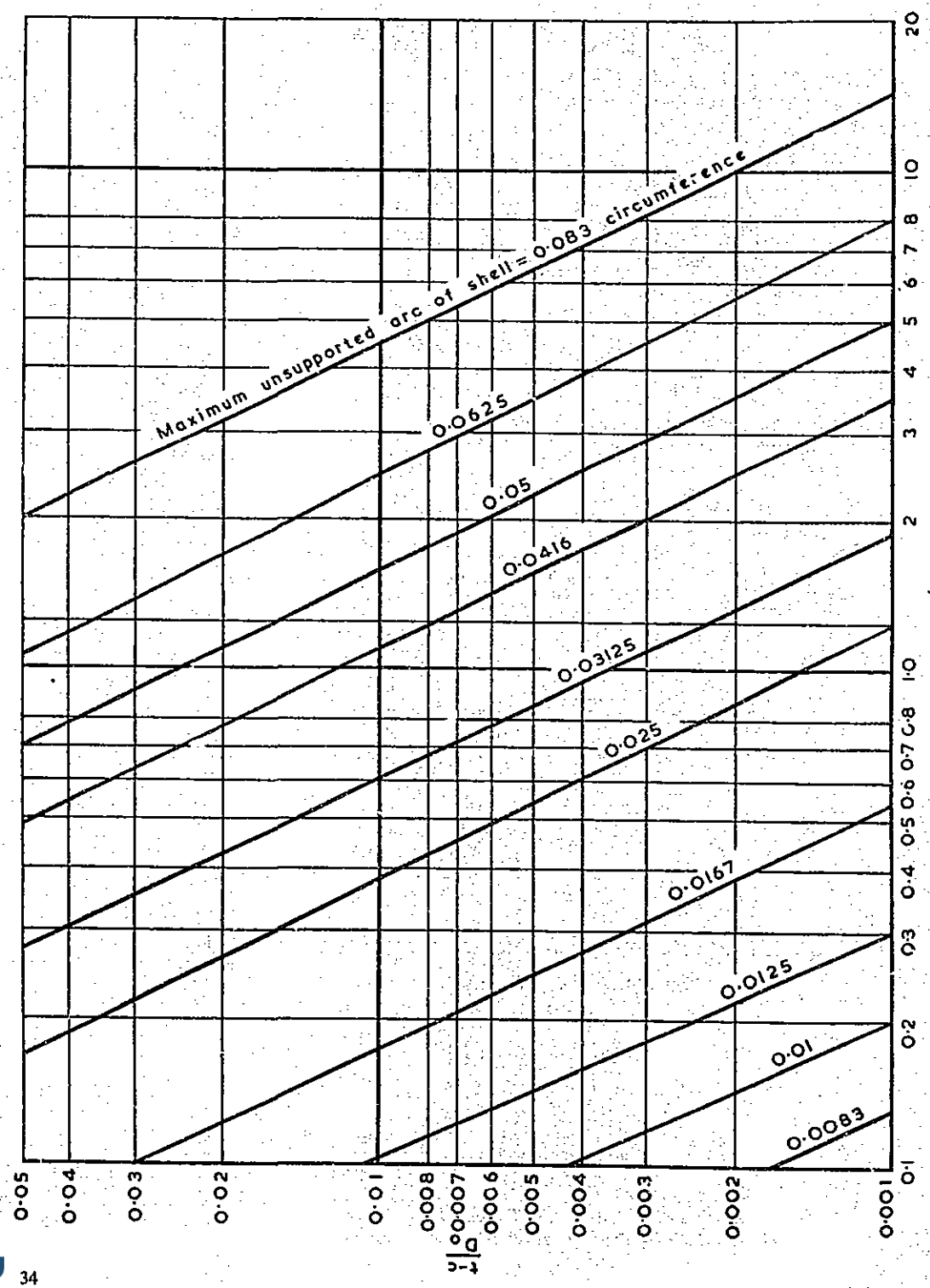


Fig. 15. Maximum permissible arc of shell unsupported by stiffening rings etc.





this stress in the expression $p = \frac{2f(t-c)}{R_o}$. If the value of $p$ is obtained so 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.

*d. Combined loadings.*

(i) *Vertical vessels.* For vertical vessels subject to external pressure the maximum direct axial compressive stress, resulting from the effects mentioned in Clause 2*d* above, shall be calculated as follows:

$$f = \left[\frac{pD_o}{4(t-c)} + \frac{48M}{\pi D_o^2(t-c)} + \frac{W}{\pi D_o(t-c)}\right] \qquad \text{Equation (11)}$$

where:

$p$ = design external pressure, lb/sq. in.

$M$ = bending moment at point under consideration in pounds feet.

$W$ = total weight at point under consideration in pounds.

The value of maximum axial stress obtained from Equation (11) shall not exceed the design stress at operating temperature taken from Table 2, and in no case shall it exceed the value of

$$0{\cdot}125E\left(\frac{t-c}{D_o}\right) \text{ lb/sq. in.}$$

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

$$f = \left[\frac{pD_o}{4(t-c)} + \frac{48M}{\pi D_o^2(t-c)}\right] \qquad \text{Equation (12)}$$

The value of maximum axial stress obtained from Equation (12) shall not exceed the design stress at operating temperature taken from Table 2, and in no case shall it exceed the value of

$$0{\cdot}125E\frac{(t-c)}{D_o} \text{ lb/sq. in.}$$

## STIFFENING RINGS

4. *a.* Intermediate stiffening rings composed of structural shapes welded solid to the inside or outside of the shell shall have a moment of inertia $I$ not less than that required by Equation (13):

$$I = \frac{0{\cdot}18\, D_o^3\, p\, L_s}{E} \qquad \text{Equation (13)}$$

The moment of inertia shall be calculated taking into account an effective length $L_s$ of the shell plate (see Fig. 14*f*):

$$L_s = 0{\cdot}75\sqrt{(t-c)\,D_o} \qquad \text{Equation (14)}$$

or $L_s = L$ (see Figs. 8 to 12), whichever is the smaller.

*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 (13). 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 Fig. 14, shall be the subject of special consideration, but in no case shall the length of unsupported shell plate exceed the value given in Fig. 15.

*e.* Rings for supporting trays, etc., in fractionating columns and similar constructions, may be used as stiffening rings provided that they are adequate for the duty.

*f.* Loads from internal bracing or supports shall be distributed through the vessel wall by means of a substantially continuous ring.

## PROTECTIVE LININGS

5. 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.



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# F. END PLATES

## GENERAL

1. Domed ends of hemispherical, semi-ellipsoidal or dished shape shall be designed in accordance with the following formulae. The calculated thickness shall be increased, if necessary, to provide for additional loading, enumerated in Section Three, A.

## LIMITATIONS OF SHAPE

2. End plates shall conform to one of the following shapes. (See B.S. 1966*):

*a.* Hemispherical. (See Fig. 16*a*).

*b.* Semi-ellipsoidal. (See Fig. 16*b*).

The ratio of major axis to minor axis should not be greater than 2·6 : 1.

*c.* Dished and flanged. (See Fig. 16*c*).

(i) The inside radius of dishing $R_i$ shall not be greater than the outside diameter $D_o$.

(ii) Wherever practicable, the inside corner radius $r_i$ shall be not less than 10 per cent of the inside diameter, but if this is impracticable it shall be not less than 6 per cent of the inside diameter, unless otherwise agreed between purchaser and manufacturer.

*d.* Conical. (See Fig. 20).

*e.* Flat and flanged. (See Fig. 23*a* and *b*).

*f.* Flat. (See Fig. 23*c* to *j*).

## SYMBOLS

3. The following symbols and values are used in the formulae. (See Fig. 16):

$t$ = minimum thickness, in inches, of ends.
$p$ = design pressure, in lb/sq. in.
$D_i$ = 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.
$h_i$ = 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.
$S_f$ = length of straight flange, after edge preparation.
$R_i$ = 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 internal corrosion allowance has been added.
$f$ = design stress, in lb/sq. in. from Table 2.
$J$ = welded joint factor as shown in Table 1.
= 1·00 for ends made from one plate.
$c$ = a corrosion allowance as agreed between purchaser and manufacturer but not less than as recommended in Section Three B.
$K_1$ = a factor depending on the ratio $h_o/D_o$ and obtained from Fig. 17.
In no case shall $K_1$ be taken as less than 1·15 $R_i/D_o$ or 0·12 $D_o/r_i$.
$K_2$ = a factor depending on the size and position of any openings in the end:

(i) For ends without holes or having only reinforced holes
$K_2 = 0{\cdot}94$.

(ii) For ends having unreinforced openings which satisfy the requirements of Section Three J
$K_2 = 1{\cdot}00$.

(iii) For ends having a central uncompensated opening, flanged inwardly or outwardly, which does not satisfy the requirements of Section Three J, and which shall not exceed 16 in. in any direction
$K_2 = 1{\cdot}15$.

(iv) For ends having an eccentrically located uncompensated opening, flanged inwardly or outwardly, which does not satisfy the requirements of Section Three J, and which shall not exceed 16 in. in any direction
$K_2 = 1{\cdot}15$,
plus an amount to be agreed between purchaser and manufacturer. It is recommended that such ends be subjected to a hydrostatic proof test in accordance with Section Five D.

(v) When a flanged opening is supported by an attached flue, etc.
$K_2 = 0{\cdot}94$.

$K_3$ = a factor for ends subject to internal pressure and depending on method of manufacture:
= 1·00 for ends spun or pressed in full or ring die in one or more operations.
= 0·95 for ends dished and sectionally flanged.

## THICKNESS OF DOMED ENDS CONCAVE TO PRESSURE

4. The thickness of the ends shall be determined by Equation (15):

$$t = \frac{p\,D_o\,K_1\,K_2}{2\,f\,J\,K_3} + c \qquad \text{Equation (15)}$$

or

$$p = \frac{2\,f\,J\,K_3\,(t - c)}{D_o\,K_1\,K_2} \qquad \text{Equation (16)}$$

NOTE 1. The external height of dishing $h_o$ may for ends of partially spherical form be determined from Equation (17) (see Fig. 16*c*).

$$h_o = R_o - \sqrt{\left(R_o - \frac{D_o}{2}\right) \times \left(R_o + \frac{D_o}{2} - 2r_o\right)} \qquad \text{Equation (17)}$$

NOTE 2. The limitation of the flange edge thickness (see Section Three K, Clause 4*c* (i).) may, for deep ends with low values of $K_1$ and particularly for hemispherical ends, actually determine the end thickness and overrule the thickness $t$ resulting from Equation (15).

## THICKNESS OF DOMED ENDS CONVEX TO PRESSURE

5. The symbols and values listed in Clause 3 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 Clause 3*c* of Section Three E.

(ii) The thickness $(t - c)$, as determined for a domed end under internal pressure by Clause 4 above, multiplied by the factor appropriate to the design

* B.S. 1966 Domed ends for tanks and pressure vessels.





Fig. 16a.

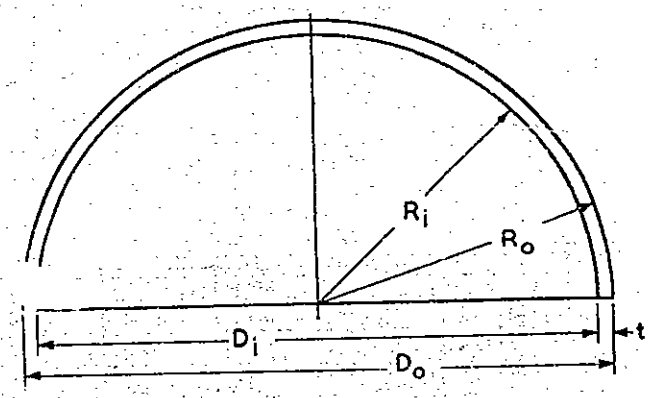


Fig. 16b.

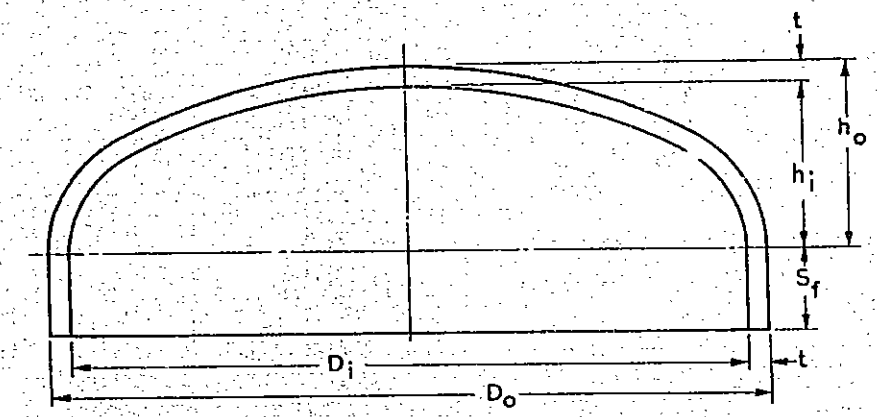


Fig. 16c.

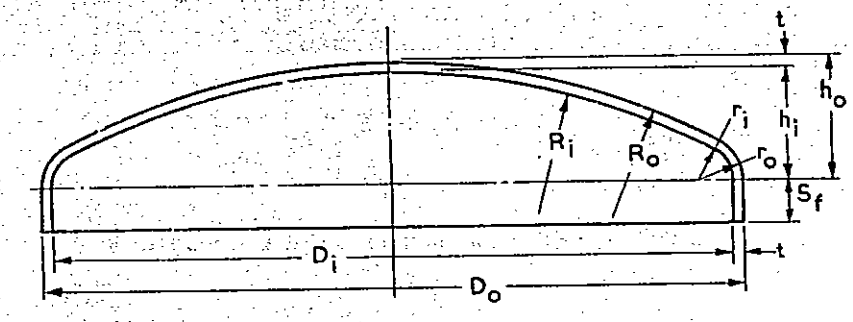


Fig. 16. Domed ends



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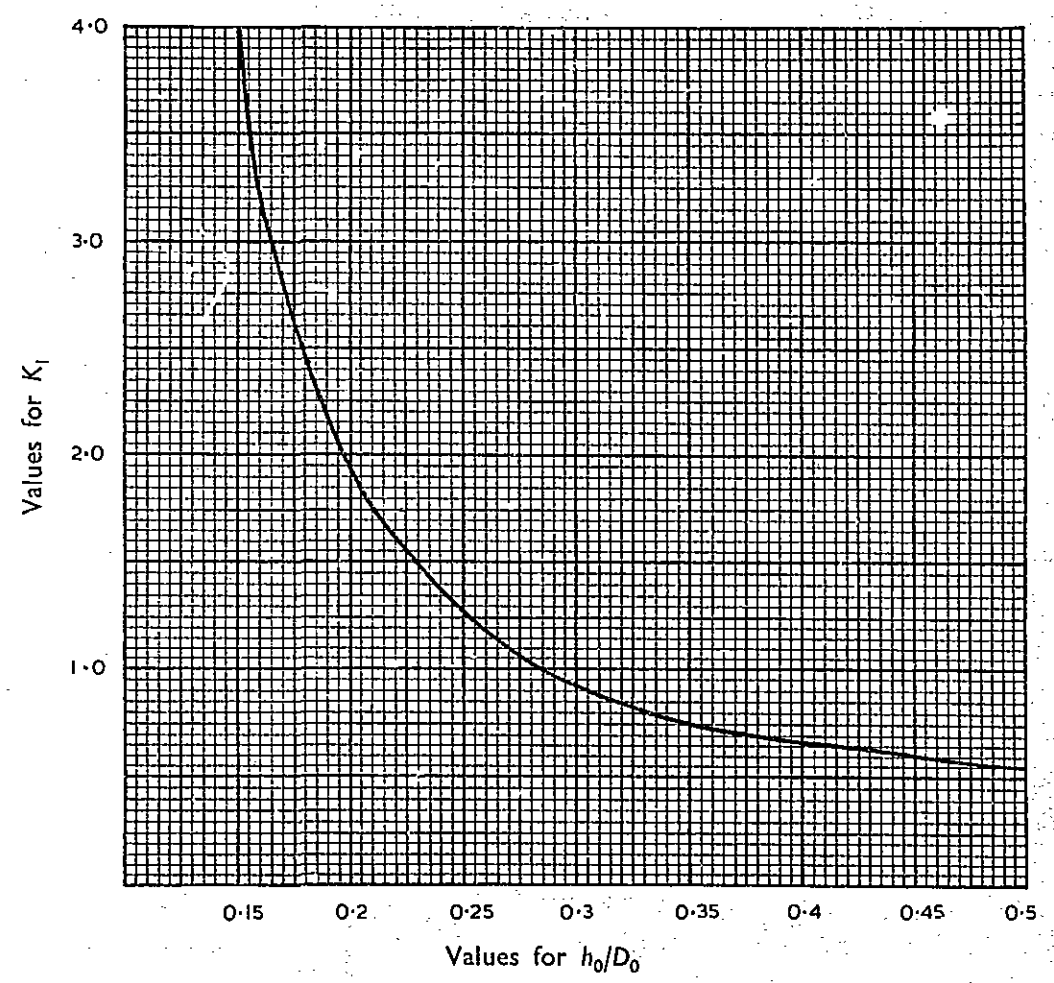



Fig. 17. $K_1$—A factor for use in equation 15, Clause 4, Section Three F





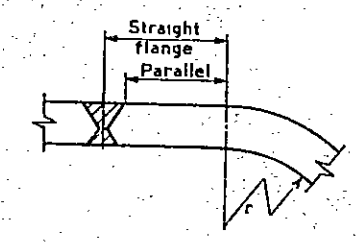
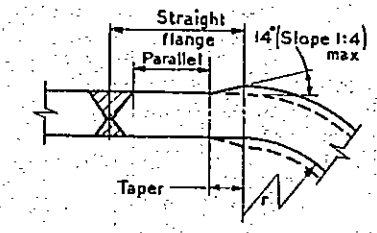
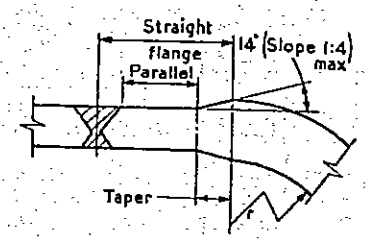
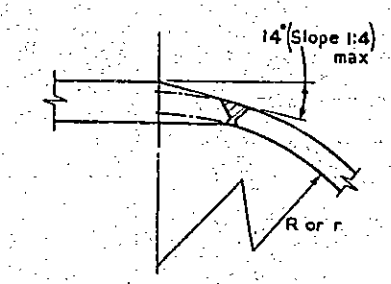
| Fig. | Connection to end of shell | Remarks |
|---|---|---|
| 18 a |   | A parallel portion with a minimum length of 1¼ in. shall be provided when the full length of the circumferential seam is to be fully radiographed, otherwise the length of the parallel portion shall be ½ in. minimum. See Fig. 4 for recommended weld preparation. |
| 18 b |   | A parallel portion with a minimum length of 1¼ in. shall be provided when the full length of the circumferential seam is to be fully radiographed, otherwise the length of the parallel portion shall be ½ in. minimum. See Fig. 4 for recommended weld preparation. |
| 18 c |   | A parallel portion with a minimum length of 1¼ in. shall be provided when the full length of the circumferential seam is to be fully radiographed, otherwise the length of the parallel portion shall be ½ in. minimum. See Fig. 4 for recommended weld preparation. |
| 18 d |   | Suitable for the connection of a hemispherical end thinner than the shell.<br><br>See Fig. 4 for recommended weld preparation. |

Fig. 18. Typical end connections to shells



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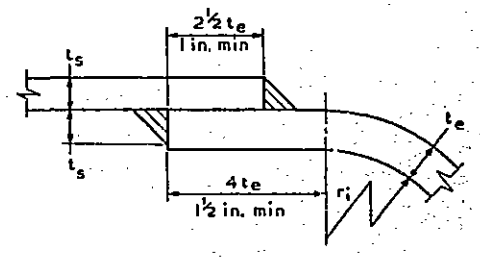
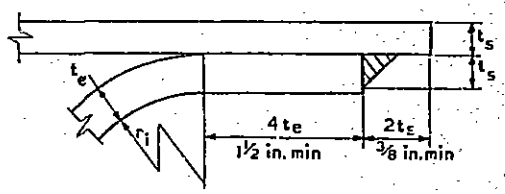
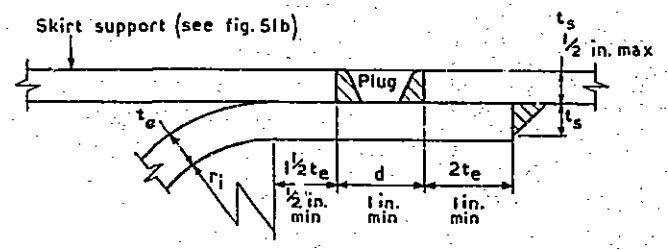
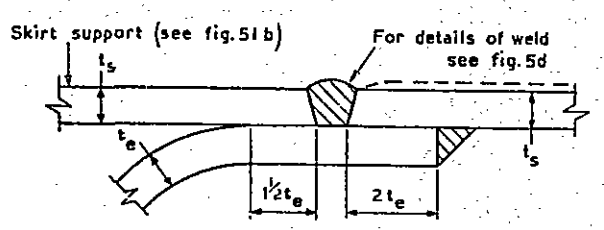
| Fig. | Connection to end of shell | Remarks |
|---|---|---|
| 18 e |   | Permitted on shells not exceeding ⅝ in. thick. For Class III vessels only. |
| 18 f |   | Permitted on shells not exceeding:<br>(i) 24 in. internal diameter<br>(ii) ½ in. thick.<br>For Class III vessels only. |
| 18 g |   | Permitted on shells not exceeding:<br>(i) 24 in. internal diameter<br>(ii) ½ in. thick.<br>For Class III vessels only.<br>Welds: Plugs and one fillet |
| 18 h |   | Permitted on shells not exceeding:<br>(i) 24 in. internal diameter<br>(ii) ½ in. thick.<br>For Class III vessels only.<br>May also be used as intermediate partitions. |

Fig. 18. Typical end connections to shells





temperature obtained from Table 5, which is intended to give a constant factor of safety based upon the yield point of the material.

TABLE 5

| Design temperature °F | Factor |
|---|---|
| Up to 450 | 1·0 |
| 500 | 1·10 |
| 600 | 1·30 |
| 700 | 1·475 |
| 800 | 1·30 |
| 850 | 1·125 |
| 900 | 1·0 |

*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 Clause 3*c* of Section Three E. 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 |
| $R_o/D_o =$ | 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 | |
| $R_o/D_o =$ | 0·81 | 0·73 | 0·65 | 0·57 | 0·50 | |

(ii) The thickness $(t - c)$ as determined for a domed end under internal pressure by Clause 4 above, multiplied by the factor appropriate to the design temperature obtained from Table 5.

LOCAL FLATS

6. When ends are pressed with local flat areas for fittings, etc., the diameter of such flats should not exceed that allowed for flat ends in Section Three H.

STAYED ENDS

7. When ends are of thickness less than that required by the formulae and rules in this section, they shall be stayed as flat surfaces in accordance with Section Three M.

CONSTRUCTIONAL DETAILS

8. *a.* Fig. 18 shows permitted types and limitations for connections between ends and shells.

*b.* The single-welded lap connections as shown on Fig. 18*f*, *g* and *h*, should be discouraged. They should only be used where there is no practicable alternative.

*d.* Forms of connection which are prohibited under this code are illustrated in Fig. 19.

*e.* Hemispherical ends may be fitted without a parallel portion.

*f.* The depth of flange $H_3$, as shown in Fig. 29, measured on the minor axis, shall be not less than:

$\sqrt{tW}$

where:

$t$ = thickness of end in inches.
$W$ = width of minor axis in inches.

The corner radius between flange and shell shall be not less than 1 in.

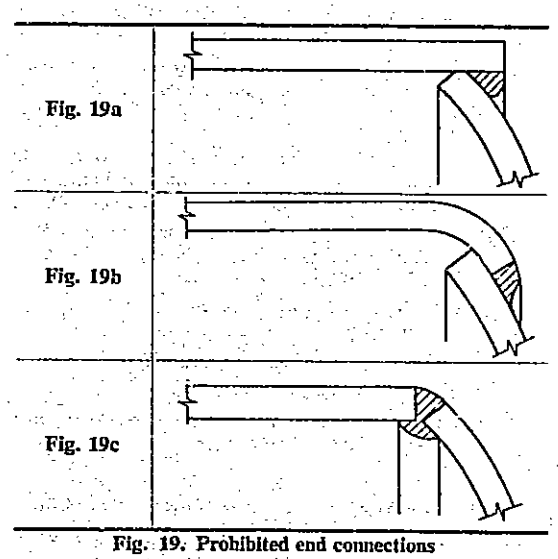



Fig. 19. Prohibited end connections



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# G. CONICAL ENDS

## GENERAL

1. Conical ends and conical reducing sections, as shown in Figs. 20 and 21, shall be designed in accordance with the following formulae. The calculated thickness shall be increased, if necessary, to provide for additional loading, enumerated in Section Three A and B.

Conical ends may be constructed of several ring sections of decreasing thickness as determined by the corresponding decreasing diameter.

## SYMBOLS

2. The following symbols and values are used in the formulae:

$t$ = minimum thickness of plate, in inches.

$p$ = design pressure, lb/sq. in.

$f$ = design stress in lb/sq. in. as listed in Table 2.

$J$ = joint factor of any longitudinal welded seam as given in Table 1.

$D_c$ = inside diameter in inches of the conical end or section at the position under consideration, before corrosion allowance is added (See Figs. 20 and 21.)
When ends are constructed of several ring sections, $D_c$ is the maximum internal diameter of the section whose thickness is to be determined.

$D_i$ = inside diameter in inches of the conical end or section at the point tangential to the knuckle. (See Figs. 20 and 21.)

$D_o$ = outside diameter in inches of the conical end or section at the position under consideration.

$r$ = inside radius of transition knuckle, in inches.

$c$ = corrosion allowance, in inches.

$\alpha$ = one-half of the included apex angle of the conical head or section. (See Figs. 20 and 21.)

## CONICAL ENDS SUBJECT TO INTERNAL PRESSURE

3. *a.* The thickness of conical ends subject to internal pressure shall be determined by:

$$t = \frac{pD_c}{\cos\alpha\,(2fJ - p)} + c \qquad \text{Equation (18)}$$

or

$$p = \frac{2fJ\,(t - c)\cos\alpha}{D_c + (t - c)\cos\alpha} \qquad \text{Equation (19)}$$

*b.* When angle $\alpha$ exceeds 30°, conical ends shall be attached to the shell by a knuckle having an inside radius preferably not less than 10 per cent of the internal diameter of the shell to which it is attached, but in no case less than 6 per cent of the shell internal diameter or three times the thickness of the knuckle. The knuckle thickness shall be at least equal to the thickness required for a domed end as determined by Section Three F, substituting,

$$\frac{D_i}{2\cos\alpha}$$

for the inside spherical radius $R_i$, where $D_i$ is the inside diameter of the conical end at the point tangential to the knuckle as shown in Fig. 20*a*.

*c.* When angle $\alpha$ does not exceed 30°, and provided the design conditions of $p/fJ$ are contained within the area above the curve of Fig. 22, conical ends may be attached with or without transition knuckles.

*d.* When angle $\alpha$ does not exceed 30° and the design conditions of $p/fJ$ fall below the curve of Fig. 22, the use of transition knuckles is recommended. Under these conditions the attachment of conical ends not provided with transition knuckles may require compression rings of the type shown diagrammatically in Fig. 20*b*, and such cases shall be the subject of special agreement between purchaser and manufacturer.

## CONICAL ENDS SUBJECT TO EXTERNAL PRESSURE

4. 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 Clauses 1, 2 and 3 above are applicable, except that the thickness shall be determined as prescribed below.

*a.* The thickness of a conical end or conical section under external pressure, when half of the included (apex) angle of the cone is less 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 equal to the slant height of the cone or conical section, or slant height between the effective stiffening rings whichever is less.

*b.* When half of the included (apex) angle is 75° or greater the thickness shall be determined as for a flat cover as prescribed in Section Three H.

## CONSTRUCTIONAL DETAILS

5. *a.* Fig. 18 shows permitted types of and limitations for connections between ends and shells.

*b.* The single-welded lap connections as shown on Fig. 18*f*, *g* and *h* should be discouraged. They should only be used where there is no practicable alternative.





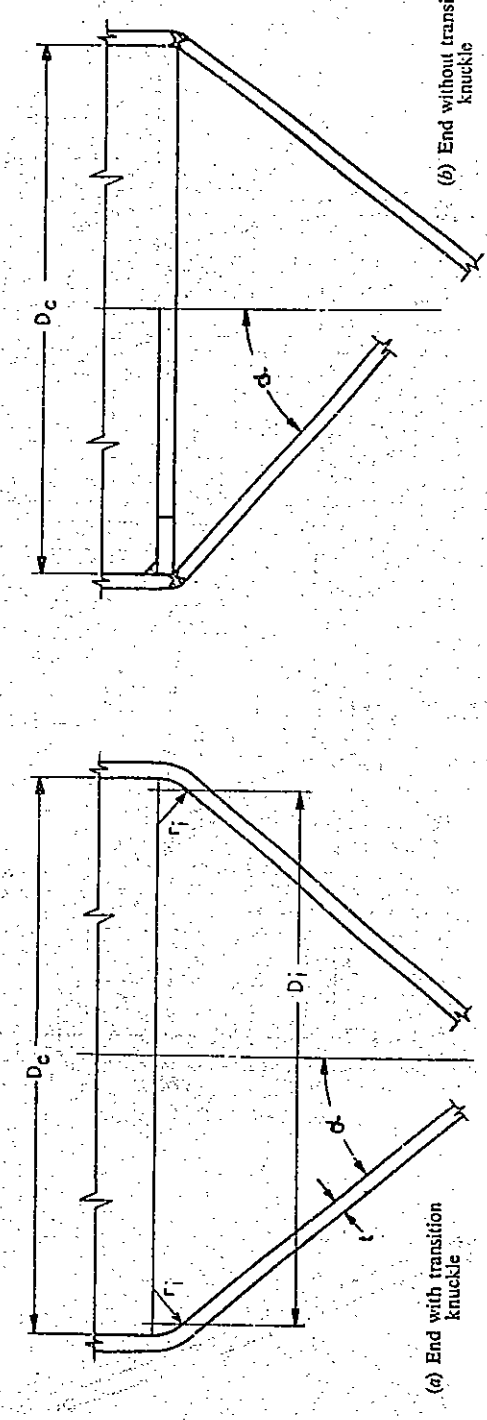


Fig. 20. Alternative forms of conical ends



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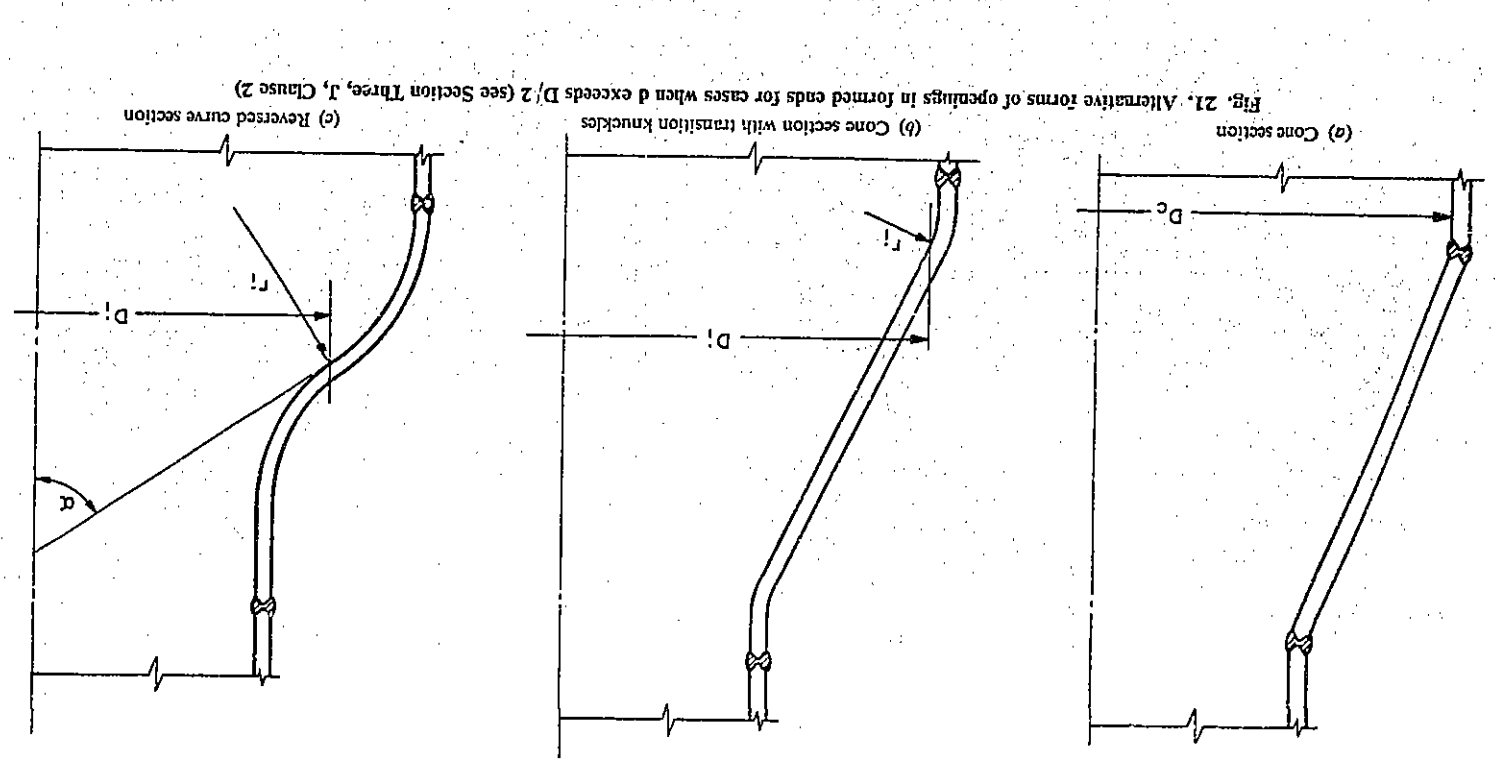


Fig. 21. Alternative forms of openings in formed ends for cases when d exceeds $D_i/2$ (see Section Three, J, Clause 2)





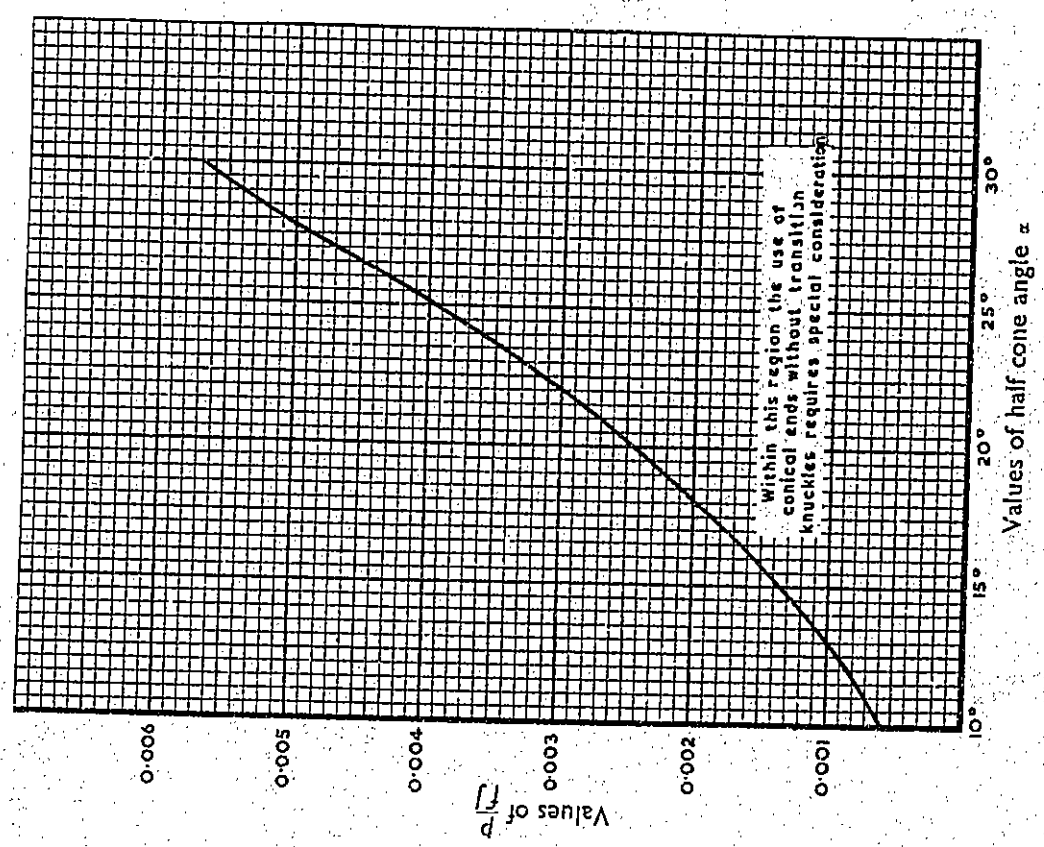


Fig. 22.



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# H. FLAT COVERS AND ENDS

## GENERAL

1. *a.* Flat covers and end plates shall be designed in accordance with the following formulae, and the thickness shall be increased, if necessary, to carry the additional loads enumerated in Section Three A.

*b.* Fig. 23 shows typical flat ends and covers, with approved methods of attachment to the shell.

## SYMBOLS

2. The following symbols and values are used in the formulae:

$t$ = thickness of end in inches.

$p$ = design pressure in lb/sq. in.

$D$ = diameter in inches of the plate measured as indicated in Fig. 23.

$f$ = design stress in lb/sq. in. from Table 2.

$c$ = a corrosion allowance as agreed between purchaser and manufacturer but not less than as recommended in Section Three B.

$K$ = a constant which depends upon method of attachment to shell and is given in Column 4 of Fig. 23.

## THICKNESS

3. The thickness of flat covers and end plates shall be calculated from Equations (20) and (21):

$$t = D\sqrt{\frac{p}{Kf}} + c \qquad \text{Equation (20)}$$

$$p = \frac{Kf(t-c)^2}{D^2} \qquad \text{Equation (21)}$$

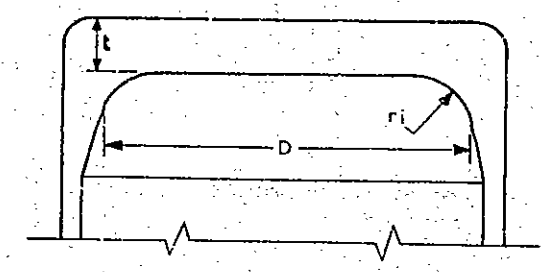
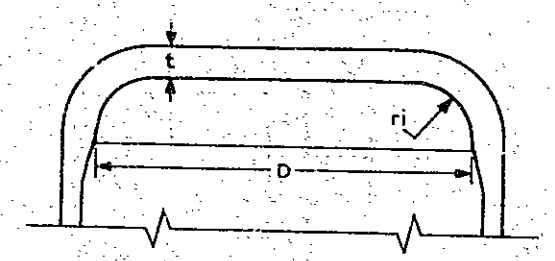
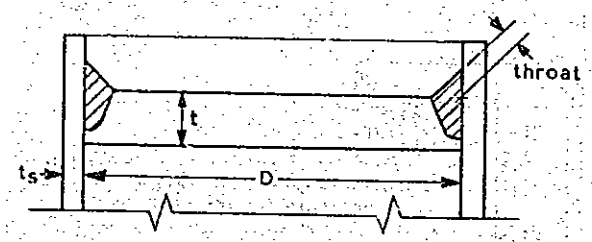
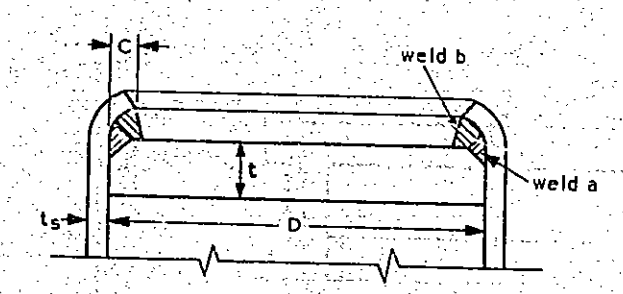
## TYPES OF FIXING

4. *a.* Attachments as shown in Fig. 23 shall be designed to have a minimum factor of safety of 4, against failure in tension or shear.

*b.* Where bolted connections are used as in Fig. 23 these shall comply with Section Three M.





| Fig. | Type | Limitations | K |
|---|---|---|---|
| 23 a |   | $K=\frac{4}{\left[D-r_i\left(1+\frac{2r_i}{D}\right)\frac{1}{12}\right]^2}$<br>$D$=Inside diameter of end in inches<br>$r_i$ =Inside corner radius in inches, not less than 3 times flange thickness | |
| 23 b |   | | |
| 23 c |   | Fillet throat not less than 1·25 times $t$ or $t_s$ whichever is smaller | 4·0 |
| 23 d |   | $C=\frac{t_s}{3}$ but not less than $\frac{3}{16}$ in. Weld 'a' is made before the shell is crimped over. | 3·0 |

In the forms of construction illustrated in Figs. 23 *c*, *d* and *g*, the hazard of weld cracking must be taken into consideration

**Fig. 23. Typical flat ends and covers**

Weld profiles are diagrammatic only



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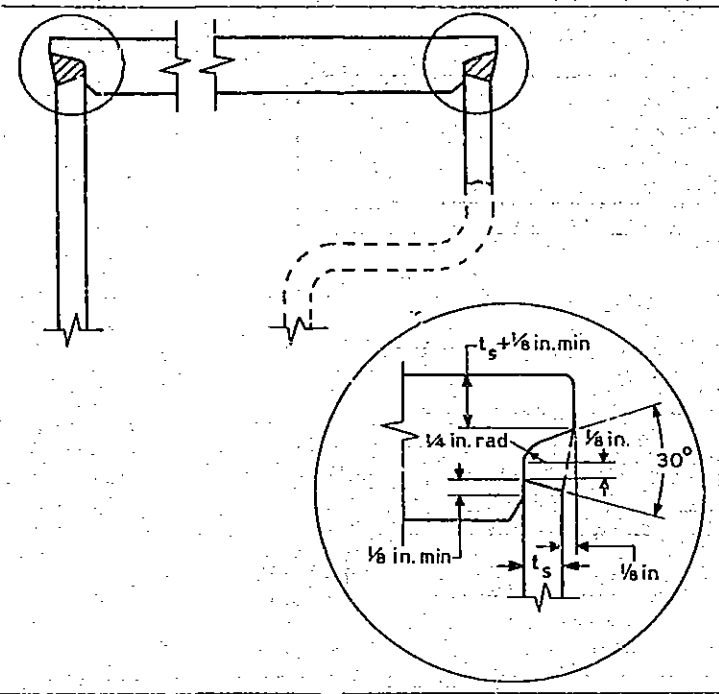
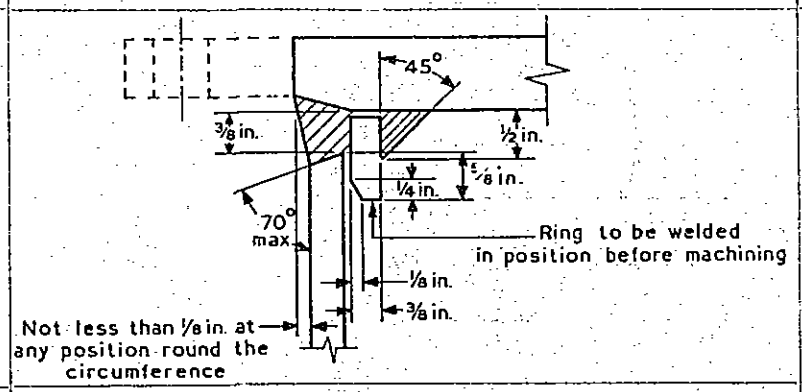
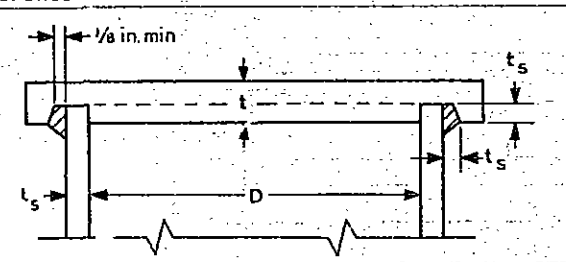
| Fig. | Type | Limitations | K |
|---|---|---|---|
| 23 e |   | | 5·0 |
| 23 f |   | | 4·0 |
| 23 g |   | $t$ not less than $2t_s$ | 3·0 |

In the forms of construction illustrated in Figs. 23 *c*, *d* and *g*, the hazard of weld cracking must be taken into consideration

**Fig. 23. (contd.) Typical flat ends and covers**

Weld profiles are diagrammatic only





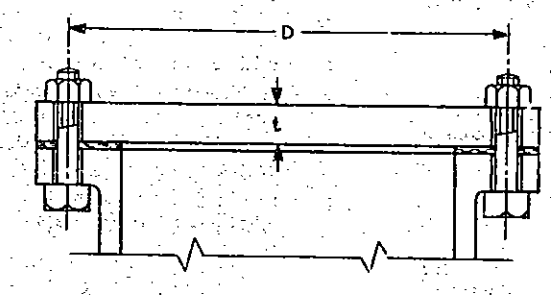
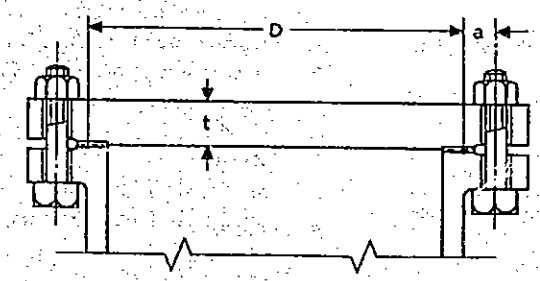
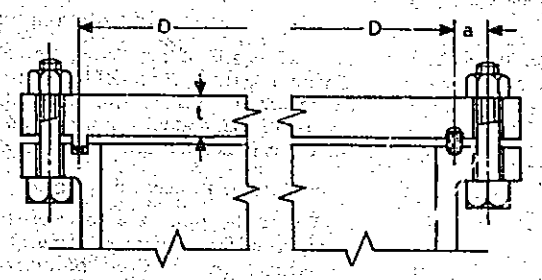
| Fig. | Type | Limitations | K |
|---|---|---|---|
| 23 h |   | Gasket shall be full width of flange. | 6·0 |
| 23 i |   | $K = \dfrac{1}{\left[0{\cdot}30 + \dfrac{1{\cdot}5 \times F_b \times a}{F \times D}\right]}$<br>$F_b$ = Total bolt load in lb<br>$a$ = Lever arm of bolts about centre line of ring or gasket<br>$F$ = Total internal pressure load on area bounded by mean diameter of joint ring or gasket.<br>$D$ = Mean diameter of gasket. | |
| 23 j |   | | |

**Fig. 23. (contd.) Typical flat ends and covers**



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